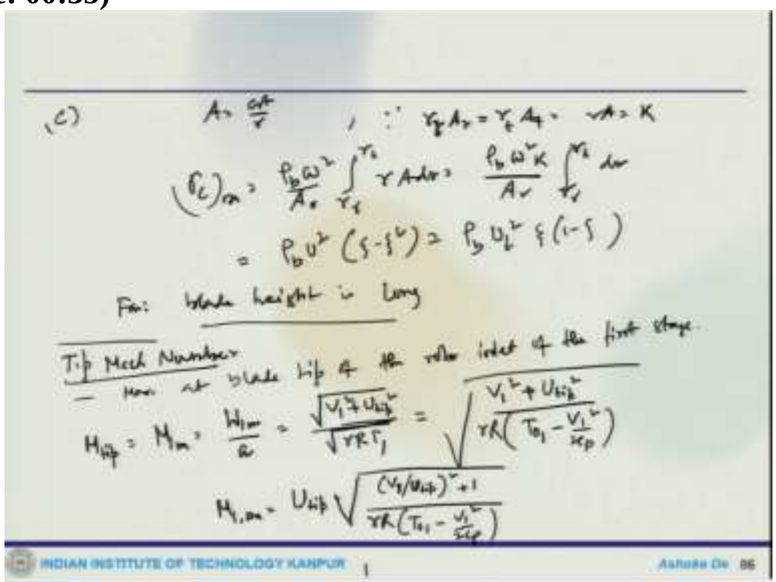
Int1roduction to Airbreathing Propulsion
Prof. Ashok De
Department of Aerospace Engineering
Indian Institute of Technology – Kanpur

Lecture - 52
Axial Compressor (Contd.,)

So, let us look at this factor which actually causes this design of the axial flow compressor. We have looked at how the different parametric basic design parameters like rotational speed blade and all these and what the kind of forces they are exposed to.

(Refer Slide Time: 00:35)



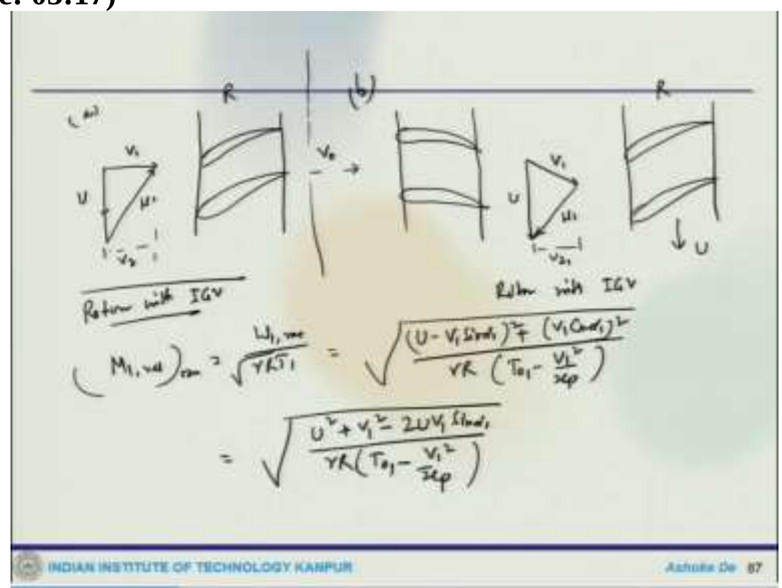
So, we are looking at now the tip Mach number. So, this I said that because this is another important parameter, because the maximum Mach number in the axial flow compressor occurs at the blade tip of the rotor inlet on the first stage. So, this is essentially maximum at blade tip of the rotor inlet of the first stage. So, this is due to the maximum value of the relative speed W 1 and the maximum static temperature of the sonic inlet.

So, that for the first the axial velocity is equal to the inlet absolute velocity and has a constant value along the annulus. So, the corresponding maximum Mach number which you can estimate Mach tip is a

$$M_{tip} = M_{1max} = \frac{W_{1max}}{a} = \frac{\sqrt{V_1^2 + U_{tip}^2}}{\sqrt{\gamma R T_1}} = \sqrt{\frac{V_1^2 + U_{tip}^2}{\gamma R \left(T_{01} - \frac{V_1^2}{2C_p}\right)}}$$

So, from this particular equation for a specified tip Mach number and rotational speed the actual velocity of the inlet can be determined. Now, for a transonic compressor an acceptable value of tip Mach number would be around 1.3 while the fan in turbofan engines much higher Mach number is expected and allowed, so, which would be typically 1.5 to 1.7. So, the challenging design is to increase the rotational speed U and keep tip Mach number with acceptable limits. So, this can be achieved by inlet guide vanes.

