25/4/19

Module-5

Axial flow Turbines

* Working Principle & parts (fig from text)

* Work can be extracted from a fluid at a higher inlet pressure to a lower back pressure by allowing the fluid to flow through a turbine. In a turbine as the gases passes through it, the gases expands, the work done by the gas is equivalent to the change in its enthalpy.

The axial flow turbines operate on the momentum principle. The part of the energy of the gas during expansion is converted into kinetic energy in the stator [also reffered as stator nozzles]. The gas leaves these stationary nozzles (stator) at a relatively higher velocity. Then it is made to impinge on the blades over the turbain rotor, thus the momentum is imparted to the rotor blades which turns the turbine.

The two primary parts of a turbine are

(1) The stator nozzle (stator)

(2) Rotor blades

Most of the turbine poses more than one stage so as to achieve effective expansion and to produce sufficient power to drive the engine. A turbine stage consists of a fixed set of nozzle blades (stator nozzle) followed by a set of rotor blades.

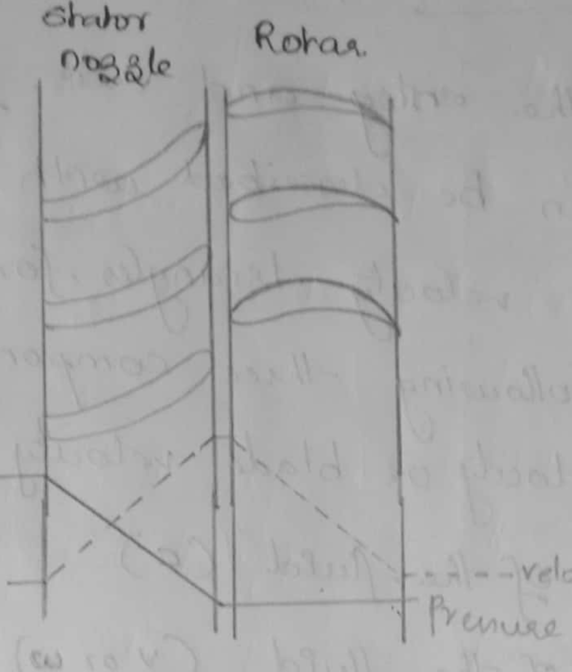
Generally a turbine stage is classified as follows;

(1) Impulse stage

(2) Reaction stage.

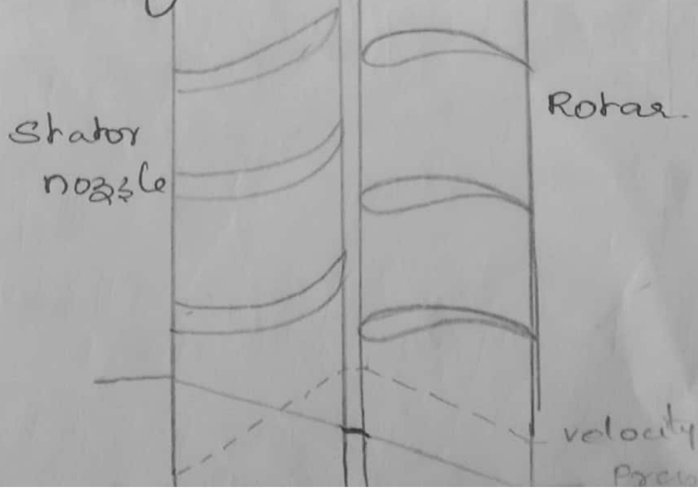
① Impulse Stage:-

Impulse machines are those in which there is no change in pressure of the working fluid in the rotor. The rotor blades only transfer the energy and there is no energy transformation. The energy transformation takes place in the stator nozzles, The transfer of kinetic energy to the rotor in an impulse machine from a high velocity ~~noz~~ fluid occurs only due to the impulsive action of the fluid on the rotor. The figure shows an impulse turbine stage. It can be seen from the figure that in the rotor blade passages of an impulse turbine, there is no acceleration of the fluid i.e, there is no energy transformation in the rotor. In this type the chances for boundary layer seperation is greater which results in losses, there by reducing the stage efficiency.



② Reaction stage:-

The reaction machines are those in which the changes in pressure occurs both in the Stator and rotar blades In this case the energy transformation occurs both in the fixed as well as moving blades. The rotar experiences both energy transfer as well as energy transformation, this is mainly due to the continuous accelaration of the flow with reduced losses which results in a higher efficiency. The figure shows a reaction turbine stage along with the pressure and velocity changes as the fluid pass through it.



