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3. Axial Flow Compressor

* ~~Pr~~

The basic requirements of compressor generally includes,

i) high airflow capacity per unit frontal area.

ii) high pressure ratio per stage.

iii) high compression efficiency.

iv) Discharge direction suitable for multistaging

* A high level of aerodynamic performance must be maintained over a wide range of mass flow rate and speeds. The compressor should be designed in such a way that, it should have a minimum length and also its weight should also be minimum as possible.

* Geometry & Working Principle of axial flow compressor

The axial flow compressor is a pressure producing machine. The energy level of air flowing through it is increased by the action of rotor blades which exerts a torque on the fluid. This torque is supplied by an external source such as an electric motor or a gas turbine.

An axial flow compressor consists of an alternating sequence of fixed ~~and~~ and moving set of blades. The set of fixed blades are spaced around an outer stationary casing and the set of blades are refered as stator blades. The set of moving blades are fixed to a spindle ~~refered~~ rotor and the blades are refered as rotor blades

The rotor and stator blades ~~are~~ must be as close as possible so as to ensure smooth and efficient flow. The radius of the rotor hub and the length of the blades are designed ~~as~~ so that there is only a very small tip clearance at the end of the rotor and stator blades, one set of stator blades and one set of rotor blades is refered as a stage.

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The number of stages depends upon the pressure ratio required by the engine. The basic principle of operation of the axial flow compressor is same as that of the centrifugal compressor. In an axial flow compressor the working fluid (air) enters the engine in axial direction and leaves the compressor in an axial direction. ~~and~~ The rotating blades impart kinetic energy to the air which is then converted into pressure & rise. The stator serves to recover the part of KE imparted to working fluid. It also redirects the fluid at an angle suitable for entry into the rotating blade.

Usually at the entry one or more stator is provided so as to guide the air the air correctly into the first set of rotor blade, these blades are refered as "Inlet guide Vein" (IGV). In many compressors there are one to three rows of diffusers after the last stage so as to straighten and slow down the air before it enters into the combustion chamber

* Stage Velocity Triangle

The flow geometry at the entry and exit of a compressor stage to stage can be described by the velocity triangles at various stations. A minimum number of data on velocity vectors and their directions are required to draw a set of velocity triangles. The magnitude of the velocity vector and their directions vary between these sections. A single pair of velocity triangle generally represents one dimension flow through a stage.

The velocity triangles for a compressor stage contains

i) Peripheral velocity or blade velocity (u)

(ii) Absolute velocity (c)

(iii) Relative velocity (w or v)

These velocities are related by the expression,

$$\vec{c} = \vec{u} + \vec{v}$$

The velocity triangle shown in the figure receives the air with an absolute velocity (c_1) and at an angle α_1 (with respect to axial direction) from the previous stage. In the case of first stage in a multistage machine the axial direction of the approaching flow is changed into the desired direction by providing a IGV, therefore the first stage experiences additional losses from the flow through the guide veins. For a general stage the stations are denoted as 1, 2 & 3 respectively.

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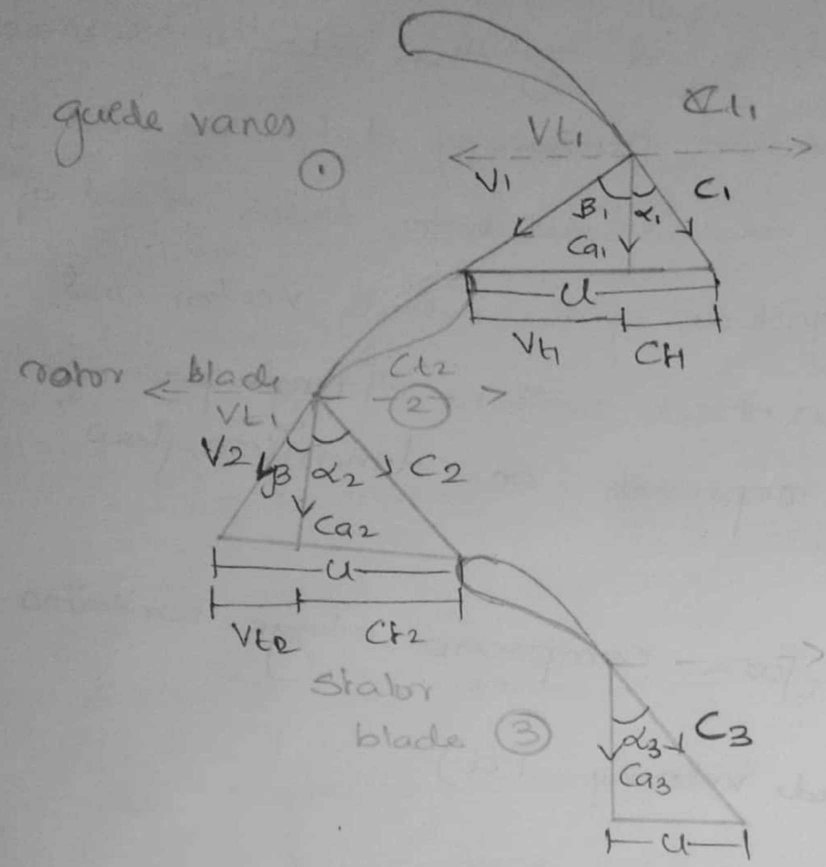

