Turbo machinery Aerodynamics

Prof. Bhaskar Roy
Prof. A M Pradeep
Department of Aerospace Engineering
Indian Institute of Technology, Bombay

Lecture No. # 34

Design of Centrifugal Compressors: Impellers, Vaned/Vaneless diffusers/Volutes

We are talking about centrifugal compressors. You have already done through couple of lectures earlier on various theories of centrifugal compressor. You have done a tutorial on how to solve standard problems of centrifugal compressor and you know basically how centrifugal compressor operate, its fundamental principles and of course, the basic theories that govern the operation of centrifugal compressors.

In today's lecture, we will take a look at how the centrifugal compressors are indeed designed. Now, as you have already learnt, the centrifugal compressors have been around much longer. It actually predates axial flow compressors by long distance and this is because the centrifugal compressor principles are rather simple and have been known to mankind for a long long time. As a result of which, they have been around much longer than axial flow compressors, which as we have seen are little more fragile kind of aerodynamic machines. While a centrifugal compressor is a very robust machine and as a result, aerodynamically, as well as, structurally and because of that it has been around for a very long time.

The principles you have already learned, and we will try to put together all the basic principles that you have done, in the earlier lectures, into a set of design considerations, and lead you towards how to put all the theories together, principles together, into designing centrifugal compressors, albeit, the ones which are more modern ones.

In fact, the centrifugal compressors were sort of put in the back burners, in terms of research and development for many years, when the axial flow compressors came into existence. Of course, the multi staging of axial flow compressors are, especially in gas turbine engines and aero engines, allowed people to raise the overall pressure ratio of the engine, which for a centrifugal compressor was kind of reaching a limit, because more than a two stage of multi staging, the efficiency of the centrifugal compressor as you learned, suffers a lot and that is an unacceptable for modern engines.

Whereas, axial flow compressor does not have that deficiency and because of that the centrifugal compressor were kind of put in the back burner in terms of research and development.

However, for the last ten years or so, the centrifugal compressors have come back into the research arena all over again. A lot of developments are now taking place in centrifugal compressors, because of which specially in last fifteen years and because of which a huge amount of improvement; both in efficiency as well as its working capability has happened in centrifugal compressors. What are the modern design choices that we have? We will discuss in today's lecture.

So, centrifugal compressor is somewhat simpler machine. It is an aerodynamic machine; it uses aerodynamic principles to compress the air. In that sense, it is similar to axial flow compressors, but it does not actually does a positive displacement compressing; compressing which the piston engine does.

So, it is still an aerodynamic machine and a very robust machine. So, let us take a look at what are the design choices and design considerations that go into design of centrifugal compressors, especially those which are meant for modern applications like gas turbine engines and aircraft engines.

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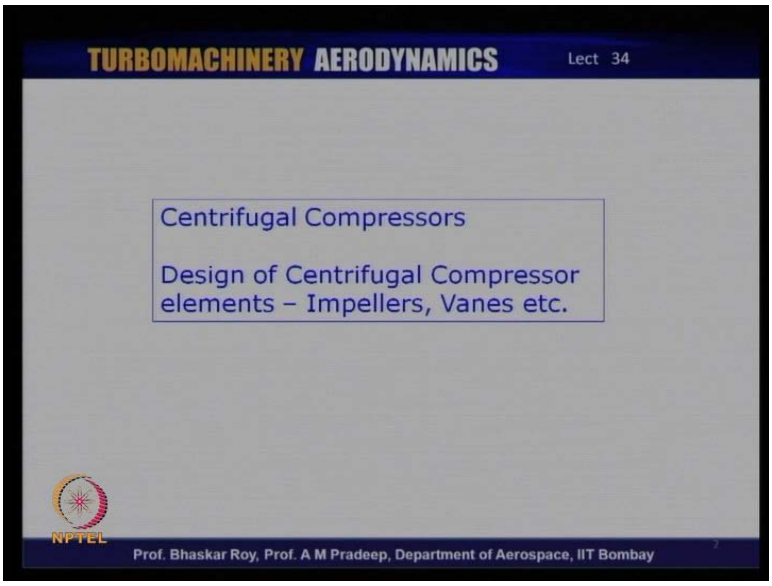

The centrifugal compressors that we are looking at and the principle that you have done actually has a few elements and those elements are the impellers, the diffusing vanes. Sometimes, we have guide vanes in front of the centrifugal compressors.

So, you could have two sets of vanes. One in front of the centrifugal compressor at the entry, and the other at the exit of the centrifugal rotor impeller, which is the diffusing vanes, which converts high kinetic energy to pressure, as you have done in the earlier lectures. This principle can be put together into making of centrifugal compressors.

We would need to look at possible design of not only impeller, but at least one set of vanes and probably two sets of vanes, and then we will look at some of the other design choices that we have in modern centrifugal compressors.

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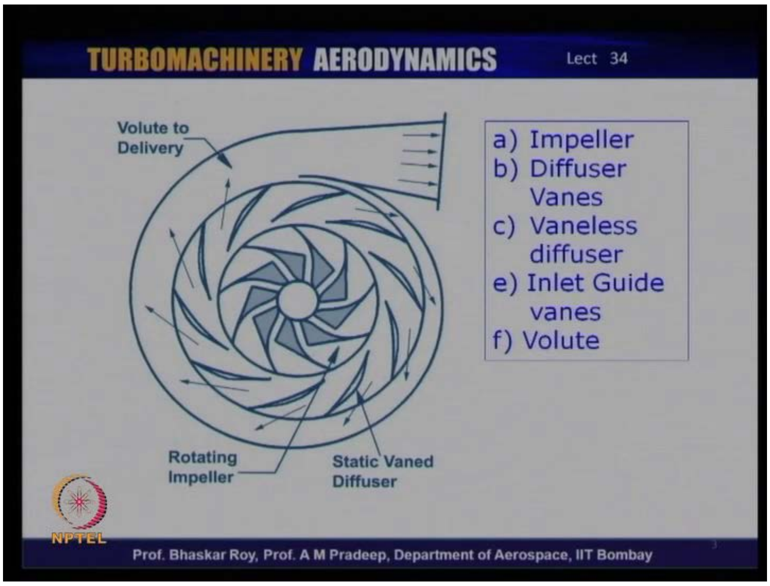

Now, the centrifugal compressors which we are looking at, typically has a rotating impeller, which is normally, called impeller over the years, and of course, the static vanes at the exit to the impeller, and these are two necessary ingredients of a centrifugal compressor.

In the modern centrifugal compressor, the possibilities people are looking at, through research and development is creation of a vaneless diffuser and you have learned that if you have vanes you have a more controlled diffusion. On the other hand, the vanes also create a little problem in terms of narrow operating range, because the vanes are made of aerofoil kind of elements, and the shape of that restricts angle of incidence at which the flow can get into the vanes, and that restriction applies to the diffuser vanes, and as a result of which the mass flow through the machine gets again somewhat limited in its operating range.

By nature of the centrifugal compressor has a much wider operating range compared to that of an axial flow compressor. Its mass flow operating range is much wider and it has normally or naturally, a much larger stall margin, because of which as I mentioned it is a more robust and more reliable machine.

The modern centrifugal compressors, which have gone a very high speed and the impeller tip speed of the flow is gone supersonic. In such cases, the vanes often again restrict the operating range to a narrower mass flow operating range. So, in centrifugal compressor, the range of operation and many of the stall and surge related problems are actually associated with the vanes; the diffuser vanes and not the impellers.

In case of axial flow compressor, we have seen most of the stall and surge are associated with a rotating blades; the rotors. In case of centrifugal compressor, it is mostly associated with the static vanes or the stators. So, the modern thinking is that if you do not have stators, if you have a stator less or vaneless diffusion system at the exit from the impeller, you probably can rid of the entire problem of surge and stall.

Now, this is a thinking of course, if you do not have the vanes, you lose control over the flow guidance and that is another issue that needs to be looked into. This is normally looked into case by case. That means every design has to be analyzed in detail to find out whether you can do without a set of stator vanes. So, that is one of the choices and then of course, you need to make a choice whether you are going to have an inlet guide vane. Normally, inlet guide vane used when you need to provide an inlet swirl into the flow of the inlet of the centrifugal impeller.

