Turbomachinery Aerodynamics

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Lecture No. #11
Axial Compressor Characteristics
Single Stage, Multi Stage and Multi Spool Characteristics

Hello and welcome to lecture number eleven of this lecture series on turbomachinery aerodynamics and today, we are on the eleventh lecture of lecture series and we are going to discuss a very important aspect of axial flow compressors today. And in today's class, we are going to talk about performance characteristics of single and multistage axial compressors and the reason why I said it is very important is because performance characteristics are extremely important information that are required by designers and; obviously, for the engine designers as a whole because it is the compressor performance which determines the limits of operation of the engine itself. That something which we are going to discuss in today's class which means that the limit of operation of an entire aero engine will in some sense be dictated by the compressor performance itself. That is beyond certain levels, the compressor cannot operate and that will puts a limit on the engine operation itself.

And hoping that in the previous few lectures; seven or eight odd lectures which you have undergone so far, you must have had a fairly good amount of information about two dimensional as well as three dimensional aspects of flow through axial compressors. So, today's lecture is kind of an over view of the whole thing with out of course, looking at what is happening on the blade at individual stage level, but trying to find out how certain compressor which has been designed is going to perform when the operating conditions change. So, as the operating conditions of the compressor changes, how does a compressor react to such change in operating condition?

So, that is the aspect of discussion which we going to take up in today's class. So, we are going to talk about two distinct aspects which are related to performance characteristics of axial compressors. We will begin with the discussion on a single stage axial compressor performance and we will extend that to a multistage axial compressor. They are the basic trends are identical for both of them, but they are distinct differences between what happens in a single stage axial compressor as compared to a multistage axial compressor. So, these are two aspects of compressor performance that we going to discuss in today’s class.

We will start with a single stage axial compressor. Now you might be aware that a stage of an axial compressor is constituted of a rotor and a stator. So, rotor precedes the stator. So, a rotor and stator combination together put together is what is known as a stage of an axial compressor and it is this that we are going to analyze in today’s lecture on how we can estimate the performance of a single stage and how the single stage performance varies as the operating conditions change. So, in today’s class, we will basically be talking about single and multistage axial compressor characteristics. Let us start with the single stage axial compressor.

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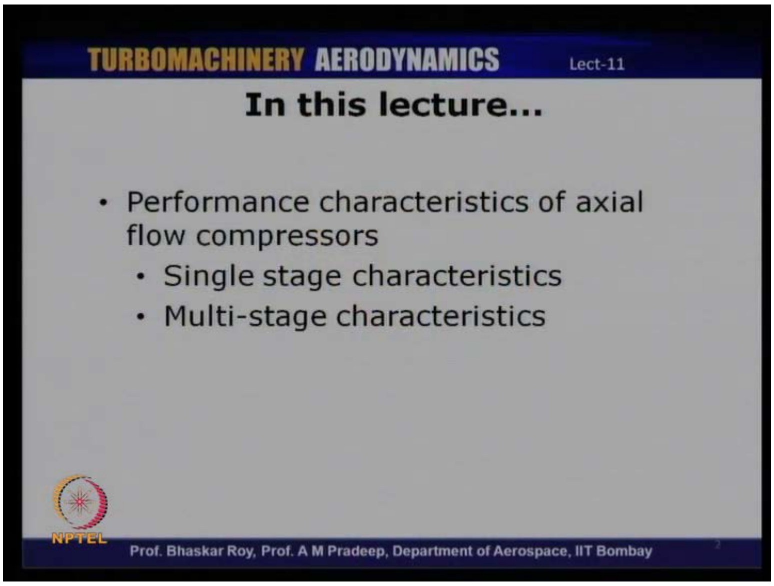

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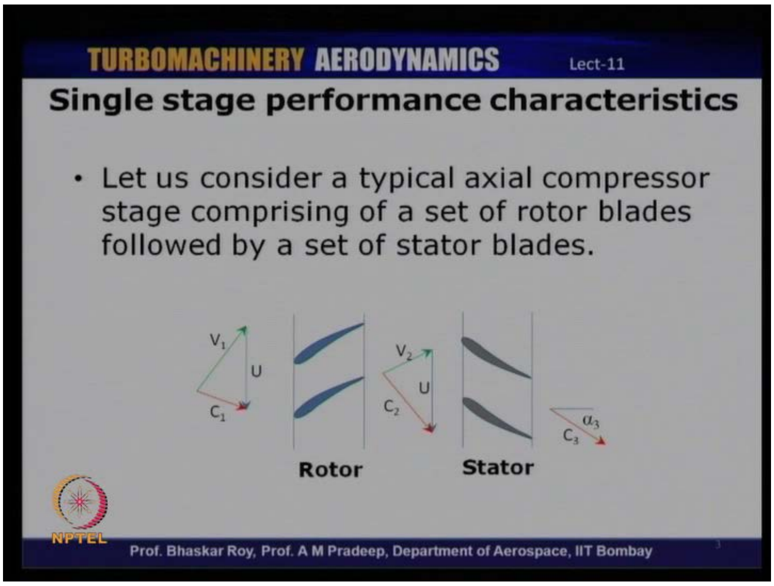

So, I mentioned that a combination of a rotor and a stator put together is a stage. So, this is this these put together constitute a stage of an axial compressor and this is what we shall analyze and see how the performance changes as the operating conditions of this particular stage changes.

So, to understand that better, let us construct the velocity triangles. Now this is the inlet velocity triangle for the rotor. The air is entering at an angle of C1 absolute velocity, relative velocity is V1 which enters this blade at tangential direction. Blade rotates at a peripheral speed of U. This is the velocity triangle at the inlet of the rotor and this is the velocity triangle at the exit of the rotor. Here velocity leaves the rotor at an a velocity of V2 which is again tangential to rotor, C2 is the absolute velocity leaving the rotor, U is the blade speed and C2 is going to be the velocity with which the flow enters the stator So, the velocity leaving the stator is C3. There is no relative component for the stator because the stator is stationary. Relative component is true only for rotating components like a rotor and that is why we have relative velocities at the inlet and exit of the rotor.

So, this a typical axial compressor stage comprising of rotor and a stator and we will see how we can estimate the performance or what are the parameters on which the performance of this single stage axial compressor would depend upon and for which, we will need to analyze the velocity triangles a little more carefully. So, if we look at the velocity triangles little closely and also mark the various velocity components on the velocity triangle, we get velocity triangle combination like this.