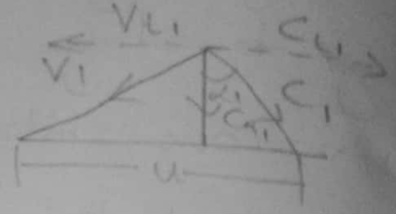

From the velocity triangle at station 1 we can write,

$$\tan \alpha_1 = C_{t1}/C_{a1} \longrightarrow (1)$$

$$\Rightarrow \quad C_{t1} = C_{a1} \tan \alpha_1$$

$$\tan \beta_1 = V_{t1}/C_{a1} \longrightarrow (2)$$

$$\Rightarrow \quad V_{t1} = C_{a1} \tan \beta_1$$

If the axial component of absolute velocity (C_a) is considered same at all locations, then we can write.

$$C_{a1} = C_{a2} = C_{a3} = C_a$$

Now the above expression can be written as,

$$C_{t1} = C_a \tan \alpha_1 \longrightarrow (3)$$

$$V_{t1} = C_a \tan \beta_1 \longrightarrow (4)$$

Now the blade velocity or peripheral velocity at station ①

can be represented as, $u = V_{t1} + C_{t1} \longrightarrow (5)$

(3) & (4) in (5),

$$u = C_a(\tan\beta_1 + \tan\alpha_1) \longrightarrow (6)$$

$$\frac{u}{C_a} = \tan\beta_1 + \tan\alpha_1 \longrightarrow (7)$$

Similarly for velocity triangle at station ② we can write,

$$\tan\alpha_2 = C_{t2}/C_{a2} \longrightarrow (8)$$

$$C_{t2} = C_{a2}\tan\alpha_2 \longrightarrow (9)$$

$$\tan\beta_2 = V_{t2}/C_{a2} \longrightarrow (10)$$

$$V_{t2} = C_{a2}\tan\beta_2 \longrightarrow (11)$$

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From velocity triangle at station ② we can write.

$$u = V_{t2} + C_{t2} \longrightarrow (12)$$

Now substituting the expression for V_{t2} and C_{t2} from (9) and (11), we get,

$$u = C_a(\tan\beta_2 + \tan\alpha_2) \longrightarrow (13)$$

where, $C_{a1} = C_{a2} = C_{a3} = C_a$,

$$\frac{u}{C_a} = \tan\beta_2 + \tan\alpha_2 \longrightarrow (14)$$

Now equating equation no (7) and (14) we get,

$$\tan\alpha_1 + \tan\beta_1 = \tan\beta_2 + \tan\alpha_2 \longrightarrow (15)$$

Similarly from eqn (5) & eqn (12) we can write.

$$V_{t1} + C_{t1} = V_{t2} + C_{t2}$$

$$C_{t2} - C_{t1} = V_{t1} - V_{t2}$$

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$$C_a(\tan\alpha_2 - \tan\alpha_1) = C_a(\tan\beta_1 - \tan\beta_2)$$

$$\tan\alpha_2 - \tan\alpha_1 = \tan\beta_1 - \tan\beta_2 \longrightarrow (16)$$

* Work input in a compressor.

The work input in a compressor is derived in terms of velocity and blade angles. The derivation is based on the assumption that the axial velocity (C_a) remains constant through out the machine. This assumption simplifies the design calculations as well as the expressions for work input and pressure rise.

From (7) and (14) we can write

$$\frac{u}{C_a} = \tan\alpha_1 + \tan\beta_1 = \tan\alpha_2 + \tan\beta_2.$$

The work ~~done~~ input to the compressor can be expressed as the product of change in the tangential components of the absolute velocity and the blade peripheral velocity.

ie,
$$\boxed{W = u(C_{t_2} - C_{t_1})} \longrightarrow (17)$$

From (9)
$$C_{t_2} = C_a \tan\alpha_2.$$

from (8)
$$C_{t_1} = C_a \tan\alpha_1$$

$$\rightarrow \boxed{W = uC_a(\tan\alpha_2 - \tan\alpha_1)} \longrightarrow (18)$$

Similarly in terms of angle β we can express the work input as follows,

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W =

from (16) we're,

$\tan\alpha_2 - \tan\alpha_1 = \tan\beta_1 - \tan\beta_2$

$$\Rightarrow \boxed{W = UC_a[\tan\beta_1 - \tan\beta_2]} \rightarrow (19)$$

To obtain higher efficiencies the work input should be as minimum as possible, to achieve this the blade and the flow geometry has to be designed properly.