Now, this is normally choice to be made at the design stage of the machine and as you have learned from the basic theory that if you put an inlet swirl or pre swirl into the inlet, you actually get a little less work done. So, the work done suffers a little, because you have the c w 1, that comes in, which reduces the work done capability, but it provides a certain amount of comfort to the flow which go in. Sometimes, it avoids the flow going into the impeller becoming supersonic.

Most centrifugal compressors, even to these days, for most normal applications, conventional centrifugal applications, try to avoid flow going supersonic into the inlet of the machine. The flow does go often supersonic at the exit of the impeller, but at the inlet they try to avoid going supersonic because you have shocks attached to the impeller face, which could create problem in terms of a flow separation and would have a lot of losses to bother about.

So, some of those things are avoidable and most people would like to avoid supersonic flow going into the impeller. Of course, there are special applications again, in terms of rocket vehicles or space vehicles where we may have a small turbo pump in, which you may actually have a centrifugal compressor that is completely supersonic. Those possibilities exist and they are used for very special applications, normally for a short duration of operation or single point operation for commercial applications. Especially, in aero engines, quite often people try to avoid shocks right in the face of the impeller inlet, and to do that quite often inlet guide vane is one of the methods by which the shockers avoided at the inlet.

Some of these elements need to be designed into the shape and size of the inlet. The size of course, is decided by the overall engine design criteria, as we have done in case of compressors and turbines. There is another element, which centrifugal compressors often need. The centrifugal compressor, as you know, is exiting the flow radially, and the static vanes or stator vanes or diffuser vanes are also arranged in radial annulus arrangement.

So, the flow is kind of going out in a radially. Now, that flow if it is to be supplied, let us say to combustion chamber, in a gas turbine engine, it has to take a radial turn and then get into the combustion chamber. Now, that requires a change of direction, at least by 90 degrees, sometimes more. Then that requires a guided flow passage. This is sometimes simply called a volute, and this volute also needs to be designed, and typically the losses and other aerodynamic features of the volute are needed to be factored into the centrifugal compressor design, because whatever is being supplied from the centrifugal compressor and before it is delivered to the combustion chamber. For example, everything is considered to be part of the centrifugal compressor. The efficiency of the entire process would have to be factored into the overall machine efficiency and then of course, the overall engine efficiency.

So, volute or some kind of a passage that takes the exit flow from the centrifugal compressor and delivers its somewhere needs to be factored into the centrifugal compressor design. These are the various elements, you have the inlet guide vane, you have the impeller that needs to be designed, and you have the exit vanes that need to be designed. The diffuser vanes very important ones and then in many cases or in most cases some kind of a passage or volute that supplies the flow towards final destination.

So, let us take a look at some of these elements specially the vanes and the impellers how they are factored into the design of centrifugal compressors. Now, as you have done, there is couple of varieties of centrifugal compressor, which you may need to decide or factoring at the beginning of the design. One of them, as you know, is simply called a double sided impeller.

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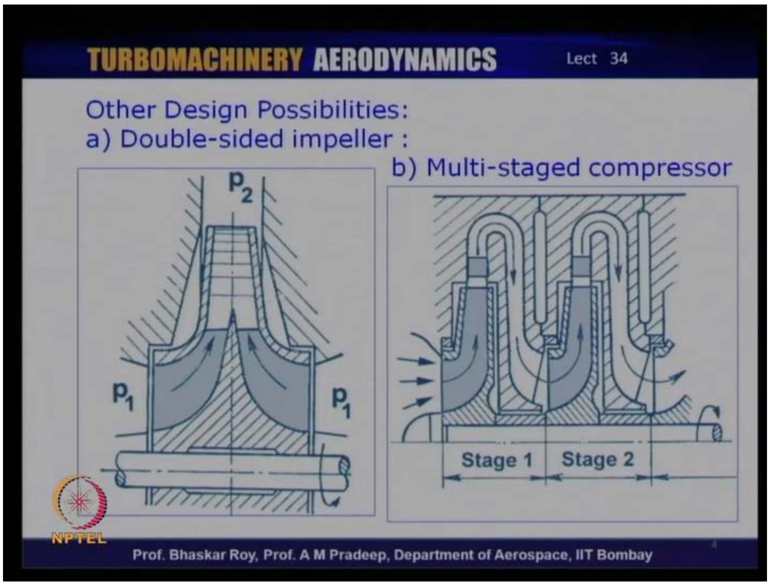

Now, a double-sided impeller is essentially one in which the flow is actually being brought into the centrifugal compressor from two sides. This is essentially done to increase the mass flow through the centrifugal compressor and as the mass flow through centrifugal compressor is somewhat restricted by its inlet, which is smaller than its a outer diameter.

Now, this is compensated by allowing the flow to come from two sides. So, if you allow the flow to come from this side as well as from that side, you are almost doubling the mass flow and then of course, it goes to actually one centrifugal impeller. So, the compression is done over the two mass flows together and then is supplied into the stator vanes and then on to the delivery passage. So, this is often called simply a double-sided impeller, which is essentially done to increase the mass flow, almost double the mass flow that is possible. This, of course, has to be decided right in the beginning at the design stage. The other kind of impeller in which you can improve the or increase the compression ratio is the multi-staging.

The multi-staging, essentially as you can see, in this picture, essentially means that the flow is delivered through the impeller, through the stator vanes, then as I was mentioning, a passage or some kind of a volute, and then through this, it is delivered on to the next stage.

So, it takes 100 and 80 degree turn, through this passage and then it has to be a well designed passage and then it gets into the next centrifugal stage and then goes to the impeller and then the stator vanes and then goes into the next delivery passage, which could be another volute or in some very special cases there are centrifugal compressors, which going to 3 or 4 or 5 stages. These are land based applications; very special kind of applications, where a lot of pressure ratio is required for a comparatively small amount of fluid flow, and in such cases multi-staging is done in this vane.

So that requires a design of these passages, which are indeed very important, because as I said, the losses, fluid mechanic losses through these passages need to be factored into the design of the centrifugal machine, so that the overall efficiency of the machine would be then decided after these losses are factored in. And of course, as you have already learned, the losses through these passages often are of very high order, so that the overall efficiency of centrifugal compressor is often somewhat on the lower side, compared to that of a typical axial flow compressor.

So, these are the fundamentally different kind of centrifugal compressors that you may need to decide upon, before you initiate the design process, as to which kind of centrifugal compressor you are designing or embarking on your design.

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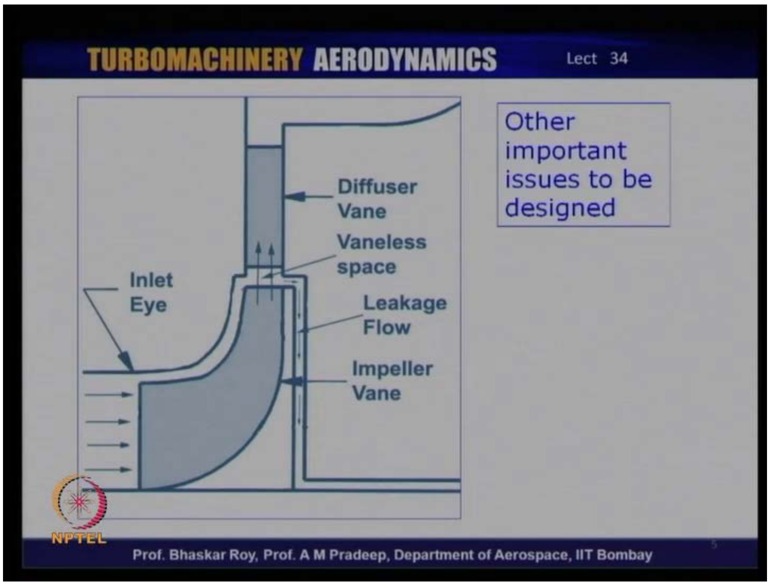

Now, some of the details that you would need to decide when creating a centrifugal compressor. As we mentioned, there is a inlet eye, through which the flow comes in. It gets into the centrifugal volute and centrifugal impeller, which has the shape something like this. Typically, if we consider conventional centrifugal compressor and this allows the flow to kind of get in and go out radially. Typically, there will be a small gap to allow the rotor to rotate, because the casing on the overall covering is a stationary element. Then the rotor has some kind of a back plate on which the vanes are indeed designed. This back plate also rotates with the vanes. So, to allow the back plate to rotate there is a small gap over here from the machine body. So, all around the impeller there has to be small gap.