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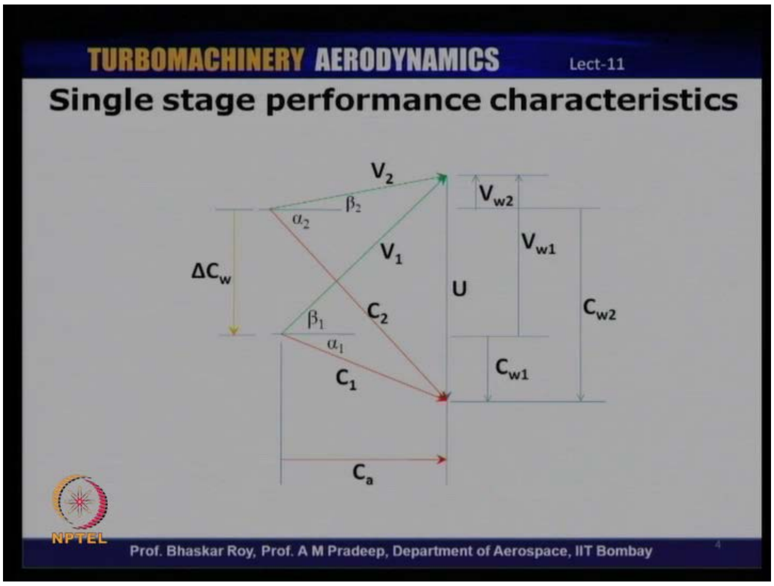

So, this is the velocity triangle at the inlet and exit put together. Since the blade speed is common for both the rotor entry and rotor exit that is the common point here. This is the inlet velocity triangle comprising of C1 V1 and angles alpha1 and beta1. At the exit we have C2 V2 and angles alpha2 and beta2.

The corresponding velocity components are Cw1 which is the tangential component of C1. Cw2 is the tangential component of C2. Vw1 is tangential component of V1. Vw2 is the tangential component of V2. Ca is the axial component of absolute velocity and delta Cw is the difference between Cw2 and Cw1. Delta Cw is important as we have seen in earlier, because the power required to drive the compressor primarily depends upon delta Cw. So, since the power require depends primarily on delta Cw, that is why in the analysis that will the change in tangential component will play significant role.

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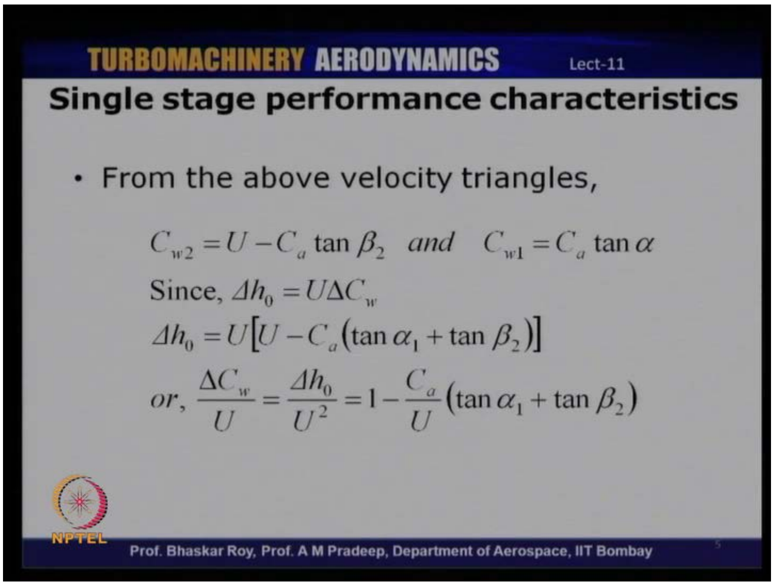

Now, from these velocity triangles, what we can see is the at the exit Cw2 should be equal to U minus Ca tan beta2 and Cw1 is Ca tan alpha1. Let us take a look at the velocity triangle. Cw2 is this component. Cw2 is U minus Ca tan and beta2 and similarly Cw1 which is this component is basically equal to Ca tan alpha1 and since the net change in enthalpy; that is delta h naught is equal to U times delta Cw, we have delta h naught as U times delta Cw which is U minus Ca times tan alpha1 plus tan beta2 or delta Cw divided by U is equal to delta h naught by U square is 1 minus Ca by U into tan alpha1 plus tan beta2.

So, what we see here is that, we get a parameter which is in terms of the delta Cw to the velocity ratio which is a function of certain angles which come from the velocity triangle tan alpha1 and tan beta1; the axial velocity and the peripheral speed. So, these are certain parameters which will kind of influence. This parameter of delta Cw by U, the significance of which is what we will discuss very soon, that how this that this ratio delta Cw by U will play any role in the performance characteristics. So, what we see from this equation that we have now derived is that, the performance of a single stage axial compressor will depend upon a certain set of parameters. One of them of course, becomes is the axial velocity the blade speed and the angles.

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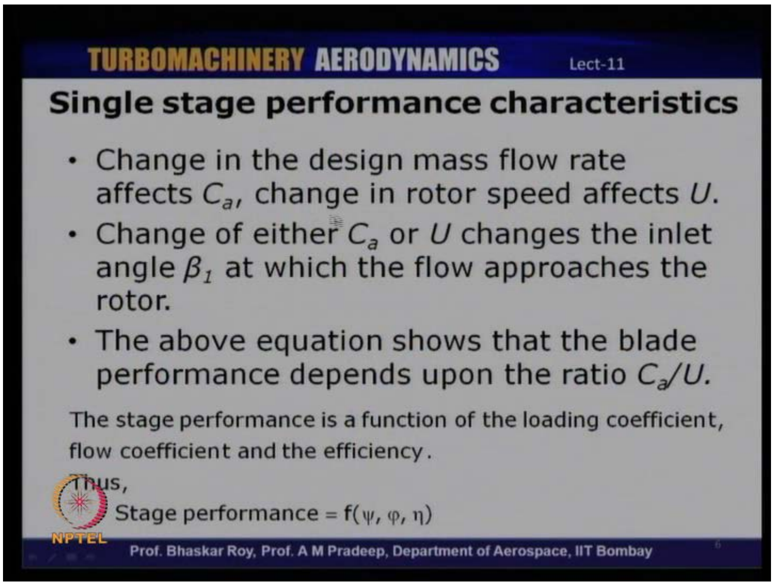

So, this means that any change in the design mass flow rate; obviously, will affect axial velocity Ca and change in rotor speed; obviously, affects U. So, change in either Ca or U will change the inlet angle beta1 because let us go back to the velocity triangle here. So, if either Ca or U changes, it will cause a change in beta1 and; that means, that the blade performance is strongly a function of this ratio Ca by U. That is this ratio of axial velocity to the blade speed will significantly affect the blade compressor performance. And it can be deduced that the stage performance is a function of three parameters; the loading coefficient psi, the flow coefficient phi, and the efficiency. That is, there are three parameters which of course, will also include the flow coefficient which is Ca by U. Besides Ca by U, the performance will depend upon the loading coefficient and also it will depend upon the efficiency of the compressor.

So, there are three parameter significant parameters on which a single stage axial compressor performance will depend; loading coefficient which is denoted by psi, flow coefficient which is Ca by U and the efficiency eta. So, these are parameters on which a single stage performance will depend. Now when you go to a multistage as we will later, on the host of other parameters which will also be playing a significant role in the performance which of course, we will discuss in later slides. Now if I were to plot the loading coefficient which was delta Cw by U with reference to the flow coefficient and also the efficiency with reference to flow coefficient, let us now look at how that affects the performance. Before I go to that, let me also emphasize this particular point that the flow coefficient Ca by U is in some sense, a measure of the mass flow rate. Because mass flow rate is directly a function of the axial velocity Ca and rotational speed of course, being fixed, the mass flow rate is directly proportional or flow coefficient is directly proportional to the mass flow rate.