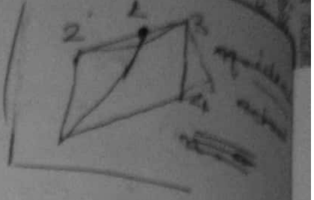
Imp
* Degree of Reactions. [R or Λ]

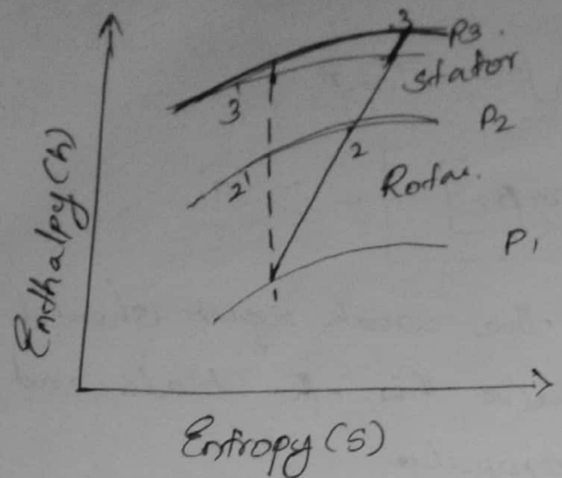
The degree of reaction defines the distribution of the stage pressure rise between the rotor and the diffuser blade rows, this determines the cascade losses in each blade rows,

The degree of reaction is defined as the ratio of enthalpy change in the rotor to that of enthalpy change in a stage. For a compressor the degree of reaction can be defined as the ratio of enthalpy increase in the rotor to that of enthalpy increase in the stage.

$$R = \frac{\text{Enthalpy increase in rotor.}}{\text{Enthalpy increase in stage.}}$$

26/3/17 The enthalpy-entropy diagram of an axial flow compressor is as shown in the figure ~~is as shown in the figure.~~





In the above graph, station 1 represents inlet of the rotor, station 2 represents exit of the rotor. station 3 represents exit of stator. Therefore, station 1-3 a compressor stage.

The degree of reaction for an axial flow compressor can be now be expressed as,

$$R = \frac{h_2 - h_1}{h_3 - h_1} \qquad \{h = C_pT\}$$

$$R = \frac{T_2 - T_1}{T_3 - T_1}$$

The enthalpy increased in the rotor, can be expressed in terms of energy equation, considering, the tangential Component of relative velocity $[V_t]$

$$h_1 + \frac{1}{2} V_{t1}^2 = h_2 + \frac{1}{2} V_{t2}^2$$

$$h_2 - h_1 = \frac{1}{2}\left[V_{t1}^2 - V_{t2}^2\right] \longrightarrow (1)$$

The enthalpy increase in a compressor stage can be considered as same as that of work input in a compressor stage,

$$h_3 - h_1 = u(C_{t2} - C_{t1}) \longrightarrow ②$$

$$R = \frac{V_{r1}^2 - V_{r2}^2}{2u(C_{t2} - C_{t1})}$$

from velocity triangles,

$V_{r1} = C_a \tan\beta_1$ $\quad$ $C_{t1} = C_a \tan\alpha_1$

$V_{r2} = C_a \tan\beta_2$ $\quad$ $C_{t2} = C_a \tan\alpha_2$

$$R = \frac{C_a^2(\tan\beta_1^2 - \tan\beta_2^2)}{2uC_a(\tan\alpha_2 - \tan\alpha_1)}$$

from (16) we've

$$\tan\alpha_2 - \tan\alpha_1 = \tan\beta_1 - \tan\beta_2$$

$$\Rightarrow \quad R = \frac{C_a^2(\tan\beta_1^2 - \tan\beta_2^2)}{2uC_a(\tan\beta_1 - \tan\beta_2)}$$

$$= \frac{C_a^{\cancel{2}}\cancel{(\tan\beta_1 - \tan\beta_2)}(\tan\beta_1 + \tan\beta_2)}{2u\cancel{C_a}\cancel{(\tan\beta_2 - \tan\beta_1)}}$$

$$\boxed{R = \frac{C_a(\tan\beta_1 + \tan\beta_2)}{2u}}$$

The term C_a/u is refered as flow coefficient 'ϕ'

