CENTRIFUGAL COMRESSOR

It is a radial flow m/c in which air enters through the eye of impeller in an axial direction and moves through the impeller in a radially outward direction.
In the inlet casing there is a little drop in pressure with increase in velocity. while flowing through the impeller energy is transferred which increases the static pressure along with absolute velocity. Kinetic energy obtained in the impeller is transformed into pressure energy in the diffuser.

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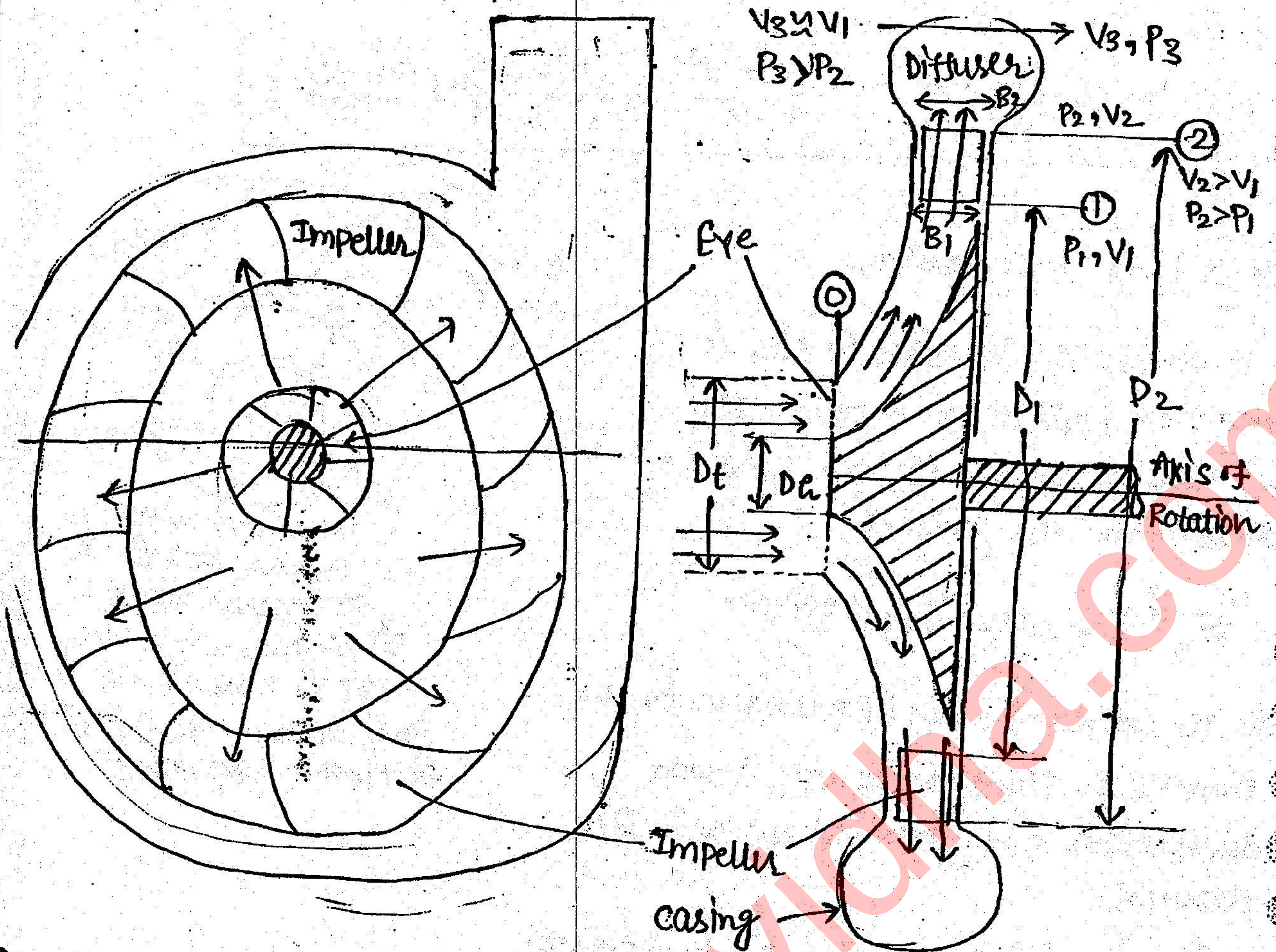

$D_1, D_2 \rightarrow$ Impeller Diameter at inlet and outlet

$D_t \rightarrow$ tip Diameter

$D_h \rightarrow$ Hub Diameter.