So, as the compressor is throttled or mass flow rate is changed, it changes the flow coefficient. Similarly for the same mass flow rate, as the speed is changed that also changes the flow coefficient or Ca by U. Similarly the loading coefficient will depend upon how this mass flow rate is changing. So, there is a dependence of the loading coefficient delta Cw by U on the flow coefficient. There is also a dependence of the compressor efficiency on the flow coefficient. So, let us now take a look at how these two parameters change as flow coefficient changes.

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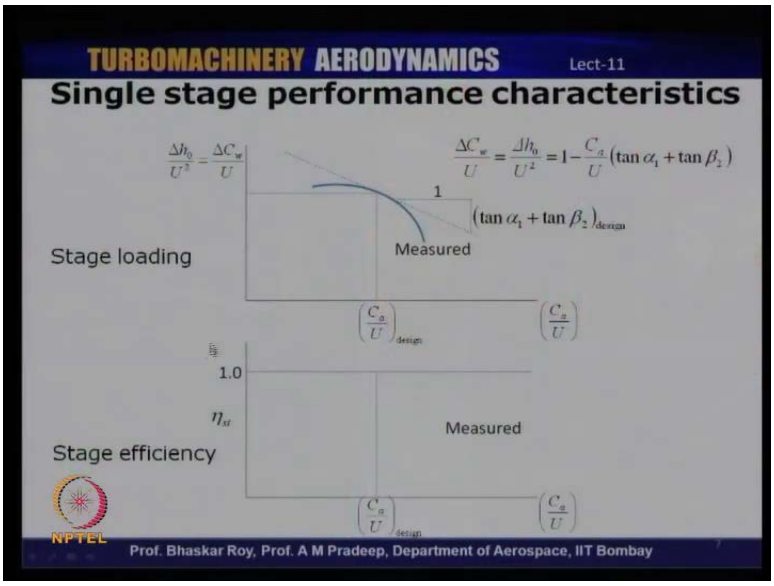

So, if I were to plot, let us take a look at loading coefficient first. So, I have loading stage loading on the y axis which is delta h naught by U square which is also equal to delta Cw by U. On the x axis, we have Ca by U which is the flow coefficient and we have seen this expression, we have derived this delta Cw by U is equal to delta h naught by U square is 1 minus Ca by U into tan alpha1 plus tan beta1.

So, there is direct correlation between the loading coefficient and the flow coefficient. If you were to plot this variation, you should expect it to be a linear line. That is what is given by this dotted line here. One would have expected a linear variation of Cw by delta

Cw by U with the flow coefficient. Of course, in an actual practice what is shown here is given by the solid line that is the measured performance? You can see the measured performance is not necessarily a linear variation. It is equal to the design point at which the compressor has been designed which is Ca by U design at which both the measured and the actual is the ideal are the same. At any other point, you can see that the performance is different from what it has it is supposed to be. And if you were to draw it tangent basically the actual performance, the slope of this line is basically given by this angles; tan alpha1 plus tan beta1 toward the design point.

So, this is basically giving this flow which comes from this equation here. So, what is to be noted here is that, as the flow coefficient changes from the design point, it also affects the loading coefficient correspondingly. So, there is a change in loading coefficient. There is a deviation of the loading coefficient from the design as the mass flow rate or the flow coefficient changes. Now if we similarly look at this stage efficiency, the stage efficiency also has a similar variation with reference to the ideal performance. Stage efficiency would have approached in efficiency of 1 for a Ca by U design and since when it is actually operating, the stage efficiency deviates from the design point and there is a variation in stage efficiency as compared to the design point variation. So, there is also a difference of the stage efficiency or dependence of stage efficiency on the flow coefficient. Similarly there is a strong dependence of the loading coefficient on the flow coefficient.

So, there are three parameters as we have seen which kind of dictates the single stage performance. The first parameter is the stage loading coefficient which also has a dependence on the flow coefficient phi. Then we also have the efficiency which again depends upon phi in some sense that, as phi varies that affects the efficiency as well. So, these are parameters which drastically affect a single stage performance. Let us now look at what happens as we change the mass flow rate or as the ratio Ca by U deviates from the design point. At the blade level, how does it affect the performance, what happens to rotor performance as Ca by U changes from its design point.

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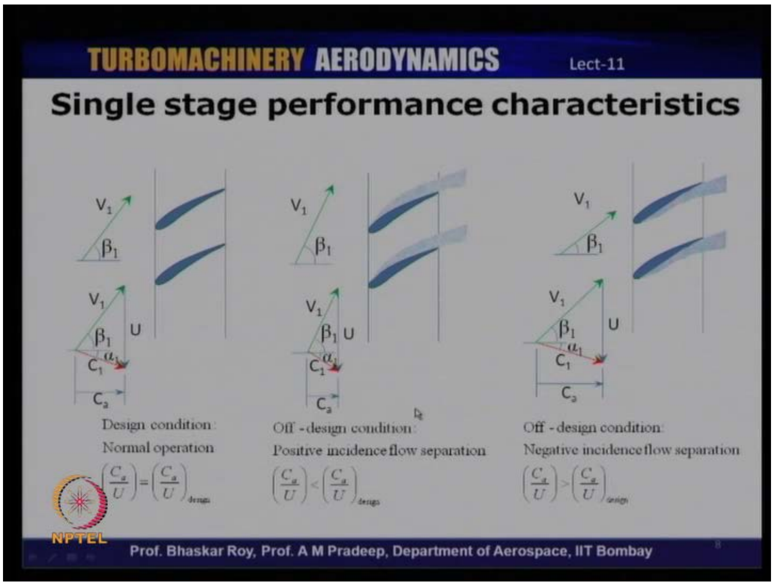

So, for that we have the velocity triangles for three different cases which have been shown. One is a design condition and two off design conditions. The design condition which is normal operation, we have Ca by U is equal to Ca by U design. This is the rotor blade and we have flow entering the rotor at an angle of V1 which is relative velocity at an angle of beta1.

So, this is the velocity triangle at the inlet, V1 at an angle beta1 to axial direction, C1 at an angle of alpha1 and a blade speed of U. So, this is when the flow coefficient is equal to flow coefficient at design condition. Now if let us say, for a constant speed, the mass flow rate is reduced, then this means that this ratio Ca by U also reduces and this necessarily an off design condition. When Ca by U is less than Ca by U design that is axial velocity is now decreases for same U. So, keeping U fixed, if we reduce axial velocity because mass flow rate is reduced, it leads to an increase in beta1 and as beta1 increases beyond a certain angle, it leads to what is known as positive incidence flow separation. So, there could be flow separation taking place on the suction surface of the rotor blade. So, this is a suction surface of the rotor blade. There will flow separation from the suction surface when beta1 is greater than beta1 design which occurs when flow coefficient is less than flow coefficient design. The other counter part of this off design condition is the negative incident separation which will occur when Ca by U exceeds the Ca by U design. That is when Ca is greater than Ca design for constant U, beta1 becomes very low and as beta1 decreases there could be chances of flow separation from the pressure surface of the blade of the rotor blade. You can see that flow as separation from the pressure surface of the rotor blade and this basically a negative incidence flow separation. So, both these cases of flow separation can occur either positive incidence or negative incidence separation when you know, the flow coefficient is different from its design value. When flow coefficient is lower than design value, it leads to what is known as positive incidence separation. When flow coefficient is greater than the design value, it may lead to negative incidence flow separation.

