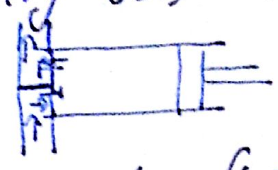
① Compressor: To compress gases and vapour from ↓ to ↑ pressures
→ P ↑ → T ↑ → ρ ↑.

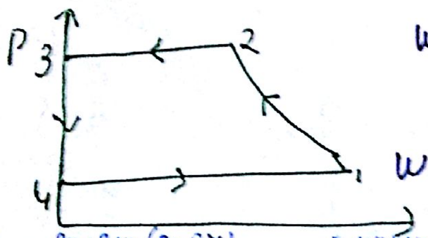
+ve displacement : Rec, Roots, vanes blower
[Pr. ↑ by ↓ing volume of gas]
Rotary steady flow : Centrifugal, Axial.
[↓ velocity gas (rotor) ↑ in K.E. ↑ in pressure (diffuser) ↑ in pr. by ↓ in K.E.]

② (i) Fan (<1.1) (ii) Blower (1.1–2.3) (iii) Compressor (>2.3)
* If we take the K.E. given by it to diffuser — Also a compressor. * Depends on pr. Ratio

③ Pressure [10–80–1000 bar] Volume [~9–3000– m^3/min]

④ Rec. Compressor : (2-stroke cycle)
→ Inlet outlet valve (pressure operated, sucⁿ in, delivery out)
→ A.C, fridge, mining, etc.
→ Slow speed, ↑ pressure, ↓ volume, ↑ losses.



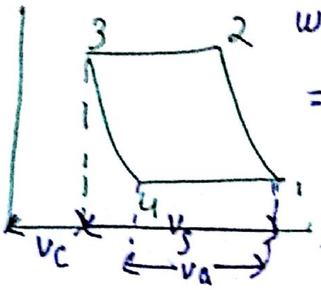


$$W = \frac{n}{n-1} \cdot P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W = \frac{n}{n-1} \cdot mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$W_{1-2} = \frac{P_1V_1 - P_2V_2}{n-1}$ (By Air)
$W_{1-2} = \frac{P_2V_2 - P_1V_1}{n-1}$ (on comp.) V

* volume of air delivered = V_1
* mass " " " = $\frac{P_1V_1}{RT_1}$



$$W = \frac{n}{n-1} \cdot P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$
$= W_C - W_E$
$$W = \frac{n}{n-1} \cdot mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

* now volume delivered = $V_1 - V_4$

Work I/P = work in comp 1-2 + work in discharge − work in expansion − work in sucⁿ.

⑤ Why V_c → [Damage to cylinder, valves, piston Ring seize] (in case of Thermal expansion)

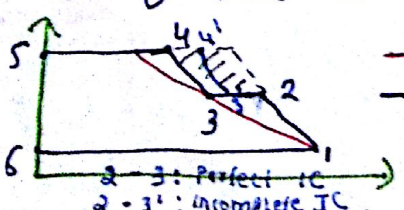
Effect of V_c: work Ind of it ∵ it ↓ volume ∴ No. of cycles for same work ↑.
* For same qty. of air to be delivered, total work would be same, only no. of cycle will be ↑ in 2nd case as vol./cycle ($V_1 - V_4$) is less than V_1.

⑥ Heat Transfer to cyl. wall during compression → $\frac{\gamma - n}{\gamma - 1} \cdot dW_{poly}$ or $h_1 + Q = h_2 + W_{net}$ → don't use only W_{1-2}
* or by $dQ = du + dW_{1-2}$
(−) (+) (−)

⑦ Volumetric $\eta_V = 1 + C - C \cdot \left(\frac{P_2}{P_1}\right)^{1/n}$ * $C = \frac{V_C}{V_S}$
→ 100% $P_1 = P_2$ but no compression
→ ↓ with ↑ in P_2. * till $P_2/P_1 = \left[\frac{1+C}{C}\right]^n$ with $\eta_V = 0$
→ 100% if C = 0
→ ↑ with ↑ n **INDEX OF EXPANSION**