* Velocity triangle & Work Output.

The flow geometry at the entry and exit of a turbo machine stage can be described with the help of velocity triangles. The velocity triangles for a turbine contains the following three components

i) The peripheral velocity or blade velocity (u)

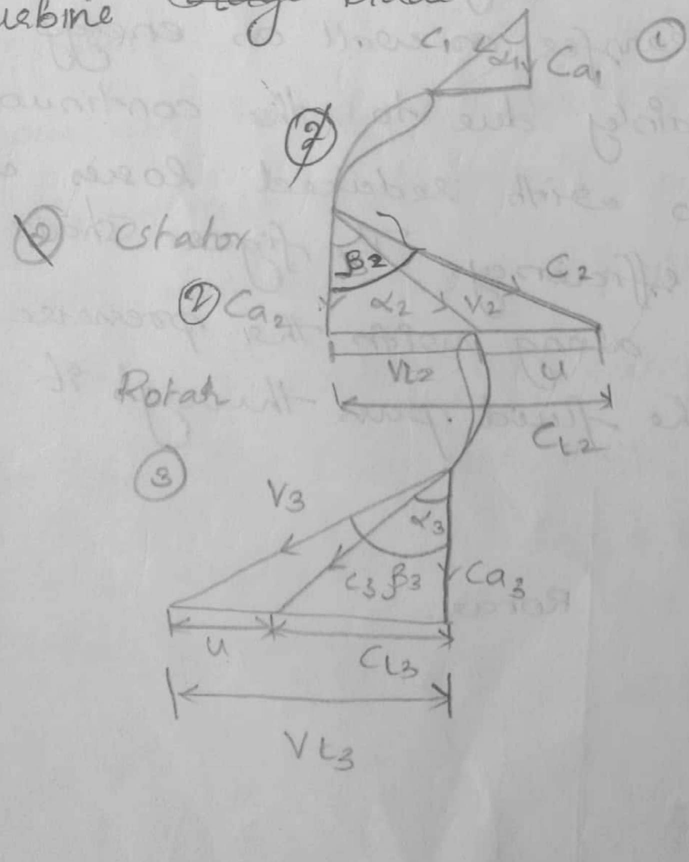
ii) Absolute velocity of the fluid (c)

iii) Relative velocity of the fluid (v or ω)

The above three velocities can be related as follows,

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

The velocity triangles at the entry and exit of a turbine stage blade is as shown in the figure;



From the figure,

$$\tan\alpha_2 = \frac{C_{t2}}{C_a} \longrightarrow (1)$$

$$\tan\beta_2 = \frac{V_{t2}}{C_a} \longrightarrow (2)$$

$$\tan\alpha_3 = \frac{C_{t3}}{C_a} \longrightarrow (3)$$

$$\tan\beta_3 = \frac{V_{t3}}{C_a} \longrightarrow (4)$$

From the above (4) equations we get

$C_{t2} = C_a \tan\alpha_2 \longrightarrow (5)$

$V_{t2} = C_a \tan\beta_2 \longrightarrow (6)$

$C_{t3} = C_a \tan\alpha_3 \longrightarrow (7)$

$V_{t3} = C_a \tan\beta_3 \longrightarrow (8)$

The axial component of absolute velocity, (C_a) is comstant in a stage, therefore we can write,

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

It can also be seen that, the peripheral velocity or the blade velocity (u) remains constant, for velocity triangles

In the rotar blades the energy is transfered from the fluid which is turn produces the shaft power. The stage work in an axial flow turbine can be expressed as follows,

$$W = u(C_{t2} - C_{t3})$$

Now considering the direction for tangential compo[nent] of absolute velocity we can write,

$C_{t3} \rightarrow +ve$

$\therefore$ direction to relative while resolving (

$$W = u(C_{t2} - C_{t3})$$

$$= u(C_{t2} - -C_{t3})$$

$$\boxed{W = u(C_{t2} + C_{t3})} \rightarrow (9)$$

from (5) & (7) we can substitute the expressions for C_{t2} & C_{t3} in (9)

$$\boxed{W = u\,C_a(\tan\alpha_2 + \tan\alpha_3)} \longrightarrow (10)$$

26/4/19

from velocity triangle at station (2) we can wri[te]