So, this is how the performance of a single stage compressor can vary. You have dependence on three parameters; the loading coefficient, the flow coefficient and the efficiency and as flow coefficient changes, we have seen how it affects the performance of a rotor and there are two extreme cases possible. You may have a positive incidence separation when the flow coefficient is much lower than the design value, leading to flow separation from the suction surface of the rotor and negative incidence separation which occurs when the flow coefficient is greater than the design flow coefficient, leading to flow separation from the pressure surface of the rotor blades and these are two different possibilities wherein the performance of the compressor can be drastically affected. We have also seen how phi versus psi and loading coefficient versus flow coefficient changes and how efficiency changes with the flow coefficient.

Now having understood the single stage performance characteristics, we will now proceed towards a multistage axial compressor and see how a multistage axial compressor performance changes or varies as the operating conditions change or what are the parameters on which a multistage axial compressor performance will depend upon. So, multistage compressor as we know, consist of a series of stages of axial compressor which means you will have a several combinations of rotor stator and if you put all of them together, that constitutes a multistage axial compressor. And what we are going to do is that, we will denote the inlet station of a multistage compressor by station1 and exit of the compressor by station2.

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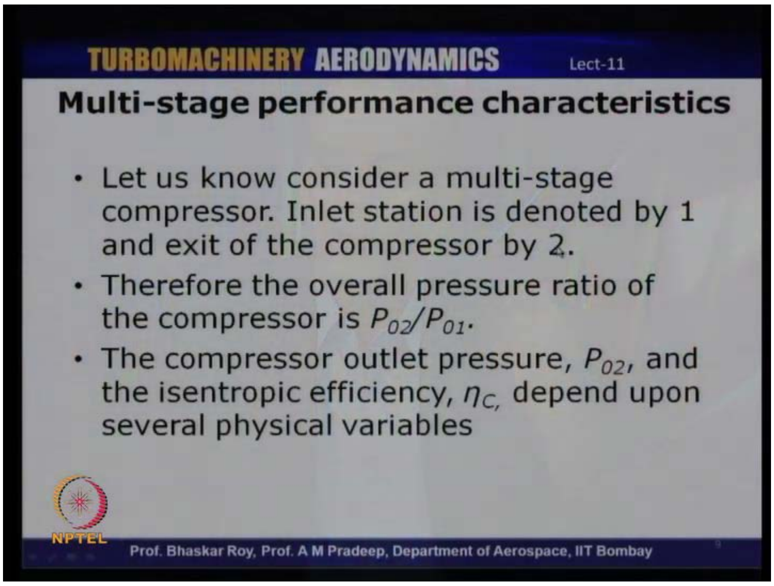

Therefore, the overall pressure ratio of the compressor, we will denote as P 02 by P 01 where P 02 is the compressor outlet pressure and P 01 is the compressor inlet pressure. So, the compressor outlet pressure P 02 and efficiency will depend upon several physical parameters or variables. So, we are going to look at what are these different parameters or variables on which the performance will depend. So, we will list all these different parameters on which the performance of a compressor is going to depend.

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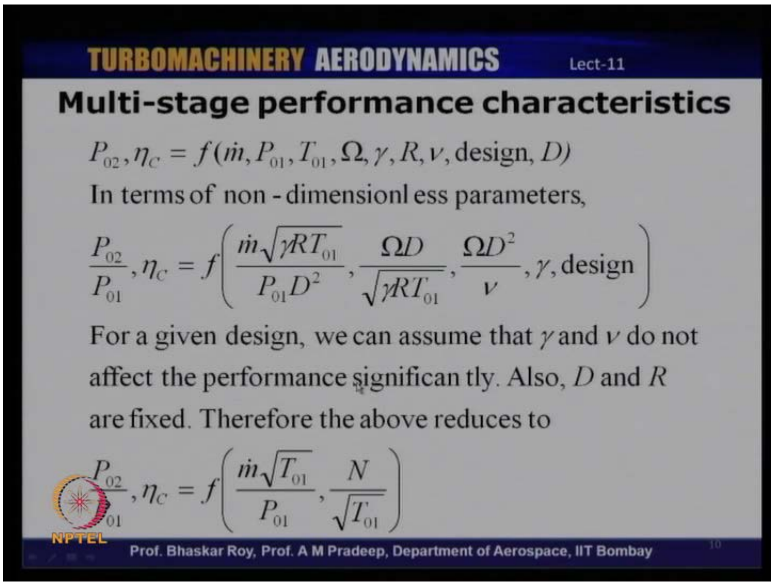

So, let us list all these parameters and what we see is that, P 02 which is exit stagnation pressure and efficiency were is a function of these mini parameters and what are these parameters? We have mass flow rate, inlet stagnation pressure, inlet stagnation temperature, rotational speed omega, the ration of specific caves gamma, the gas constant R, viscosity of the air, then the design itself and the diameter D. So, these are the different parameters on which the pressure ratio will depend. So, now, if we non-dimensionalise these parameters, we can do that using Buckingham pi theorem probably have learnt about phi theorem earlier on. So, if you non-dimensionalise this and express these parameters in terms of non-dimensional clusters, then we have P 02 by P 01 and efficiency; both of which are functions of these mini non-dimensional parameters which is one of them is mass flow rate time square root of gamma R T 01 divided by P 01 D square. Then we have omega D divided by square root of gamma R T 01, then omega D square by new, then gamma and design.

So, these are anyway non-dimensionalise. So, the other non-dimensional parameters are these three. Now for a particular design, we can safely assume that gamma and new do not affect the performance significantly. Similarly D and gas constant are fixed. So, since D is fixed, gas constant R is fixed, design is fixed, new is fixed and the gamma are fixed. These non-dimensional parameters can be simplified and expressed as P 02 by P 01 and efficiency are functions of m dot root T 01 by P 01 and speed as N by root T 01. So, here we have the pressure ratio and efficiency expressed in terms of two distinct parameters. I will still not call them non-dimensional because if you look at the dimensional, if you if you look at the dimensions of these two parameters, they are strictly not non-dimensional. That is because we have taken off other parameters which would have indeed made it non-dimensional. P 02 by P 01 pressure ratio and efficiency are functions of two parameters, one is mass flow rate time square root of T 01 by P 01, the other is speed divided by square root of T 01.