Mean piston speed = 2LN/60
– C ↑es with ↑ in mps
$\left(\frac{P_2}{P_1}\right)_{max}$ for $\eta_V = 0$ ↳ $= \left[1 + \frac{1}{C}\right]^n$

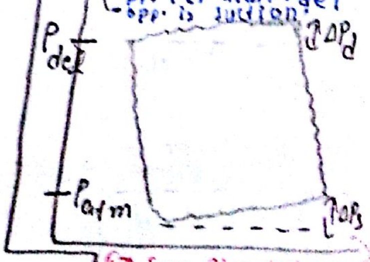
⑧ MULTISTAGING WITH INTERCOOLER:
→ To ↑ P_2 w/o ↓ η_V → To ↓ outlet temp.
→ To ↓ size of cylinder →



2 – 3 : Perfect IC
2 – 3' : incomplete IC

— Isothermal	Single	5.6 bar
— polytropic	Double	5.6–30 "
$T_2 = T_3$	Triple	30–120 "

⑫ Actual cycle:
– Del. valve open at Pr. ↑er than P_{del}
– opp. is suction.



⑨ # $P_2 = \sqrt{P_1 \cdot P_3} \cdot \left(T_3/T_1\right)^{1/2x}$ * For min. work, $x = \frac{n-1}{n}$
$W = n \cdot W_I$ if perfect IC
$\frac{P_{n+1}}{P} = K^n$
$\frac{P_2}{P_1} = K$.

⑬ Free Air delivered: Vol. of Air delivered when Reduced to 1 bar and 15°C.
$\frac{1.013 \times V_f}{288} = \frac{P_1(V_1 - V_4)}{T_1}$

⑩ $\eta_{V_1} \cdot D_1^2 \cdot P_1$ = Constant.

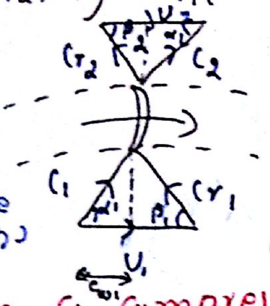
⑪ After cooler : To ↓ Receiver size

① **Energy Transfer** : only in moving part (E.g. mech. Energy (Shaft) → fluid Energy (fluid))
- change of carrying medium

Energy Transformation : Both in moving and fixed (E.g. K.E. → P.E.)
- change in forms of Energy

② Rotating m/c or Eulers principle : Torque $\tau = \dot{m} \cdot C_{w_2} \cdot r_2 - \dot{m} C_{w_1} r_1$

- Fluid enters & exits Rotor at r_1 and r_2.
- Relative = Abs - Abs



α = Absolute vel ∠.

β = Blade ∠.

$U_1 = \omega r_1 \quad U_2 = \omega r_2$

$\vec{C_1} = \vec{U_1} + \vec{C_{r_1}} \quad \vec{C_1} = \vec{C_{w_1}} + \vec{C_{f_1}}$

$C_2 = \vec{C_{r_2}} + \vec{U_2} \quad \vec{C_2} = \vec{C_{w_2}} + \vec{C_{f_2}}$

Power $= \tau \cdot \omega$

Power $= \dot{m} \cdot C_{w_2} \cdot U_2 - \dot{m} C_{w_1} U_1$

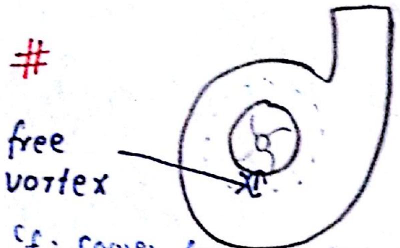
Power/kg $= C_{w_2} U_2 - C_{w_1} U_1$.