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

$$u = C_a(\tan\alpha_2 - \tan\beta_2) \rightarrow (11)$$

from velocity triangle at station (3) we can write

$$u = V_{t3} - C_{t3}$$

$$u = C_a(\tan\beta_3 - \tan\alpha_3) \longrightarrow (12)$$

Now equating (11) & (12) we get,

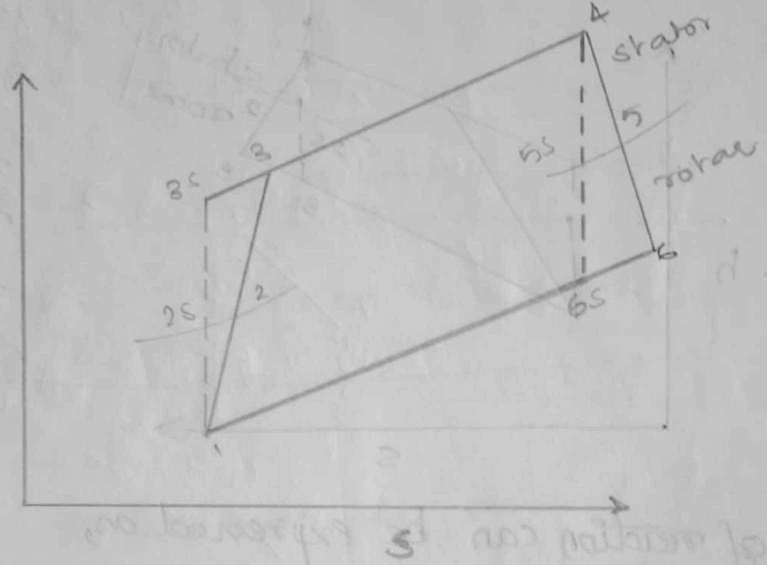
$$\tan\alpha_2 - \tan\beta_2 = \tan\beta_3 - \tan\alpha_3 \longrightarrow (13)$$

$$\tan\alpha_2 + \tan\alpha_3 = \tan\beta_2 + \tan\beta_3 \longrightarrow (14)$$

Now substituting the expression for 'α' from equation (14) into eqn (10) we get,

$$\boxed{W = UC_a(\tan\beta_2 + \tan\beta_3)} \longrightarrow (15)$$

The work done by an axial flow turbine can also be expressed in terms of change in enthalpy as follows,



The work done 'w' can be expressed as,

$$\boxed{\begin{aligned} W &= h_4 - h_6 \\ W &= h_{04} - h_{06} \end{aligned}}$$

$$\boxed{\begin{aligned} W &= C_p(T_4 - T_6) \\ W &= C_p(T_{04} - T_{06}) \end{aligned}}$$

* Degree of Reaction of an axial flow turbine.

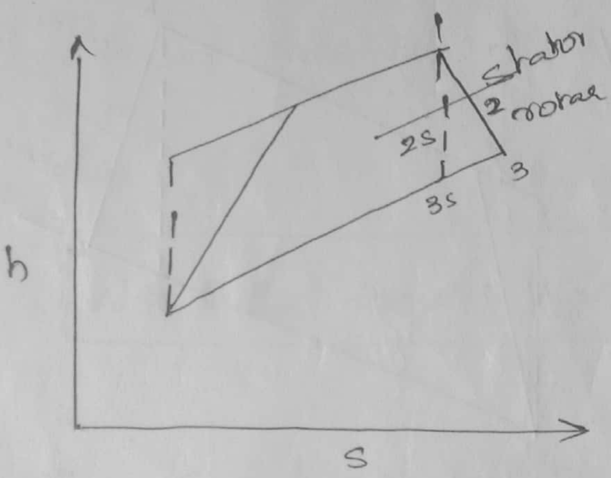
For an axial flow turbine the degree of reaction R is defined as the ratio of enthalpy ~~dege~~ decrease in the rotor to that of enthalpy decrease in the stage.