And so; that means, that there is one parameter which is a function of mass flow rate, other parameter which is function of speed. So, pressure ratio and efficiency are functions of mass flow rate and the rotational speed. We will further simplify this and see how the performance changes. We will take a look at how the pressure ratio changes as mass flow rate changes and speed changes. How efficiency changes as mass flow rate and speed changes. But the bottom line is that the performance of axial compressors multistage axial compressor in terms of pressure ratio and efficiency can be expressed as functions of two distinct groups; one is to do with mass flow rate that is m dot root [P 0 1] T 01 by P 01 and the second parameter is N by root T 01 which does not make it non-dimensional because there were other parameters which we have neglected for a given design like the diameter D, gas constant and so on.

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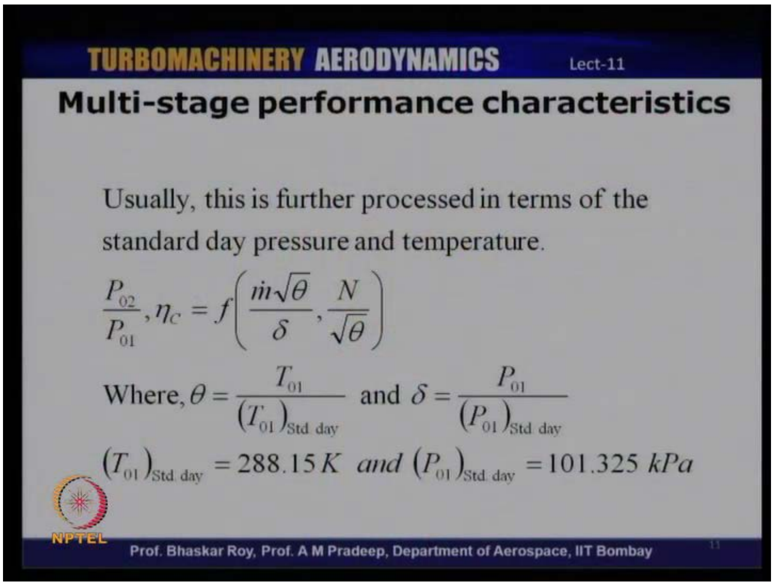

Now, let us simplify this expression further. What is normally done is that the temperature and pressures are expressed in terms of standard day pressure and temperature. We will non-dimensionalise temperatures and pressure with reference to standard day conditions. Therefore, P 02 by P 01 and efficiency can be expressed as functions of m dot root theta by delta and N by root theta. Here theta should be equal to T 01 by T 01 standard day and delta is stagnation pressure divided by stagnation pressure of a standard day. So, here theta and delta refer to temperature and pressure ratios for a standard day. I will explain the significance of why this non-dimensionalization is also is required for temperature and pressure. Basically because when we are designing a compressor for a particular condition, one normally designs it for a certain ambient condition, but this compressor may be operating in an engine which is used in an ambient condition which is entirely different from what it has been designed for.

So, what is the guarantee that this compressor is going to perform the same way as it has been designed for a different condition?

So, the way to account for this is to express the pressure and temperature in a non-dimensionalised form and so usually this is also referred to as corrected pressure and corrected mass flow rate and corrected speeds because mass flow rate has been corrected for the standard day pressure and temperature. So, even if let say the engine is operating at a temperature and pressure which is drastically different from what it has been designed for, because of this correction it can partly take care of this variation in pressure and temperature. So, with this background that we had so far on how we can express the performance of multistage compressors, we will now proceed towards looking at how the variation of a compressor performance itself is expressed.

So, basically a compressor performance variation is also referred to as performance map. So, compressor performance is expressed in the form of a map and performance map forms a significant part of a very significant role in the design of an aero engine or for that matter any gas turbine engine because as I mentioned, a compressor performance puts limits on the whole engine performance as a whole because they certain limits of operation for a compressor beyond which it cannot operate or there are instabilities which are introduced.

Therefore an engine will not be able to operate satisfactorily beyond these operating ranges and that is why a compressor map forms a very significant input for an engine designer who will need to know, what are these limits between which this compressor is going to operate. That is if I am designing an engine and I will need to know that this compressor has certain limits in terms of mass flow rate and pressure ratio. So, this is amount of limit that I have during which between which the engine has to operate because beyond that the compressor cannot operate and therefore, the engine as whole will not operate at all. And therefore, compressor performance map will form a significant input for an engine designer who will need to know what these limits are.

So, compressor performance is expressed in terms of two parameters; as a function of the mass flow rate. So, one the parameters which we are interested is what is a pressure ratio developed by the compressor. So, the pressure ratio P 02 by P 01 expressed as a function of mass flow rate m dot root theta by delta. And similarly what is this variation with different speeds. As speed changes that N by root delta or N by root theta, how the performance changes. This is one of the parameters which we are interested in. The other parameter is variation of efficiency with the non-dimensional mass flow rate and the non- dimensional speed how the efficiency changes.

So, I will now show you one typical compressor map which basically tells us how this variation can be tracked. Now if you look at what is shown here, I have a typical compressor map which is shown here.

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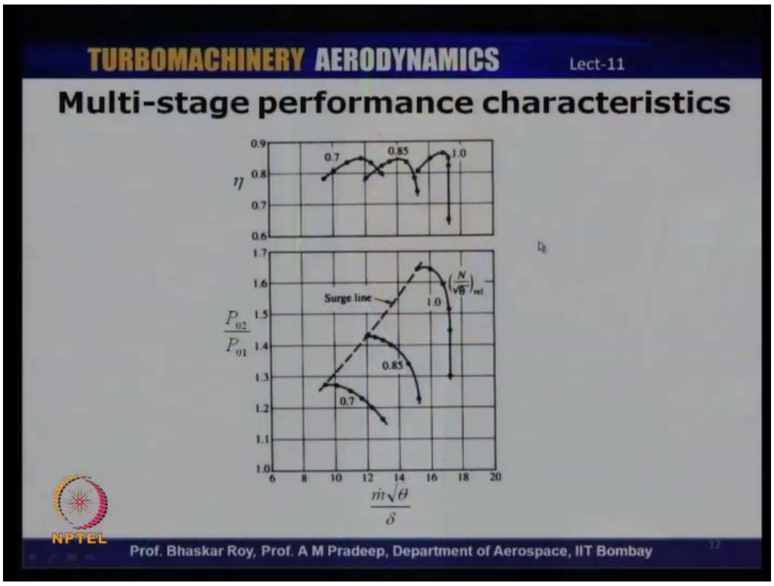

On the y axis, we have P 02 by P 01 and efficiency x axis we have mass flow rate which is expressed in terms of m dot root theta by delta where this is the temperature standard day ratio and this is the pressure ratio standard day with reference to standard day and they are series of course, which you will see and each of these lines correspond to constant speed line and the speeds are expressed in terms of N by root theta and that is, as the speed changes, as you change the speed of the compressor from very low speed where this ratio will be close to 0 and as you keep increasing that and it approaches the actual speed the design speed which is 1.

