Turbomachinery Aerodynamics

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Lecture #31

Centrifugal Compressors: Thermodynamics and Aerodynamics

Hello and welcome to lecture number thirty-one of this lecture series on turbomachinery aerodynamics. We have discussed quite a lot on different aspects of turbo machines over the last several lectures. We actually started off with very introductory topics and very fundamentals of some, the turbo machines, starting with thermodynamics and its analysis as applicable to turbo machines.

We then moved on to discuss about axial compressors and we had quite a lot of discussion, detailed discussions on axial compressors and the various design aspects of axial compressors and how an analysis, the detailed analysis of axial compressors can be carried out. Subsequently, we spend some time on discussion on turbines and we first started off with the axial turbines, the design issues related to axial turbines, performance analysis of axial turbines and so on.

What we going to discuss today is a slightly different topic in the sense, that the, the component that we are talking now about is not really an axial turbo machine. We are going to discuss about centrifugal compressors starting from today's lecture and we will continue with this on 2 more lectures. We will discuss some details about centrifugal compressors, but in contrast to axial compressors, our discussion would be rather superficial in the sense, that we will not be really taking up very detailed discussion on the design aspects of centrifugal compressors.

The basic reason being, that centrifugal compressors are not really widely used in modern day aircraft engines as compared to the axial compressors. They are still used, but they are, their usage has been limited to rather smaller sized aero-engines and some other applications. And that is the reason, why we shall not really be discussing too many details about, especially to do with design aspects of centrifugal compressors. But of course, we will spend some time discussing about the fundamental issues related to centrifugal compressors, the different aspects of analysis of centrifugal compressors, as well as the performance characteristics in quite some detail, over the, over the next 2 or 3 lectures. Also, of course, we will also be having a tutorial session, where we will get a chance to solve some problems associated with centrifugal compressors.

So, in today's lecture, we will start with the fundamentals, we will talk about the thermodynamics of centrifugal compressors and subsequently, we will be discussing about the different elements of centrifugal compressors. What are the different components, which constitute a centrifugal compressor, are and what is the flow characteristic associated with the flow through these different components?

So, let us begin our discussion with the thermodynamics of centrifugal compressors, but of course, we will also have a quick introduction to centrifugal compressors as a whole. And see, what centrifugal compressors are and why are it, that these compressors are not really used in that much popularly as compared to the axial compressors? So, that two distinct aspects of centrifugal compressors.

As we can see, thermodynamics, as well as the components of centrifugal compressors, so it will be very interesting to know, that centrifugal compressor, in fact, has a longer history than axial compressors.

In fact, the earliest jet engines that flew, one by Frank Whittle in, in England and the other, where German engineer named Fanzhoean, both of them developed the jet engine independently and both these initial developments of jet engines used centrifugal compressors. In fact, that continued for a very long time and even do the, the, the very common aircraft, which were used, the fighter aircraft which were used during the 2nd world war, all of them had, in fact, most of them had centrifugal compressors.

There are certain inherent advantages with centrifugal compressors, but of course, the disadvantages are also equally substantial and that is the reason why, as we look at larger and larger sized aero-engines, use of centrifugal compressors becomes rather disadvantageous, in the sense, that centrifugal compressors require a larger frontal area. That, is probably the most important disadvantage of the, of a centrifugal compressor, that the pressure ratio, that is developed per stage from a centrifugal compressor, very much depends upon the overall diameter of the jet of the centrifugal compressor, which means, that if you look at a larger sized aero-engine, which is used in, let us, a modern day passenger or fighter aircraft, they all require the thrust requirement from such an engine is tremendous. And to be able to develop such a high level of thrust, it is necessary, that the compressor develops the corresponding pressure ratio required for developing or generating this kind of a thrust.

From a centrifugal compressor if you were to develop such a high level of thrust, the frontal area required by such an aircraft of such a compressor would be substantial. And obviously, such an engine would also have a huge amount of drag.

So, an aero-engineer, an aircraft, airframe engineer would not really want an aero-engine with a large frontal area because that is going to increase the overall drag of the aircraft and obviously, that is not a good idea. And so, if you compare this with an axial compressor, to develop the same pressure ratio and axial compressor requires a smaller frontal area, but of course, axial compressor requires multiple stages to develop the same pressure ratio.

Centrifugal compressors can generate far higher pressure ratio per stage as compared to axial compressor and that is of course one advantage, a huge advantage of centrifugal compressor, that they can generate a large, much larger pressure ratio per stage than an axial compressor.

We will, we will of course, be exploring the reasons for this a little later. When I will explain thermodynamics of centrifugal compressor, we will see that centrifugal compressors have a slightly different mechanism of pressure rise than axial compressor. And that is the reason why they can generate much higher pressure ratios per stage.

Yet, there is a yet another difference or disadvantage, so to say, between centrifugal and axial compressor. For larger sized engines, axial compressors have slightly higher or better efficiencies than centrifugal compressor and that is, of course, if you look at an aero-engine perspective, the efficiency is of utmost importance and even a slight improvement in efficiency means a big deal in terms of the overall engine performance. And therefore, that is where axial compressor scores yet again over centrifugal compressor and these are primarily the two reasons, why centrifugal compressors are not really used in larger sized aero-engine.

They are used very much in smaller sized engines because for smaller engines, axial compressors, in fact have certain disadvantage, because as you reduce the engine overall diameter to smaller and smaller levels, the losses associated with tip clearance and so on increased substantially. And therefore, in fact, for smaller sized engines, centrifugal compressors may have performance or efficiencies as high or in fact, even better than axial compressors. And so for smaller engines, it is a common practice to use centrifugal compressors and therefore, smaller sized engines, which are used in, let us say business jet or smaller airplane, they still have engines, which have centrifugal compressors.

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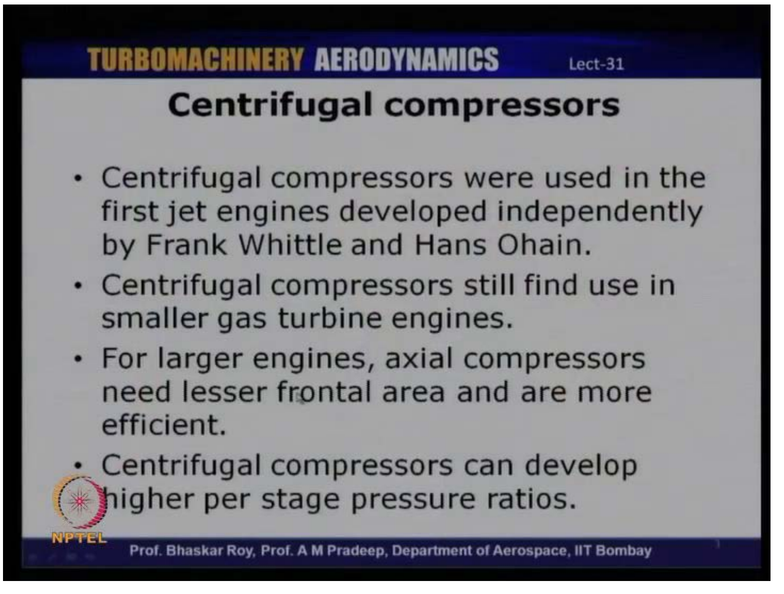

So, to sum up, centrifugal compressors obviously can generate much higher pressure ratio per stage. They, obviously, have larger frontal area and therefore, they are not as commonly used as axial compressors and of course, they are little less efficient.