Scanned by CamScanner

$$R = \frac{\text{enthalpy decrease in rotor}}{\text{enthalpy decrease in a stage}}$$

The h-s diagram for an axial flow turbine is as shown in the figure,

$$\cancel{R = \frac{h_2 - h_3}{h_1 - h_3}} = \cancel{\frac{T_2 - T_3}{T_1 - T_3}}$$



In this case ① denotes inlet to the turbine & ③ denotes exit of the turbine

The degree of reaction can be expressed as,

$$R = \frac{h_2 - h_3}{h_1 - h_3} = \frac{T_2 - T_3}{T_1 - T_3}$$

work

From the h-s diagram the work stage ① of an axial flow turbine can be written as,

$$W = h_1 - h_3$$
$$= C_p (T_1 - T_3)$$

Now considering the Bernoulli's theorem [enthalpy & tangential component of absolute velocity] for the rotor blades we can write,

$$h_2 + \frac{1}{2} C_{t2}^2 = h_3 + \frac{1}{2} C_{t3}^2$$

$$h_2 - h_3 = \frac{1}{2}\left[C_{t3}^2 - C_{t2}^2\right] \quad \text{—} ①$$

The work done can be expressed in terms of α in follows, from eqn (10)

$$W = h_1 - h_3 = U C_a (\tan\alpha_2 + \tan\alpha_3) \rightarrow (2)$$

$$R = \frac{\frac{1}{2}(C_{t3}^2 - C_{t2}^2)}{U C_a (\tan\alpha_2 + \tan\alpha_3)}$$

$$R = \frac{\frac{1}{2} C_a^2 (\tan^2\alpha_3 - \tan^2\alpha_2)}{U C_a (\tan\alpha_2 + \tan\alpha_3)}$$

$$\boxed{R = \frac{C_a}{2U}[\tan\alpha_3 - \tan\alpha_2]}$$

NOTE

Whenever the DOR is to be derived for a (compressor or turbine). The expression should be in terms of 'β'. To derive the relation in terms of β, the Bernoulli's theorem applied to rotor should be expressed in terms of 'V_t' as follows,

$$h_2 + \frac{1}{2} V_t^2 = h_3 + \frac{1}{2} V_{t3}^2$$

$$h_2 - h_3 = \frac{1}{2}(V_{t3}^2 - V_{t2}^2) \longrightarrow$$

$$W = h_1 - h_3 = U C_a (\tan\beta_2 + \tan\beta_3)$$

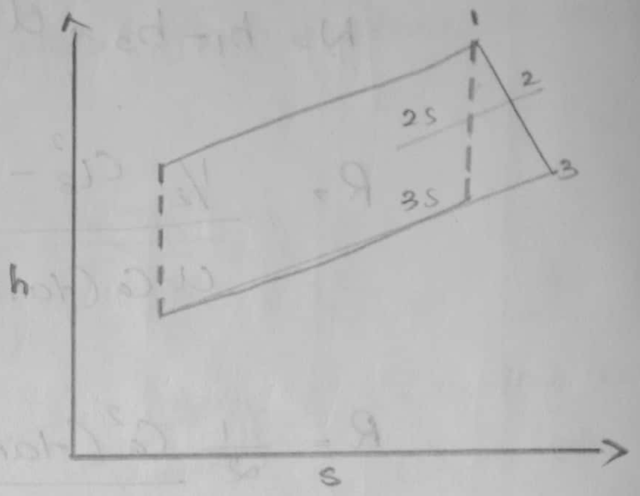
$$R = \frac{\frac{1}{2} C_a^2 (\tan^2\beta_3 - \tan^2\beta_2)}{U C_a (\tan\beta_2 + \tan\beta_3)}$$

$$\boxed{R = \frac{C_a}{2U}(\tan\beta_3 - \tan\beta_2)}$$

Scanned by CamScanner

26/4/19

* Efficiency Of Axial flow turbine



The efficiency of an axial flow turbine can be defined as the ratio of the actual work output to that of ideal work output and it can expressed as follows,

$$\eta_{tt} = \frac{h_{01} - h_{03}}{h_{01} - h_{03s}}$$ (total - total efficiency)

$$\eta_{ts} = \frac{h_{01} - h_{3}}{h_{01} - h_{3s}}$$ (total - static efficiency)

$$\eta_{ss} = \frac{h_{1} - h_{3}}{h_{1} - h_{3s}}$$ (static - static efficiency)

The efficiency can be expressed in term of pressure ratios as follows;

$$\eta_{tt} = \frac{C_p (T_{01} - T_{03})}{C_p (T_{01} - T_{03s})}$$

$$= \frac{T_{01} - T_{03}}{T_{01}\left(1 - \frac{T_{03s}}{T_{01}}\right)}$$

from isentropic relation, $\frac{T_{03s}}{T_{01}} = \left(\frac{P_{03}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}}$

$$\eta_{tt} = \frac{T_{01} - T_{03}}{T_{01}\left[1 - \left(\frac{P_{03}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

2017- (10)

* Turbine & Compressor Matching

The problem of matching turbine and compressor has a great importance in jet engines which must operate under conditions involving large variations in thrust, inlet pressure, temperature and flight MACH number, Mach Matching the components of turbo fan and turbo prop engines involves a similar procedure but in the case, the procedure for turbojet engine is explained.