One has to decide, how much this gap should be, because the gap actually promotes or you know facilitates leakage a flow and the leakage a flow is the loss to the machine in terms of the compressing of work that needs to be done. The other thing that would have to be decided is the vaneless space that inevitably is provided between the impeller and the defusing vane.

Now, the defusing vane actually converts kinetic energy to pressure and as you have learnt that it is a hugely important element in a centrifugal compressor design. This vane less spaces typically, one is you have to have a vaneless space. The rotor needs to be separated from the stator. On the other hand, we will see, as you have seen a little, and we will see again today, these vaneless spaces often used for very intelligently to control the flow that is going from a rotor to the stator. This is to reduce the losses on one hand and also, sometimes to ensure that the flow is guided from rotor to stator in a manner that avoids all kinds of stall, surge and even a supersonic flow related problems.

So, some of these geometric parameter that is the little bit of gap that needs to be left all around the impeller and a very importantly the vaneless space, between the rotor and the stator, needs to be decided by design, so that the flow during its operation, is always under some amount of control and avoids losses and a various other disturbing or instability related issues.

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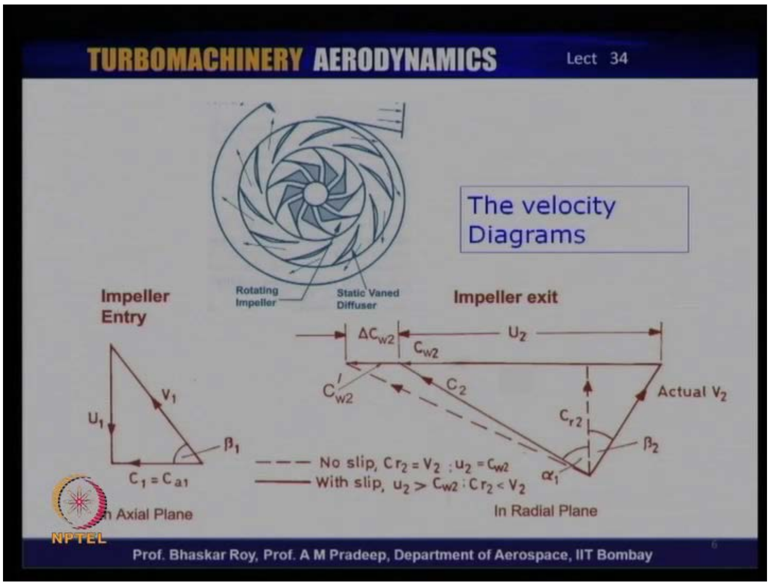

Now, let us take a look at something that you have done in great detail is the velocity diagrams that typically are operative at the entry and exit of the impeller; impeller, being of course, the most important element that needs to be designed. Now, at the impeller entry, as you have done the flow often is expected or allowed to come in axially.

So, the axial flow comes in and then there is a rotation of the inlet rotor. Let us say, its medium velocity U 1 is somewhere at the mid radius of this inlet eye or for that matter at the mid radius over here (Refer Slide Time: 20:50) because you would be varying from here to here, from root of the eye to the tip of the eye. So, design starts at the mean of the eye, and at the mean let us construct the velocity triangle, and we have an axial velocity and then we have the rotational speed, blade speed and the resultant of the two is the relative velocity with which the flow indeed is getting into this impeller high inlet.

As a result of this, the impeller eye needs to be curved; the eye inlet face needs to be curve to meet this velocity at this angle, which is beta one. This angle as you can now very well see would change from root to the tip of the eye because u would be changing from root to the tip of the eye. As a result, beta 1 would be changing from root to the tip of the eye and hence both beta 1 and V 1 would change from root to the tip of the eye. And this is where in case of modern centrifugal compressors rotating at very high speeds; it is entirely possible that V 1 actually indeed could go supersonic.

Now, by design, the designers try to avoid the supersonic flow, going into the inlet to the centrifugal impeller. When V 1, at the eye tip, so V 1 eye tip, if it goes supersonic, one may consider putting an inlet guide vane there to turn the flow away. That means the axial flow is turned away this way so that the velocity V 1 is reduced from supersonic to subsonic. This is a simple trick that people have been doing for quite some time. That means, if you have a very high speed rotor and there is a possibility the calculation shows that V 1 could go supersonic, the inlet guide could turn the flow the other way so that the V 1 could again become subsonic.

At the impeller exit, as you have done, the ideal in our velocity diagram is the right angle triangle, which is given by C w 2 bar as C w 2 prime and which is the right angle triangle and C r 2 of course, is the exit velocity from the impeller. Now, in reality, most of the time the flow does not go out quite radially even of a radial vane impeller. It has an tendency to go out at some angle and this angle is sometimes you know at some angle beta 2, which is to be decided through some analysis. As you know, the slip factor is one parameter, which attempts to capture this deviation from radial exit. As a result of which you have a small difference in delta C w 2 between ideal and real.

This delta C w 2 is nothing but, different between the ideal C w 2, which is equal to and which would have been equal to U 2. In reality, what happens is C w 2 is now only so much and as a result of which U 2 is actually more than C w 2. The ratio is often referred to as slip. How much this deviation away from the radial should be? It is decided by the slip factor to begin with. The choice of the slip factor, prime of a c upfront at the design time is something decided by number of formulae or number of expression that have been created. We will look at some of the expressions and again what those choices are?

Once the machine’s first cut design has been created the geometry has been created, this can be subject to CFD analysis. The CFD analysis would tell us the average flow direction with which the flow is going out from the impeller exit. In the impeller exit over here, may have a flow direction, which is an average flow direction from the impeller exit. That will tell us what kind of angle beta 2 is taking. That can be then factored back into refinement of design after the CFD has given a reasonable prediction.

These are the methods by, which the velocity triangle, which you have done in great detail and solve some problems also is brought back into the design and the design considerations.

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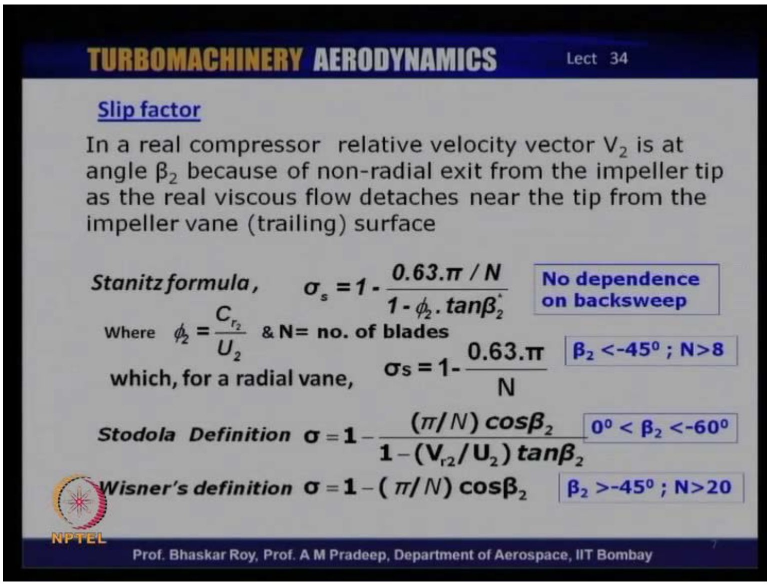

Now, let us take a look at one of the important elements of design that is deciding the slip factor. Slip factor, as you know is essentially is typical of centrifugal compressor; it did not appear in axial flow compressor at all. As you have learnt, the most popular one is decided or quite often by using Stanitz formula, which is expressed in terms of 1 minus 0.63 pi by N and the whole thing divided by 1 minus phi 2 into tan beta 2.