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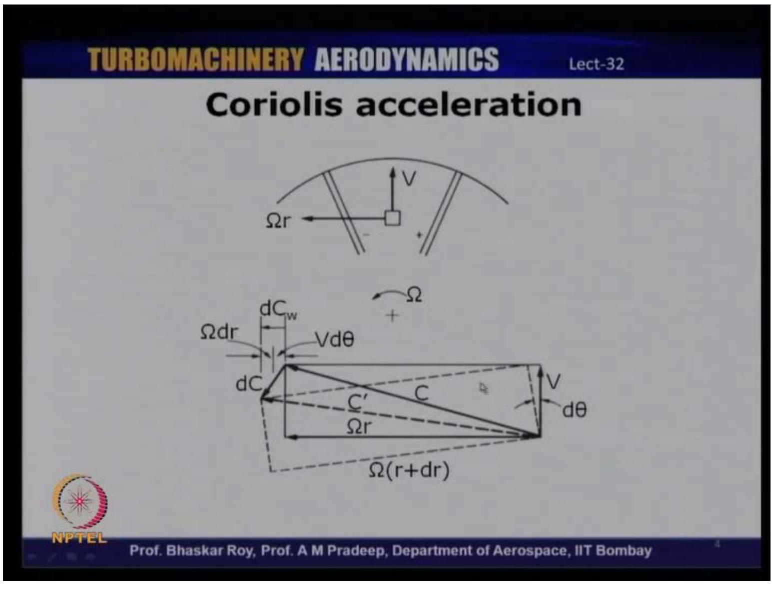

And centrifugal compressors are used in auxiliary power units, the APUs of many aircraft. Some aircrafts also have a centrifugal compressor as part of the air conditioning system, which is used in the aircraft and in some engines, centrifugal compressors are used in combination with an axial compressor, that is, centrifugal compressor would form the last stage of a set of axial compressor stages and such engines are also quite popularly used, which are of course the medium or smaller sized engines; some examples being the T700 from GE, PT6 from Pratt & Whitney or the Annie Wetly 53 engines.

So, these are some engines, which have centrifugal compressors, which are used in combination with an axial compressor, and the, the inherent advantage here is, that one axial compressor can replace multiple centrifugal compressor stages because it is possible to generate a higher pressure ratio per stage by a, from an axial, from a centrifugal compressor as compared to axial compressor. And that is why, having a centrifugal compressor as a last stage would actually replace several axial compressor stages, and that is a big advantage because it leads to a lot of saving, possible savings in weight and part count and so on. So, there are some advantages of trying to do that kind of a configuration.

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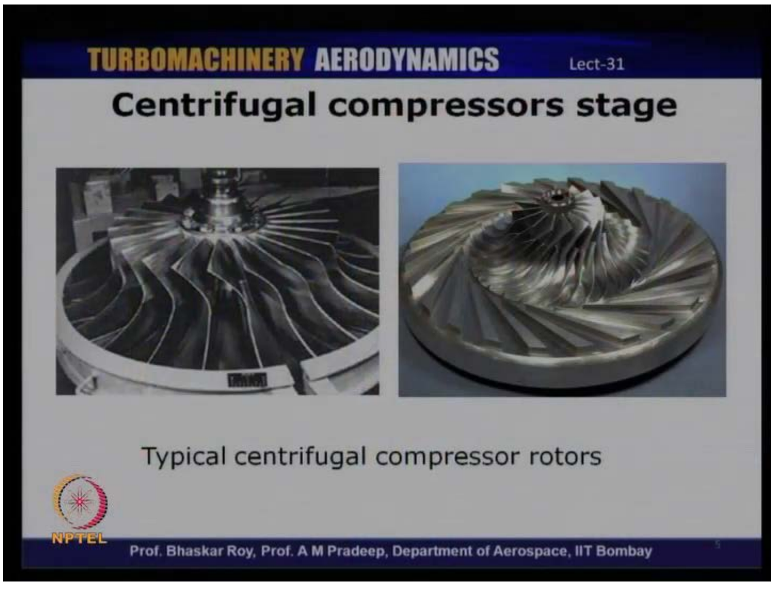

So, let us now take a look at some typical centrifugal compressor rotors. So, I have here two different types of rotors, the one you see on the left hand inside is a so called classical centrifugal compressor rotor, which has a straight radial blade. You can see, this is the impeller of the centrifugal compressor and you can see, that these blades are radial and straight, and of course, there is a bend here, at what is known as an inducer. We will discuss little more details about this later on. So, an inducer actually turns an axial flow and guides it into the impeller and makes the flow radiant.

On the right hand side, you can see, any, the rotor of a centrifugal compressor, which is much more complicated than what you see on the left hand side and this is a rather recent development. And you can see that these blades are much more complicated than what you see in a conventional centrifugal compressor rotor. And you can also see that at the exit of the impeller, you, these are the diffuser vanes, of course that is not really shown here, these also would have diffuser vanes. So, these are the vanes of the diffuser and of course, we will discuss more details on why a diffuser is used in, in a centrifugal compressor.

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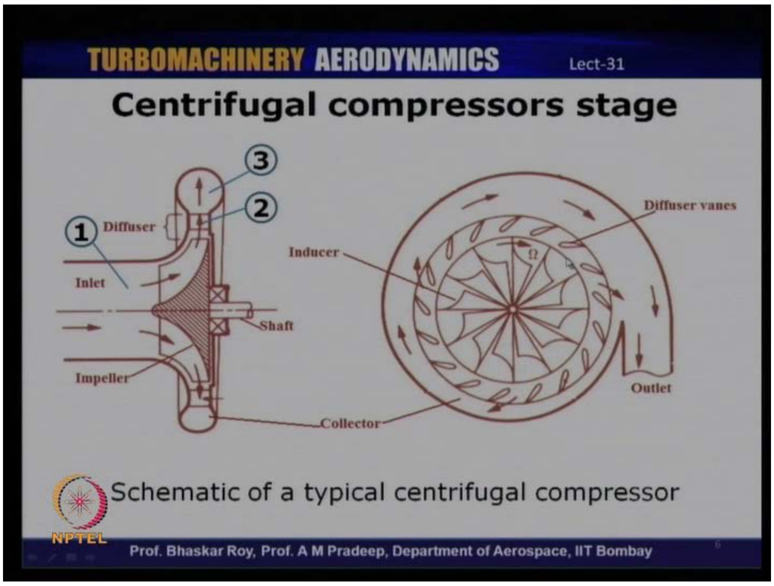

So, having understood or at least please take a look at two different classes of centrifugal compressor rotor. Let us take a look at the schematic and understand what constitutes a centrifugal compressor.

So, there are primarily 3 distinct components in centrifugal, in a typical centrifugal compressor. It has an inlet and followed by a rotor, which is also referred to as an impeller in a centrifugal compressor and the impeller exhausts or discharges the flow into what is known as a diffuser, and from the diffuser there is a collector or a volute, which guides the flow towards the outlet.

So, the flow enters the impeller in an axial direction and then, the impeller deflects the flow and turns it into a radiant flow and this radial flow exceeds or exhausts the rotor, exhausts the compressor through the collector or volute.

And most of the impellers also have, what are known as, inducers. Inducers basically, guide an axial flow and allow the flow to enter the impeller smoothly in the absence of inducer as we will see later, the flow can actually, say, there is a tendency for the flow to separate from the impeller vanes if one does not have any inducers. So, inducer is sort of like an inlet guide vane that we have seen in the case of an axial compressor.