The studystate engine ~~perp~~ performance at each speed is determined by two conditions,

(1) Continuity of the flow

(2) Power balance

The turbine mass flow must be the sum of the compressor mass flow and the fuel flow from which the compressor bleed flow should be subtracted. Also the power output of the turbine, must be equal to the power demanded by the compressor.

For given flight MACH number, ambient conditions, diffuser and nozzle efficiencies and flow areas we can determine the performance of a jet engine from the performance characteristics of the turbine and compressor as shown below, The matching procedure of turbine and compressor are as follows,

1) Select the operating speed.

2) Assume turbine inlet temperature.

3) Assume compressor pressure ratio.

4) Calculate the compressor work per unit mass. (of air)

5) Calculate the turbine pressure ratio [expansion ratio] to produce this work.

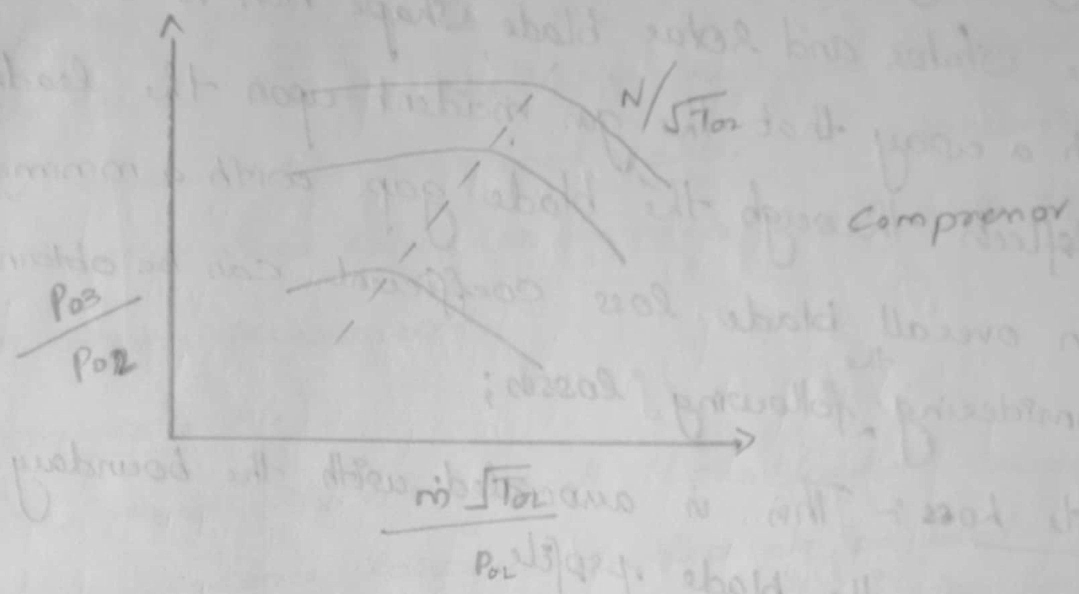
6) Check to see that if the compressor mass flow plus the fuel flow equals to that of turbine mass flow, if it is not the case assume a new value of the compressor pressure ratio, and repeats the steps 4, 5 & 6 until it is satisfied.