(Refer Slide Time: 03:17)



So, we can see how I mean already we have looked at that, but again just to reiterate the fact by showing some so the velocity triangle would be like this. So, this is V_1, U, this is W_1 and this is V_z, this is a. Now, that V_0 which goes in, so, these are the blades here. So, when it comes out that comes out this is U, this is W_1, so that is V_{z1} and this is U, V_1 then again, so this goes back so that is how the U also. So, this is 2 different scenarios.

This section is a, so this is a situation where rotor, this is rotor without IGV Inlet Guide Vanes. So this is how it happens? And this is a case with rotor this is rotor with IGV. So with the presence of Inlet Guide Vanes, you can actually increase the speed, but at the blade speed, but at the same time that tip Mach number can be under within kept within the acceptable limit. So, if we also add a swirl component to this V_1 here, then this can reduces the relative speed of W_1.

Now, one has to note that the passage of the inlet guide vanes are nozzles to increase the absolute speed of V_1 which is greater than V_0 does keep a constant axial velocity of V_z. Now, the maximum tip Mach number would be

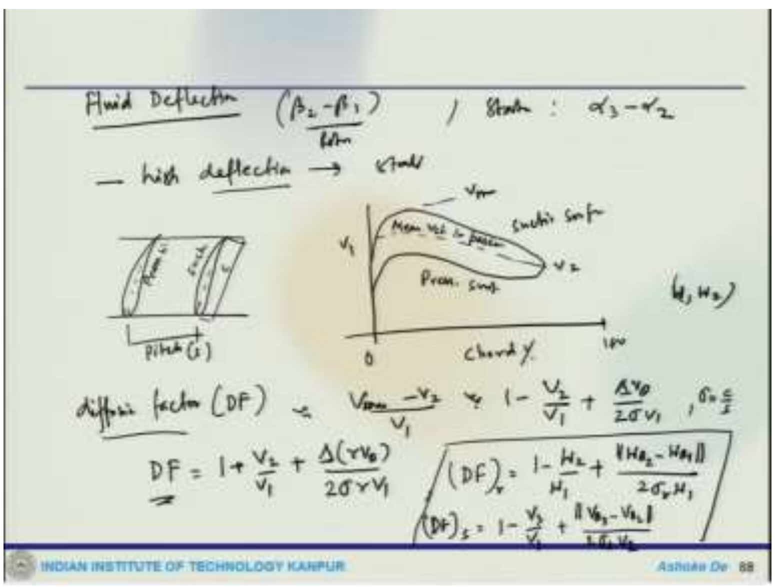
$$M_{tip_{max}} = \frac{W_{1max}}{\sqrt{\gamma R T_1}} = \sqrt{\frac{(U - V_1 \sin \alpha_1)^2 + (V_1 \cos \alpha_1)^2}{\gamma R \left(T_{01} - \frac{V_1^2}{2C_p}\right)}}$$

where alpha is the angle between the absolute velocity and the axial direction. The maximum relative Mach number can be written as

$$M_{rel_{max}} = \sqrt{\frac{U^2 + V_1^2 - 2UV_1 \sin \alpha_1}{\gamma R \left(T_{01} - \frac{V_1^2}{2C_p}\right)}}$$

So, this is what it happens.

(Refer Slide Time: 06:53)



Now, the other condition is the fluid deflection. So, that means $\beta_2 - \beta_1$. So, this is the difference between the outlet and inlet angles this is for rotor, and for stator it could be $\alpha_3 - \alpha_2$. So, excessive deflection in the flow angle which means a high rate of diffusion will lead to blade stall. So, high deflection leads to stall. So, to maximize the specific work and the difference between the flow angles in rotor must attain a maximum allowable value.