So, there are, there is a, there is a set of stationary components as well as one rotating component in a centrifugal compressor. The impeller forms the rotating component or the rotor of a centrifugal compressor. The inlet and the diffusers are the stationary components of stator of a centrifugal compressor.

So, once we have an, so now, that we have understood the, the working or basically, the components are constituents of a centrifugal compressor, let us now try to understand the operation of the centrifugal compressor from a little more fundamental sense, from a thermodynamics perspective. We will try to understand what really happens as the flow passes through these different components of a centrifugal compressor.

Now, I had shown the, the other three distinct components, which I had marked as 1, 2 and 3. Let us take a look at them once again. 1 is here being referred to as the inlet of the centrifugal compressor; 2 is the diffuser and 3 is the exit of the diffuser or the volute or collector of the centrifugal compressor.

Now, on a T-S diagram, this is temperature entropy diagram, you can see, that of course, it looks quite complicated here. Let us try to understand what are, what is basically being implied by the set of a constant pressure lines that are shown here.

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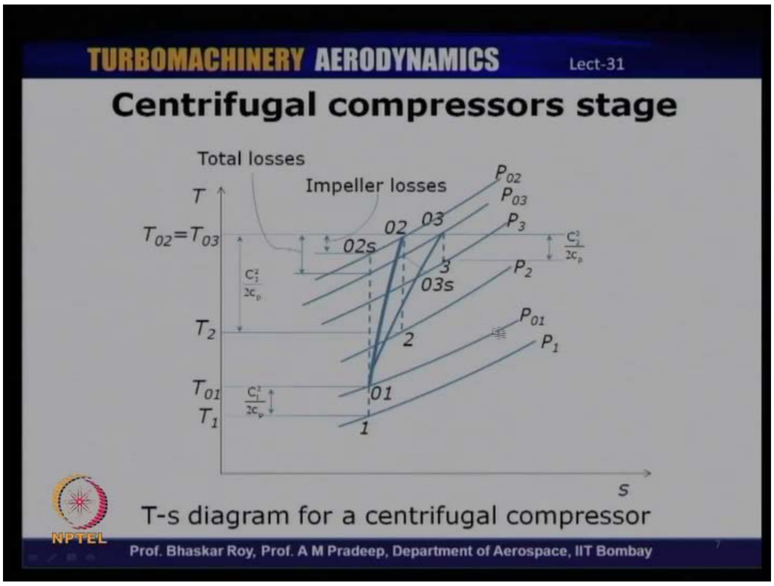

So, station 1, as we have seen, is this inlet of the compressor; 2 correspond to the diffuser and 3 is for the exit of the diffuser or the volute. So, there are static pressure lines as well as the corresponding total pressure lines, which have been shown. Energy is added, as you know, in the impeller and so the stagnation pressure rise takes place between 0 1 and

0 2 pressure, between stations P 01 and P 02. And in the volute, there is a stagnation pressure loss that is why you can see, that there is a certain amount of loss, which is being associated here because of the loss of stagnation pressure in the volute.

Now, from station 1 static pressure rises between station 1, P 1 to P 0 1 and this is the corresponding stagnation parameters, C 1 square by 2 C p. So, T 1 plus C 1 T 1 is T 1 plus C 1 square by 2 C p. There is no change in stagnation temperature between station 2 and 3 because there is no energy added after the impeller. So, energy is added between station 1 and 2 and that is why, we see a change in stagnation temperature from T 0 1 to T 0 2, where, that is where the energy is added.

And actual process they are shown in 2 different lines here. There is a thicker one, which is showing the process between 0 1 and 0 2 and there is another line, which is showing the process between, 0 2 and, 0 1 to 0 3. So, it, it is possible of course, that in some centrifugal compressor, there may not be a vane less, could be or may not really have this vane less space between the inducer of an impeller exit and the diffuser, and that is why these two distinct lines have been shown.

But of course, they both lead to the same stagnation temperature; there is no change in stagnation temperature here. If we were to look at the losses, which are in, occurred in centrifugal compressor, then the total losses, as shown here, is between the stagnation temperatures, which it would normally achieve minus the stagnation temperature that it should have achieved if the process was isentropic.

So, this is corresponding to the process if the whole were to be isentropic and in which case, there are, of course, no losses taking place. And so, the corresponding dynamic parameters are also shown here between station for, let us say, station 3, between P 3 and P 0 3. We have c 3 square by 2 C p and C 2 square by 2 C p between which basically, takes the temperature from d 2 to T 0 2.

So, if you look at, if you compare this with the temperature entropy diagram that we had discussed for an axial compressor, you will see, you can quickly figure out the similarities between both these compressor operations, at least in a thermodynamic sense. In, even in an axial compressor we have seen that energy is added in the rotor and you may actually have a total pressure loss taking place in the stator. Even though static pressure continues to rise in the stator, one may have total pressure loss taking place because of frictional effects.

So, they are quite similar in the sense, that in, in the case of centrifugal compressors also we have an impeller or the rotor, where energy is added and that is followed by a diffuser, where there is obviously, no more energy addition taking place.

One may have, one continues to have static pressure rise, which is why P 3 is actually greater than P 2, but there could be some amount of total pressure loss taking place in the diffuser due to frictional losses and, and that is why, we have P 0 3, which is less than P 0 2.

So, what we will do next is to look at the working of the centrifugal compressor, as well as trying to estimate the work done or work required for driving centrifugal compressor rotor, and we will relate that to the velocity components; very similar to the analysis we have done for an axial compressor as well.

We will try to relate the work done or work required for driving the compressor to the velocity components because it would be easy for us to construct the velocity triangles and develop and calculate the work done or work required for a centrifugal compressor.

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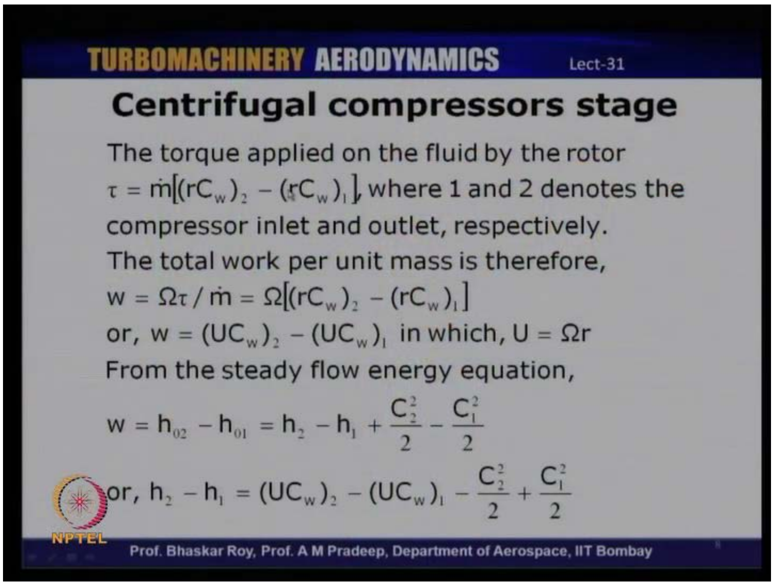

So, let us look at the governing equations for centrifugal compressor stage. Now, in a centrifugal compressor rotor, the torque required for or torque applied on the fluid by the rotor is a function of the mass flow rate of course, and components of the tangential velocity. So, here, we have torque required is equal to m dot into r times C w at station 2 minus r times C w at station 1 and so here, the stations 1 and 2 are denoting compressor inlet and outlet respectively.

