

(1) Compressor: To compress gases and vapour from T_1 to T_2 pressures
 $\rightarrow P_1 \rightarrow T_1 \rightarrow P_2$.

+ve displacement : Rec., Roots, vane blower
 [Pr. n by ving volume of gas]

Rotary steady flow : Centrifugal, Axial.
 [Velocity $g\omega$ \rightarrow in pressure \rightarrow in pr. & in a.e.]

(2) (i) Fan (1.1) (ii) Blower (1.1-2.3) (iii) Compressor (> 2.3)

* If we take also a compressor * depends on pr. ratios
 * It to diffuser given

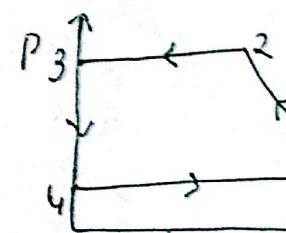
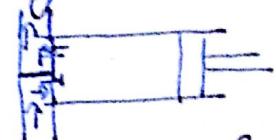
(3) Pressure [10 - 80 - 1000 bar] Volume [- 9-3000 - m³/min]

(4) Rec. Compressor : (2-stroke cycle)

\rightarrow Inlet outlet valve (pressure operated, suc'n in, delivery out)

\rightarrow A.C., fridge, mining, etc.

\rightarrow slow speed, \uparrow pressure, \downarrow volume, \uparrow losses.



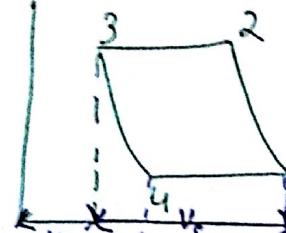
$$W = \frac{n}{n-1} \cdot P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} \cdot MRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$W_{1-2} = \frac{P_1 - P_2 V_2}{n-1} \text{ (by a.v.)}$$

$$W_{1-2} = \frac{P_1 - P_2 V_1}{n-1} \text{ (on comp.)}$$

Work $W_p = \text{work in comp } 1-2 + \text{work in discharge} - \text{work in expansion}$
 - work in suc'n.



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

$$= W_c - W_e$$

$$W = \frac{n}{n-1} \cdot MRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

* now volume delivered $= V_1 - V_4$

(5) Why $V_c \rightarrow$ [Damage to cylinder, valves, piston ring seize]
 (in case of thermal expansion)

Effect of V_c : work ind of it \because if \downarrow volume \therefore No. of cycles

for same work \uparrow .

* For same qty. of air to be delivered,
 total work would be same, only no. of cycles
 will be \uparrow in and case as vol./cycle $V_1 - V_4$ i.e.

OR $h_1 + \theta = h_2 + W_{net}$ \rightarrow less than V_1 .
 * or by $d\theta = du + dw$ \rightarrow don't use only W_{1-2}

(6) Heat Transfer $\rightarrow \frac{\gamma - n}{\gamma - 1} \cdot dW_{poly}$

to cyl. wall during compression

* $C = \frac{V_2}{V_1}$

$\rightarrow 100^\circ \text{ if } P_1 = P_2 \text{ but no compression}$
 $\rightarrow \downarrow \text{ with } \uparrow \text{ in } P_2 \cdot \text{ with } n_v = 0$

(7) volumetric $\eta_v = 1 + C - C \cdot \left(\frac{P_2}{P_1} \right)^{1/n}$

mean piston speed = $2\pi N / 60$

- c per with \uparrow in mps $\left(\frac{P_2}{P_1} \right)_{max}^{1/n} = (1 + \frac{C}{n})^n$

(8) MULTI STAGING: WITH INTERCOOLER:

$\rightarrow T_0 \uparrow P_2 \text{ w/o } \downarrow n_v$

$\rightarrow T_0 \downarrow \text{ size of cylinder}$

$\rightarrow 100^\circ \text{ if } C = 0$

$\rightarrow \uparrow \text{ with } \uparrow \text{ in } n$

INDEX OF EXPANSION

$\rightarrow T_0 \downarrow \text{ outlet temp.}$

\rightarrow

single 5.6 bar

double 5.6-30 "

Triple 30-120 "

Quadruple 120-240 "

Quintuple 240-480 "

Septuple 480-960 "

Decuple 960-1920 "

Quadruple 1920-3840 "

Octuple 3840-7680 "

Sextuple 7680-15360 "

Quintuple 15360-30720 "

Quadruple 30720-61440 "