You can see that initially the speed lines are flatter. As you look at compressor performance at lower speeds very low speeds, one would see a flat variation rather flat variation of the pressure ratio with mass flow rate. That is as mass flow rate changes pressure ratio also changes, but that is over a wide range. But as the speed increase as we go towards higher and higher speeds, the speed curves become sharper and sharper. For example, if you look at this speed curve at N by root theta is equal to 1, you can see that speed curve is extremely sharp. And therefore, the pressure ratio varies drastically with mass flow rate, but that is over a very narrow range of mass flow rate, beyond which there is certain line which is shown here a dotted line of course, there is curve beyond this also, but that is a curve which a designer would never want his engine to operate on.

I will explain what the curve is a little later. Now I what you will notice is that, as the speed increase from very lower speeds and as we proceed towards higher and higher speeds, the performance curve which was initial flat starts becoming sharper and sharper to the extent that at very high speeds, that is the design speed of 1, the curve becomes very sharp; which means that there is a very narrow range of operation of the compressor here. And the sharp curve we are basically means that mass flow rate does not really influence or for the pressure ratio versus mass flow rate is kind of a constant here and this basically refers to what is known as its choking point, where you are trying to pass the maximum mass flow rate which this compressor can generate.

So, beyond which mass flow rate does not change much and there is a significant drop in efficiency which I will come to little later. So, what does line here means is that, under this operating condition, even if we change the mass flow rate substantially, there is a the variation in pressure ratio is very drastic for a very narrow change in mass flow rate, beyond which the pressure ratio drops as you try to operate the compressor for mass flow rate beyond that because mass flow rate is fixed. You might recall the concept of choking which you would have learned in your gas dynamics solid fluid mechanics that under certain operating conditions, mass flow rate attains a peak level and mass flow rate becomes maximum.

This is exactly in the case that is happening here, that maximum mass flow rate has taken place and no further mass flow rate can be passed through by this compressor and if you try to pass more and more mass flow, what will happen is that it will effect two parameters; one is the pressure ratio which drops and also the efficiency which drops drastically.

So, the curves which are shown here are the efficiency curves for the corresponding speeds. So, for lower speed as expected, one would see a flatter efficiency curve and as the speeds approach 1, the efficiency curves also becomes sharp just like the pressure ratio curves and you can see that the change in efficiency is very drastic and very narrow as the speeds approach the design speed. Now on this each design line or in each speed line, you can see multiple points which have been shown here. These are the different operating points of the compressors.

So, compressor may be operating on any of these points or in between these points. So, when a designer embarks upon designing a compressor, what he does is that he tries to design it for a particular operating condition and then one would also need to evaluate what happens when the compressor is going to operate in conditions which are different from what it has been designed for or what are known as off design conditions. For an aero engine for example, the design the operating conditions can vary so drastically that the designer has to ensure that even if the off design condition is it its extreme, the compressor still operates safely. Because as you have seen here in the curve, on the left hand side there is a dotted line which has been shown on the pressure ratio versus mass flow rate. There is dotted line which is shown and it is indicated as surge line. Now what is surge is something we will discuss in detail in a later lecture, but let me tell you that the compressor operation is affected drastically affected by two instabilities which are likely to occur. One is known as rotating stall and the other is known as surge.

Both of these are instabilities which can drastically affect the performance of the compressor to the extent that if surge occurs, the compressor may fail and lead to flame blow out in the commercial chamber and the engine may shut down if the compressor is surging.

So, the left, the dotted line shown on the left hand side of the pressure ratio versus mass flow rate curve is sort of a limit for operation of the compressor. That is, though if you extend that line, you will still see a line on the left hand side, but that is a line on which you just cannot operate the compressor because that is a region of instability for the compressor. The compressor cannot operate in a stable manner if it is on the left hand side of what is known as the surge line.

So, if compressor tries to operate on the left hand side, it would undergo what is known as surge during which the entire operation of the compressor breaks down and engine as a whole gets affected drastically and it might lead to failure of the compressor and engine shut down. So, surge is the phenomenon which can affect the performance of a compressor drastically. So, let me take a closer take you to a closer look of what is the surge line and how it affects the performance.

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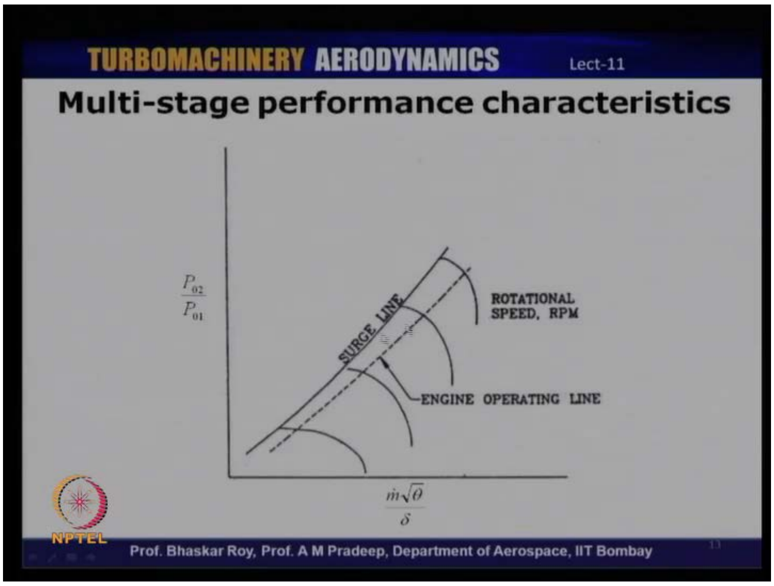

Now the same performance curve that you saw here, pressure ratio versus mass flow rate is being shown in a better way here. Now here we have the speed lines which are these different lines which are shown. The surge line is shown here and what is also shown is the engine operating line. Now engine operating line is the line which is something that the designer would like to use and how does one at arrive at the operating line? Operating line is the line through which the engine is accelerated from 0 speed all the way to design speed. Operating line would ideally have to be a line where the efficiency is maximum because you would always like to operate the compressor in a condition where it as maximum efficiency. If you join all those points ideally, join all the points maximum efficiency, you can get the ideal operating line, but it may so happen that many a times the maximum efficiency is very close to the surge line which is a risky affair because if you are operating very close to the surge line, any off design operation might push your engine into surge. That is a risk which the designers are not willing to take, will not absolutely be willing to take because that is too higher risk to be taken to operate the engine very close to this surge line.

So, the engine designer always wants to a keep a certain margin between the operating line and the surge line. This is known as the surge margin. Surge margin is a certain margin or buffer which the designer wants to put for the engine to ensure that even if the engine undergoes an off design operation, a transient operation, the engine still does not touch the surge line. Because if the engine were to indeed were towards the surge and touch the surge line, that can lead to catastrophic effects which are something which the designer will always want to avoid.

So, surge margin is something which is kind of a protection for the engine provided by the by the designer to ensure that the engine does not reach the surge condition even if there is an off design operation of the engine. And most of the modern day engines have inbuilt surge warning systems and mechanism which will prevent a pilot from accidently operating the engine in a such way it can surge.