Unlike an axial compressor where the, there is hardly any change in these tangential velocities between, for a given station, here what we will see little later as well, that even the blade speed is not really a constant because the flow is a radial; the blade speed also changes with every radial location.

And so, we have mass flow rate m dot m dot times the r m, the velocity component, tangential velocity C w at station 2 minus r times C w at station 1. So, the total work done per unit mass is w, is basically a function of the rotational speed omega times the torque divided by the mass flow rate. This is in turn equal to omega times r times C w at station 2 minus times C w at station 1. So, if U, multiply omega times r, we get the blade speed U. Therefore, work is equal to U times C w at station 2 minus U at C U times C w at station 1.

And so, in axial compressors we had actually written this as U times delta C w because U at inlet and exit of the compressor was assumed to be the same. Here, it, it does not remain the same and it changes with, because the flow is indeed radial.

Now, if you now look at the energy equation, the steady flow energy equation and compare that with what we have written here for a centrifugal compressor rotor, we have work is equal to the change in enthalpy stagnation enthalpy, h 0 2 minus h 1, this is equal to the static conditions and the dynamic conditions. So, h 2 minus h 1 plus C 2 square by 2 minus C 1 square by 2.

We have already calculated the change in enthalpy stagnation enthalpy in the previous equation here. So, if you substitute that here, we have h 2 minus h 1 is equal to w minus this velocity changes, where w has already been calculated as shown here. So, h 2 minus h 1 change in static enthalpy is equal to U times C w at station 2 minus U times C w at station 1 minus C 2 square by 2 plus C 1 square by 2, where C 1 and C 2 are the absolute velocities entering the rotor, entering the compressor and leaving the compressor respectively.

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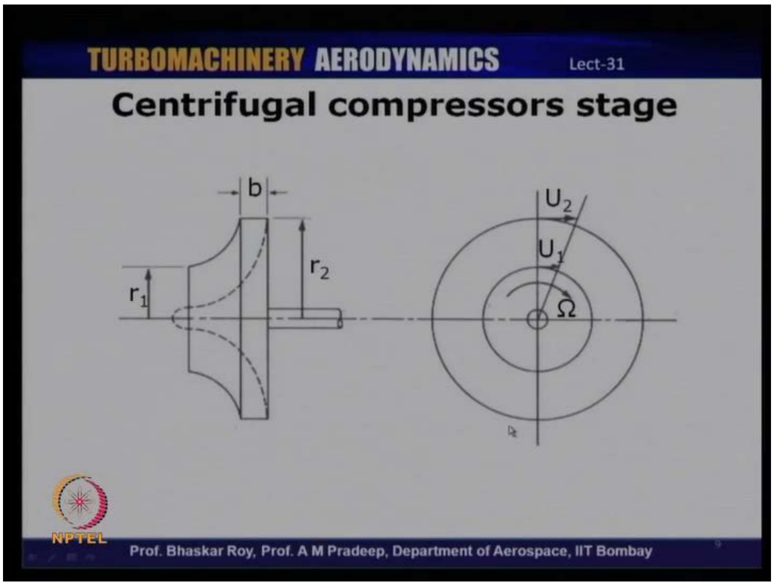

Now, if you look at the impeller for example, let us take a look at the schematic of an impeller and we have an impeller inlet tip radius, as shown here, as r 1 and impeller outlet radius as r 2 and the corresponding blade speeds would be u 1 at this location and u 2 at the exit of the impeller.

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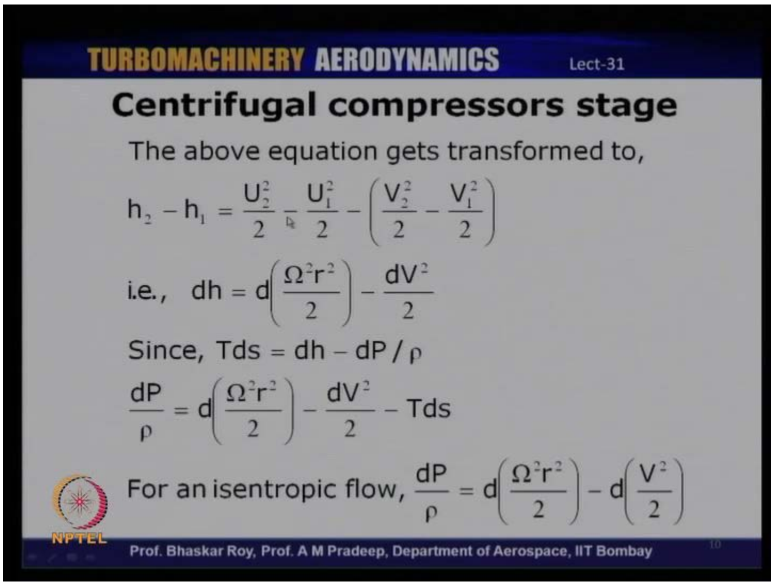

The velocity components for an impeller, then the above equation, which we have written earlier, that is, h 2 minus h 1 is the difference of U times C w at station 2 minus U C w at station 1. And the velocity components, that gets transformed into h 2 minus h

1 is U 2 square by 2 minus U 1 square by 2 minus the relative velocities V 2 square by 2 minus V 1 square by 2. So, this in differential form, we can write as dh is equal to d into omega square r square by 2 minus d into V square by 2.

Now, if you recollect the basic thermodynamics, the Tds equation, Tds is also equal to dh minus dP by rho and therefore, we have on the left hand side dh, we shall replace by dP by rho. For an isentropic process Tds is 0, so dP by rho is equal to d into omega square r square by 2 minus dV square by 2 minus Tds, which for an isentropic flow becomes 0. Therefore, dP by rho is d into omega square r square by 2 minus d into V square by 2.

So, what we have just now written down is an expression, which relates the pressure, change in pressure across a compressor, two, two distinct parameters or terms. One is proportional to the rotational speed and the radius or radial location, the other is proportional to the change in relative velocity.

Now, this is a generalized form of an equation, that I have written down, which also could be extended to an axial compressor, in which case we have assumed, that there is no, there is no change in the axial, well, the radial location for given analysis. So, if you take up one radial plane, then there is no change. The d omega square r square term becomes basically 0 for an axial compressor, which means, that the pressure rise would now be equal to minus dV square by 2 for an axial compressor, that is, the pressure rise in an axial compressor is proportional to the amount of deceleration taking place at the compressor in the relative velocity, in terms of the relative velocity?

But in an a centrifugal compressor, as we have just seen, the pressure rise is a function of one additional parameter, which is d of omega square r square by 2, that is, even if there is no deceleration taking place in a centrifugal compressor rotor, which means, that if the 2nd term is, is equal to 0, one can still at in a certain amount of pressure rise simply because of the 1st term, that is, d omega square r square by 2.

That, it is possible to achieve pressure rise in a centrifugal compressor even if there is no deceleration taking place, which means, that centrifugal compressors ideally should not be affected by boundary layer flows because we are saying, that even if there is no deceleration taking place, one can still achieve pressure rise because of the centrifugal effect and that is why, it is called the centrifugal compressor.

But most of the modern centrifugal compressors also have a certain amount of the deceleration taking place, that is, pressure rise also because of deceleration or diffusion as well as because of displacement of the centrifugal flow field.