So, if the inlet relative velocity maintains a constant angle beta1 then the outlet relative to control stall on in other condition. So, one can look at the situations where let us say you have a passage of cascade you can look at how things looks like this. So, you have essentially blade like this. So, this is the pressure side, then this is the suction side, and typically this is the chord and this is the pitch.

So, now, so this is chord percentage, so this is 100 percent 0, so that V_{max}, this is suction surface, this is V_2, this is mean velocity in passage and this is pressure surface. So, what happens that constant axial velocities assumed, and then the relative angle is measure of a magnitude of the relative velocities W_1 and W_2. So, the criteria which is called the diffusion factor of the DF. So, this was developed by NACA which can be adopted here and another.

So, this can be defined the diffusion factor. So, typical diffusion factor is

$$DF = \frac{V_{max} - V_1}{V_1} = 1 - \frac{V_2}{V_1} + \frac{\Delta V_\theta}{2\sigma V_1}$$

where

$$\sigma = \frac{C}{S}$$

So, a more precise expression on can write in the mean radius. So, that would be written as

$$DF = 1 + \frac{V_2}{V_1} + \frac{\Delta r V_\theta}{2\sigma r V_1}$$

Now values of DF in excess upon 6 are throughout to indicate blades stall and the value upon 4 pi might be taken typical design choice.

So, these diffusion factors is important, because if you have too much of DF that means this can lead to high deflection could lead to the stall. So, for both rotor and stator it can be combinely expressed that

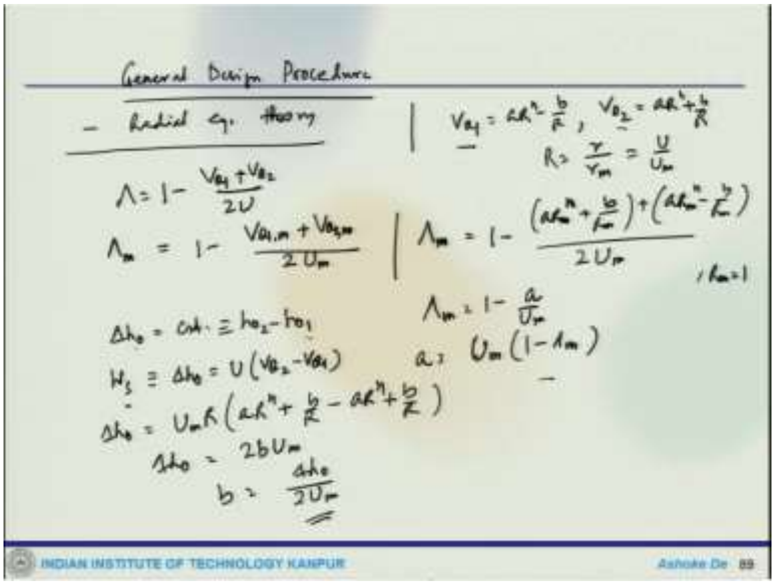
$$(DF)_r = 1 - \frac{W_2}{W_1} + \frac{W_{\theta 2} - W_{\theta 1}}{2\sigma_r W_1}$$

So, this is rotor and the stator it could be

$$(DF)_s = 1 - \frac{V_3}{V_1} + \frac{V_{\theta 3} - V_{\theta 2}}{2\sigma_s V_2}$$

So, these are 2 diffusion factors for I mean the way it is defined for the rotor and stator and that can be used. Now, these are the some of the discussion on the basic design parameter. Now, if we look at how we go about the basic design procedure.

(Refer Slide Time: 12:34)



Now, or rather general so, we can quickly look upon that thing, the general design procedure. So, that already we have discussed the disadvantage of the free vertex method and now the design is based on arbitrarily chosen radial distribution, any 2 blades which is essentially the radial equilibrium theory. So, that is radial equilibrium theory or which is called the vortex energy equation.