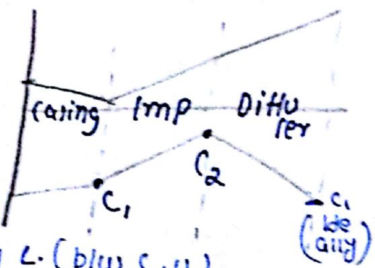
③ In C. compressor: $C_{w_2} U_2 - C_{w_1} U_1 = \left[\dfrac{C_2^2 - C_1^2}{2}\right] + \left[\dfrac{U_2^2 - U_1^2}{2}\right] + \left[\dfrac{C_{r_1}^2 - C_{r_2}^2}{2}\right]$

Ext. effect — Int. Effect

→ Impulse + centrifugal + Diffusion (Due to diverging area)

ΔP in Diffuser — ΔP in m. blade

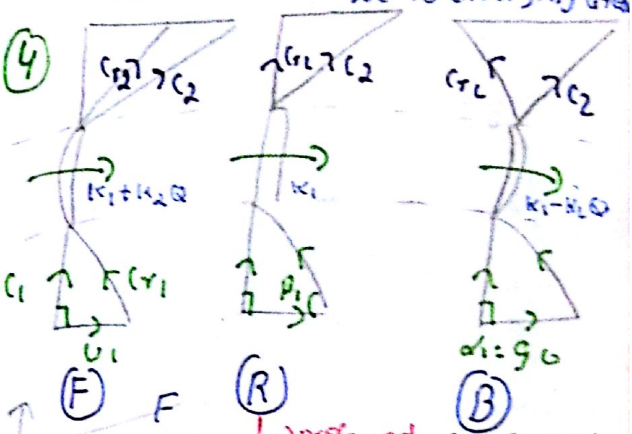




- C_f : Causes flow
- C_w : " Rotn

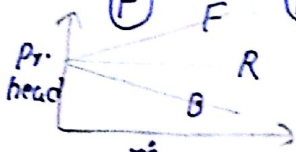
α = Abs. velocity ∠. (b/w C, U)
β = Blade ∠. (b/w C_r, U)

④



⑤ work, P. Ratio, I/P : F > R > B

η, stable, operating Range : B > R > F



(R) → preferred for ↑ speed b/c in F, B cen. effect of curved blade produces stress.

In (B) Pr. head $= \dfrac{C_{w_2} \cdot U_2}{g} = \dfrac{U_2}{g}\left(U_2 - C_{f_2} \cot\beta_2\right) = \dfrac{U_2^2}{g} - \dfrac{U_2 \cot\beta_2}{g \cdot A_2} \cdot Q = K_1 - K_2 Q$ $\quad Q = A_2 \cdot C_{f_2}$

⑥ slip in Radial blade : ∵ T.Edge (↑ pr), L.Edge ↓ (pr.)

∴ fluid deviates from its path (C_{w_2} less than U_2 for Radial blade)

→ Blade becomes like backward $(U_2 \cdot \phi_s = C_{w_2})$

Power I/P $= \dot{m} \cdot C_{w_2} \cdot U_2 = \dot{m} \cdot \phi_s \cdot U_2^2$ * per stage

$\phi_s \leq 1 \quad \phi_w > 1$

⑦ work I/P factor → $P = \dot{m} \cdot \phi_s \cdot \phi_w \cdot U_2^2$

>1 ∵ Actual work > Theoretical work due to loss

Mass flow Rate → $C_1 \cdot A_1 \cdot v_{f_1} = C_2 \cdot A_2 v_{f_2}$

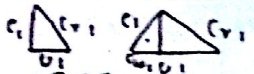
Also $A_1 = \frac{\pi}{4}\left[D_1^2 - D_h^2\right]$

Impeller (Shape of blade) → Vaneless space (- static pr. ↑ - free vortex flow) → Diffuser → Spiral casing called volute → dis. pipe

$A_1 = \pi D_1 b_1$

⑧ pre-whirl : In inlet blade to reduce C_{r_1}.