So, the modern day computer which operates an engine which is also known as the FADEC that is the full authority digital engine control, basically has inbuilt functions which will prevent a pilot from making such mistakes that will lead to surge of an engine. And even if there is a possibility, the surge warning sensors which will give a warning to the pilot saying that there is a possibility that the engine can surge if you operate it so and so.

So, in this operating multistage compressor performance map, there are two distinct lines that we should be a familiar with; one is of course, the surge line, the other is the operating line. Operating line is the line on which the engine is designed to be operating for.

There is also another line which may not be that significant. That is on the right hand side right most side. You may also join all those points on the right hand side to achieve what is known as the choking line. Choking is not really significant for a compressor, but it may be significant for a turbine which we will see later on because turbines usually operate under choked condition. So, that we will discuss a little later when we take up the turbines. For a compressor, the choking line is not really a matter of that concern, but they could still be a line which represents choking in axial compressor.

So, surge line and operating line; two distinct parameters or lines that we need to be aware of. Now based on these parameters that we have discussed, we will now look at how the performance changes as let us say mass flow rate changes which we have also seen for a single stage compressor. We have seen that as the flow coefficient changes from the design condition, it drastically effects the performance of a stage. Let us also look at how flow coefficient changes can affect the performance of a multistage axial compressor.

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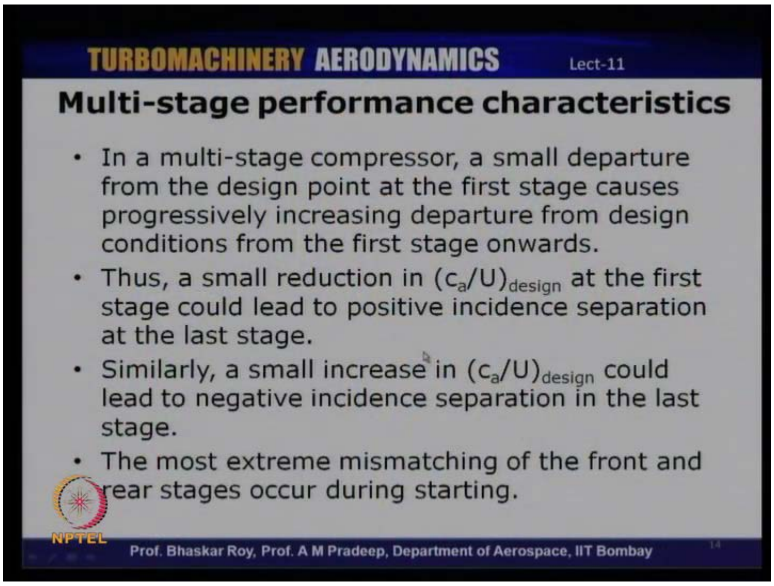

Now, in multistage axial compressor, the problem is that a small departure from the design point in the first stage can cause progressively increasing departure from the design from the first stage onwards. That is, a small reduction in the ratio Ca by U at the design point at the first stage could lead to positive incidence separation at the last stage. Similarly a small increase in Ca by U design could lead to negative incidence separation in the last stage. And the most extreme mismatching of these front and rear stages occurs during starting. Now during starting what happens is that the compressor has just about began rotating.

So, as that rotation begins, the net change in density across the compressor is not very significant. So, change in density from the first stage to the last stage is insignificant. So, there is hardly any change in the densities. So, what happens as a result of that is that as the density changes are not very high, whatever changes occurs at the inlet can have a very significant effect on what is happening at the exist.

So, there is a very significant effect of the flow from the inlet all the way to the exit and this is especially through during starting when the density development has not really taken place. Its the pressure ratio and the density changes have not really been initiated because the compressor has just about the started.

So, we will take a look at what happens during starting what happens to the flow conditions or velocity triangles for the inlet or the first stages and what happens to the flow velocity triangles at the exist or the last stages.

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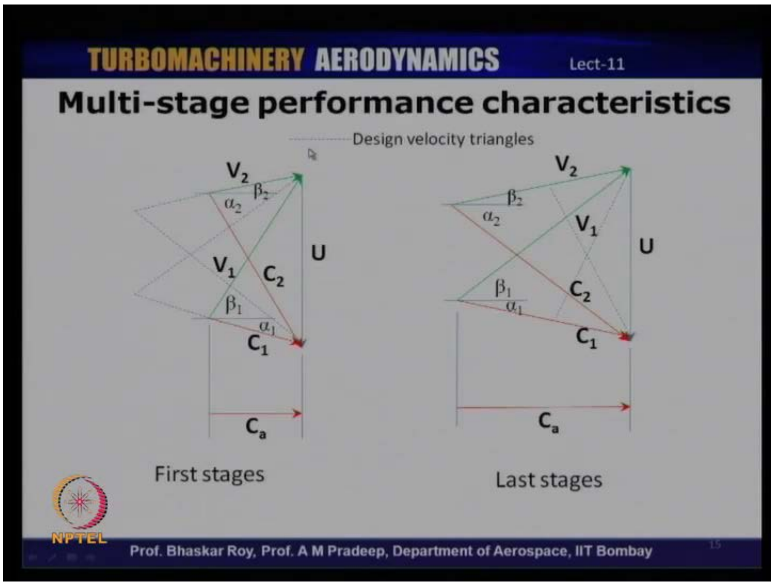

So, I have two sets of velocity triangle here; one corresponding to the first stages and one set corresponding to the last stages. What is shown here by the dotted line correspond to the design velocity triangle.

So, these dotted lines are supposed to be the so called design velocity triangle and what happens is that, in the first stages, axial velocity to the ratio of Ca by U is lower than what it should be for design condition. And what happens in the last stages is that Ca by U ratio is higher than what it is for the design condition because mass flow rate is fixed. Now mass flow rate being fixed, the area is fixed and the axial velocity is only parameter which can change because density is also fixed during starting.

So, from the inlet to the exit, if you have seen an axial compressor geometry, the area progressively reduces. From the inlet at the inlet you have large area and that area progressively reduces and you have a smaller and smaller area at the exit. Since the area is reducing, density is fixed for a constant mass flow, axial velocity has to increase. That is what is happening during starting. Now once it starts and the compressor operates, the density increases and this problem does not take place once the compressor has is fully operational.

So, this is basically a problem which occurs during transient operation during starting of the compressor. So, what happens in this case is that there are few stages in the beginning where Ca by U ratio is lower than the design ratio which means in first few stages may encounter positive incidence separation and towards the last stages Ca by U is greater than Ca by U design and it may lead to negative incidence separation in the last stage. There is a huge mismatch between the initial stages and the later stages and that happens just during starting.