Octuple 61440-122880 "

Sextuple 122880-245760 "

Quintuple 245760-491520 "

Quadruple 491520-983040 "

Octuple 983040-1966080 "

Sextuple 1966080-3932160 "

Quintuple 3932160-7864320 "

Quadruple 7864320-15728640 "

Octuple 15728640-31457280 "

Sextuple 31457280-62914560 "

Quintuple 62914560-125829120 "

Quadruple 125829120-251658240 "

Octuple 251658240-503316480 "

Sextuple 503316480-1006632960 "

Quintuple 1006632960-2013265920 "

Quadruple 2013265920-4026531840 "

Octuple 4026531840-8053063680 "

Sextuple 8053063680-16106127360 "

Quintuple 16106127360-32212254720 "

Quadruple 32212254720-64424509440 "

Octuple 64424509440-128849018880 "

Sextuple 128849018880-257698037760 "

Quintuple 257698037760-515396075520 "

Quadruple 515396075520-1030792151040 "

Octuple 1030792151040-2061584302080 "

Sextuple 2061584302080-4123168604160 "

Quintuple 4123168604160-8246337208320 "

Quadruple 8246337208320-16492674416640 "

Octuple 16492674416640-32985348833280 "

Sextuple 32985348833280-65970697666560 "

Quintuple 65970697666560-131941395332160 "

Quadruple 131941395332160-263882790664320 "

Octuple 263882790664320-527765581328640 "

Sextuple 527765581328640-1055531162657280 "

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Quadruple 2111062325314560-4222124650629120 "

Octuple 4222124650629120-8444249301258240 "

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Octuple 67553994410065920-13510798882131840 "

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Sextuple 216172782114109440-432345564228218880 "

Quintuple 432345564228218880-864691128456437760 "

Quadruple 864691128456437760-172938225691287520 "

Octuple 172938225691287520-345876451382575040 "

Sextuple 345876451382575040-691752902765150080 "

Quintuple 691752902765150080-1383505805530300160 "

Quadruple 1383505805530300160-2767011611060600320 "

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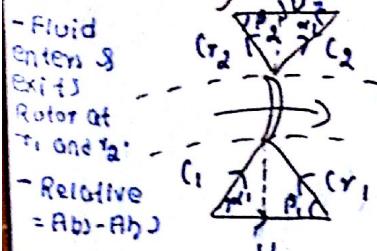
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① Energy Transfer : Only in moving part (e.g. mech. energy \rightarrow fluid energy)
 - change of carrying medium (e.g. shaft \rightarrow fluid)

② Energy Transformation : Both in moving and fixed (e.g. K.E. \rightarrow P.E.)
 - change in forms of energy

③ Rotating m/c or Euler's principle : Torque $T = m \cdot c_{w_2} \cdot r_2 - m(c_1 \omega_1)$



α = Absolute vel L.

$$\text{Power} = T \cdot \omega$$

β = Blade L.

$$\text{Power} = m \cdot c_{w_2} \cdot u_2 - m(c_1 \omega_1)$$

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

$$C_1 = U_1 + C_{1r} \quad C_2 = U_2 + C_{2r}$$

$$C_2 = C_{1r} + U_2 \quad C_2 = C_{2r} + U_2$$

$$\text{Power}/\text{lsg} = (c_{w_2} u_2 - c_1 \omega_1) \cdot$$

$$\left[\frac{C_2^2 - C_1^2}{2} \right] + \left[\frac{U_2^2 - U_1^2}{2} \right] + \left[\frac{C_{1r}^2 - C_{2r}^2}{2} \right]$$

due to diverging area

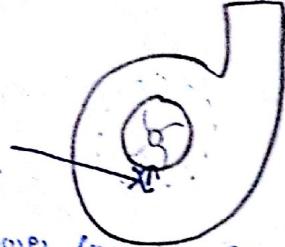
③ In. C. Compressor : $(c_{w_2} u_2 - c_1 \omega_1) = \text{Ext. effect} + \text{Int. Effect}$
 \rightarrow Impulse + Centrifugal + Diffusion

DP in Diffuser

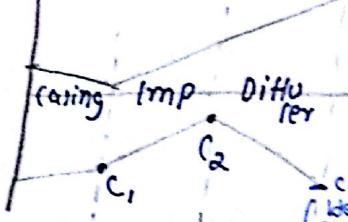
DP in m. blade

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free vortex



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⑤ WORK, P. Ratio : $F > R > B$

⑥ stable operating Range : $B > R > F$

$$\text{Pr. head} = \frac{c_{w_2} \cdot U_2}{g} = \frac{U_2}{g} (U_2 - Q_2 \cdot \cot \beta_2) = \frac{U_2^2}{g} - \frac{Q_2 \cdot \cot \beta_2}{g \cdot R_2} \cdot g = K_1 - K_2 \cdot g \quad g = R_2 \cdot Q_2$$

⑥ slip in Radial blade : \therefore Edge (P. pr.), C. Edge L (P. pr.)