$$\therefore \boxed{R = \frac{\phi}{2}(\tan\beta_1 + \tan\beta_2)}$$

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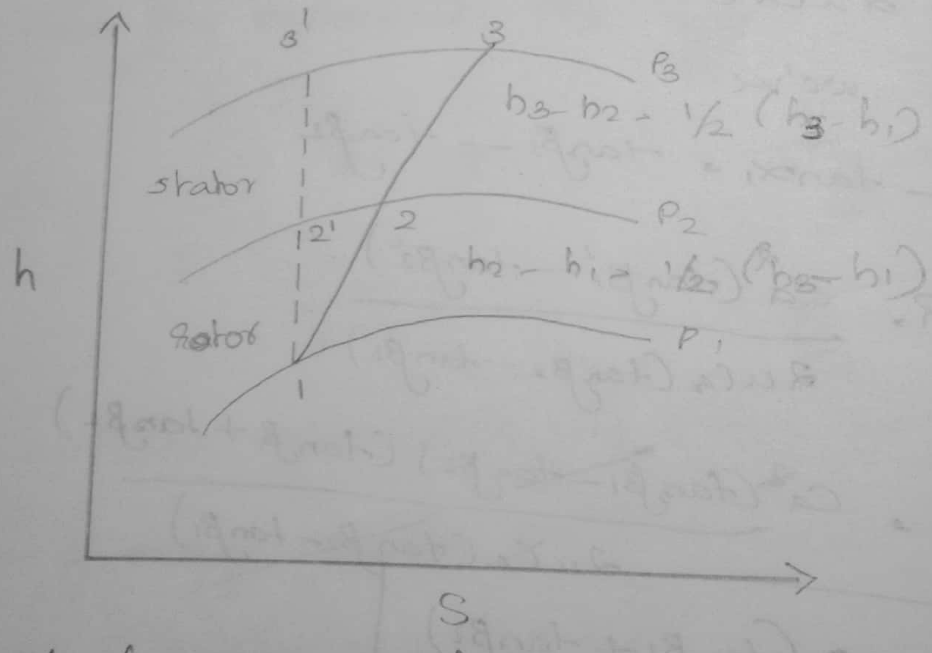
→ 50% Reaction Stage

This implies that the enthalpy ↑ in the rotor is 0.5 times that of enthalpy ↑ in the stage, which mean that 50% of the enthalpy increase in a compressor stage is contributed by the rotor. The h-s diagram (enthalpy, entropy diagram) for a 50% reaction stage is as shown in the figure.



→ For high Reaction Stage

For a higher reaction stage it implies that the contribution of the rotor to the enthalpy increase in a compressor state is higher & it can be shown in figure,

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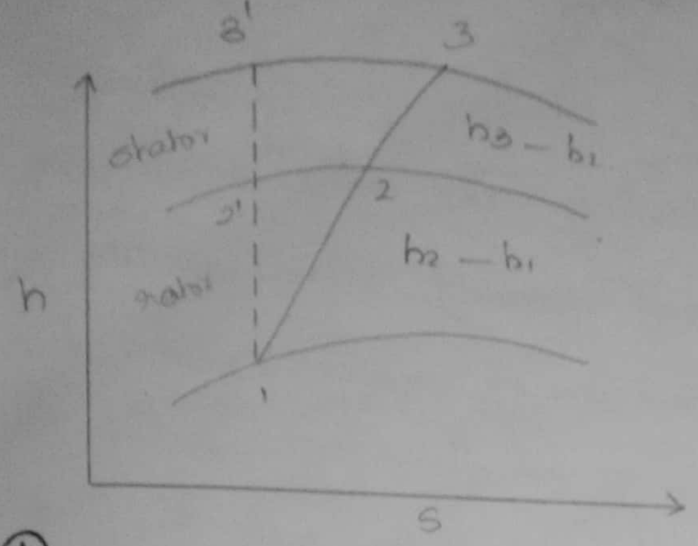

(A)

* Flow losses in axial flow compressor

The aerodynamic losses occuring in the turbo machines arise due to the growth of boundary layer & its seperation from the blade & blade passages. The non uniform velocity profile at the exit of the cascade leads to another type of loss refered as mixing losses. The aerodynamic losses in a turbo machine can be grouped as follows;

1) Profile loss

As the term indicates this loss is associated with the growth of the boundary layer on the blade profile. The seperation of the boundary layer occurs when adverse pressure gradient on the surface becomes too steep and this increases the profile loss. The pattern of the boundary layer growth & its seperation depends upon the geometry of the blade and the flow. If the flow is initially supersonic or becomes super sonic on the blade surface additional losses occur due to the formation of shock waves.

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② Annular loss

The majority of the blade rows in turbo machines are housed in a casing. The axial compressor stage has a pair of fixed and moving blade rows. In stationary blade rows a loss of energy occurs, due to the growth of the boundary layer on the end walls, this also occurs in the rotating row of blades but the flow on the end walls in this case is subjected to effects associated with the rotation of cascade.

③ Secondary loss.

This loss occurs in the region of flow near the end wall which is due to the presence of unwanted circulatory flow or cross flow. Such secondary flows develop on account of turning of the flow through the blade channel in the presence of annulus wall boundary layer. The secondary vortices in the adjacent blade channel induce vortices in the wake near region. These trailing vortices leads to additional losses. The magnitude of loss due to secondary flow depends upon the passage height of the flow.

