Introduction to Airbreathing Propulsion
Prof. Ashoke De
Department of Aerospace Engineering
Indian Institute of Technology – Kanpur

Lecture – 46
Centrifugal Compressor (Contd.,)

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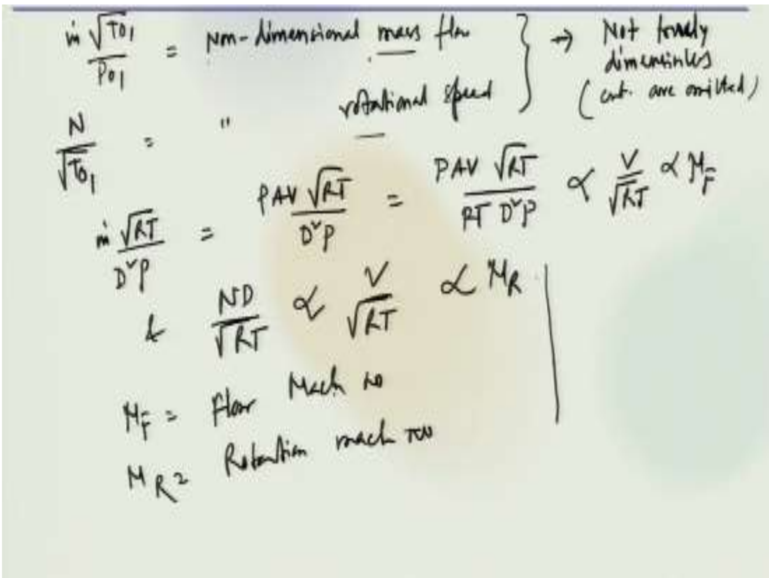

Okay, so we are in the middle of the discussion of the compressor characteristics and this is what we have been doing and this we obtain these two non dimensional group and the total non dimensional group here 4 group of non dimensional parameter and then we have got this non dimensional mass flow rate and rotational speed and from there we have obtained these two flow mach number and the rotational mach number. So now how we will get the characteristics curve.

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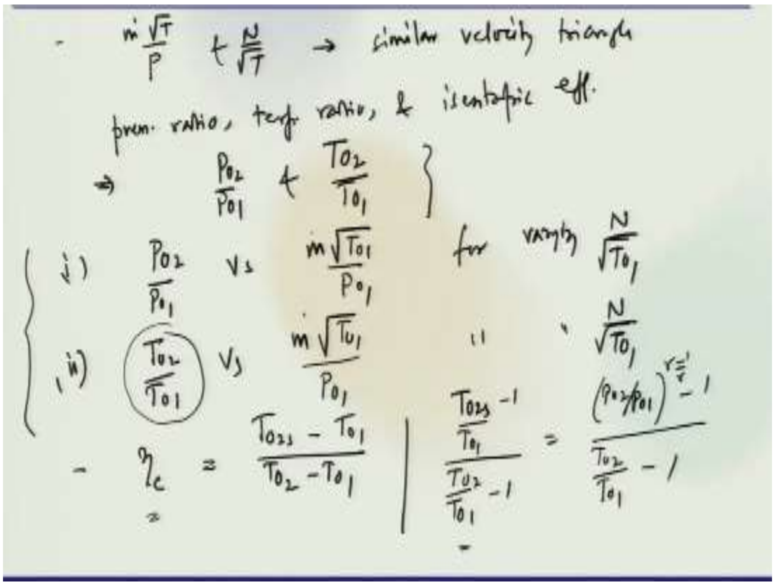

Now all operating conditions for a pair of $\frac{\dot{m}\sqrt{T}}{p}$ and $\frac{N}{\sqrt{T}}$ so this will provide similar velocity triangle. So that vane angle and the air flow direction will match and the compressor will yield the same performance in terms of pressure ratio, temperature ratio and isentropic efficiency. So this is what the non dimensional method of putting the characteristics implies. So this will yield the same $\frac{p_{02}}{p_{01}}$ and $\frac{T_{02}}{T_{01}}$

Now, one can look at two plots number one $\frac{p_{02}}{p_{01}}$ versus $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ for varying $\frac{N}{\sqrt{T_{01}}}$ and second plot is the $\frac{T_{02}}{T_{01}}$ versus $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ for varying $\frac{N}{\sqrt{T_{01}}}$. So this would be this two plot would be sufficient to provide the entire characteristics of the compressor. Now also we can obtain the isentropic efficiency like η_c which is defined as

$$\eta_c = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}}$$

And we can get

$$\frac{\frac{T_{02s}}{T_{01}} - 1}{\frac{T_{02}}{T_{01}} - 1} = \frac{\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1}$$

So, typically if you see instead of plotting these $\frac{T_{02}}{T_{01}}$ this η_c is plotted.