Now, phi is C r 2 by U 2. Here, in this, C r 2 is the velocity with which the flow is indeed going out of the impeller exit (Refer Slide Time: 26:38), and U 2 is the rotating speed at the impeller exit. So, the normalizing parameter is exit blade velocity or vane speed U 2, and that gives us the flow coefficient. N is here the number of blades. Now, this formula has been found to be somewhat independent of various parameters especially the back sweep and as a result the Stanitz formula gives a somewhat steady value of slip factor. Its simpler version, which also is valuable and you have done in the earlier lecture, is simply given by 1 minus 0.63 into pi by N.

Now, this has been found to be especially for radial vanes. It is more effective when the beta 2 is actually less than minus 45, which is the backward swept kind of a blade and the number of blades is definitely more than 8. So, in this kind of situation this simple formula is actually found to be quite valid. However, if the backward sweep is more than this 45 degree or the number of blades in a very simple machine is less than 8, then this formula may not quite be a very useful indication of this slip factor.

It is generally found that the slip factor does vary little with the value of the beta 2, especially the modern compressors are going for backward swept blade rather than the radial vane machines, which have been actually used for more than half a century. But, modern centrifugal compressors are going for backward swept blades, because of various advantages that you already know of. We will have a look at few of them again today. As a result of which, the utility value of the use of this slip factor definitions or the expressions or simple formula where people have designed earlier, for design purposes are need to be relooked at or re-visited.

The other available formula for slip factor is given by Stodola and the Stodolas definition is 1 minus pi by N into cos beta 2 divided by 1 minus V r 2, which is same as C r 2, divided by U 2 into tan beta 2, is similar looking, but slightly different. Now, this is found to be good in terms of backswept blade typically from 0 to minus 60.

The slip factor values that you get changes vary slightly. For example, if it is 0, it could be around 0.9, when it is minus 60, it could be of the order of minus 0.92 or 0.93. So, there is very small change in slip factor that can happen with change of the beta 2 if the beta 2 changes substantially.

Another variation, which has been created by Wisner, which is a variation of Stodolas expression, and that is given as simply as one minus pi by N into cos beta 2, and that is actually found to be valid at high backswept compressor applications, where the number of blades is distinctly more than 20, and beta 2 is more than 45 degree. That is backward sweep so it is minus 45 degree.

Now, we have seen that the forwards swept blades of which we have argued earlier that the forwards swept blades are normally not used in modern centrifugal machines, because of the fact that they have a inherent tendency to be unstable in operation. Hence, typically modern centrifugal compressors do not even consider forward sweep as an alternative design choice. The choices are mainly from radial vane one to backwards swept and more and more designers are going for backward swept blades.

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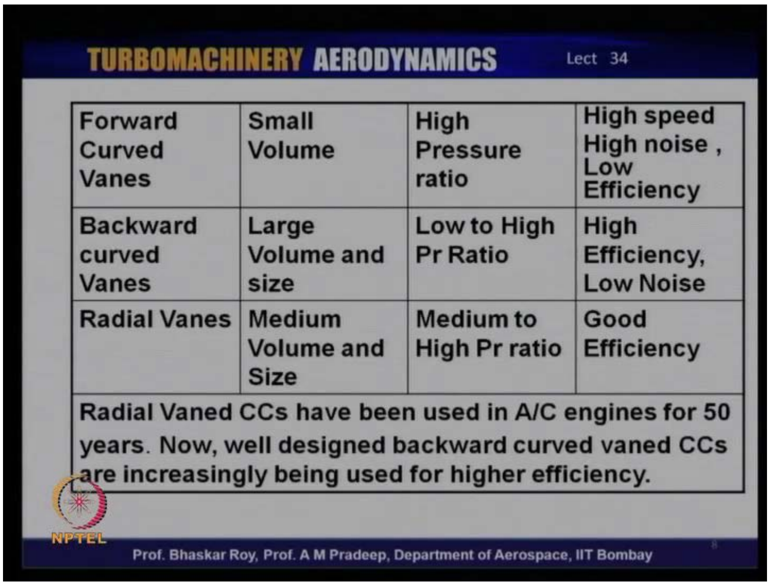

Forward Curved Vanes	Small Volume	High Pressure ratio	High speed High noise , Low Efficiency
Backward curved Vanes	Large Volume and size	Low to High Pr Ratio	High Efficiency, Low Noise
Radial Vanes	Medium Volume and Size	Medium to High Pr ratio	Good Efficiency
Radial Vaned CCs have been used in A/C engines for 50 years. Now, well designed backward curved vaned CCs are increasingly being used for higher efficiency.			

Let us take a quick look at what I was just talking about; the forward swept blades do have to operate with small volume. They can theoretically create very high pressure ratio. They operate at high speed. They create a lot of noise and are very noisy and normally low efficiency, because the high speed exit from the impeller exit creates a lot of losses in the stator vanes, and in the volutes before the high pressure ratio is achieved.

In the process of achieving high pressure ratio, its efficiency actually goes down and in fact, we have seen earlier that it very quickly gets into an instability problem, because the high speed exit from the impeller exit, when it goes into the stator vanes often the incidence is slightly higher. It creates separation and instability in the diffusing vanes. As I mentioned earlier in centrifugal compressor is that diffusing vanes which precipitate stall and surge.

So, forward curve vanes are normally not used in modern centrifugal compressor because of low efficiency and it is tendency towards getting into unstable operation. The backward curve vanes, on the other hand, normally can operate with large mass flow, and can accommodate higher sizes. Originally, it was thought to be of low pressure ratio; however, modern designs can actually increase its pressure ratio to substantially high values. It actually produces a high static pressure at the impeller exit and somewhat low exits velocity that is for example, C r 2 or C 2 going out from the impeller exit would be somewhat lower, and as a result it has a low noise and the static diffuser vanes operate with high efficiency.

So, the overall efficiency of the machine is much higher for backward curve vanes. This is the attraction, which was known earlier, because of which the modern centrifugal compressors are leaning more and more towards backward curve vanes, so that it produces high efficiency machines with low noise. The noise being a very strong regulation requirement these days in industry, as well as in aero engines, because noise is not allowed to go beyond a certain value strictly by regulations.

So, those are the attractions, because of which modern designers are going towards backward curve vanes. Now, radial vane, as I mentioned have been around for a long time and they are the mean between forward and backward. They do not have the instability of the forward curve vanes and they did not have the some of the initial drawbacks of the backward curves.

So, they operate with certain medium mass flow processing capability and the medium size operations; very large size it cannot. It produces reasonably high pressure ratio that have been useful for more than half a century of operations especially in aero engines. Of course, they produce reasonably good efficiency level. They are lower than the axial flow compressors but, still reasonably good and competitive in view of the fact they are robust, aerodynamic, compressing machines. The backward curve vanes, which are be increasingly used higher efficiency as of the attraction and the low noise is the other attraction, because of which, lot of backward curve vanes are being designed these days.

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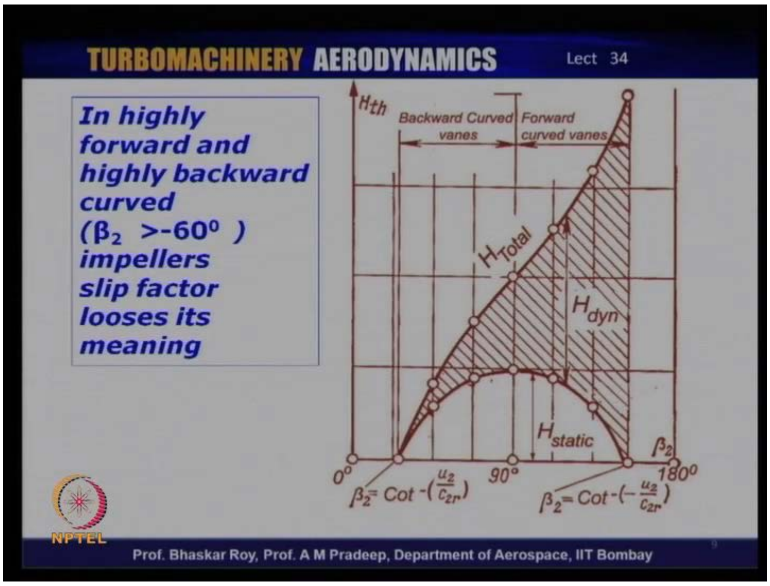

Let us look at some of the issues that are typically important for a forward and backward curve. Highly, a forward and backward curve, the slip factors you know tends to start lose its kind of meaning. Its fundamental definition and if you look at this diagram over here, it tells you what happens in a typical centrifugal machine in a very theoretical ideal sense is that the static enthalpy change sort of varies with beta 2 that is the exit flow angle in this manner.