B_1 and $B_2 \rightarrow$ width of impeller at inlet and outlet

$$A_0 = \frac{\pi}{4}(D_t^2 - D_h^2)$$

$$A_1 = \pi D_1 B_1$$

$$A_2 = \pi D_2 B_2$$

If blade thickness is considered and let n is the no. of blade and t is the thickness of each blade

$$A_1 = (\pi D_1 - nt) B_1$$

$$A_2 = (\pi D_2 - nt) . B_2$$

If Blade area coefficient is given at inlet and outlet.

$\boxed{A_1 = \pi D_1 B_1 K_1}$ $\boxed{A_2 = \pi D_2 B_2 K_2}$ $\boxed{K = 0.9 \text{ to } 0.95}$

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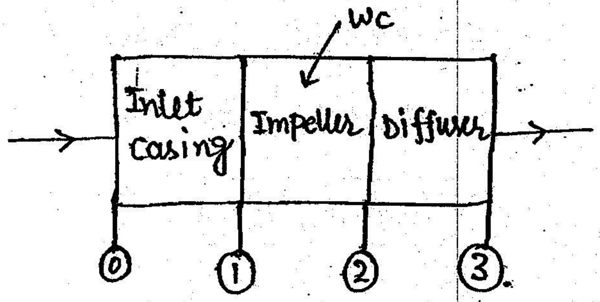
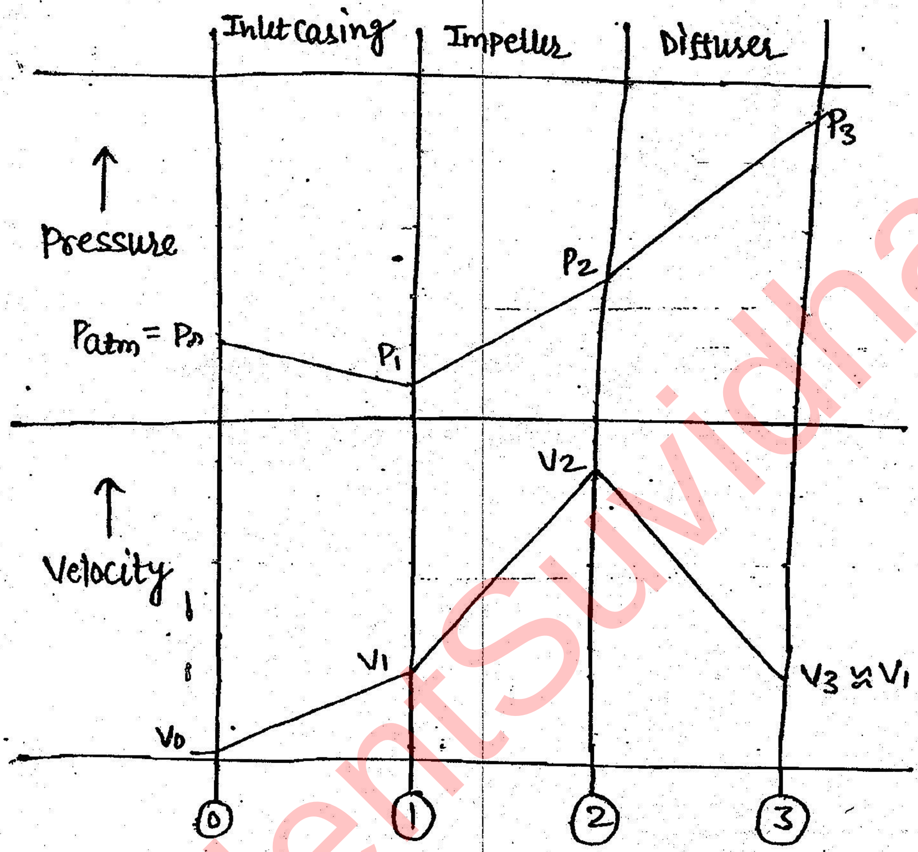
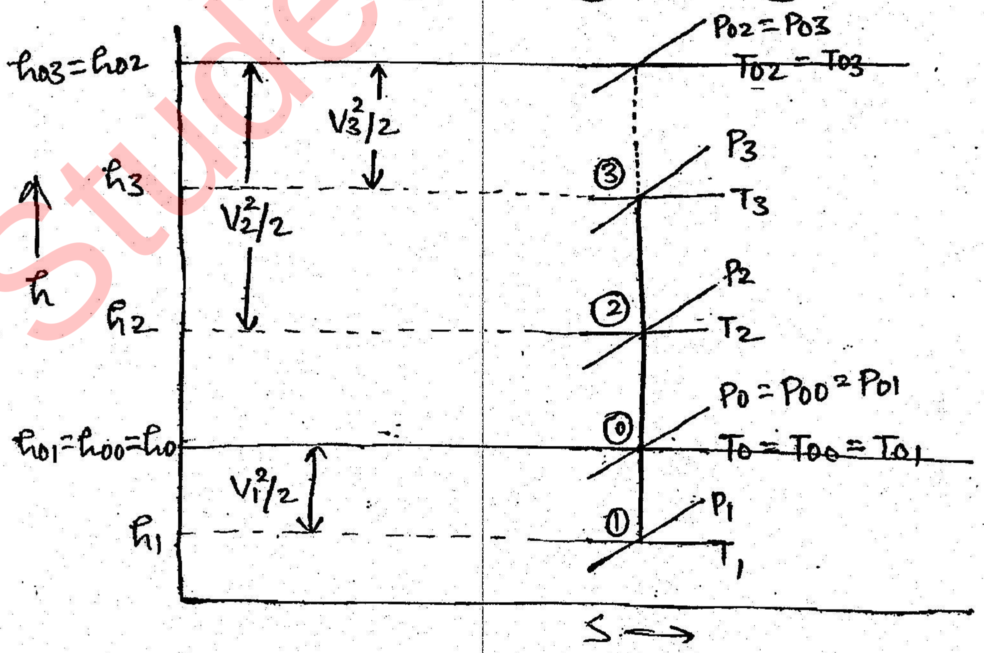
$$\boxed{\dot{m} = \rho A V_f}$$