So, there is a component of both in a centrifugal compressor and which is why, even centrifugal compressors are indeed affected by boundary layer flows, and it is not possible to eliminate the boundary layer effects all together, even in a centrifugal compressor, but of course, it is possible to achieve much higher pressure ratios per stage in a centrifugal compressor as compared to an axial compressor.

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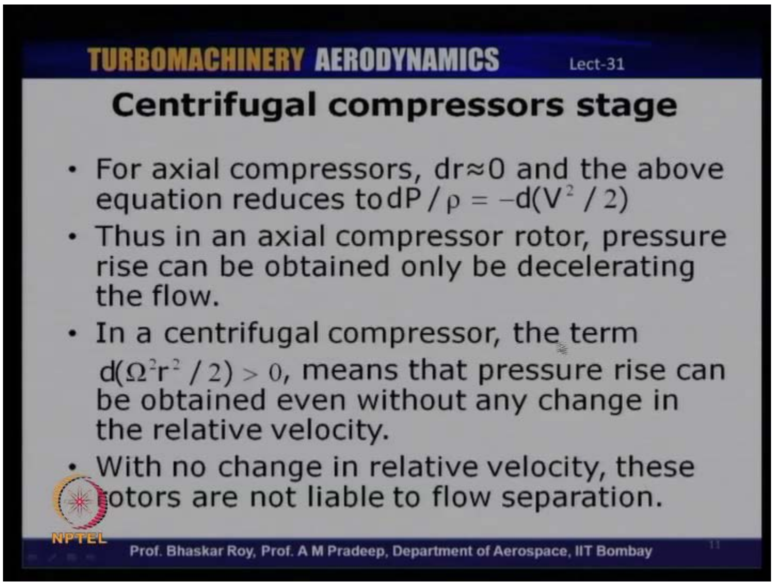

So, in an axial compressor, where we can assume, we have assumed the d r, that is change in the radius is equal to 0.

The equation, that we have written here, dP by rho is d omega square r square by 2 minus dV square by 2 is simply reduced to dP by rho s minus dV square by 2, that is, in an axial compressor rotor, the pressure rise can be obtained only by the decelerating the flow. In a centrifugal compressor, the first term is basically greater than 0 because omega square r square by 2, change of that is always greater than 0.

Therefore, pressure rise can be obtained even without any change in relative velocity and which means, that it is possible to have a rotor, which does not have any deceleration and still develops a certain amount of pressure.

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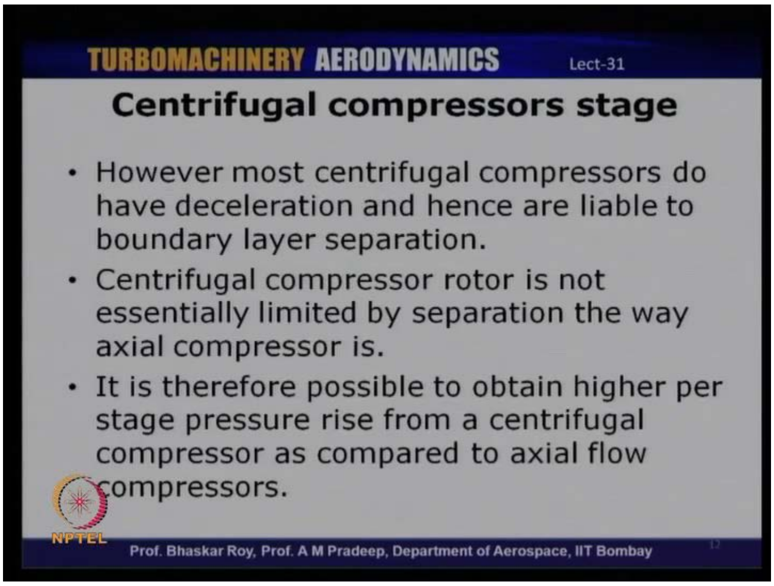

But most of the modern compressors, as I have mentioned, do have deceleration and also this means, that centrifugal compressors are indeed affected by flow separation. But not to the extent, that axial compressor is and therefore, it is possible to achieve much higher pressure ratio per stage from a centrifugal compressor, as compared to axial compressors.

Now, that is one set of governing equation that we have discussed. There is one more aspect of centrifugal compressor, which is also true with some of these radial flow machines. We shall discuss about what that equation of conservation is and that is also valid for a centrifugal compressor.

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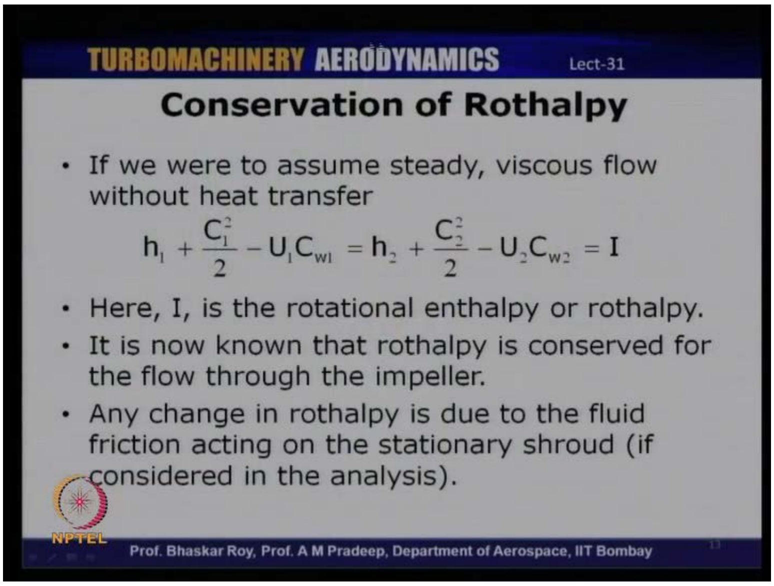

So, this is to do with what is known as rothalpy.

Now, if you, let us say, assume steady, viscous flow without any heat transfer, then in radial flow scenario we have this particular conservation equation, which is also valid for these radial flow machines, that is, h 1 plus C 1 square by 2 minus U 1 C w 1 is equal to h 2 plus C 2 square by 2 minus U 2 C w, that is, h plus c square by 2 minus delta u C w. It is basically conserved parameter in a centrifugal compressor or even in any other radial flow machines.

So, this is usually denoted by symbol I and this is known as the rotational enthalpy or rothalpy for short. It is called rotational enthalpy because it combines enthalpy in the conventional sense, that is, h plus c square by 2 and the component, which is associated with the, component which is associated with U times C w, which is to do with the tangential velocity and therefore, it has been observed, that this parameter generally is conserved as it, as the flow take place through an impeller.

And the change in rothalpy in some cases, one does see that there is a change in, in rothalpy is primarily because of fluid friction, which is acting on the stationary shroud. So, that is, if you consider the impeller plus the shroud, there could be certain amount of losses taking place as the result of the shroud and that probably, would explain the amount of loss in rothalpy, which might be observed in some analysis, but in general, in radial flow machines rothalpy is a conserved quantity. So, in centrifugal compressor also, in most of the analysis conservation of rothalpy is assumed to be satisfied as the flow takes place through the impeller.

So, having understood some of the fundamental governing equations of a centrifugal compressor, let us now look at the different components of a centrifugal compressor and also, look at how the flow develops as it passes through these different components. So, we will be discussing about 3 distinct components: one is the impeller, 2nd is the inducer, which is also a part of the impeller in fact, and the 3rd is the diffuser, which is the other stationary component of the stator in a centrifugal compressor. So, let us begin with the impeller, which is the rotor of a centrifugal compressor and the most crucial component in a centrifugal component, compressor.