So, the beta 2 changes from 90 to 0 or from 90 to 100, so this is a forward curve side (Refer Slide Time: 35:51). Right hand side of this line is 90 degree line is the forward curve, on the left hand side is the backward curve. Now, typically in a forward curve, the attraction was that it carries a very high dynamic head or kinetic energy, which if it can be in converted to pressure; we could get very high pressure. As we know from experience that conversion of this high kinetic energy to pressure is often not very efficiently done. As a result of which, the overall efficiency of the compression system actually suffers a lot.

So, forward curves vanes are indeed used in small industrial usages in line based industries, because of the fact that it carries a lot of kinetic energy, which is often useful for throwing the gas or the air or the fluid with a lot of momentum in the process of throwing them out. But, as a compression it suffers from efficiency and sometimes instability.

On the other side, you have the backward curves. Now, the backward curve once as you can see carry less of kinetic energy and most of it is indeed carried in static forms. So the backward curve, typically as I mentioned earlier, has a high static pressure at the impeller exit, compared to many of the other kinds of machines. So, by design you could indeed try to achieve very high static pressure and then a little bit of kinetic energy that it carries can be efficiently converted to pressure.

So, effective pressure that you can get in a backward curve, if properly designed, can actually be a very good, very high and in fact effectively, it can be better than forward or radial curve centrifugal machines and this is the reason. This is an ideal graph, but the logic of this has been taken up by the modern centrifugal compressor designers to create backward curve machines that are very efficient and make very less or comparatively much less noise.

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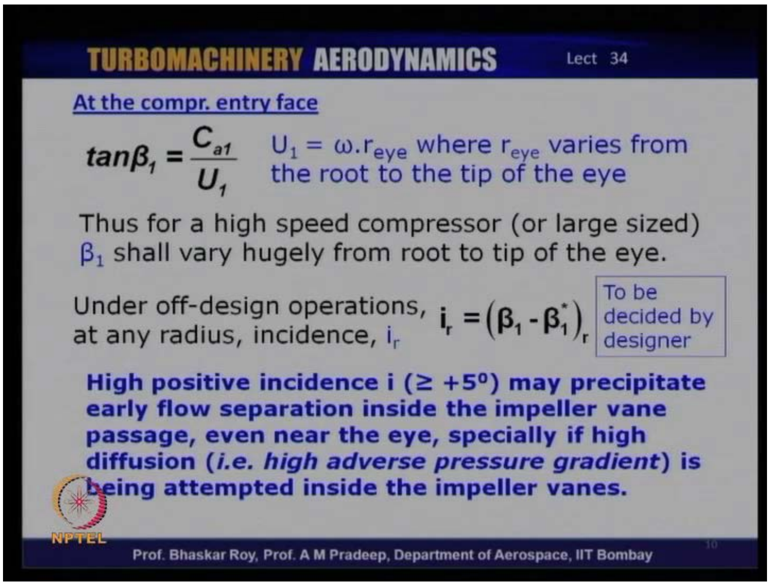

Let us start at the beginning of the centrifugal compressor, where you need to create the inlet flow and the inlet vane shape. The flow is coming in with tan beta 1 which is C a 1 by U 1, as we saw in the velocity diagrams. U 1 is of course, is omega into r of the eye which as we saw varies from root to the tip of the eye.

U 1 would indeed vary, the beta 1 would indeed vary and as a result you get it twisted inlet eye. So, the eye of the impeller is actually fairly, highly, twisted in a high speed compressor or if the compressor is not very high speed, but large in size, which is normally used in industrial applications. So, invariably they tend to be highly twisted and this twist has to be factored into the vane shape of the centrifugal impellers.

Impeller entry is often a twisted vane and then this twisted vane you need to provide as we have seen axial compressor and incidents by design and this design value needs to be factored at the time of design. Now, this incident should not be very high by design. At the point of design it should be very close to as close to 0 as possible, so that beta 1, which is the flow angle and beta 1 star, which is the design blade or vane angle, and the different between the two is the incidence.

This should be as close to 0 as possible. Essentially looking at the off design operation of the centrifugal compressor during which incidence would definitely go other than 0; it could be positive or negative. When it goes to positive or even negative, it could precipitate a separation right at the inlet of the impeller and that could actually impair the efficiency of operation, increase the losses.

High positive incidence, more than five degree, even during off design operation, could precipitate early flow separation, inside the impeller vane passage. It could happen, as early as near the eye of the impeller, especially if the flow has gone transonic or supersonic. So, in conjunction with the shocks and the separation, it could actually create a rather unstable flow condition right at the inlet. This is most avoidable for centrifugal flow operation.

When a centrifugal compressor is designed specially in aero engines for high diffusion; this is to be a designed very properly and very accurately, so that this kind of separation is not precipitated right at the impeller eye, because it will carry all through the impeller vane and would very adversely affect the centrifugal flow operation.

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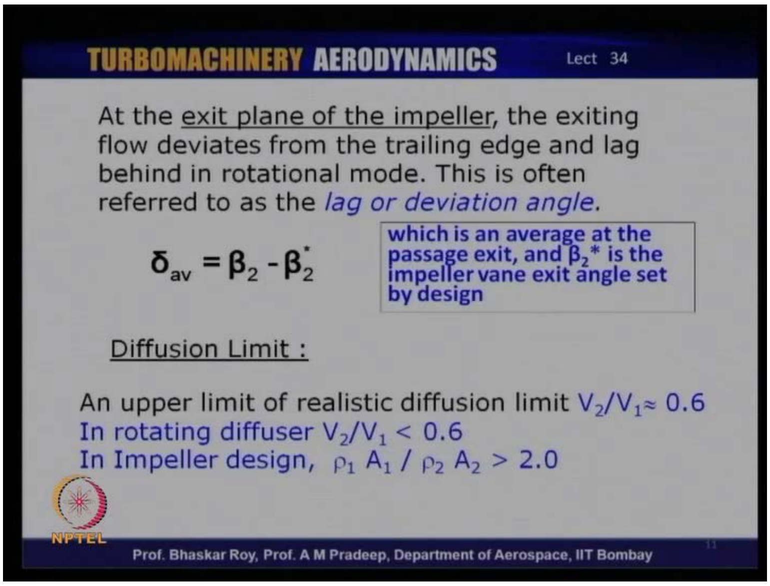

Then, at the exit plane of the impeller, as we have seen, in the flow actually, it has the two vanes. One of them is essentially a trailing edge and the other one can be called the leading edge. So, depending on the direction of rotation, the flow often has a tendency as we have seen, deviate from the trailing edge, because of this deviation, of course, we had the slip and the name slip comes from the fact that the flow so to say, slipped from the trailing surface of the blade.

So, when it deviates from the trailing edge, it creates what is known as a lag angle in the rotational mode. This lag or deviation angle also needs to be factored into the design and how much should be the lag, this lag angle, with the variation of mass flow needs to be factored into the design as far as possible. After the machine is created of course, you have the advantage of CFD to try to get an average value of lag angle between the trailing surface and the leading surface.

So, the lag angle, as you can well imagine, would vary from leading to the trailing surface. On the leading surface, it is likely to be close to 0 or 0. On the trailing surface, it will be some positive value and this would give an average lag or deviation angle. The other thing that you would need to decide at the design time is the amount of diffusion you would like to accomplish. Now, the upper limit of realistic diffusion is typically 0.6, as we had seen in case of axial flow compressor or diffusion factor of 0.7 was normally used.

A similar diffusion limit of about again V 2 by V 1, of the order of 0.6 is normally used. In most of the rotating diffusers, this diffusion limit is quite less than 0.6 to be on the safe side, so that during off-design operation it close to 0.6 and does not precipitate a large separation or surge. In the impeller design, the ratio of density into area at the inlet and at the outlet and the two ratios, which is essentially the inverse of the velocity ratio, is often a close to 2, which means the velocity ratio would be less than 0.5. Typically, 0.5 velocity ratio or the density area product ratio is often used for impeller design diffusion limit even by the modern centrifugal compressor designers.