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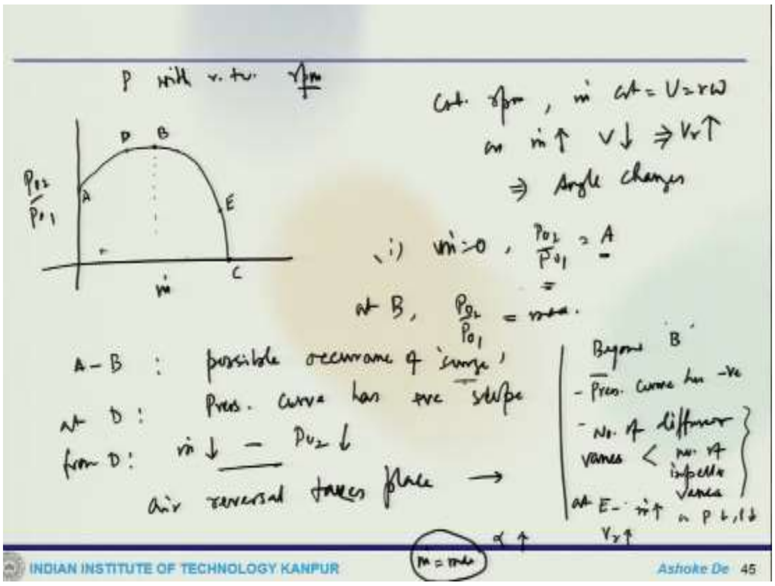

So before now considering the characteristics plot consider the variation of P with respect to rpm. So let us say the compressor is running at constant speed let there be valve at the so this is what would happen so this is $\frac{p_{02}}{p_{01}}$. Let there be a valve at the delivery line of the compressor and let us slowly open the valve. So we have constant rpm m dot constant which is $U = r\omega$ as $\dot{m}$ increases v decreases which means V_r increases.

So there would be change in angle changes. Now the first thing that would happen is that the valve is closed. When the valve is closed so the m dot would be 0 and so this is a situation where let us say $\dot{m} = 0$ so $\frac{p_{02}}{p_{01}} = A$. So corresponding to the centrifugal pressure rise because of the impeller on air trapped between the vanes. However, at this condition no flows through the diffuser hence no pressure rise in the diffuser.

Now as the valve is opened the flow starts in the diffuser hence diffuser starts adding some extra pressure and then $\frac{p_{02}}{p_{01}}$ starts increasing. So this is what happens. So this is a point where let us say the curve is like this and this goes like this and then come like this and come like this and somewhere like that. So this is A which we are just talking about then when the valve is open the flow start going through the diffuser there is added this thing so this starts increasing and reaches a point somewhere B.

Now when this is so called the efficiency probably the maximum and this point B at B, $\frac{p_{02}}{p_{01}}$ is also maximum so it reaches the maximum value. Now at very high $\dot{m}$ flow angles are quite different from the design vane angle and the air separation occurs at the vanes and the pressure drops so this curve will start dropping from there this maximum peak and any further increase is $\dot{m}$ so this is where $\dot{m}$ is increasing.

So further increasing beyond that point result in the decreasing in the $\frac{p_{02}}{p_{01}}$ and probably it reaches some point c. Now hypothetically let us say when the valve is fully open the pressure ratio may drop to unity because of the very high losses so this will be here. So all the power absorbed to overcome the internal frictional losses. So this is hypothetically this may happen. In actual practice point A can be obtained.

But the most curve between point A and B could not be obtained due to phenomena called. So between A and B there is a possible occurrence of surge. This is another new term and we are going to talk about this because these are all connected with the characteristics of the compressor. So as I said most of the situation between A to B this curve may not be obtained because of the possible occurrence of surge or surging.

So surge means there is an immediate or sudden drop in P_{02} with violent aerodynamic instability. Let us say the compressor is operating somewhere here point let us say D in between A and B it is operating somewhere point D. So the pressure curve at point D so at D the pressure curve has positive slope. Now if when it is here at point D if the mass flow rate drop so it goes in this direction.

If the mass flow rate drops so it will decrease in the pressure so P_{02} so from D if $\dot{m}$ decreases so P_{02} also drops out. So this mass flow rate decrease is accompanied by the fall in the pressure rise. Now what may happen if the pressure of the air downstream of the compressor does not fall quickly enough so there would be a positive pressure gradient exist between compressor outlet and the downstream air compressor.