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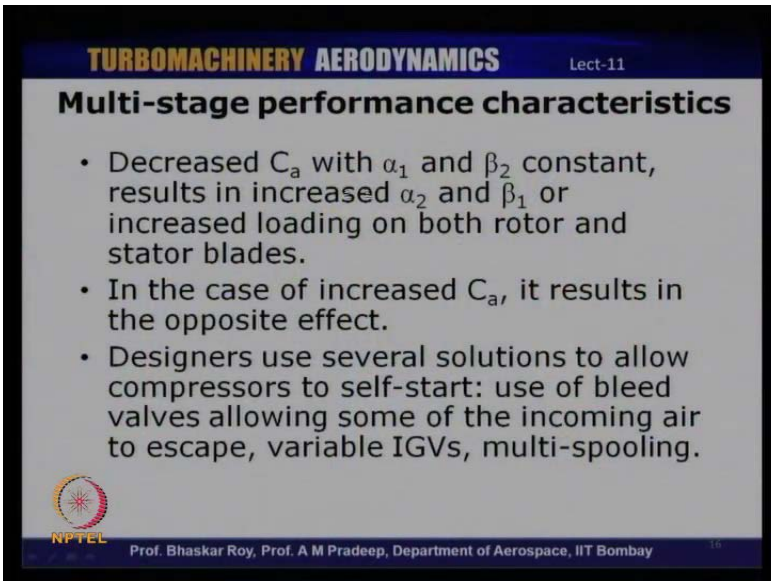

So, how can we overcome this problem? So, there are different ways of overcoming this problem and what basically happens is that. A decreased Ca with alpha1 and beta2 constant results in increased alpha2 and beta1 and similarly it leads to increase in loading in both rotor and stator. Similarly case of increased Ca, it leads to an opposite defect. So, there are different ways in which designers have configured for self starting of compressor. One of the ways is to use bleed valves which will allow some of the incoming air to escape.

So, this is the most common way one of the common ways of elevating this problem, that is use of the blade valve some way mid way between the multi stage compressor that will allow some of the mass flow rate to escape. So, as the mass flow rate escapes, later stages will have lower mass flow and therefore, it will have Ca by U ratios which are closer to the design value and therefore, the problem of a extreme mismatch between the initial stage and the later stage does not really happen. Other way of course, is to use variable guide mains which can change the inlet angles, flow angles to ensure that the flow is matched to the design Ca by U values and the third option is of course, to use multi spooling which is probably the most common thing which is used now, that is you spit the compressor into different stages so that the later stages of operates at different speeds as compared the initial stages which is why most of the modern day engines have multiple spools. It could be twin spool or a three spool engines and therefore, you have initial fan followed by low pressure compressor and then a high pressure compressor which are driven corresponding by different stages of turbines.

So, these are different ways of trying to ensure that the compressor operation specially during starting is also taken care of and there are no extreme mismatch is taking place between the first few stages and the last stages of multistage axial compressor.

Now, besides this, I mentioned that there are two distinct problems or instabilities which can affect compressor performance. One is known as rotating stall as I mentioned and the other is known as surge.

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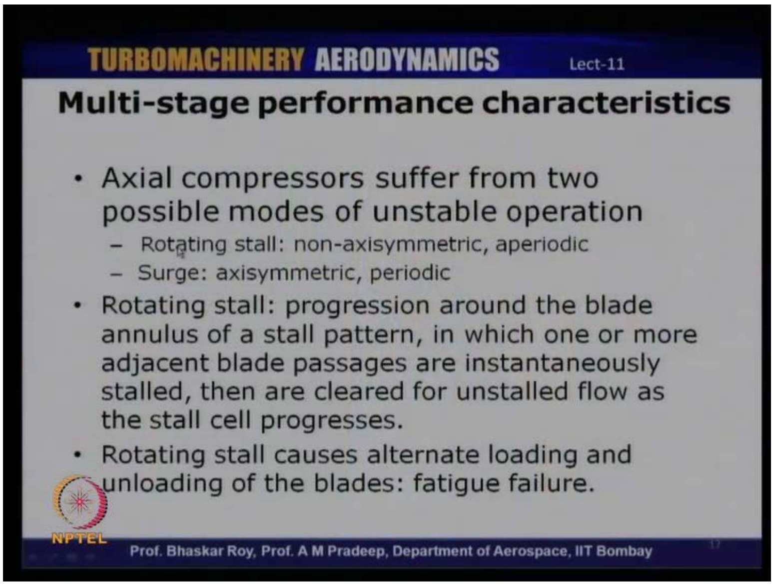

Now we will discuss in detail about rotating stall and surge in a later lecture. Let me just give you a just quick introduction to what is mean by rotating stall and surge.

Rotating stall is basically a non axisymmetric phenomenon and it is aperiodic. So, it is not a periodic phenomenon. Surge on the other hand is axisymmetric which means it affects the entire annulus of the compressor and it is periodic. Rotating stall is basically a progression around the blade annulus of a stall pattern in which one or more adjacent blade passages are instantaneously stalled then cleared for unstalled flow as a stalled progress. Rotating stall obsessively causes alternate loading and unloading of blades may be leading to a fatigue failure, but it is not fatal. Surge on the other hand is a low frequency oxidation of the entire annular flow and it is periodic, but it leads to it is a kind of fetal phenomenon for the engine because onset of surge can almost always lead to engine failure and therefore, surge is much more severe phenomenon as compared to rotating stall.

Now, let me just go to the pressure ratio versus mass flow characteristic to explain this little bit more. Now in this pressure ratio versus mass flow characteristics, on the left hand side that you see here is what is meant by surge. When though theoretical there is a curve which extents, if you draw this curve on the left hand side you still extend their way, but a practical limitation in terms of initiation of surge will prevent compressor from operating on this side of the curve and that is why this is known as the surge line and it is important that they clearly demark it between the surge line and the engine operating line.

So, that is why there is a clear difference between engine operating line and the surge. Rotating stall is often considered to be a precursor to surge, that is as the engine approaches the surge line, there is a possibility that rotating stall initiates and rotating stall then allow to proceed further and propagate further, can evolve into surge and the engine can eventually lead to surge, but rotating stall is something which can be prevented and controlled, but surge once initiated is very difficult to control and that is that is why designer would also always want to avoid that the engine even approaches surge.

So, surge is something that we will need to be avoided and under all circumstances and that is why it is important that the designer understands the surge margin very well. We will discuss details of surge and stall, that is instabilities in detail in one of the later lectures.

So, let me conclude today’s lecture where we have discussed about very important aspect of performance characteristics of axial compressors. We began our lecture today with discussion on the performance of single stage characteristics and extended that to multi stage characteristics. We have seen that pressure ratio for single stage; its the loading coefficient versus the flow efficient and efficiency versus flow efficient that one would be interested in. In a multi stage characteristics, we have pressure ratio versus mass flow rate non-dimensional mass flow rate as a function of different speeds which is also non-dimensionalized with the temperature and efficiency versus mass flow rate at different speeds. So, these are different parameters based on which one can construct the performance characteristics or performance map of multistage axial flow compressor.

So, these were the topics that we have discussed in today’s lecture. So, I hope you have been able to grasp some of the effects of aspects of performance characteristic and what is the significance of performance map of an axial flow compressor. So, these were the topics which we have discussed in today’s lecture and we will continue discussion on instabilities and in flow conditions and their effect on performance of axial compressor in future lectures.