$M = \dfrac{C_{r_1}}{\sqrt{\gamma R T_1}}$



$P_{I/P} = \dot{m}\left[C_{w_2} \cdot U_2 - C_{w_1} \cdot U_1\right]$

* Initial Rotn to fluid by fixed guide vanes at inlet.

η ↓, $\dot{m}$ ↓, Pr per stage = ↑, ↓ U_1, stable Range of op.

⑨ work by SFEE ∵ $h_1 + \dfrac{C_1^2}{2} = h_2 + \dfrac{C_2^2}{2} + W$

∴ $W_{I/P} = h_{0_2} - h_{0_1}$

* NOTE : if stagnation given use → ; if static given use $W = h_2 - h_1$ (Assume ΔK.E. very ↓)

→ Equate both

* Isentropic η use $2s$

C. compressor : - Entry : Axially ($\alpha_1 = 90°$) thru impeller eye at hub
- Exit : Radially at outer tip of impeller

* In comp., pr. Rise due to cen. (U) and Diff (C_r) and K.E. ↑ due to impulse (C) which ↑ Pr. in diffuser

* No multistaging : ∵ In every stage (Axial → Radial) ∴ 90° turn

* on entry : little pr. drop due to such which causes ↑ to C_1 at entry.

* Acc. Nozzle (Accelerate from inlet to IGV) → IGVs (Directs flow in desired direcn to Entry of impeller) → Impeller (Transfer shaft work to the fluid energy - Inducer Receives flow b/w axial and passes to Radial portion of Imp. - flow to Imp. may be with or w/o swirl)

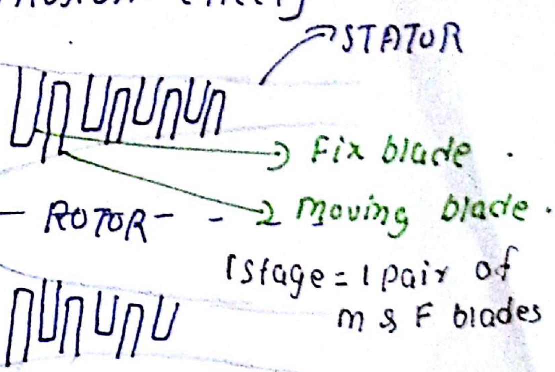
① A. Compressor → $U_1 = U_2$ so only [Impulse + Diffusion Effect]
* AXIAL FLOW * no centrifugal effect

→ Pr. Ratio (1·3 – 1·4 in each stage)

→ Adv: staging can be done

→ volume: ↑

→ principle: Reaction.

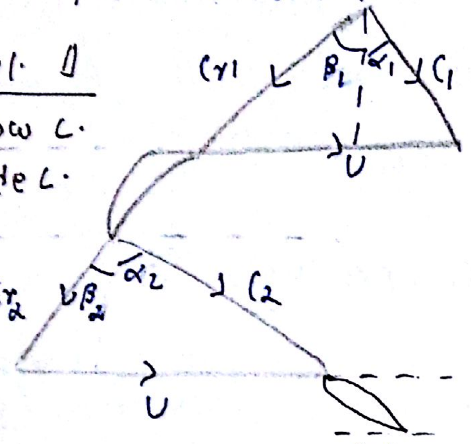


moving → $C_1 \xrightarrow{\text{Tes to}} C_2$ KE↑
moving → $C_{r1} \xrightarrow{\text{des to}} C_{r2}$ Pr.↑ – ROTOR –
Fixed → $C_2 \xrightarrow{\text{des to}} C_1$ Pr.↑

[Stage = 1 pair of M & F blades

② vel. △

α = Flow ∠.
β = Blade ∠.