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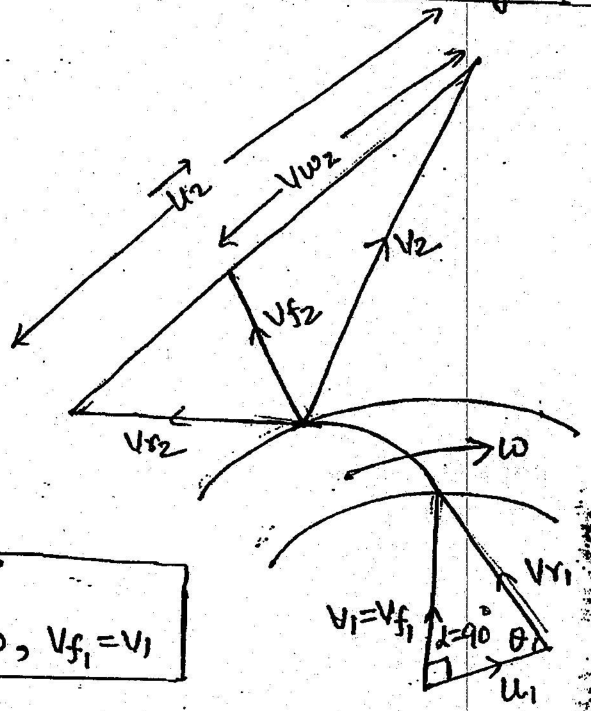
Velocity Diagram for Centrifugal Compressor :-



$P = (V_{w_2} \cdot u_2 - V_{w_1} \cdot u_1)\dot{m}$

Now as $V_{w_1} = 0$

$$\boxed{P = \dot{m}\, V_{w_2} \cdot u_2}$$

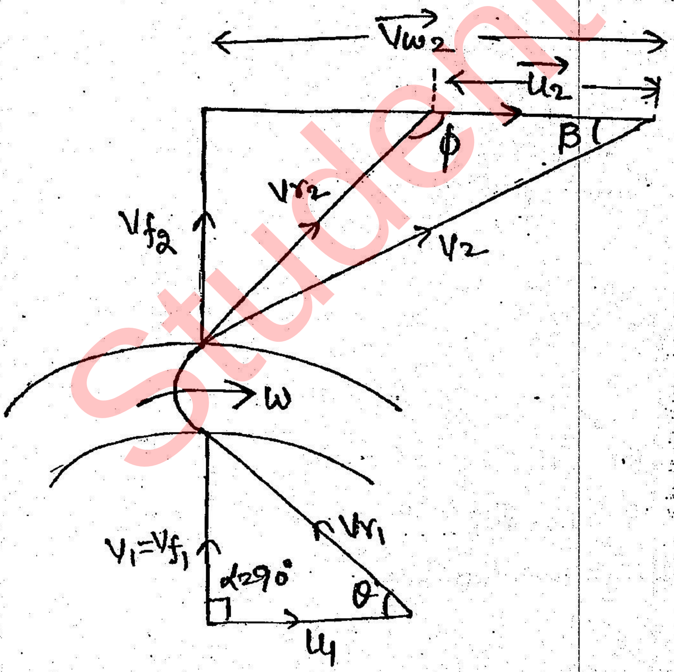
$\alpha = 90°$

$V_{w_1} = 0,\ V_{f_1} = V_1$

a) At the inlet to the impeller fluid always travel minimum distance and therefore it enters the impeller in radial direction i.e $V_{w_1} = 0$ and $\alpha = 90°$.