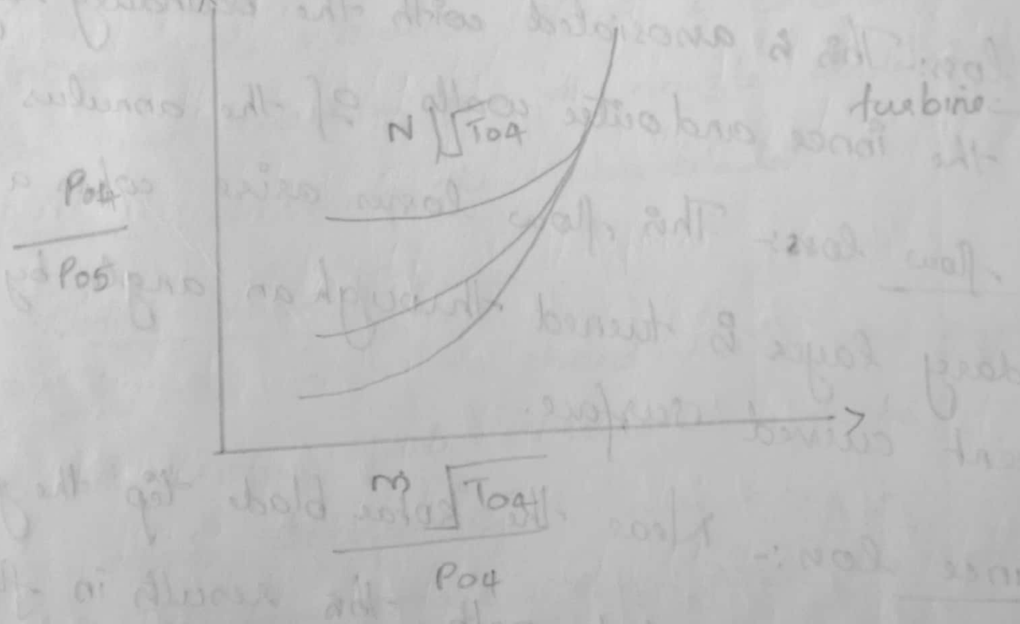
7) Now calculate the pressure ratio across the jet nozzle from the pressure ratios across diffuser, compressor, combustion chamber & turbine.

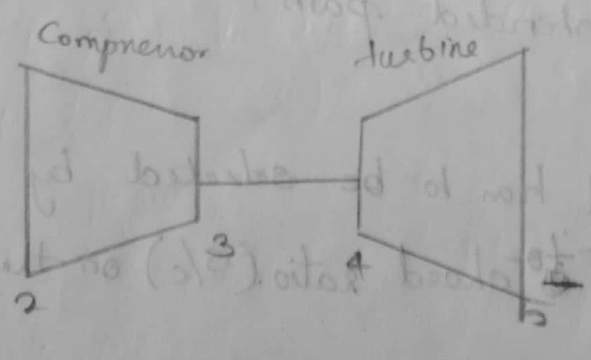
8) Calculate the area of the jet nozzle [exhaust nozzle] which is necessary to pass the turbine mass flow calculated in step 6. If the calculated area does not match the actual exit area then assume, a new value of turbine inlet temperature and repeat the entire procedure.

—

The designer will try to match the compressor and turbine so that the compressor is operating at its peak efficiency through the entire range of operation as shown in the figure,







27/4/19

98 Imp * Basic Blade Profile Design Considerations or choice of blade profile, pitch and chord

If the gas angles at various blade heights is determined then the stator and rotor blade shape has to be determined in such a way that the gas incident upon the leading edge deflects through the blade gap with a minimum loss. An overall blade loss coefficient can be obtained by considering the following losses;

(1) Profile Loss :- This is associated with the boundary layer growth over the blade profile.

(2) Annulus loss :- This is associated with the boundary layer growth on the inner and outer walls of the annulus

(3) Secondary flow loss :- This flow losses arise when a wall boundary layer is turned through an angle by an adjacent curved surface.

(4) Tip clearance loss :- Near the rotor blade tip the gas does not flow the intended path, this results in flow losses

Now the pitch and chord has to be selected by considering

(1) The effect of pitch to chord ratio (S/c) on the blade loss coefficient.

(2) The effect of chord on the aspect ratio [b/c]

(3) The effect of rotar blade chord on the blade stresses

(4) The effect of rotar blade pitch upon the stress at the point of attachment of the blades to the turbine disc

(1) Optimum pitch-to-chord ratio.

The greater the gas deflection required the smaller must be the optimum pitch-to-chord ratio. The optimum value of Pitch-to-chord ratio, can be identified by estimating its effect on stage performance.