④ Tip clearance loss

This loss arises due to the clearance between a moving blade and the casing. On account of static pressure difference the flow leaks from the pressure side towards the suction side. The tip clearance and secondary flow are closely related to each other and it is generally estimated together

* (B) Stage losses in axial flow Compressor.

The stage work is less than the energy supplied to the shaft by the prime mover on account of bearing and ~~twist~~ disc friction loss. All the stage work ~~done~~ doesnot appear as energy at the stator entry due to aerodynamic losses in the rotor blade row. After deducting the stator blade losses from the energy at its entry the value of ideal i.e. isentropic work required to obtain the stage pressure rise is calculated. The cascade losses in the rotor and stator will depend upon the degree of reaction.

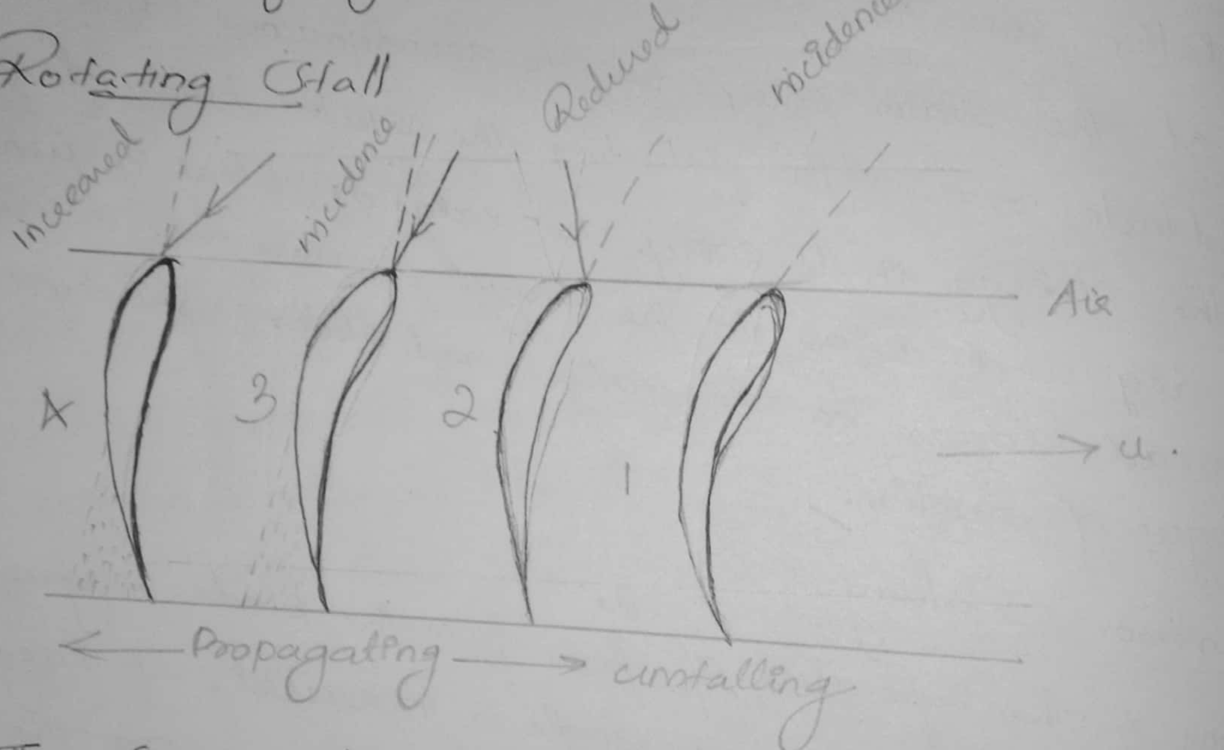
Imp * Compressor Stalling

Stalling is the seperation of the flow from the blade surface. At low flow rates the incidance angle is increased, at large values of incidence the flow seperation occurs on the suction side of the blades which is refered as positive stalling.

The negative stalling is due to the seperation of flow occuring on pressure side of the blade due to large values of negative incidence.

In a high pressure ratio multistage compressor the axial velocity is relatively small due to higher density. In such high pressure stages a small deviation from the design point causes the incidence angle to exceed its stalling value and the stall cells, first appear near the hub and tip regions. The size and number of cells increases with decreasing flow rate. At very low flow rate the stalling cells grow larger and affects the entire blade height, this causes a significant drop in pressure, which can lead to flow reversal or surge. The stage efficiency also reduces considerably on account of higher losses.

* Rotating Stall



The figure shows 4 blades in a compressor rotor, it can be seen that the third blade receives the flow at an increased incidence and this causes blade (3) to stall. On account of this the passage between the 3rd & 4th blade is blocked which causes deflection of the flow in the neighbouring blades. Therefore the stalling occurs also on the 4th blade. Thus, the stall cells move towards the left hand side at a fraction of blade speed. The blades are subjected to forced vibrations on accounts of their passage through stall cells at certain frequency. The frequency &

amplitude of vibration depends on the extend of loading and unloading of the blades, and the number of stall cells.