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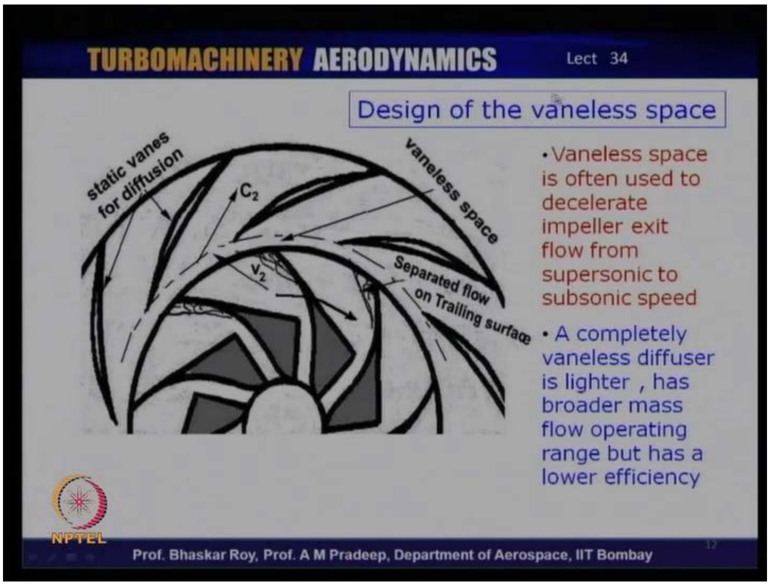

Now, one of the things, I was talking about earlier is that many of the machines often have a tendency to have a vaneless space. This vaneless space is essentially is often intelligently used by the designers. The flow going out from the impeller has this little space to diffuse just a little and this diffusion is often used by the designers to deliberately allow the flow to decelerate from supersonic to subsonic. That means your exit from here (Refer Slide Time: 44:21) and the C 2 that it creates could be actually supersonic exactly at the impeller exit.

In the vaneless space, the small space or the space the can be controlled by design in such a manner, so by the time it goes from here to here, the slight deceleration that is taken place, actually reduces the value of C 2 from supersonic to subsonic, going into the stator vanes of the diffusion process. This vaneless space is often a very important design consideration, because in the modern centrifugal compressors, it allows the designer to avoid certain problems related to supersonic flow going into the stator vanes. This also given rise to the confidence of the centrifugal designers in the modern era that you could actually have a completely vaneless diffuser.

In aero engine application if you get rid of all these vanes, the engine or the compressor becomes so much lighter and as a result of which you can have a much lighter engine. The other issue is that if you do not have this static vanes, the incidence of the flow going into the static vanes, which limits the operating range of the centrifugal compressor in terms of mass flow, actually goes away. That limit goes away, in terms of mass flow operating range, normally related to the stator vanes. So, if you do not have stator vanes that limit is nonexistent and as a result you have a much broader mass flow operating range of the centrifugal machine.

However, since you do not have the stator vanes in that case, the flow would be unguided diffusion. It will diffuse by natural law of diffusion, as you have done in the lectures earlier, and hence the efficiency of the diffusion process will be lowered and the efficiency of the centrifugal compressor would be somewhat lower.

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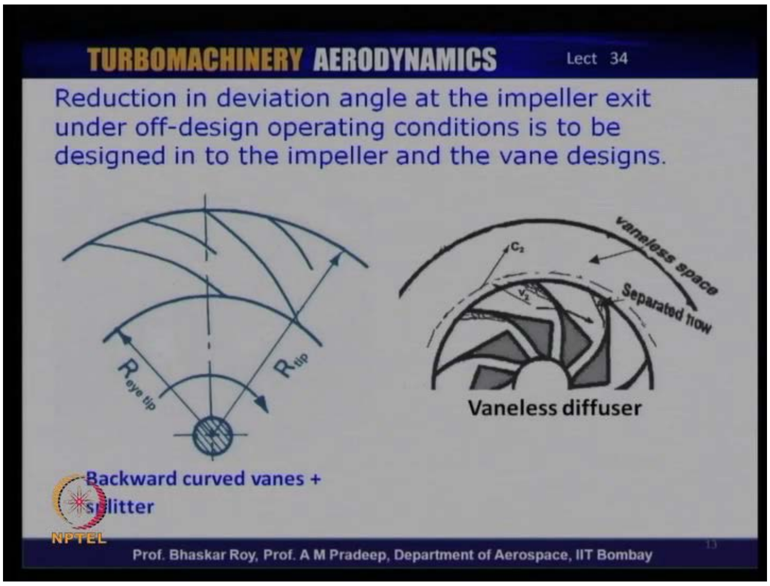

The choice of the diffusion process after the impeller is, you can have a vaneless space, so the vaneless space, we are talking about is completely extended over the entire vaneless diffusion area. You get a broader mass flow but, you have a somewhat lower efficiency.

Lot of research is going on in this respect, so that you have a vaneless centrifugal compressor, but you have a reasonably competitive efficiency of that and research is going on in this area quite a lot. The other area in which research is going on, but people are already applied is to have a splitter vane. So, your centrifugal impeller, which is going as I mentioned backward curve attempts to go for a very high diffusion, in the impeller vanes only and as we have seen their diffusion limit is 0.6 or something.

If you come very close to that kind of limit, there is a danger that under off design operation at lower mass flows, the flow could indeed separate on this trailing edge. This is the leading edge and this rotating this way (Refer Slide Time: 47:40), so this is the leading edge and this is the trailing edge. So, flow could separate away from here and this was the danger. So, this separation from this surface it stick to the surface, but separate from the surface, so they put the splitter vanes halfway through the passage so that the diffusion process is now split in two passages.

Now, you have a better guidance of the diffusion process in the impeller itself. So, the outer part of the impeller, which has a large diffusion being attempted now, has a splitter passage. This splitter then controls the diffusion flow and guides it gently in proper control diffusion and exits the impeller. So, splitter vanes, backward curve, centrifugal compressors, have been put into operation and they are found to be very useful. The exact size, the exact curvature of the splitter, the exact point, at which the splitter is to be deployed or positioned, is to be decided by the designer through analysis and research. So, that is where research and analysis helps the modern designer in a very big way.

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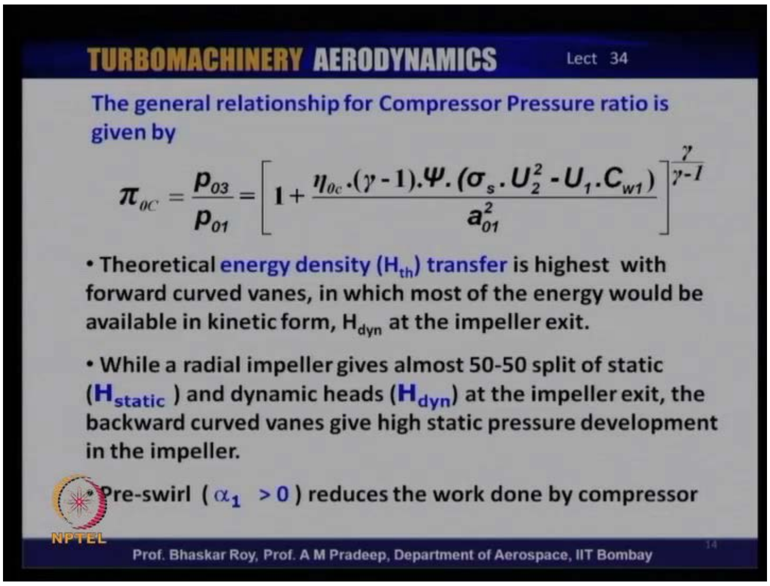

The general relationship of the compression pressure ratio is as you have done earlier depends on the efficiency. It depends on the blade loading, the losses, and the slip factor and of course, the velocity triangle that we have done before. As we have seen, the theoretical energy density transfer is highest in forward curve but, it has all kinds of other problems and hence normally is not used in modern centrifugal compressor especially in aero engines.

The radial impeller gives a 50-50 split of a static and dynamic head at the impeller exit. The backward curve vanes give a very high static pressure development right in the impeller itself, especially if you have the advantage of using splitter vanes.