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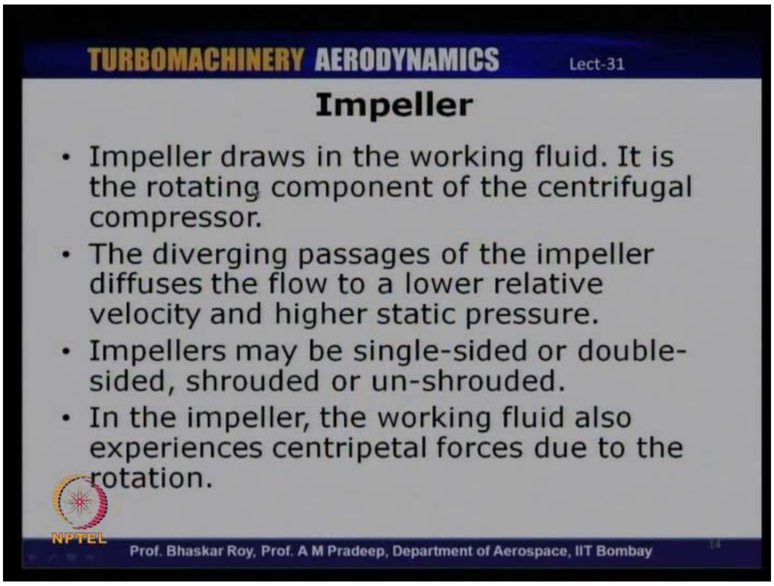

So, impeller is the component, which draws the working fluid and it is like the rotor as we have discussed for an axial compressor. And impeller has diverging passages, which diffuses the flow to a lower static pressure and well, higher static pressure and lower relative velocities. And that different ways or different configurations of an impeller, it could be either single sided or double sided, shrouded or un-shrouded and so on. Now, in impeller, the working fluid, besides deceleration, it will also experience centripetal forces because of the rotation itself and displacement of the rotational, of the fluid elements. And therefore, besides the fact, that the fluid decelerates and diffuses, there is also a certain amount of centripetal force acting on the fluid elements as it passes through an impeller.

Now, there are 3 different types of impellers or impeller blades that are possible. Now, simplest type is the straight radial type or one might have forward leaning blades or backward leaning blades. Now, forward leaning blades are considered to be inherently unstable and we will see the reason for its unstability probably in the next class, where we will be talking about performance analysis or performance characteristics of centrifugal compressor. So, we will see the reason for, why forward leaning blades are not really used; that is probably something, which I will be explaining in next class.

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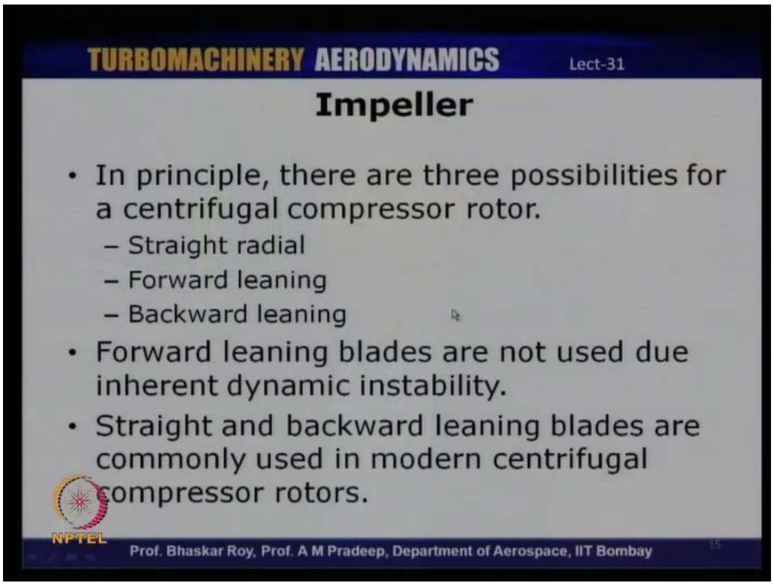

The two other configurations, which are commonly used, are the straight radial blades and the backward leaning blades. So, both these configurations are commonly used in modern day compressors centrifugal compressors; so, straight, radial backward leaning and forward leaning blades.

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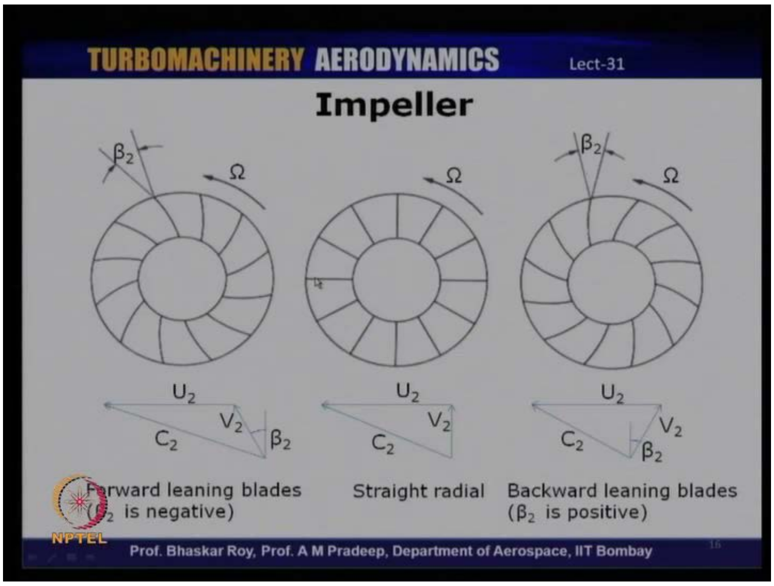

Let us take a look at what these 3 different blading configurations are. We will start with straight radial first. So, this, the one that is shown in center, is a typical straight radial blade, this is the direction of rotation and this is the velocity triangle of the flow as it leaves the impeller. So, V 2 is the relative velocity leaving impeller, C 2 is the absolute velocity and U 2 corresponds to the blade speed at the tip of the impeller.

And so, as you can see, the flow leaves the blade radially as V 2 is indeed radial, that is why it is called straight radial blade. If you look at the forward leaning blades, V 2 leaves the blade tangentially, that is why V 2 is at a certain angle to the impeller vanes at the exit and it leaves at an angle of beta 2, which is negative. So, forward leaning blades have a negative blade angle of beta and the velocity triangle is what you see here, C 2 and U 2. Backward leaning blades, on the other hand, had a positive beta, beta 2. And so you have beta 2 and so you have U 2 in this direction, which has a positive blade angle beta 2 and C 2 is the absolute angle and U 2 is the blade speed at the tip.

So, these are the three different configurations of impellers, which are possible and two of them, as I mentioned, straight radial and backward leaning are the ones, which are commonly used. Of course, the, the straight radial blades are, have conventionally been commonly used because it is simpler to construct as well as the fact, that the blade should not have to undergo lot of stress because as you bend the blades, the stress on the blades become substantial, but a modern day design and materials permit us to use these complicated shapes as well. In fact, some of the blades have a combination of straight radial and backward leaning geometry. So, there are blades, which have a combination of backward and straight radial as well.