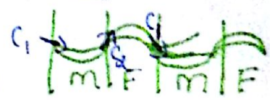
Imp:

$U = C_f(\tan\alpha_1 + \tan\beta_1)$

$U = C_f(\tan\alpha_2 + \tan\beta_2)$

$W = \dot{m} \cdot C_f \cdot U(\tan\beta_1 - \tan\beta_2)$

$= \dot{m} \cdot C_f \cdot U(\tan\alpha_2 - \tan\alpha_1)$



* C_f constant to keep Axial Thrust 0

$F_{axial} = \dot{m}(C_{f1} - C_{f2})$

③ flow coeff. $\phi = C_f/U = 1/(\tan\alpha_1 + \tan\beta_1) = 1/(\tan\alpha_2 + \tan\beta_2)$

④ R = Degree of Reacⁿ = $\frac{\Delta h_m}{\Delta h_s} = \frac{\Delta h_m}{\Delta h_m + \Delta h_f}$

$$R = \frac{\frac{C_{r1}^2 - C_{r2}^2}{2}}{C_{w2} \cdot U_2 - C_{w1} U_1} = \frac{C_f^2(\sec^2\beta_1 - \sec^2\beta_2)}{2 \cdot U \cdot C_f(\tan\beta_1 - \tan\beta_2)} = \frac{C_f}{2U}[\tan\beta_1 + \tan\beta_2]$$

$\dot{m} = \rho_1 \times C_f \times \pi D h_1$

⑤ For R = 1/2 (50%) $\alpha_1 = \beta_2$, $\alpha_2 = \beta_1$, $C_1 = C_{r2}$, $C_2 = C_{r1}$.
* pressure Rise equally divided b/w Rotor & stator stages

⑥ Frontal Area = $\pi(r_t^2 - r_h^2) = \pi \cdot D_m \cdot H$

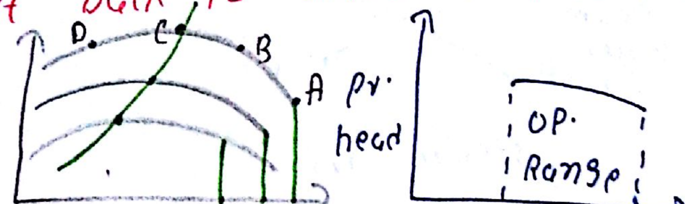
$H = r_t - r_h$ = ht. of blade
$D_m = r_t + r_h$

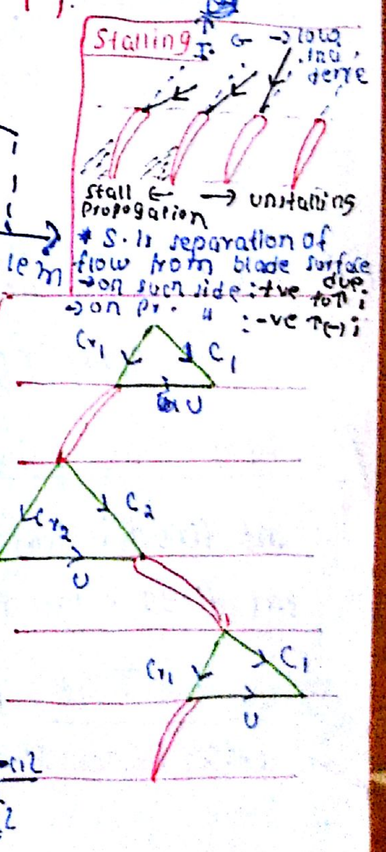
$\dot{m} = \rho_1 A_1 V_{f1} = \rho_2 A_2 V_{f2}$ For ρ_1 use $P_1\ T_1$ not $P_{01}\ T_{01}$

⑦ may have to convert back to static to find P_1.