The pre-swirl that is the inlet guide vane in front of the rotating impeller; it reduces the work done but, it avoids the supersonic speeds at the impeller eye, inlet eye and avoids all kinds of separation shock bound related separation, related separation, related issues right at the impeller eye, so it is a useful thing to have. If you do not need it, you do not need to put it there. So, you have to decide whether you need it or not.

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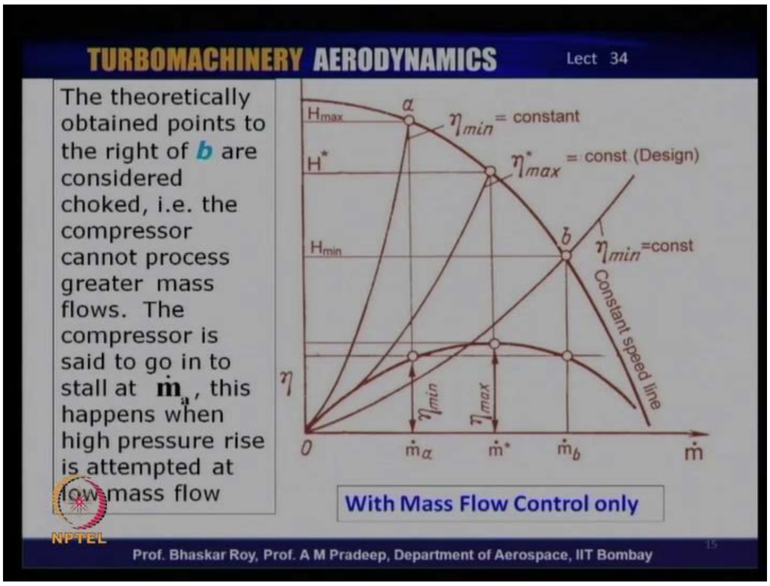

This is the graph where you have a mass flow control that means you only have a mass flow throttle control of the centrifugal machine. In which case, your operation is between over this speed line and b is where the machine is supposed to be choked or maximum mass flow. And a is where you have the minimum mass flow, where the flow could get into stall. So, between a and b the operation is normally to be a carried out. Somewhere, in between you have the maximum efficiency line, which is where you have the maximum efficiency operation over this speed line, which and then if you go away from that, the efficiency could go down.

Then you could have stall here and then you could have choking here, where again the efficiencies in deed somewhat on the lower side. So, maximum efficiency line is over this and this is what decides the compressor operation in a natural manner.

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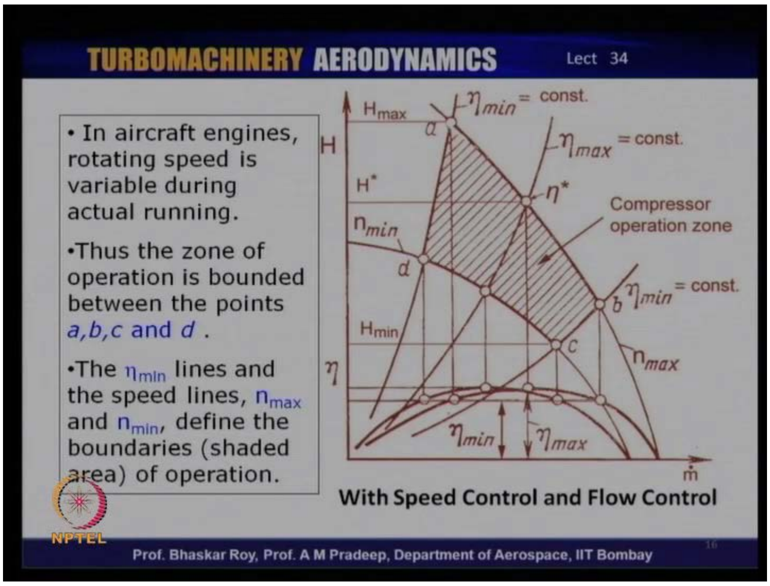

However, if you like to have more controls then what can be done is you have a speed control and flow control together, in which case, you have a zone of operation, which is a, b, c, and d bounded by two speed lines. One is this speed line, which is let us say n max or maximum speed line. Then you have the n min ,which is the minimum speed line and then you have the eta min, which was the minimum efficiency line. So, in this bounded region, shaded region, you have the entire centrifugal flow operation.

From a single line operation, from a to b, now the centrifugal flow can be operated over this entire zone of a b c d. This is what allows if you have independent speed control and mass flow control. The earlier one was only mass flow control now you have speed control.

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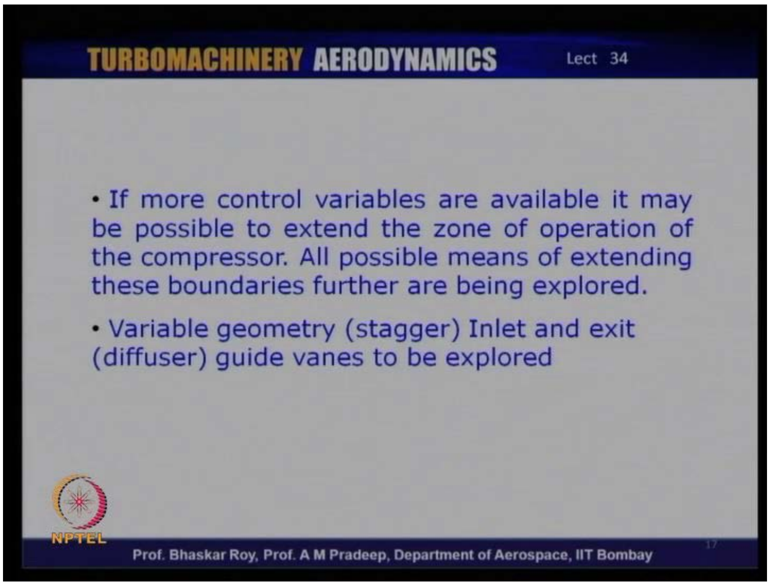

More control variables, if they are available, it may be possible to extend the zone of operation of the compressor, and sum of the possibilities are variable geometry, inlet guide vane and even variable geometry exit diffuser guide vane, which is very difficult because the flow there is at high speed, but inlet guide vane variable geometry is indeed quite possible.

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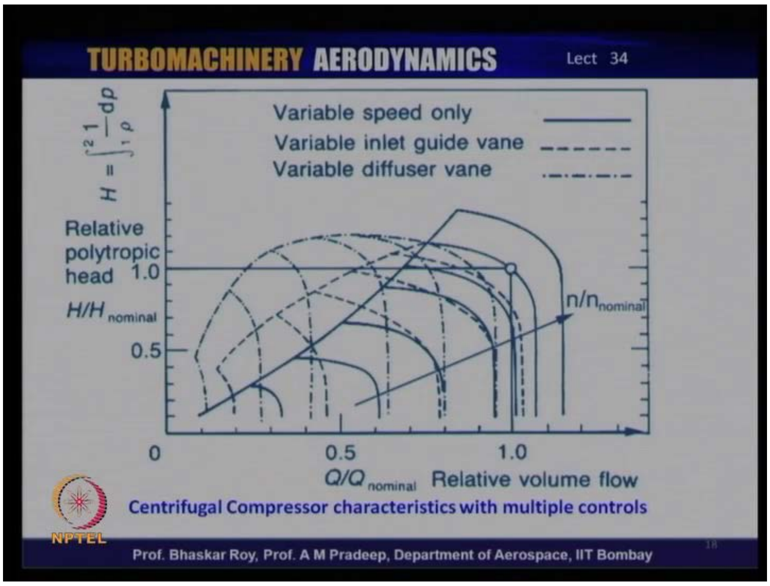

If you do have a variable geometry inlet guide vane, you can extend this stall line further to the left, and extend the operation of the centrifugal compressor. This is a typical centrifugal compressor characteristic map and you can extend it far to the left.

If you have an inlet guide vane variable geometry, you can extend it up to this (Refer Slide Time: 53:19). This is a normal stall line only with variable speed and of course, variable mass flow; variable mass flow is the primary one. Then your variable speed gives you this operating line. If you have additional variable, inlet guide vane, you get this as a stall line and addition to all that, if you variable geometry diffuser vane, the stall line can be extended far to the left and your operation can go to the extent that its mass flow operating range can be extended hugely, by the help of these variable operations. Those are mechanically and by design especially the variable diffuser vane is still in the research in the realm of research and not really been materialized as yet.