So, that is about the impeller, where, which is probably the more crucial component of a centrifugal compressor. And we have seen the velocity triangle corresponding to these 3 different configurations of a centrifugal compressor impeller.

Now, there is another component, which is usually part of the impeller itself. We have already seen that in the pictures I had shown in the earlier, probably the 3rd slide in today's class, where there, the initial part of the impeller, as you have probably have noticed is, is bent. So, there is curvature given to initial part of the impeller and that is known as an inducer. And an inducer is basically a component, which is just ahead of the impeller or in; in fact, it is almost, always initial part of the impeller itself. The basic function of an inducer is to guide the flow smoothly into the impeller in the absence of an inducer when the impeller is rotating. There is a relative component at the impeller inlet and in the absence of the inducer, the relative velocity enters the impeller at a certain angle and there could be flow separation taking place from the impeller (( )). To avoid that, one uses set of blades, which are like guide vanes; these are known as inducers.

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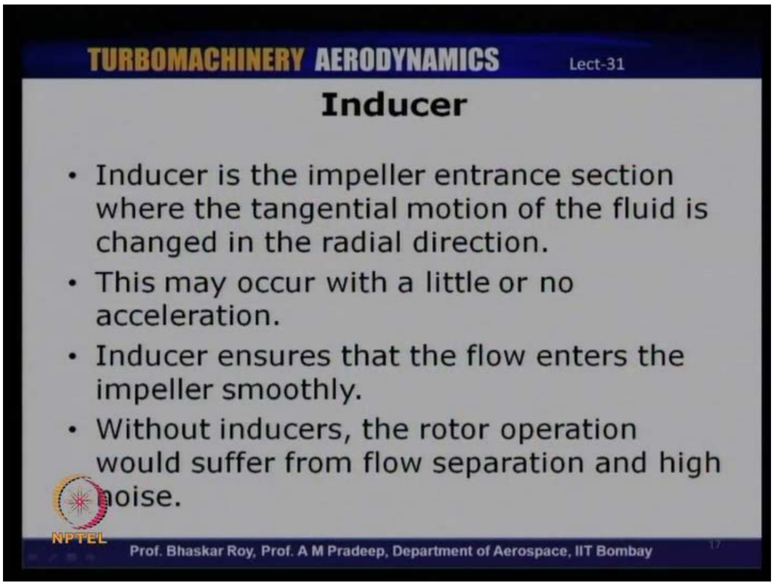

So, inducer is the inlet, impeller entrance section, where the tangential motion of the fluid is basically changed to the radial direction. And one may have either the acceleration or some amount of acceleration within the inducer, as we will see it later.

Inducer, the basic function of an inducer is to ensure, that the flow enters the impeller smoothly; without inducer, the flow operation is likely to suffer from flow separation and high levels of noise because of the flow separating from the impeller.

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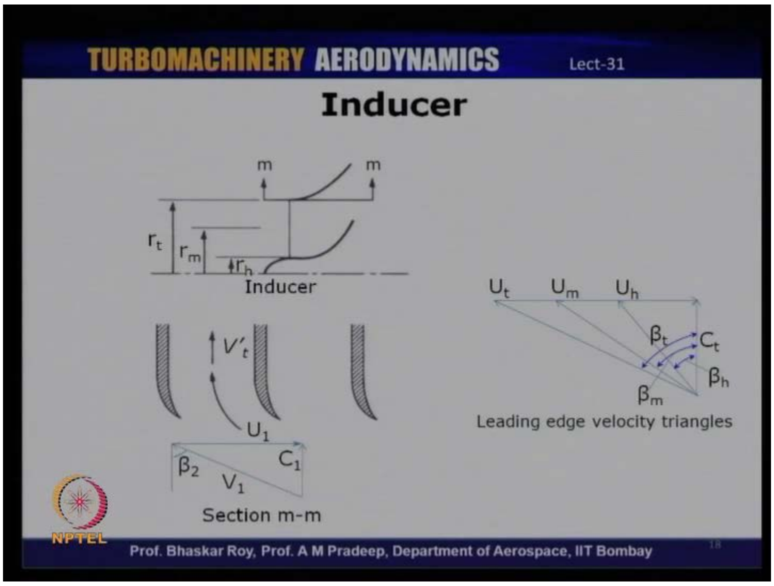

So, this is schematic of an inducer and there are two views of the inducer shown. So, the initial part of the impeller, as you can see, this part of the impeller is the inducer and if you take a cross section of the inducer, one would see vanes, likes what is shown here.

So, the inlet velocity triangle of an impeller would look something like this that the flow, assuming that the flow is coming at an axial, in an axial direction C 1 and because of the inducer, the relative velocity is V 1 and this is at an angle of beta 1.

So, this is the inlet blade's speed, which is U 1 and the flow leaving the inducer could be a velocity of V 1 prime, let us say at the tip of the inducer. So, that is why is called V 1, denoted by V 1 prime, with a subscript t, which responds to the tip of the inducer.

Now, if you look at the velocity triangles, you can see, that since the blade's speed changes all the way, hub to the tip, the velocity triangle and the angles at the hub beta h, beta mean and beta tip, they are quite different, that is because of the fact, that the velocity, the blade speed continuously changes from the hub to tip. And therefore, if you take the velocity triangles at the hub, the mean and the tip, velocity triangles are different because of the fact, that blade angles are changing, which means, that there is certain twist to the inducer itself as it, as we look at the inducer from the hub and trace it all the way to the tip. So, there could be a certain amount of twist.

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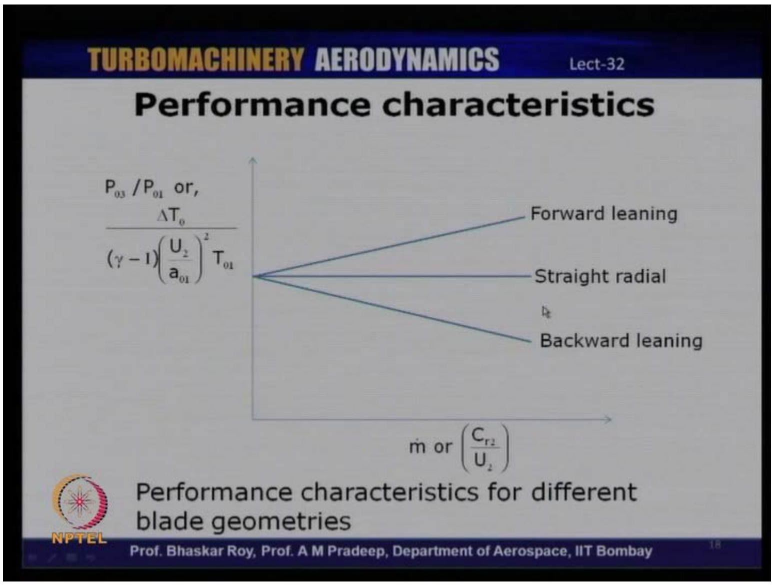

Now, from the velocity triangles we have seen, that the velocity leaving the inducer, relative velocity leaving the inducer could be, is denoted by V prime t and therefore, V prime t is a component. It is, it is related to the inlet relative velocity through the blade angle. So, V prime t is basically equal to V 1 t time cos beta 1 t and so you can see, that since, V 1 V, V prime t is equal to V 1 t times cos beta 1 t, there is a certain amount of diffusion taking place even in the inducer. That is because, V 1, V prime will always be less than V 1 because V prime t is equal to V 1 times cos beta 1 and that means, that there is a certain amount of diffusion also taking place in the inducer.