* Surging of Compressor blade

The unstable flow ~~of a~~ in an axial flow compressor can be due to the flowing reason;

1) Separation of the flow from the blade surface refered as Stalling
2) Complete breakdown of the flow [steady flow] refered as Surging.

3) Both these phenomena occur due to off design operating condition and are aerodynamically and ~~mechanically~~ ~~ae~~ mechanically undesirable.

It is generally difficult to differenciate between the operating conditions leading to stalling and surging, it may be noted that the flow in some regions stalls, without surging. Surging affects the whole machine while stalling is a local phenomena Surging of the compressor leads to ~~whole~~ vibration of the entire machine which can ultimately lead to ~~mach~~ mechanical failure. The frequency and magnitude of to and fro motion of the air (surgeing) depends on the relative volume of the Compressor, delivery pipe. and the flow rate.

Imp # Performance characteristics of an axial flow compressor.

The performance characteristics of an axial flow compressor or their stages at various speeds can be represented using the following graph and parameters.

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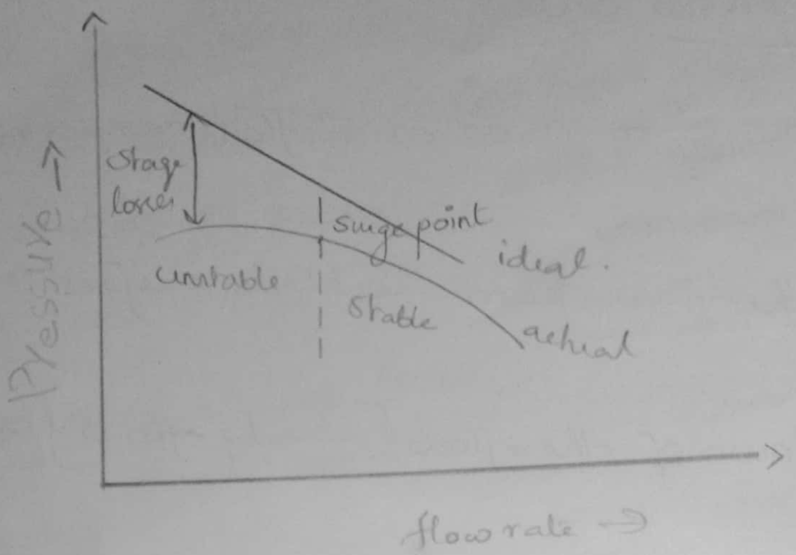
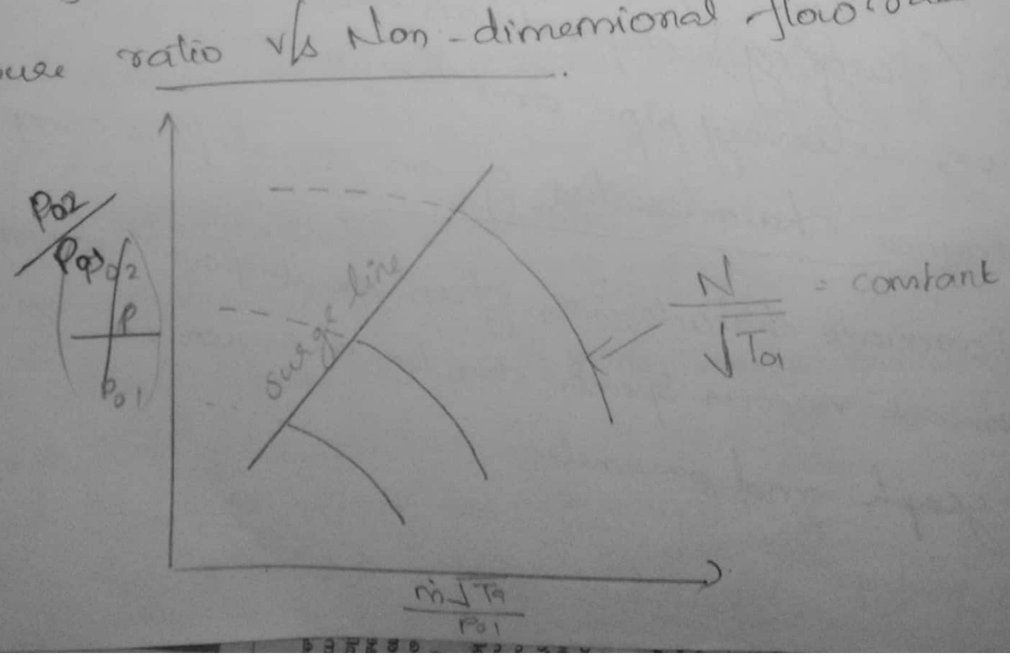
① Pressure rise v/s flow rate



The above graph shows the ideal and actual performance curve of an axial flow compressor, it can be seen that the difference between the ideal and actual performance curve is due to stage losses. The operation of the compressor to the right hand side of the surge point is stable and to the lefthand side the operation is unstable. The operation of the compressor is unstable due to reduction in the the mass flow rate.