TYPES OF BLADE :-

1) Forward Curved Blade ($\phi > 90°$)



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② Radial Blade :- ($\phi = 90°$)

LECTURE-6
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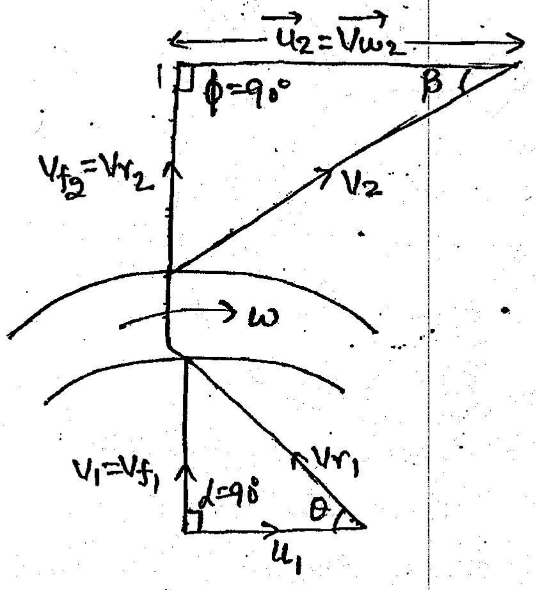
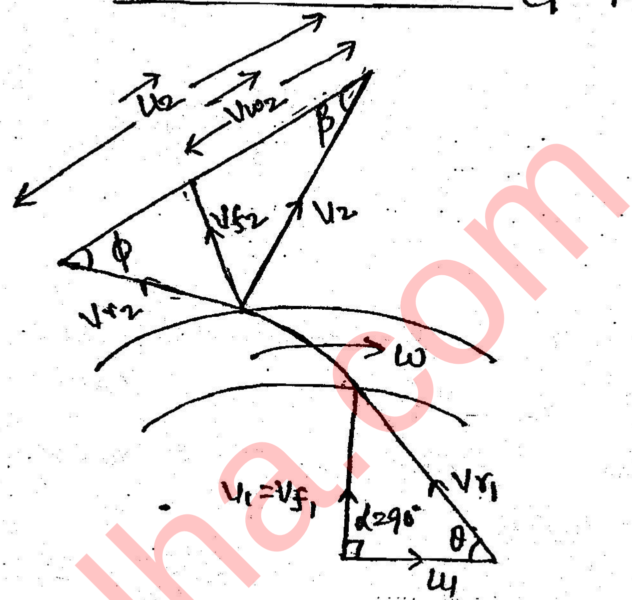


③ Backward Curved Blade ($\phi < 90°$)



NOTE :- If nothing is mentioned always assume radial blade at outlet i.e $\phi = 90°$.

* centrifugal effect of curved blade create bending moment and develops high stresses which may change the blade profile and therefor for compressors running at high RPM Radial blades are prefered.

- Forward curved blade have lowest efficiency and has a narrow operating range, but they consume highest power and develop highest pressure ratio.
- Backward curved blade have best efficieny and wide operating range. and consume less power and develop lower pressure ratio.
- Radial blade in between forward curved blade and Backward curved blac

SLIP

under ideal condition we assume that the fluid particles exactly follow the blade profile but in actual working where the No. of blades are infinite air is trapped b/w the blade passage due to inertia effect which create pressure difference across the