And so, what we can look at. So, the designer usually prescribes the velocity somewhere between a force vortex and the free vortex and according to the relation, so the relations like if

$$V_{\theta 1} = aR^n - \frac{b}{R}$$

Or

$$V_{\theta 2} = aR^n + \frac{b}{R}$$

So,

$$R = \frac{r}{r_m} = \frac{U}{U_m}$$

Now value of a is determined from the degree reactions where

$$\Lambda = 1 - \frac{V_{\theta 1} + V_{\theta 2}}{2U}$$

and the mean radius

$$\Lambda_m = 1 - \frac{V_{\theta 1 m} + V_{\theta 2 m}}{2U_m}$$

Now, substitute the values of well velocities then these tangential velocities or component. So, what do we get

$$\Lambda_m = 1 - \frac{\left(aR^n{}_m + \frac{b}{R_m}\right) + \left(aR^n{}_m - \frac{b}{R_m}\right)}{2U_m}$$

So, $R_m = 1$, then this would be

$$\Lambda_m = 1 - \frac{a}{2U_m}$$

and

$$a = U_m(1 - \Lambda_m)$$

the constant b, which will be calculated for the constant condition of constant specific work, which is Δh_0, which is constant that is $h_{02} - h_{01}$

.

So,

$$W_s = \Delta h_0 = U(V_{\theta 2} - V_{\theta 1})$$

So, what we can write

$$\Delta h_0 = U_m R\left(aR^n + \frac{b}{R} - aR^n + \frac{b}{R}\right)$$

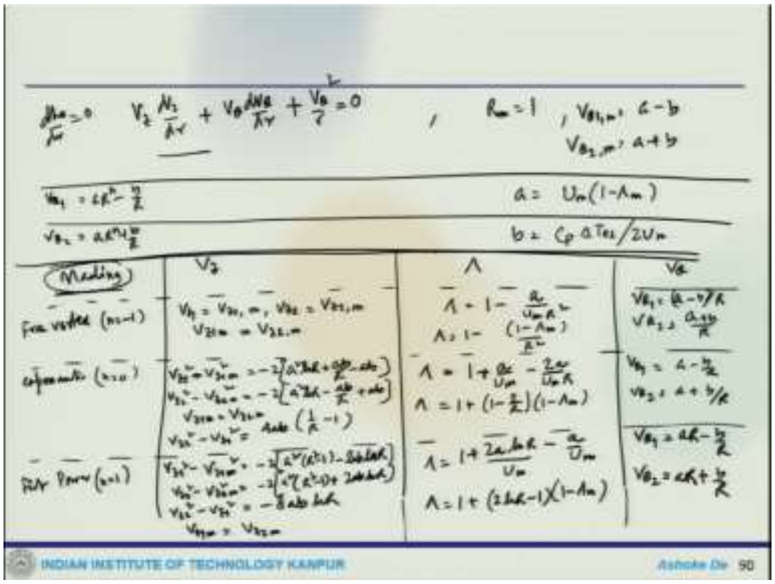
So, this looks

$$\Delta h_0 = 2bU_m$$

$$b = \frac{\Delta h_0}{2U_m}$$

So, once you have these values of a and b, so these are one can see these values are independent of r and this can be used. Now, the total enthalpy upstream of the stage h_{01} is assumed to be constant and independent of r and since work the specific work is $h_{02} - h_{01}$ work which is also assumed to be constant the downstream enthalpy h_{02} is also constant. So, if this is constant and these are the ahead of it assumed to be constant, so, these become also constant.

(Refer Slide Time: 16:47)



Now, this actually the already we have seen the radial equilibrium theory which we have obtained that. And that is what we have already got here. So, if you recall that

$$\frac{dh_0}{dr} = 0$$

so from the radial equilibrium theory, so we can put that thing back and the similar thing and write the V_z in terms of r and all this. So, then we can find out all this so

$$V_z \frac{dV_z}{dr} + V_\theta \frac{dV_\theta}{dr} + \frac{{V_\theta}^2}{r} = 0$$

So, it can be integrated between station 1 and 2 and mean radius r. So, where R_m is 1,

$$V_{\theta 1,m} = a - b$$

Or

$$V_{\theta 2,m} = a + b$$

So, the resulting values of axial velocities at radius in station 1 and 2 can be calculated. And there are big so different ways one can do that. So, for example, let us say we put