\therefore fluid deviates from its path (c_{w_2} less than U_2)

\rightarrow blade becomes like backward ($U_2 \cdot \phi_s = \omega_2$)

$$\text{Power/Ip} = m \cdot c_{w_2} \cdot U_2 = m \cdot \phi_s \cdot U_2^2$$

⑦ work/Ip factor $\rightarrow P = m \cdot \phi_s \cdot \omega_w \cdot U_2^2$

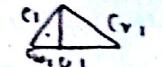
γ_1 : Actual work γ_2 : Theoretical work due to loss

$$\text{mass flow Rate} \rightarrow \rho_1 \cdot A_1 \cdot V_f_1 = \rho_2 \cdot A_2 \cdot V_f_2$$

$$\text{Abo. } A_1 = \frac{\pi}{4} (D_1^2 - D_{h1}^2)$$

⑧ pre-whirl : In Inlet blade to reduce c_{r_1}

$$m = \frac{c_{r_1}}{\sqrt{g R T_1}}$$



$$\text{Pip} = m [c_{w_2} \cdot U_2 - c_{w_1} \cdot U_1]$$

\rightarrow diff. pipe I. blade

vaneless space

Diffuser

blades

$$A_1 = \pi D_1 b_1$$

Impeller (shape of blade) \rightarrow
 vaneless space (static pr. in)
 - free vortex flow \rightarrow diffuser
 \rightarrow spiral casing called volute



⑨ work by SSEE : $h_1 + \frac{c_1^2}{2} = h_2 + \frac{c_2^2}{2} + w$

\rightarrow Evaluate both if stagnation given use $w = h_2 - h_1$ (assume s.e.f. very 1)

* NOTE : if static given use $w = h_2 - h_1$ (assume s.e.f. very 1)

* Entropic w as $w = h_2 - h_1$ (assume s.e.f. very 1)

C. compressor : - Entry : Axially ($\alpha_1 = 90^\circ$) thru impeller eye at hub * in comp., pr. rise due to

- Exit : Radially that outer tip of impeller cen. (u) and diff (cr) and \rightarrow γ due to impulse (c)

* no multistaging : \therefore in every stage (Axial \rightarrow Radial) : 90° turn which γ pr. in diffuser

* on entry : little pr. drop due to swirl which causes γ to c_1 at entry.

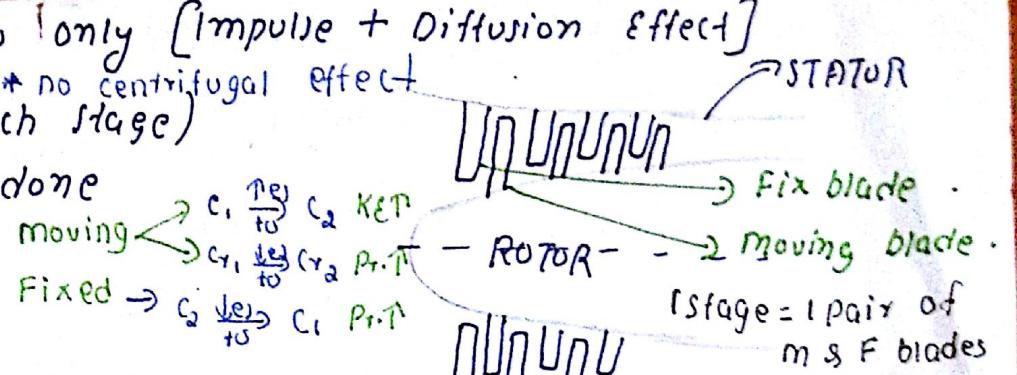
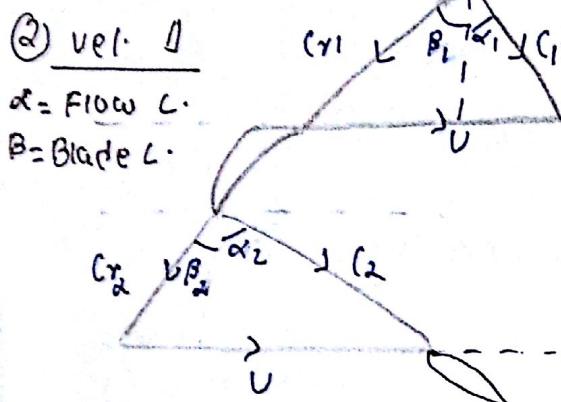
* Acc. Nozzle (Accelerate from inlet to IGV) \rightarrow IGVs (Directs flow in desired direction to entry of imp.)