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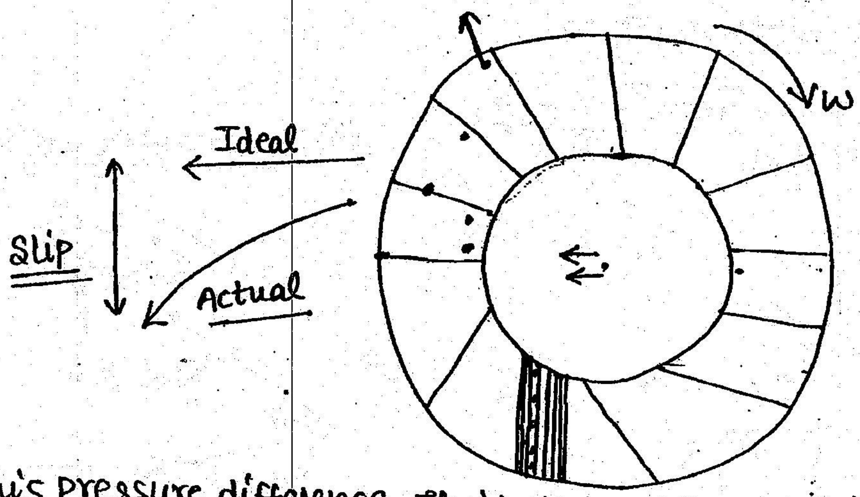
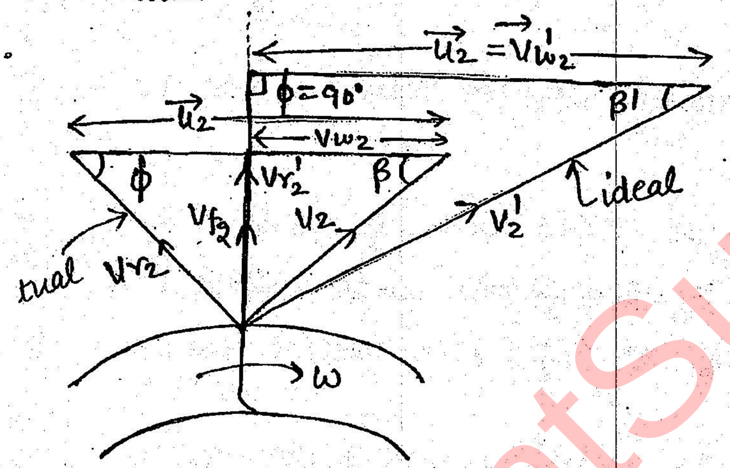


Blade. Due to this pressure difference fluid deviate from the ideal path. and this deviation is called slip. Due to slip actual blade angle at outlet always decreases which decrease the whirl component at outlet.



$$\boxed{\text{Slip} = V'_{w_2} - V_{w_2}}$$

SLIP FACTOR :- (ϕ_s)

It is defined as the ratio of workdone under ~~ideal~~ theoritical condition to workdone under ideal condition.

$$\phi_s = \frac{\text{Theoritical power}}{\text{Ideal power}} \quad \frac{(\text{Finite blade})}{(\text{Infinite blade})}$$

$$\phi_s = \frac{\dot{m} V_{w_2} u_2}{\dot{m} V'_{w_2} u_2}$$

$$\boxed{\phi_s = \frac{V_{w_2}}{V'_{w_2}}}$$

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But $V_{w_2} = u_2$

$$\boxed{\phi_s = \frac{V_{w_2}}{u_2}}$$

$$P_{theo.} = \dot{m} V_{w_2} \cdot u_2$$

$$\boxed{P_{theo.} = \dot{m}\, \phi_s\, u_2^2}$$

POWER (OR) WORK INPUT FACTOR :- (ϕ_w)

It is a power when frictional losses are taken into account. It is the ratio of actual power to theoritical power.

$$\boxed{\phi_w = \frac{P_{act}}{P_{theo.}}}$$

$$P_{act} = \phi_w \cdot P_{theo.}$$

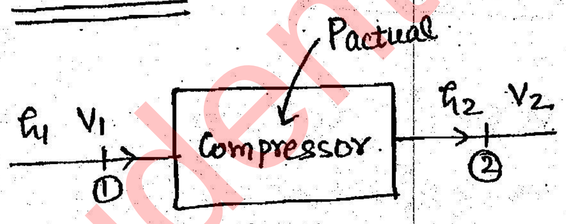
$$\boxed{P_{act} = \dot{m}\, \phi_s\, \phi_w\, u_2^2}$$

$P_{act} > P_{theo.}$

$\therefore$ $\boxed{\phi_w > 1}$

$$\boxed{\phi_w \simeq 1.035 \text{ to } 1.05}$$

BY SFEE



$V_1 \simeq V_2$

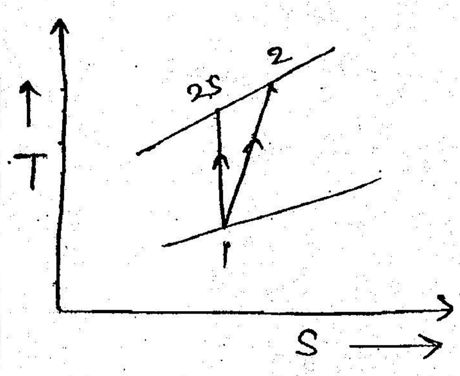
$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) - P_{act}$$

$$P_{act} = \dot{m}(h_{0_2} - h_{0_1})$$

$$\boxed{P_{actual} = \dot{m} c_p (T_{0_2} - T_{0_1})}$$

If $V_1 = V_2$

$$\boxed{P_{actual} = \dot{m} C_p (T_2 - T_1)}$$



$$\boxed{\eta_c = \frac{T_{2S} - T_1}{T_2 - T_1}}$$

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$$\boxed{P_{act} = \phi_s \cdot \phi_w \cdot u_2^2 = C_p(T_2 - T_1)}$$ $$u_2 = \frac{\pi D_2 N}{60}$$