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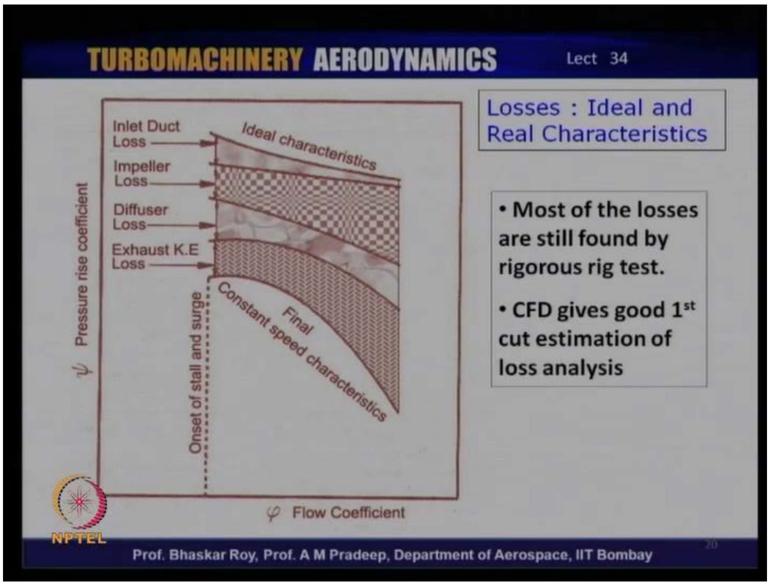

As you can see, in this ideal versus real loss characteristics, the ideal characteristic gives characteristic something like this that is loss coefficient versus flow coefficient. It slightly go down with rise of flow coefficient. However, when you start factoring in the losses, the inlet duct losses starts rubbing higher and goes down with the higher and higher flow coefficient, because the flow becomes more and more turbulent and hence the boundary layer related losses are less and less.

On the other hand, the impeller losses start going up because the friction losses are very high inside the impeller surfaces, which is large amount of metal surface and hence the losses go up with low coefficient. In the diffuser losses on the other hand, it goes down first with the rise of flow coefficient, and then when the coefficient very high, it goes up again. The exhaust kinetic energy keeps going up with the rise of flow coefficient.

At the end, you get a characteristic, which is the characteristic, we had a look at earlier in today's lecture and that you have done in the earlier lecture. So, this is the characteristic (Refer Slide Time: 55:33) we normally look at and this is the ideal characteristic, this is what we know finally get. In between are all these losses of the various elements that we are looking at and exhaust kinetic energy loss, would also be factored into the delivery passage of the volute loss that would have to be factored into the machine efficiency.

That gives us the various losses that need to be factored in. Losses are the most difficult thing to be found in aerodynamic machines and typically the losses are finally, found only after the rig tests are done. The CFD analysis gives a very good first cut estimation of the modern CFD packages, and as a result gives a reasonable idea about the losses that are occurring. Some of the design loss correlations are available in many of the textbooks or old handbooks. One can use the design value that means if the design is somewhere over here, you can factor in the loss parameters through empirical or semi empirical correlations to figure out what the losses should be or how they should be factored, in order to get a possible design performances of the centrifugal compressor.

So, these are the methods by which the losses are somehow factored into the design. Finally, analysis needs to be done, CFD and rig testing to finally, eliminate all the unwanted losses and maximize the efficiency of the centrifugal compressor.

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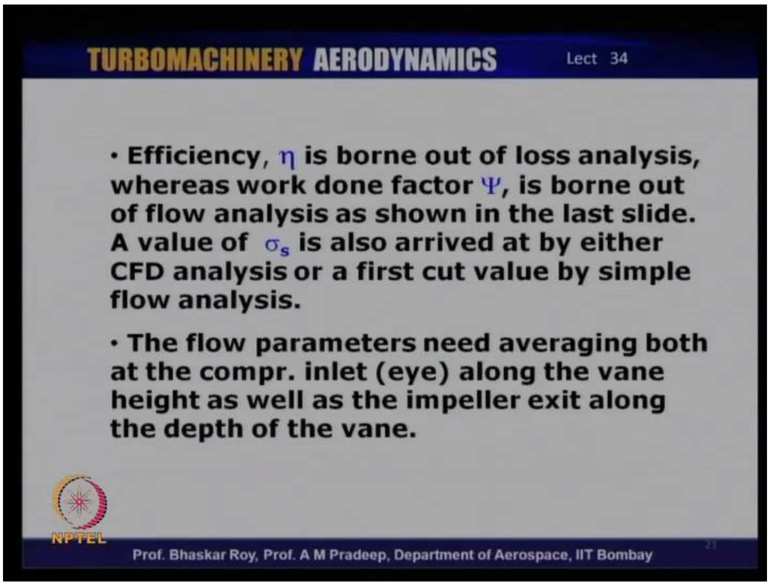

Now, efficiency is borne out of this loss analysis, experimental as well as CFD, and the work done factored that is psi, is borne out of flow analysis, which needs to be normally maximized. You would like to maximize the work done to get maximum pressure rise, so the optimization between high efficiency and high work done, needs to be done by the designer, through series of analysis. So, modern design often is accompanied by a lot of analysis, which optimizes between high efficiency and high work done factor.

This also allows the slip factor, sigma s to be arrived at either by CFD analysis or by simple flow analysis. Somewhere, along the process, you also decide the number of vanes to be decided to be used in impellers and in the static vanes.

The flow parameters that need averaging, both at the compressor inlet eye along the vane height, from the eye root to the eye tip, as well as at the impeller exit, along the depth of the vane from whatever the depth of the vane is there, of the impeller, at the exit of the impeller. Those things need to be analyzed in some detail to begin with through CFD and then later on through rig testing. This is to finalize and refine the design, before such a design can be used for gas turbine engines or for a aero engine application, where the demand for high efficiency, low noise and high work done factor is the highest of all the applications.

This is how one tries to factor in all the possibilities and tries to find out a reasonable optimization of the final design before it is sent for final fabrication and production.

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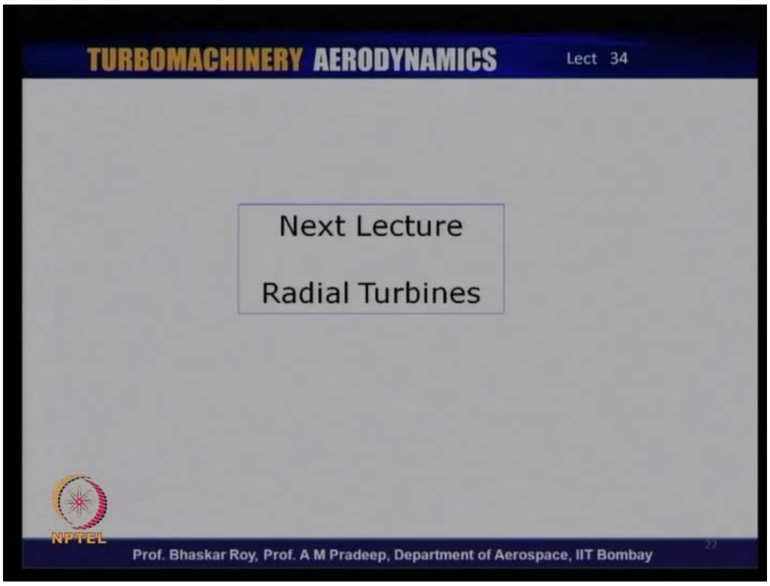

We have come to the end of the centrifugal compressor chapter. In the next chapter, we will be talking about radial flow turbines, which are called radial flow turbine, because of flow. We will see in the next lectures that they are inward flow and not outward flow, so the centrifugal action is inward and often is simply referred to as radial flow turbine.

So, radial flow turbines also have been around, because again they are robust machines like centrifugal compressors, but their application is somewhat less than that of axial flow turbines, because of various reasons. We will look into how a radial flow turbine operates, its principles, its theories, in the next few lectures. So, the next few lectures will be on radial flow turbines.