So when this happens so there is a possibility that positive pressure gradient may occur. So air reversal takes place and starts to flow towards the compressor outlet so the pressure ratio drops rapidly. Now when the pressure at the downstream of the compressor falls substantially the compressor can pick up again to repeat the same cycle. So this occurs at very high frequently and is very much detrimental to the engine and this is called the sort of surging phenomena.

And when that happens this surging does not occur immediately at operating point moves left to B because initially the downstream pressure may fall at a faster rate than P_2, but as $\dot{m}$ is reduced sooner or later the compressor will go into surge unstable between A and B. So this is what is important so that means between point A and B most of the operating procedure are not or the compressor rather do not operate between these two points because surge may occur.

Now beyond B what is happening there beyond B or between B to C if you look at the pressure curve has negative slope. So this operation between B to C is also stable. In this range if $\dot{m}$ is decreased let us say somewhere here if m dot is decreased then that can be accompanied by rising the pressure lost or the P_2. So, typically surging depends on the mass handling capacity of the turbine probably it starts to occur in the diffuser vanes.

And which is already have an advance pressure gradient due to diverging process plus frictional forces. So now the tendency to surge increases with a increase in number of diffuser vanes. So when there are several diffuser vanes for one impeller channel it is very difficult to split the air equally to all the vanes so that is another problem that if you have multiple these things.

So the channel with low mass flow rate will go into surging first and the other case. So air will then start to flow up the channel the other end and then call surging that is why the number of diffuser vanes are less than number of impeller vanes. So, you can see this is an another important criteria while designing this compressor because if that happens there is a possibility of occurrence of surge.

Now since each is supplied by several compression so this can be evened out. Now even in this case at low mass flow rate all the diffuser process will go into surge simultaneously adding a limitation between B to C. So there is still the possibility so one has to be careful. Now there could be another point where let us say E where at E if the m dot goes up here so let us say at E $\dot{m}$ goes up so as pressure goes down which means density goes down which means V_r goes up so alpha increases.

So after certain point it is possible to increase $\dot{m}$ any further. So this is called the choking in a compressor and that means this corresponds to the maximum flow rate of the particular rmp for which the curve is drawn that means that $\dot{m}$ max condition is obtained. So this is what is going to happen.

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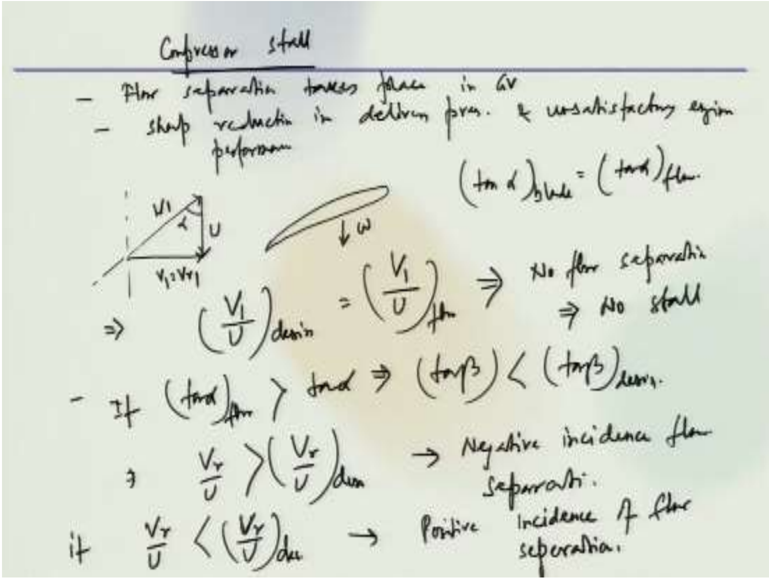

Now apart from the surge so we can look at the compressor stall. Now stall is common to both axial flow compressor and centrifugal compressor. Now centrifugal compressor both of them can go in surge and stall. Now stall is typically when the flow separation takes place. So this happens in guide vane and blade passages on blade passages and of design condition. So when the incoming flow angle is greater than the stalling angle the flow separates and leads to this stall.

So what happens with the stall there would be sharp reduction in delivery pressure and unsatisfactory engine performance. So, at design condition the flow enters the process smoothly. So if you see this is my aerofoil and this could be the velocity triangle for that. So this is U, this is W_1, this is alpha, this is $V_1 = V_{r1}$ so this is a rotation omega. So what it gives the

$$(\tan\alpha)_{blade} = (\tan\alpha)_{flow}$$

So which means

$$\left(\frac{V_1}{U}\right)_{design} = \left(\frac{V_1}{U}\right)_{flow}$$

so that no flow separation that means no stall. The same thing is true for the diffuser vane on the compressor. If

$$(\tan\alpha)_{flow} > \tan\alpha$$

$$\tan\beta < (\tan\beta)_{design}$$

So there is which means the

$$\frac{V_r}{U} > \left(\frac{V_r}{U}\right)_{design}$$

which is known as negative incidence flow separation or if

$$\frac{V_r}{U} < \left(\frac{V_r}{U}\right)_{design}$$

this is called positive incidence of flow separation.