B. Air at a temp. of 320 K enters a centrifugal compressor running at 15000 rpm. $\phi_s = 0.9$, $\phi_w = 1.05$, $\eta_c = 0.85$, $D_2 = 0.8$ m assume the blade velocity at inlet and oulet are equal then →

(1) actual temp. at outlet

(2) pressure ratio developed.

$$u_2 = \frac{\pi D_2 N}{60} = 628.3 \text{ m/s}$$

$$C_p(T_2 - T_1) = \phi_s \phi_w u_2^2$$

$$\boxed{T_2 = 691.2 \text{ K}}$$

$$\eta_c = 0.85 = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$\boxed{T_{2s} = 635.53 \text{ K}}$$

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma - 1}{\gamma}}$$

$$\boxed{r_p = 11.04}$$

.22 / Q.33. $N = 9000$ rpm

$V_1 = 600 \text{ m}^3/\text{min}$

$T_1 = 293$ K

$P_1 = 1$ bar

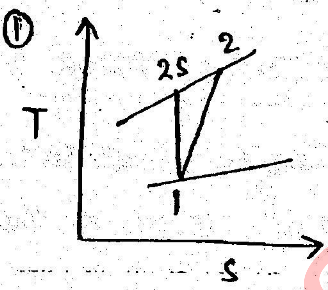
$r_p = 4$, $\phi = 90°$

$V_{f_1} = V_{f_2} = 62$ m/s

$D_2 = 2D_1$

$\phi_s = 0.9$ $\phi_w = 1$

$K_1 = 0.9$

① 

$$\left(\frac{T_{2s}}{T_1}\right) = (r_p)^{\frac{\gamma - 1}{\gamma}}$$

$T_{2s} = 435.4$ K

$$\eta_c = 0.82 = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$\boxed{T_2 = 466.65 \text{ K}}$$

② $P = \dot{m} C_p (T_2 - T_1)$

$$\dot{m} = \frac{P_1 V_1}{R T_1} \qquad \dot{m} = 11.89 \text{ kg/sec}$$

$$\boxed{P = 2075.34 \text{ kW}}$$

③ $\phi_s \phi_w u_2^2 = C_p(T_2 - T_1)$

$$0.9 \times 1 \times u_2^2 = 1.005 \times 10^3 (T_2 - T_1)$$

$$u_2 = 440.35 \text{ m/s}$$

$$u_2 = \frac{\pi D_2 N}{60}$$

$$\boxed{D_2 = 0.935 \text{ m}}$$

$$\boxed{D_1 = 0.467 \text{ m}}$$

$$u_1 = 220.17 \text{ m/s}$$

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4) $\dot{m} = \rho_1 A_1 V_{f_1}$