⑧ → Surging ($\dot{m}_{min}$)

→ Chocking ($\dot{m}_{max}$)





from A → C on ↓ing $\dot{m}$ ⇒ Pr. head ↑es ∴ $\dot{m}$ ↑er than des pr. no problem
from C → D " " " ⇒ Pr. " " ⇒ breakdown of steady flow

⑨ polytropic η : Constant throuout stages

overall η : ↑ than η_{pc} and ↑ with pressure

⑩ Half-life : Th-232 > U-238 > U-235

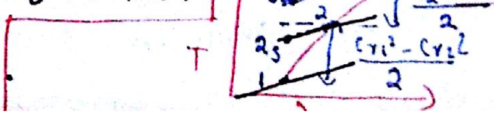
⑪ Fuel Rods clad by : Zirconium, SS, Al

Moderators : H_2O, D_2O, graphite, Beryllium

Coolant : H_2O, D_2O, CO_2, liq metal

Reflector : mod or coolant

Control Rod : Cd, B, Hafium

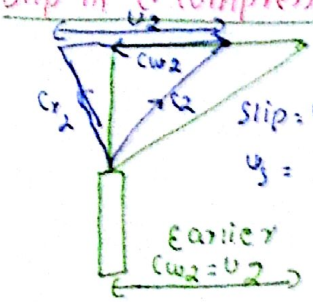


Power Plant: Fuel (che. Energy) → C.chamber → Heat Energy ($\uparrow$ pressure) → Nozzle → K.E. → Turb/Ma → mech. Power (shaft) → Gens Power factor → E. power

* Turbine = Nozzle + Rotor

* A heat Engine: Source : C.chamber; H.E. : Turbine; Sink : Ambient

Slip in C. compressor:



Slip $= U_2 - C_{w2}$

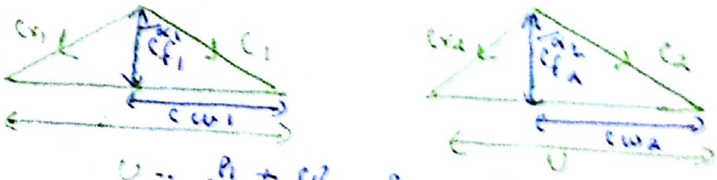
$\sigma = \frac{C_{w2}}{U_2} \le 1$

Rotating machines

Pump, Comp. $W_{i/p} = C_{w2} \cdot U_2 - C_{w1} \cdot U_1$

Turbine. $W_{o/p} = C_{w1} \cdot U_1 - C_{w2} \cdot U_2$

Axial compressor:



$C_{f1} = C_{f2} = C_f$

$F_{axial} = \Delta P \times Area$

$F_{whirl} = \dot{m}(C_{w1} - C_{w2})$

$\therefore \frac{P_1}{\rho} + \frac{C_1^2}{2} = \frac{P_2}{\rho} + \frac{C_2^2}{2}$ $\therefore (\Delta P)_0 = \rho\left(\frac{C_1^2 - C_2^2}{2}\right) = \frac{\rho}{2} \cdot C_f^2 (\tan^2\alpha_1 - \tan^2\alpha_2)$

$F_{axial} = \rho \cdot S \cdot C_f^2 (\tan\alpha_1 - \tan\alpha_2) \cdot \tan\alpha_m$ $S = Area\ m$, $\tan\alpha_m = \frac{\tan\alpha_1 + \tan\alpha_2}{2}$

$F_{whirl} = \rho \cdot S \cdot C_f^2 (\tan\alpha_1 - \tan\alpha_2)$ $\dot{m} = \rho \cdot C_f \cdot S$ $C_{w1} - C_{w2} = C_f(\tan\alpha_1 - \tan\alpha_2)$

Resultant $F_{isentropic} = \sqrt{F_a^2 + F_w^2}$ Direcn of $F_{is} \Rightarrow \tan\alpha_m = \frac{F_a}{F_w}$

With Friction: $\Delta P_0 = P_{01} - P_{02} = \Delta(\text{Stagnation pressure}) =$ pr. loss due to fricn

Pr. loss coefficient $\varepsilon_L = \frac{\Delta P_0}{\frac{1}{2} \cdot \rho \cdot C_1^2} \Rightarrow \Delta P_0 = \frac{1}{2} \cdot \rho \cdot C_1^2 \cdot \varepsilon_L$