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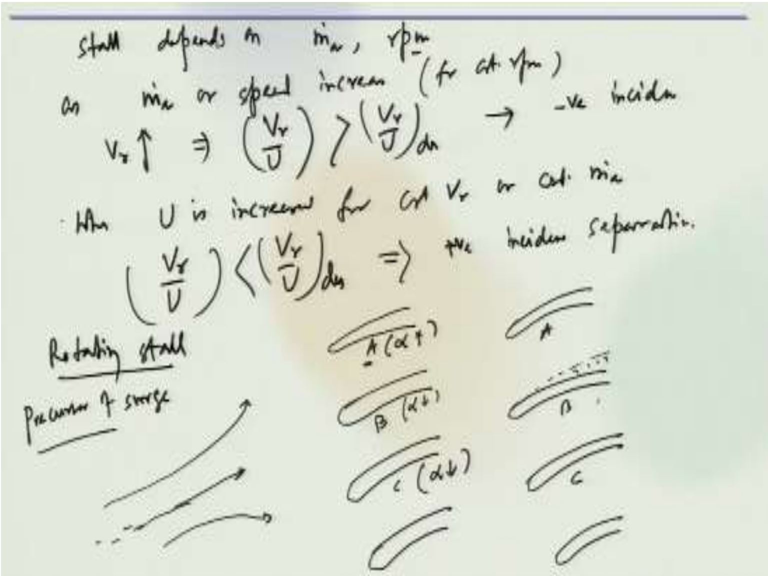

So, which means stall depends on mass flow rate rpm. So as $\dot{m}_a$ or speed increases for constant rpm V_r goes up which means $\frac{V_r}{U} > \left(\frac{V_r}{U}\right)_{design}$ so which causes negative incidence. Now when U is increased for constant V_r or constant $\dot{m}_a$ so $\frac{V_r}{U} < \left(\frac{V_r}{U}\right)_{design}$ so that is a positive incidence separation. Now this depends on all these things there could be another situation where it called rotating stall which means so since these are the rotating condition so let us say we can have passage like A.

So let us say B the airfoil C so this is a passage and then this could be A, this could be B let us say there is a separation passage C like this. So this is how the flow comes in and it distributes in this passages like this, it goes like this. So when one passage actually stalls the flow gets deflected. So essentially the flow is supposed to come in this direction, but since due to the stall the flow gets deflected towards another passage.

So when that happens so alpha for one of the neighboring passage also increases. Let us say if for A alpha increases then passage c alpha actually decreases. So this will lead to the stalling of that passage A and then that passage has already stalled originally recover since alpha in the passage B decreases. So now the stall keeps one from A it will move to B from B it will move to C so the stalls actually from one blade passage to another blade passage to another passage because of the rotation.

It goes on shifting and multiple passages can go on stalling situation simultaneously. So this is what called the rotating stall and typically this is a precursor of surge. If multiple passages go on stalling, then that can lead to the surge and this would be a detrimental effect. Now at the impeller I the stall rotates in opposite directions of the rotation of impeller. So this causes aerodynamically induced vibration so that resulting in static failure in other passages also of the gas turbine.

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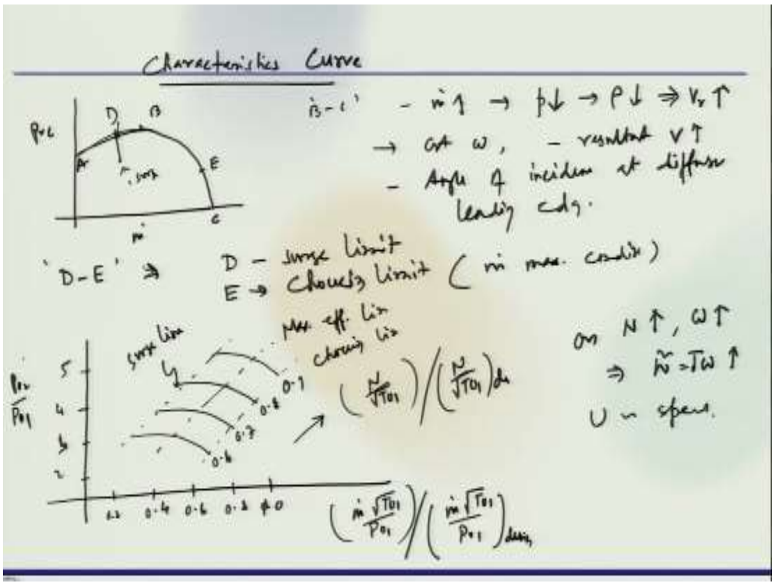