Impeller (transfers shaft work to the fluid energy - inducer receives flow blow back and passes to radial portion of imp. - flow to imp. may be with or w/o swirl)

$$\therefore W = h_{o2} - h_{i1}$$

Pre-whirl angle θ
 γ_1 , m_1 , P_1 per stage
 $= \gamma_2$, U_2 , h_2
 stable range of op.

- (1) A: Compressor $\rightarrow U_1 = U_2$ so only [Impulse + Diffusion Effect]
 * AXIAL FLOW * no centrifugal effect
- \rightarrow Pr. Ratio (1.3-1.4 in each stage)
- \rightarrow Adv: Staging can be done
- \rightarrow volume: T
- \rightarrow principle: Reaction.



(2) Vel. I

α = Flow C.
 B = Blade C.

IMP:

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

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

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

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

* C_f constant to keep Axial Thrust = 0
 $F_{\text{Axial}} = m(C_{f1} - C_{f2})$

(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 Reaction: $\frac{\Delta h_m}{\Delta h_s} = \frac{\Delta h_m}{\Delta h_m + \Delta h_f}$

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

$m = \rho_1 \times C_f \times \Delta h_i$

(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) Area: $\pi (r_t^2 - r_h^2) = \pi \cdot D_m \cdot H$ $\uparrow r_t$ $H = r_t - r_h$ = ht. of blade

Frontal $m = \rho A_1 V_1 - \rho A_2 V_2$ For P, use $P_1 T_1$ not $P_{01} T_{01}$ $\uparrow r_h$ $D_m = r_t + r_h$ to find P_1 .

(7) may have to convert

(8) \rightarrow surging (m_{\min})

\rightarrow chocking (m_{\max})

from A \rightarrow C on line $m \Rightarrow$ Pr. head $P_{e2} \therefore m$ per than det pr. no problem
 from C \rightarrow D " " " \Rightarrow Pr. " " \Rightarrow breakdown of steady flow

(9) polytropic η : Constant throughout stages

Overall η : T than η_{pc} and T with prelurp

(10) Half-life: Th-232 $>$ U-238 $>$ U-235

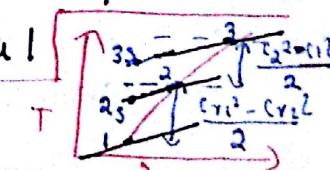
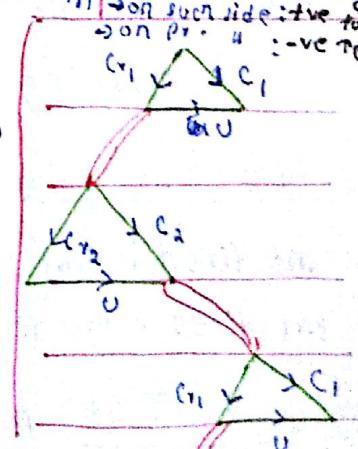
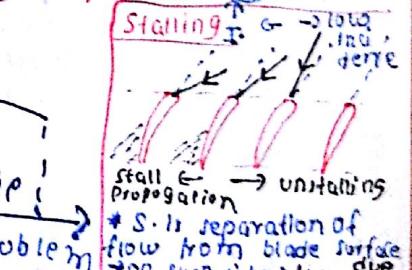
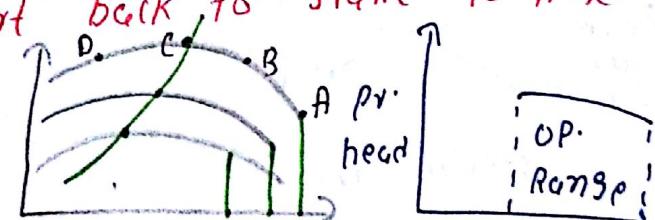
(11) Fuel Rods clad by: Zirconium, SS, Al