② Pressure ratio v/s Non-dimensional flow rate.



The pressure ratio here refer to the stagnation pressure ratio and the non-dimensional flow rate take into consideration the variation of mass flow rate, stagnation temperature and stagnation pressure. It can be seen from the figure that the stagnation pressure ratio and the non-dimensional flow rate is considered for various rpm of compressor.

A compressor gives its best performance while by operating at its design point, i.e., at the pressure ratio and flow rate for which it has been designed. However, like any other machines it is also expected to operate away from the design point. ∴ The performance characteristics gives a clear idea where the off-design operation of an axial flow compressor.

* Cascade testing of an axial flow compressors

A cascade is a stationary array of blades as shown in the figure. The cascade is constructed for measurement of performance of the blades ~~sy~~ similar to that used in axial compressors. The cascade has end walls which are usually porous which can remove the boundary layer through suction so that the flow between the blades is nearly considered 2D. In this case the end wall boundary layer effects and the radial variation in the velocity field can be excluded so that the flow the blades can be considered under controlled conditions.

The objective of the measurement is to relate the fluid turning angle to the blade geometry and to measure the losses in the stagnation pressure due to friction. The effects of boundary layer separation can be clearly understood when the fluid exit angle deviates from the blade exit angle and when the stagnation pressure losses are greater than the minimum. From cascade testing the effects of blade shape, orientation. and spacing on the degree to which the flow can be turned without flow separation from the blade surfaces.

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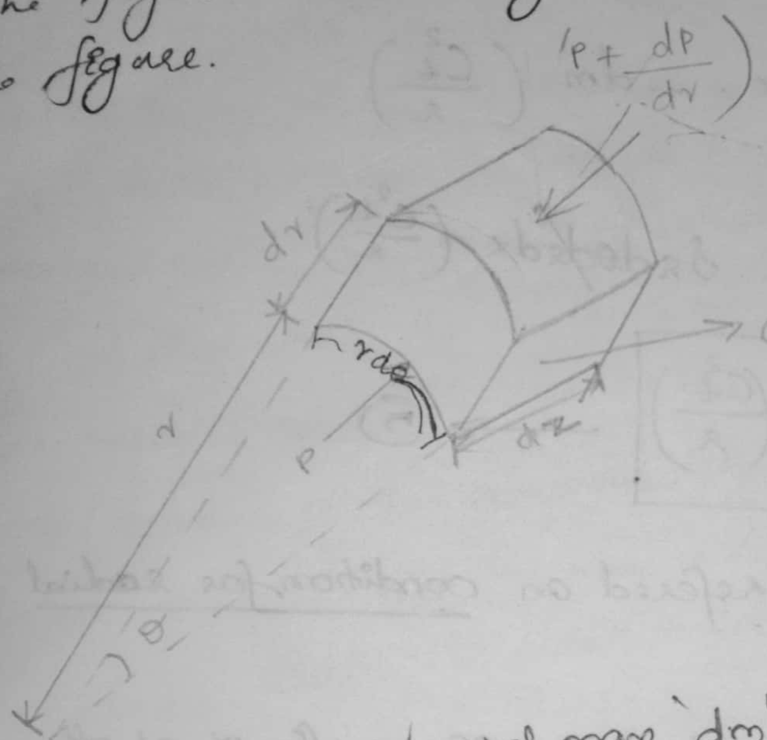
* Free vortex and constant reaction design of axial flow compressor.

While deriving the work input for axial flow compressor we've ignored the radial variation in the flow through the compressor annulus and have considered based on mean radius condition.

In order to establish a satisfactory compressor design ~~compression~~ it is neccessary to take into account, the radial variation in

1) blade speed , $u = r\omega$
2) Axial velocity , C_a
3) tangential velocity, C_t
4) Static pressure

The axial velocity at the inlet to a multistage machine will be uniform radially, but large variation can develop after the fluid has passed through 2-3 stages. The radial variation in tangential velocity depends on the way in which the blade angle vary with radius. The radial pressure gradient, depends directly on the absolute tangential velocity as shown below. Consider a small fluid element, 'dm' as shown in the figure. With tangential velocity component as shown in the figure.



angle $= \frac{arc}{r}$

arc $=$ angle $\times r$.