Now coming back to that characteristics curve so when you come back to that so we are considering operation at certain point. So let us say this is how this would go this is c somewhere E, B point A, D and this is P_{rc}. So point D which could be a point of surge so we got this curve. So between B to c $\dot{m}$ increases so p decreases which means ρ decreases which means V_r increases.

For a constant omega this means the increase in resultant v so that goes up. So the angle of incidence at diffuser leading edge. So the operation is typically limited between so D to E. So D, one can call surge limit and E one can call choking limit which is the $\dot{m}$ max condition. So now this particular curve P_{rc} / $\dot{m}$ is for a fixed rpm. Now once we change the rpm we can get similar curve and get the complete characteristics plot.

So let us say 0.2, 4, 6, 1.0 so this could be 2, 3, 4, 5 like that. We have P_2 / P_1 and this side we have $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ divided by $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ design. So what we get we get typically this is one limit so this is another limit and we get curve like this and this is called maximum efficiency line, this is surge line and this could be 0.6, 0.7. 0.8. 0.9 these are the choking lines and the variation is $\frac{N}{\sqrt{T_{01}}}$ divided by $\frac{N}{\sqrt{T_{01}}}$ design.

So this is how the speed varies so for different rpm which get this pressure line and what it does that as N goes up rotational speed goes up which means W T omega goes up. So more work involved and hence higher pressure rise and as we have discussed earlier the N is limited by the structural stress so which essentially controls the U for a particular speed.

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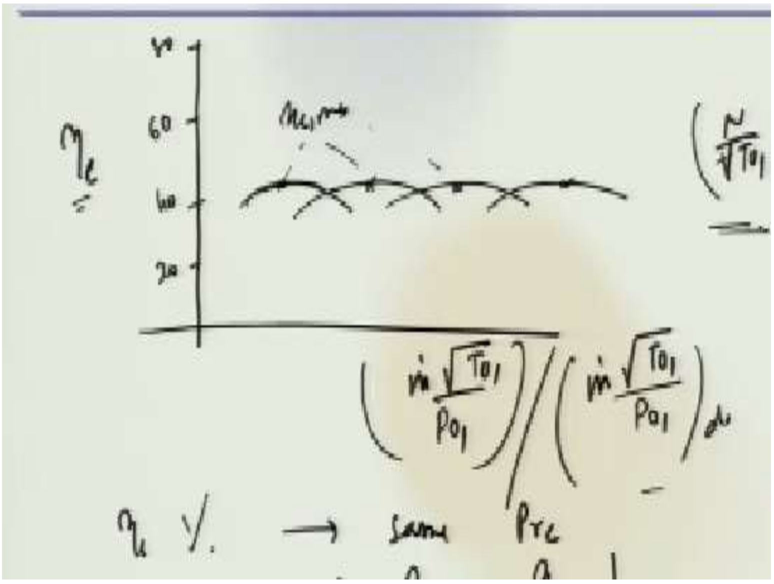

So similarly one can find the curve for η_c versus $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ divided by the $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ design. So you get typical curves like this so these are again maximum point which are varying $\frac{N}{\sqrt{T_{01}}}$ divided by $\frac{N}{\sqrt{T_{01}}}$ design. So η_c percentage varies in the same manner as P_{rc} same variation as P_{rc}. Beyond an optimum P_{rc} η_c actually decreases and if you look at these are the η_c max condition like this.

So eta c max pretty much remains almost constant with varying speed. So the compressor should be designed that it approaches along eta c max so this is sort of an gas turbine design criteria. So these are the true curve one gets so you can have 20, 40, 60 some numbers 80. So these are some particular number that has nothing to do with that so one is this pressure versus non dimensional mass flow rate and it gives you the surge line and choking line.

And other case you have the isentropic efficiency versus non dimensional mass flow rate and for varying speed and you get these two important characteristics curve and from this curve actually one can determine the characteristics of an compressor. So this is not only true for centrifugal compressor this should be also true for the axial compressor as we do the discussion in the following lecture that they also have the similar characteristics curve.

And the variation of the efficiency and pressure rise with respect to speed and non dimensional mass flow rate typically the same. So we will stop here and continue the discussion in the follow up lecture.