$$V_{\theta 1} = aR^n - \frac{b}{R}$$

If it is there, then

$$a = U_m(1 - \Lambda_m)$$

$$V_{\theta 2} = aR^n + \frac{b}{R}$$

then we get

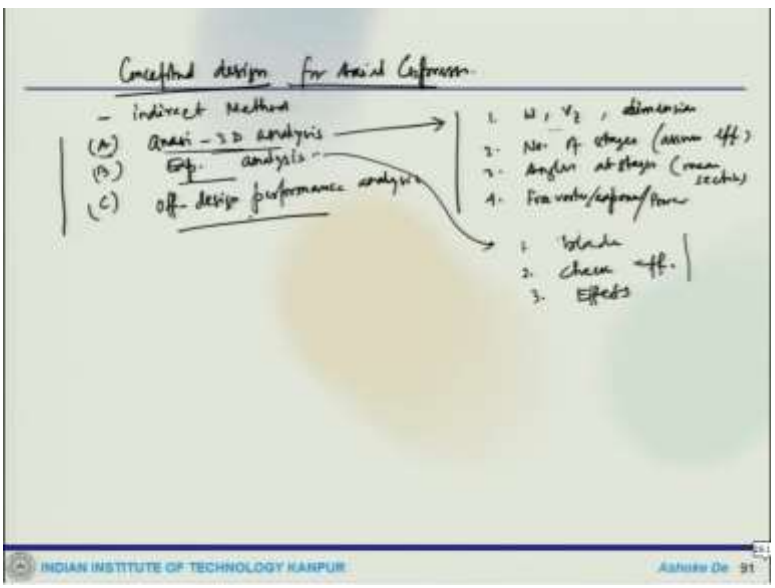
$$b = \frac{\Delta h_0}{2U_m} = \frac{C_p \Delta T_{0s}}{2U_m}$$

Or now, we can have different kind of blading and different values of, so this is what we have, this is blading V_z then we get the degrees of reaction V_θ. So, these are different, so one case it could be free vortex where n = - 1 then, $V_{z1} = V_{z1m}$, $V_{z2} = V_{z2m}$ and this would be so, and V_{z1} m is V_{z2m} . So, the degrees of reaction would be

$$\Lambda = 1 - \frac{a}{U_m R^2}$$

So, you can see when you consider the different design. So, you have to assume some of these design procedures and then follow the basic governing equations or laws to find out these different components. So, these come all in the design procedure.

(Refer Slide Time: 24:35)



Now, one can think about some conceptual design, how to get it for axial compressor so, typically the compressors are designed by indirect method. So, the first one defines the flow path that means casing and half radius and decides the thermodynamics aerodynamics forces. And then the flow path and iterative analysis is conducted the continuity momentum energy equation for such conceptual designs.

There are different stages like you can have Quasi 3D analysis or you could have experimental analysis or testing of the blades and things like that now or you have Off design performance analysis. So, the first step for Quasi 3D starts by 2D analysis which is axial tangential and this extend the; define the flow parameters. So, provide the quasi 3D analysis details of such tip one can have.

You can have a choice of compressor data like rotational speed, axial velocity and then the annulus or other dimension or dimension of the annulus. And then you can define the number of stages or determine the number of stages. So, here you have to assume the efficiency then you can calculate all the angles at stages, so, that mean section. So, once you calculate all the angles at the stages of the mean section.

Then you can find out the determine the variation of the angles from rotor to tip by different blading like free vortex or exponential or power like that. So, this is what one can do or think about the analysis which requires the 3D analysis. So, 3D Quasi analysis starts with the 2D analysis, where you assume some of these speed axial velocity dimensions and then determine some of these parameters.

Or in any other way if somebody does experimental analysis of the blades you can choose the blade from the experimental cascade data. So, then you can check efficiency which is compared to the assumed cascade data and then finally look at the other effects like compressibility and such parameters and then you go for the off design performance analysis. So, essentially this kind of conceptual design if you look at it down in the compressor design.

So, you go in 3 different stages or steps. Step 1 you have Quasi 3D analysis, step 2 you have experimental analysis and finally, the off design performance analysis. So, in a Quasi 3D analysis it starts with some 2D analysis. Is where you assume certain parameters like speed, axial velocity, dimensions and all these, then calculate some of the other parameters. And then go to the experimental analyses where as you mean those blade parameters.

You do the testing done and check the efficiencies of the compressibility effects such things and finally, the off design performance analysis. So, this is how the conceptual design is done and we will look at some of these systems in slightly in details in the subsequent lectures.