The volume of the elemental mass 'dm' can be written as,

$$dV = r d\theta\, dr\, dz. \longrightarrow (1)$$

The mass can be ~~de~~ related to volume, by,

$\rho = \frac{m}{V}$

$m = \rho V$

$$dm = \rho dV$$

$$dm = r\rho\, d\theta\, dr\, dz \longrightarrow (2)$$

The blade element has a centripetal accelaration towards the axis, therefore the radial pressure force, can be written as

$$F_r = -dm\left(\frac{C_t^2}{r}\right) \longrightarrow (3)$$

Radial pressure force can be explained as follows,

$$F_r = p(r d\theta dz) - \left[P + \frac{dP}{dr} dr\right] r d\theta dz$$

$$F_r = -\frac{dP}{dr}\, dr\, r\, d\theta\, dx \longrightarrow (4)$$

Now equating (3) & (4) we've

$$-\frac{dP}{dr}\, dr\, r\, d\theta\, dx = -dm\left(\frac{C_t^2}{r}\right)$$

$$\frac{dP}{dr}\, r\, \cancel{dr}\, d\theta\, dx = \rho\, r\, d\theta\, \cancel{dr}\, dx\left(\frac{C_t^2}{r}\right)$$

$$\boxed{\frac{dP}{dr} = \rho\left(\frac{C_t^2}{r}\right)} \longrightarrow (5)$$

The above equation is refered as <u>condition for radial equillibrium</u>.

In order to provide a uniform flow at the exit of the compressor, it is desirable to maintain a uniform work input along the radial length of the rotor. The change in stagnation enthalpy along a stream line through any stage in an axial flow compressor can be written as,

$$\boxed{\Delta h_0 = U\, \Delta C_t} \longrightarrow (6)$$

$W = U(C_{t2} - C_{t1})$
$= U \Delta C_t$
$W = \Delta h$

The radial gradient of the above eqn can be written as,

$$\frac{d}{dr}(\Delta h_0) = \omega \frac{d}{dr}(r\, \Delta C_t) \longrightarrow (7)$$

Thus from the above equation, it can be understood that in order to keep the fluid at constant stagnation enthalpy, the product $(r \Delta C_t)$ must be constant with the radius.

One of the configuration which satisfy this requirement as well as the requirement of radial equillibrium is refered as "free vortex design." In this design, the product 'rC_t' is held constant across the exit of each blade row.

4/4/19 The blades of a free vortex compressor will require excessive blade twist, unless the blade height is quiet small compared with the blade tip diameter.

Other than the undesirable blade twist, the very high absolute velocity at the exit of the rotor hub can vary the performance of the compressor. This is because it may result in larger pressure rise, it may while passing through the unequal bla stator blades. Both the blade twist and the unequal diffusion distribution can be improved while maintaing constant work if, '$r\Delta C_t$' is constant instead of 'rC_t'. Such a blade system is desirable in that it avoids the drawback encounteed by the free vortex design. The distribution of angular momentum with radius can be considered as a design choice, that will affect the velocity triangles, Therefore a design can be considered which will satisfy the requirements of radial equillibrium with no variation in the work done in radial direction, thus avoiding the short comings of free vortex design. Thus the possible designs can be as listed below:

1) Free vortex :- $rC_t = a$

2) Forced vortex :- $rC_t = ar^2$

3) Constant reaction :- $rC_t = ar^2 + b$

Where a and b are constants.

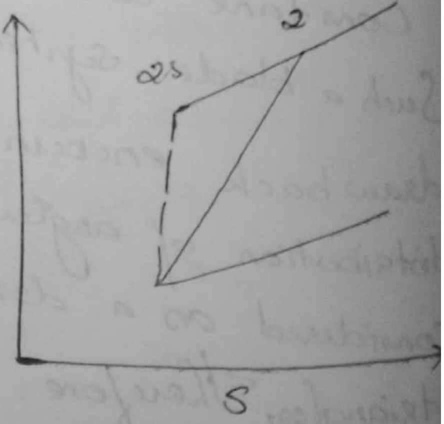
* Compressor Efficiency
(Axial flow)

⊙ The stage efficiency is defined as, the ratio of ideal work to adiabatic work for a "single stage."

⊙ The adiabatic efficiency is defined as, the ratio of ideal work to adiabatic work for the "complete compressor."

⊙ The Polytropic Compressor efficiency is defined as the ratio of ideal work adiabatic work for an infinitesimal step in the compression process.

⇒ Expression for adiabatic efficiency.



$$\eta_c = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}}$$ (total to total efficiency)

$$\eta_c = \frac{T_{02s} - T_1}{T_{02} - T_1}$$ (total to static efficiency)

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$ (static to static efficiency)

⇒ The expression for stage efficiency can be written as,

$$\eta_{st} = \frac{T_{03s} - T_{01}}{T_{03} - T_{01}}$$

$$\eta_{st} = \frac{T_{03s} - T_1}{T_{03} - T_1}$$

$$\eta_{st} = \frac{T_{3s} - T_1}{T_3 - T_1}$$