$$\frac{P_1 V_1}{RT_1} = \frac{P_1}{RT_1} A_1 V_{f_1}$$

$$\frac{600}{60} = \pi D_1 B_1 K_1 \cdot V_{f_1}$$

$$\boxed{B_1 = 0.122\,m}$$

5) $\theta = ?$

$$\tan\theta = \frac{V_{f_1}}{u_4} = \frac{62}{220.17}$$

$$\boxed{\theta = 15.72^\circ}$$

6) $\beta = ?$

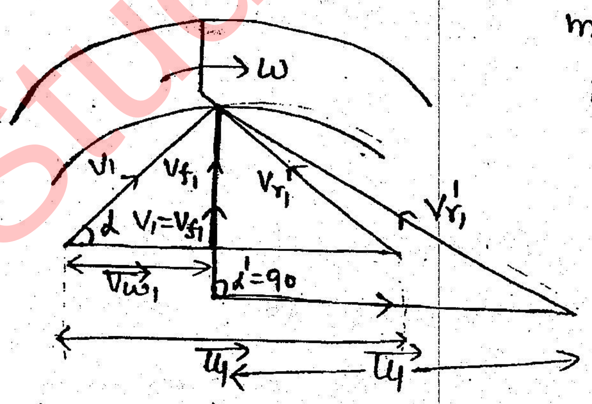
$$\tan\beta = \frac{V_{f_2}}{V_{w_2}}$$

$$\phi_s = \frac{V_{w_2}}{u_2}$$

$$\tan\beta = \frac{62}{0.9 \times 440.35}$$

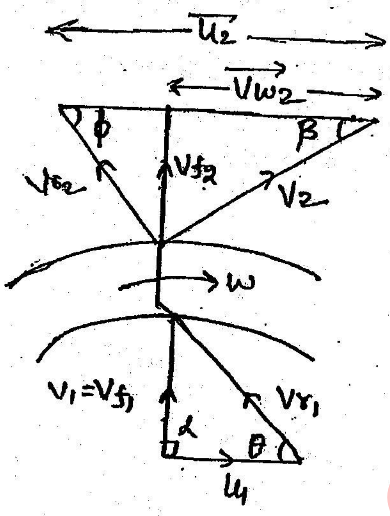
$$\boxed{\beta = 8.89^\circ}$$

PRE-WHIRL :→



Reduce the V_{r_1}' by keeping the $\overline{u_1}$ value remain same

$\alpha \neq 90^\circ$



Radial Blade

Due to slip.

7) $$\tan\phi = \frac{V_{f_2}}{(u_2 - V_{w_2})}$$

$$\tan\phi = \frac{62}{(440.35 - 0.9 \times 440.35)}$$

$$\boxed{\phi = 54.6^\circ}$$

(*) For compressors running at very high speed there is always possibility of shock wave at the inlet if the mach number is more than unity. Across the shock wave there is a sudden variation in pressure, Temp. and Velocity which may damage the blade profile. In that case with the help of inlet guide vane fluid is given a pre rotation to reduce the relative

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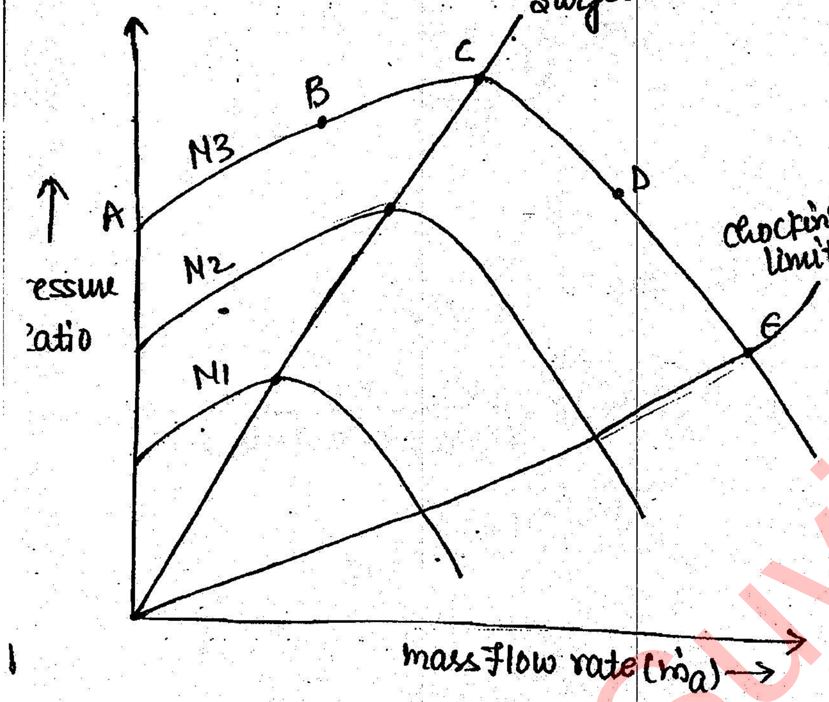
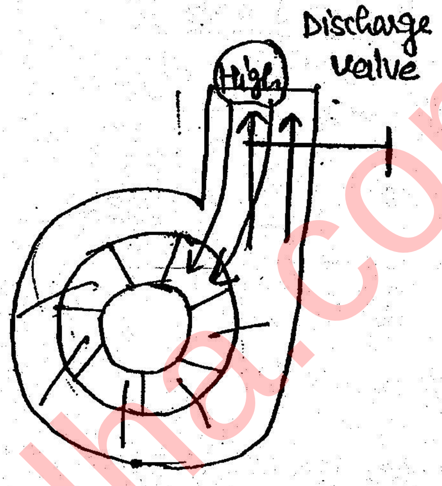
velocity at inlet. Due to this construction cost increases with increase in pressure developed.

$$\boxed{P = \dot{m}(V_{w_2}u_2 - V_{w_1}u_1)}$$

SURGING :-





$$\boxed{N_3 > N_2 > N_1}$$

It is periodic reversal of flow and it occurs during closing of valve in the operating range having mass flow rate less than value corresponding to max^m pressure ratio. This reversal of flow causes abnormal sound, vibration, decrease in η and if the intensity is more it may lead to mechanical damage.