Moderators: $H_2O, D_2O, \text{graphite}, \text{Beryllium}$

Coolant: $H_2O, D_2O, CO_2, \text{liq metal}$

Reflector: mud or coolant

Control Rod: Cd, B, Hafnium



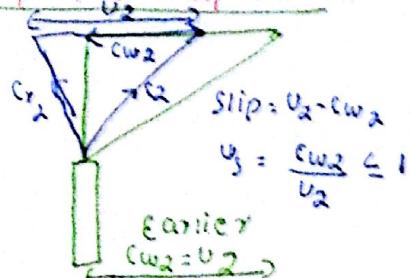
Power Plant: Fuel (che. energy) $\xrightarrow{\text{exchangers}}$ heat $\xrightarrow{\text{mech. eng.}} \text{K.E.} \xrightarrow{\text{power ratio}}$

* Turbine = nozzle + rotor

* A heat engine.

Source: H.E. Sink: Ambient

Slip in compression:



Axial compressor:

$$\text{Axial} = \rho P \times \text{Area}$$

$$F_{\text{whin}} = \dot{m}(c_{w1} - c_{w2})$$

=

$$\text{Resultant Entropic} = \sqrt{F_a^2 + F_w^2} \quad \text{Dirn of } F_w \Rightarrow \tan \alpha_m = \frac{F_w}{F_a}$$

$$\text{with Friction: } \Delta P_0 = P_0 - P_{02} = 0 \text{ (stagnation pressure)} = \text{friction due to friction}$$

$$\text{Pr. loss coefficient } \epsilon_L = \frac{\Delta P_0}{\frac{1}{2} \cdot \rho \cdot c_{w2}^2} \Rightarrow \Delta P_0 = \frac{1}{2} \cdot \rho \cdot c_{w2}^2 \cdot \epsilon_L$$

$$\Delta P_{\text{actual}} = \rho \cdot c_{f2}^2 \cdot (\tan \alpha_1 - \tan \alpha_2) \cdot \tan \alpha_m = \frac{\rho \cdot c_{f2}^2 \cdot \epsilon_L}{2}$$

$$F_a = \Delta P \cdot s \quad F_w = \rho \cdot s \cdot c_{f2}^2 (\tan \alpha_1 - \tan \alpha_2) = \frac{\rho \cdot c_{f2}^2 \cdot \epsilon_L}{2 \cdot \cos^2 \alpha_1}$$

$$\text{Lift force } L = F_w \cdot \cos \alpha_m + F_a \cdot \sin \alpha_m \quad \text{Drag force } D = F_{\text{whin}} \cos \alpha_m - F_a \cos \alpha_m$$

$$C_D = \frac{D}{\frac{1}{2} \cdot \rho \cdot c_{w2}^2} \quad C_L = \frac{2 \cdot s}{C} (\tan \alpha_1 - \tan \alpha_2) \cdot \cos \alpha_m - C_D \cdot \tan \alpha_1$$

$$\text{Cascade or diffuser } \eta = \frac{\Delta P}{\Delta P_{\text{gen}}} = \frac{P_2 - P_1}{P_{02} - P_1} = 1 - \frac{2 \cdot C_D}{C_L} \cdot \sin 2\alpha_m$$

$$\text{Dimensionless parameters: } 1 - \text{Flow coeff} = \frac{c_f / U}{1 / \tan \alpha_1 + \tan \beta_1}, \quad 3 - R = \frac{U_{\text{stage}}}{\Delta T_{\text{stage}}} = \frac{C_f}{2U} (\tan \beta_1 + \tan \beta_2)$$

D. parameters - C. (compressors):

$$2 - \text{Head coeff} = \frac{A_{\text{stage}}}{\text{K.E. at tip velocity}}$$

$$= \frac{\Delta h}{U_2^2 / 2} = \frac{2 (P_f - P_i)}{\rho U_2^2}$$

$$4 - R = \frac{\Delta T_{\text{rotor}}}{\Delta T_{\text{stage}}} = 1 - \frac{C_p \cdot \Delta T_{\text{stator}}}{C_p \cdot \Delta T_{\text{stage}}}$$

$$1 - \text{Flow coeff} = \frac{C_f / U}{1 / \tan \alpha_1 + \tan \beta_1} = \frac{C_f}{U_2}$$