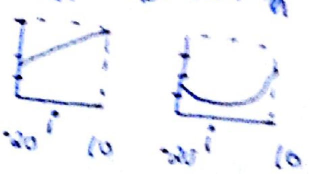
ΔP Actual pr. Rise $= \rho \cdot C_f^2 \cdot (\tan\alpha_1 - \tan\alpha_2) \cdot \tan\alpha_m - \frac{\rho \cdot C_1^2 \cdot \varepsilon_L}{2}$

$F_a = \Delta P \cdot S$ $F_w = \rho \cdot S \cdot C_f^2 (\tan\alpha_1 - \tan\alpha_2) - \frac{\rho \cdot C_f^2}{2 \cdot \cos^2\alpha_1} \cdot \varepsilon_L$

Lift force $L = F_w \cdot \cos\alpha_m + F_a \cdot \sin\alpha_m$ Drag force $D = F_w \sin\alpha_m - F_a \cos\alpha_m$

$C_L = \frac{L}{\frac{\rho}{2} \cdot c \cdot C_m^2}$ $C_D = \frac{D}{\frac{\rho}{2} \cdot c \cdot C_m^2}$ $C_L = \frac{2 \cdot S}{c}(\tan\alpha_1 - \tan\alpha_2) \cdot \cos\alpha_m - C_D \cdot \tan\alpha_m$

$C_D = \varepsilon_L \cdot \frac{S}{c} \cdot \frac{\cos^3\alpha_m}{\cos^2\alpha_1}$



Cascade or diffuser $\eta = \frac{\Delta P}{\Delta P_{isen}} = \frac{P_2 - P_1}{P_{2is} - P_1} = 1 - \frac{2 \cdot C_D}{C_{L \cdot is}} \cdot \sin 2\alpha_m$

$(\eta_c)_{max} = 1 - \frac{2 \cdot C_D}{C_{L \cdot is}}$

Dimensionless parameters: 1- Flow coeff $= C_f/U = 1/(\tan\alpha_1 + \tan\beta_1)$

2- Head coeff $\psi = \frac{C_p \cdot \Delta T}{U^2/2} = \frac{2(\tan\beta_1 - \tan\alpha_1)}{\tan\beta_2 + \tan\alpha_1}$ 3- $R = \frac{\Delta T_R}{\Delta T_{stage}} = \frac{C_f}{2U}(\tan\beta_1 + \tan\beta_2)$

D. parameters - C. compressors: 1- Flow coeff = Ratio of mass flow Rate $\dot{m}$ Referred to impeller tip $= \frac{C_{f2} \cdot \pi D_2 b_2 \cdot \rho_2}{\rho_2 \pi D_2 b_2 \cdot U_2}$ $= \boxed{\frac{C_{f2}}{U_2}}$

2- Head coeff $= \frac{\Delta h_{stage}}{K.E.\ at\ tip\ velocity} = \frac{\Delta h}{U_2^2/2} = \frac{2\, C_p \cdot \Delta T}{U_2^2}$

3- pressure coeff $= \frac{\Delta h_{ise}}{U_2^2/2} = \frac{\eta_{is} \cdot 2 \cdot C_p \cdot \Delta T}{U_2^2}$

4- $R = \frac{\Delta T_{Rotor}}{\Delta T_{stage}} = 1 - \frac{C_p \cdot \Delta T_{stator}}{C_p \cdot \Delta T_{stage}}$ * $= 1 - \frac{C_{w2}}{2U_2}$ if $C_{f1} = C_{f2}$

① C. Compressor:

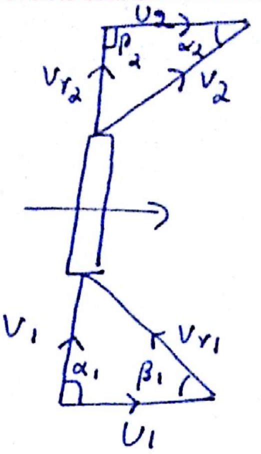
Axial Inlet ⟶ Radial outlet

⟶ [$\because V_1$ tangential].