(2) Aspect ratio (b/c)

If the aspect ratio is very small it can lead to secondary flow and tip clearance losses, which depends on blade height. On the other hand if the aspect ratio is too large, it increases the vibration effects. A common practice is to use an even number for the nozzle blade (stator ~~nozzle~~ blade) and a prime number for the rotar blades

(3) Rotar blade stresses

Once the pitch to chord ratio and aspect is selected when the rotar blade stresses has to be evaluated. The three main sources of stress are, -

(a) Centrifugal tensile stress. - when the rotational speed is specified, the allowable centrifugal tensile stress imposes a limit on the annulus area. It can be given by the equation,

$$\sigma_{ct} = \frac{\delta_b \omega^2}{a} \int_r^t a r \, dr$$

where 'ρ_b' is the density of the blade, 'ω' is the angular velocity, 'a' is the crossectional area of the blade and 'r' denotes radius of the blade

ⓑ Gas - bending stress:-

The gas bending stress is produced due to the interaction of the gas with the blade surface. It can be given the expression,

$$\sigma_{gb} = \frac{m(C_{t2} + C_{t3})}{n} \frac{b}{2zc^3}$$

where, n is the no. of blades

b - span

c - blade chord

z, section modulus.

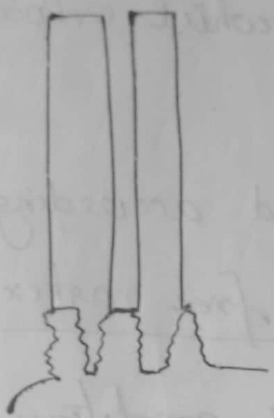
ⓒ Centrifugal Bending Stress:-

This denotes the stress induced on the blades due to the centrifugal action of the gas flowing through the vane.

④ Effect of blade pitch on blade root fixing:-

The pitch is defined as the gap between two rotar blades, if the pitch selected is so small, then the blades and the disc can be manufactured by a single forging. The disadvantage is that during maintanance it will difficult to remove the individual blades.

If the pitch selected is large then it is advantages from maintance point of view. But it may affect the aerodynamics the flares to between the blades. Which may result in energy loss: When the turbine is running the blades are held firmly in the serrations. by centripetal force



fir-tree-root fixing

29/4/19

* Free vortex & Constant nozzle angle design.

The shape of the velocity triangles must vary from root to tip of the blade, this is because the blade velocity 'U' increases with the radius of the blade. Another reason is that the tangential component of the velocity ~~causes~~ (whirl component) causes the static pressure and the temperature to vary across the annulus. With a uniform pressure at the inlet it is clear that the pressure drop across the nozzle will vary giving rise to a corresponding variation in the exit absolute velocity. Twisted blading designed to take into account of the change in gas angles is reffered as vortex blading.

① → Free vortex design

① The stagnation enthalpy h_0 is constant over the annulus, ie, $\frac{dh_0}{dr} = 0$

② The axial velocity [absolute component] is constant over the annulus.

③ The tangential velocity or the whirl velocity is inversely proportional to the radius.

A turbine stage which is designed according to the above three conditions is reffered as 'a free vortex stage' or 'a free vortex design'. Applying this conditions to a turbine stage can ensure a uniform inlet condition to the stator nozzles, and further more the nozzles are designed, so as to keep,

C_a = Constant

$C_t r$ = Constant

If all the three conditions are fullfilled then the condition for radial equllibrium is satisfied.

Therefore the work done by a stage can be represented as,

$$W_s = U(C_{t2} + C_{t3})$$

$$W_s = \omega(C_{t2}r + C_{t3}r) = \text{constant}$$

It can be seen that, a free vortex design is one in which the work done per unit mass of a gas is constant over the annulus ~~over~~ and to obtain the total work output, this specific value has to be calculated at one convienent radius and then multiplied by the mass flow rate.

② → Constant Nozzle angle design.

As in the case of axial flow compressor, it is not essential to design the blades using free vortex concept. Conditions other than

(i) C_a = constant.

(ii) $C_t r$ = constant

maybe used to produce some other form of vortex flow which can still satisfy the radial equilibrium. In particular it is desirable to make a constant nozzle angle design as one of the conditions which determines the type of vortex flow.

The constant nozzle angle design leads to a condition as shown below,

$$C_t r^{\sin^2\alpha} = \text{constant}$$

$$C_a r^{\sin^2\alpha} = \text{constant}$$

3d 4

* Performance charateristics of an axial flow turbine.

The performance of an axial flow turbine is limited by two factors

1) Compressibility

2) Stress

The compressibility limits the mass flow rate that can pass through a gas turbine and the stress limits the rotational speed 'U'. The work produced per stage of a turbine depends upon the rotational speed 'U'. As the maximum temperature increases the allowable stress limits decreases, Therefore while designing an engine it should be a compromise between the maximum temperature and the maximum blade velocity Umax.