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

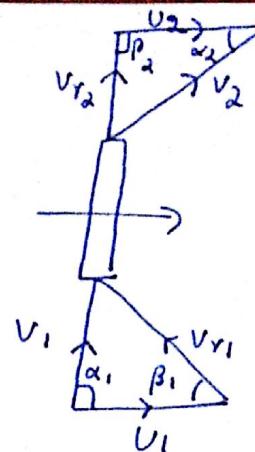
$$+ = 1 - \frac{c_{w2}}{2U_2} \text{ if } C_f = C_{fa}$$

① C. Compressor:

Axial Inlet \rightarrow Radial outlet
 $\therefore U_1 \neq U_2$ [since v_1 tangential].

$\alpha_1 = 90^\circ$ for max. η .

$$\dot{W} = m \cdot v_{w2} \cdot U_2 = m \cdot \phi_1 \cdot U_2^2.$$



② A. Compressor:

Axial Inlet \rightarrow Axial outlet

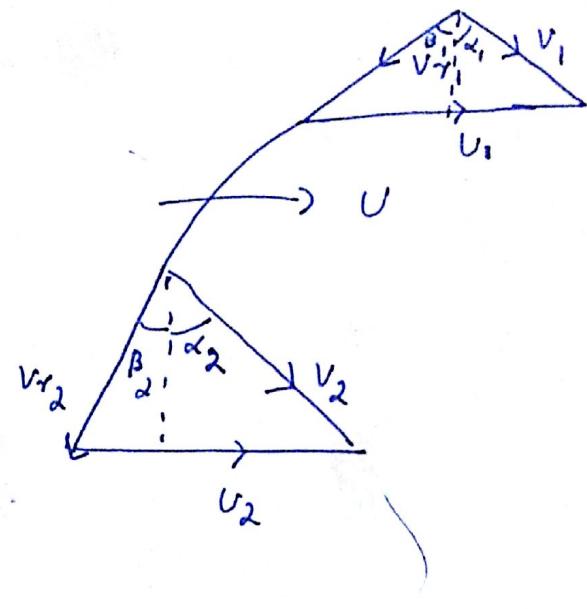
$$U_1 = U_2$$

$$V_f1 = V_f2 \quad [\text{for } 0 \text{ Axial Thrust}]$$

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

$$R = \frac{\frac{U_{r1}^2 - U_{r2}^2}{2}}{v_{w2} U_2 - v_{w1} U_1} = \frac{V_f}{2U} [\tan \beta_1 + \tan \beta_2]$$

\rightarrow same DS



③ Steam Turbines:

All are Axial flow $\therefore U_1 = U_2 = U$.

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

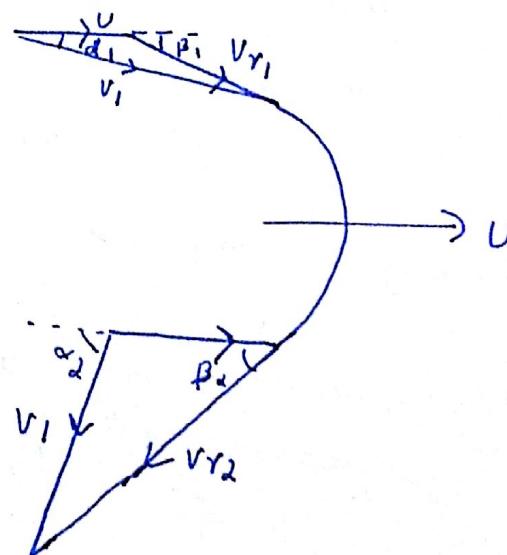
$$U_1 = U_2$$

$$V_{r2} = K \cdot V_{r1}$$

$$\dot{W} = m (v_{w1} + v_{w2}) \cdot U$$

$$U_{\max} = (U)^2 \alpha \quad \text{when } \frac{U}{V_1} = \frac{(U)^2 \alpha}{2}$$

if sym blade $\rightarrow \beta_1 = \beta_2$.



(ii) R.T.: \rightarrow same Δ \rightarrow only $V_{r2} > V_{r1}$

In S.U.R.: $R: \alpha_1 = \beta_2, \alpha_2 = \beta_1, V_{r1} = V_2, V_{r2} = V_1$

$$\rightarrow U_{\max} = \frac{2(U)^2 \alpha}{1 + (U)^2 \alpha} \quad \text{where } \frac{U}{V_1} = (U) \alpha.$$