$\therefore \quad U_1 \neq U_2$

$\alpha_1 = 90^\circ$ for max. η.

$\dot{W} = \dot{m}\, V_{w_2} \cdot U_2 = \dot{m} \cdot \phi_s \cdot U_2^2$.



② A. Compressor

Axial Inlet ⟶ Axial outlet

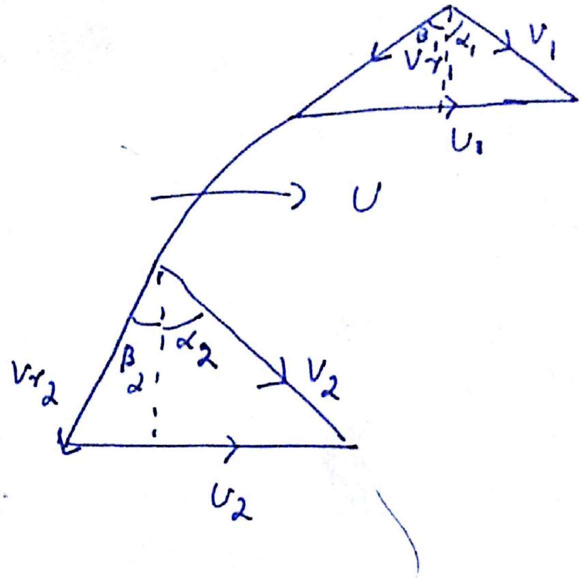
$U_1 = U_2$

$V_{f_1} = V_{f_2}$ [for 0 Axial Thrust]

$\dot{W} = \dot{m} \cdot U \cdot V_f \left(\tan\beta_1 - \tan\beta_2\right)$

$$R = \frac{\dfrac{V_{r_1}^2 - V_{r_2}^2}{2}}{V_{w_2} U_2 - V_{w_1} \cdot U_1} = \frac{V_f}{2U}\left[\tan\beta_1 + \tan\beta_2\right]$$

⟶ same as



③ Steam Turbines: All are Axial flow $\therefore U_1 = U_2 = U$.

(i) I.T. All Inlet Energy = K.E.

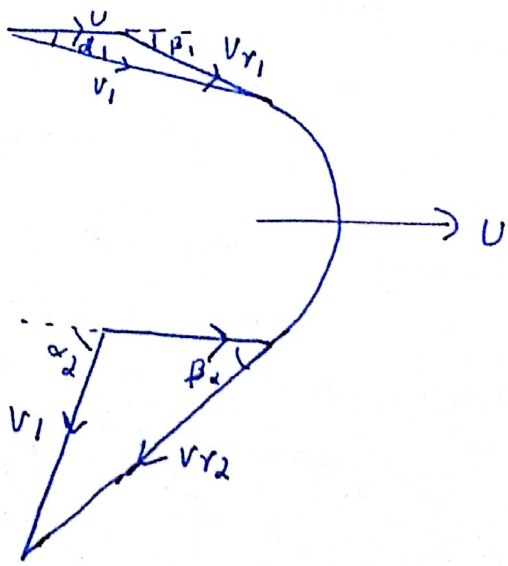
$U_1 = U_2$

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

$\dot{W} = \dot{m}\left(V_{w_1} + V_{w_2}\right) \cdot U$

$\eta_{max} = \cos^2\alpha$ when $\dfrac{U}{V_1} = \dfrac{\cos\alpha}{2}$

if sym blade ⟶ $\beta_1 = \beta_2$.



(ii) RT: ⟶ same Δ ⟶ only $V_{r_2} > V_{r_1}$

In 50% R: $\alpha_1 = \beta_2$ $\alpha_2 = \beta_1$ $V_{r_1} = V_2$ $V_{r_2} = V_1$.

⟶ $\eta_{max} = \dfrac{2\cos^2\alpha}{1 + \cos^2\alpha}$ where $\dfrac{U}{V_1} = \cos\alpha$.