CHOCKING :-

The max^m mass flow rate possible in a compressor is known as chocking flow. It occurs generally when the mach number corresponding to relative velocity become sonic. After that mass flow rate becomes constant irrespective of pressure ratio.

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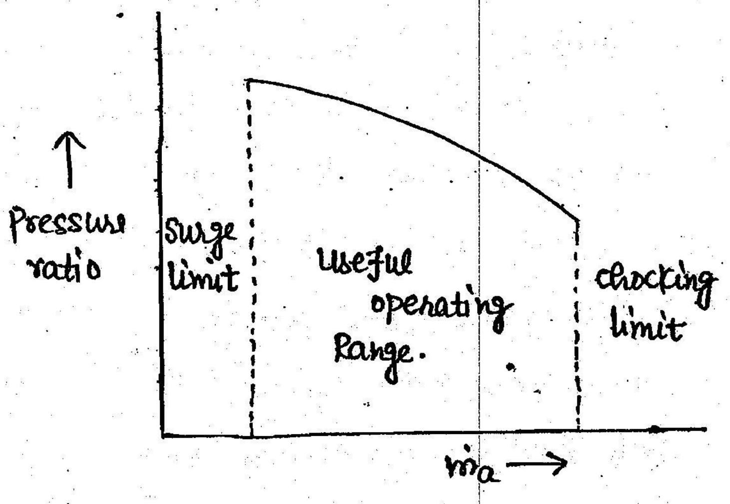
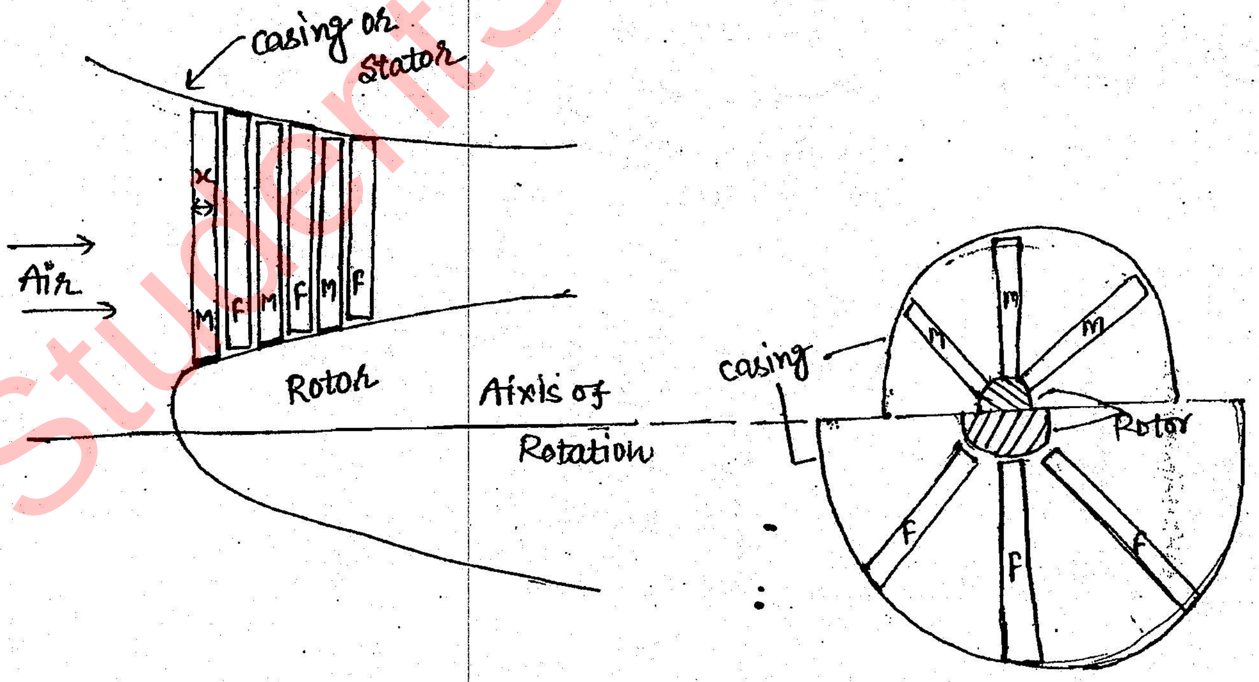


STALLING

It is simply seperation of flow from the blade surface due to improper design. It is different from slip as slip can not be avoided in actual working. Stalling is a local and temporary phenomena which can be avoided by proper blade design.

AXIAL FLOW COMPRSSOR



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