And similarly, the Mach number is related to the inlet Mach number through the blade angle beta 1. Now, the 3rd component that we will be discussing is the diffuser. So, we have seen the impeller, the inducer and now, we will talk about the diffuser.

Now, diffuser is, has a function very similar to that of a stator of an axial compressor, that is the deceleration, which was taking place in the impeller continues and that continues in the diffuser, where the flow is further decelerated. Of course, there is neither energy addition nor, of course there is a certain amount of pressure loss as I mentioned, total pressure loss taking place in the diffuser vanes, but there is static pressure rise continuously taking place in the diffuser as well.

So, the flow leaving the impeller is at a very high speed and so one can convert part of that kinetic energy into pressure rise or static pressure and that is basically through the diffuser. And the different types of diffusers, which have been used in different types of applications and there are vaned type diffusers and vaneless type pipe diffusers and channel diffusers and soon. So, we will discuss some of these aspects and different types of diffusers in the next few slides.

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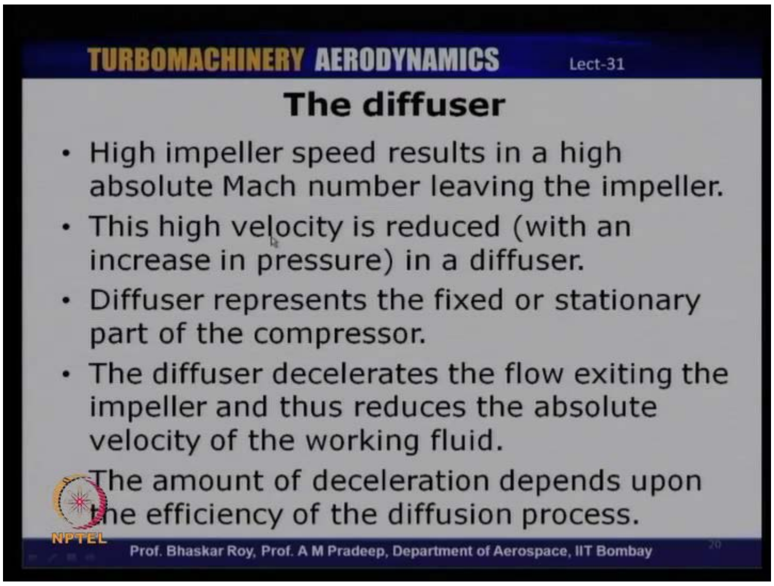

So, the high velocity, that is of the flow leaving the impeller, is basically decelerated using a diffuser and the diffuser basically decelerates the flow and thereby, it reduces the absolute velocity of the working fluid. And how much deceleration takes place depends upon, firstly, the application for which the compressor is being used, as well as the efficiency of the diffusion process itself.

And since the diffusion is, as we have discussed many times before, is a, is a process, wherein the flow has to encounter an adverse pressure gradient, the chances of or the risk of flow separation is always there when the flow encounters an adverse pressure gradient. And therefore, the amount of diffusion that one can achieve in a stationery component, like a diffuser, will depend upon how much the flow can withstand the pressure gradients.

Therefore, the diffuser flow is, is kind of always affected by or limited by the fact, that there could be chances of flow separation taking place. And so the flow basically leaves from the impeller in a radial direction and then, there is a certain space or gap between the impeller exit and the diffuser beginning, and that is called the vaneless space.

And actually, the diffuser as we will see little later, that diffusion continues even in the vaneless space and after that, the flow enters the diffuser or the vaned space and diffusion continues even further in the vaned space.

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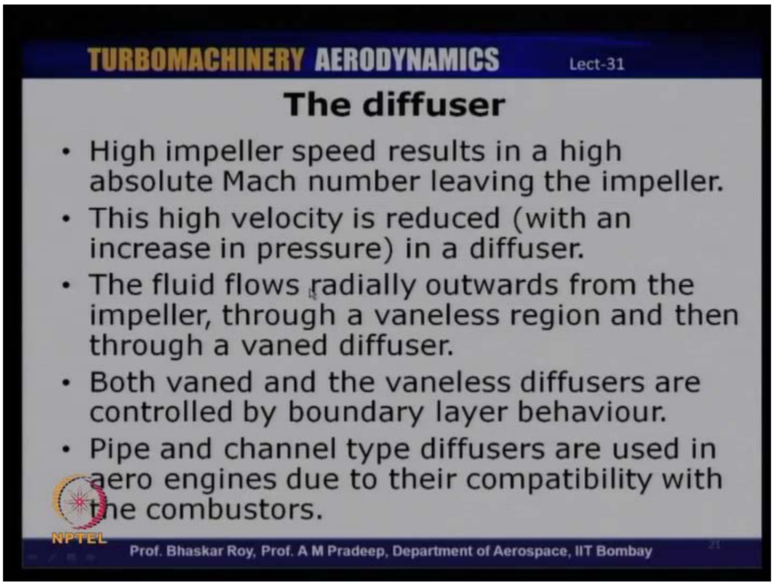

So, the fluid, which leaves the impeller, it leaves the impeller radially outward and then it passes through a vaneless region and subsequently, through a vaned diffuser. Now, there are different types of diffusers, as I mentioned, vaned type diffusers, vaneless type diffusers. These are diffusers, which are conventionally being used and if you look at compressors, centrifugal compressor, which have applications in aero-engines, these conventional types of diffusers may not release of the purpose, one need to employ better diffusion mechanisms of the flow exiting the impeller.

So, in aero-engine centrifugal compressors, one might encounter diffusers, which are known as pipe diffusers or channel diffusers, which for, which are much more efficient and more amenable to integration with the combustion chambers. For example, if you have can type combustion chambers, then it is easy for distributing the flow from these pipe diffusers and directing them towards these can type of combustors. So, these types of diffusers, which are used in aero-engines, are normally not the vane type diffusers, which are used in other applications; diffusers used in aero-engines are usually the pipe type of diffusers.

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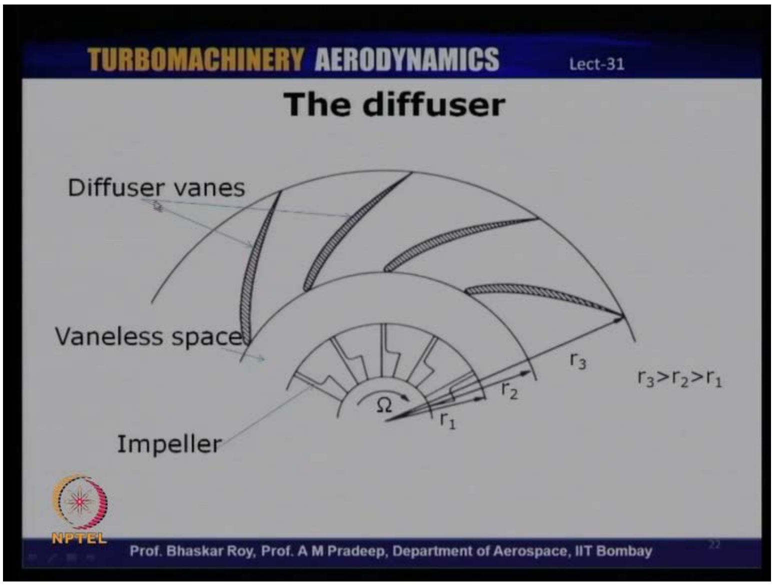

We will do now a very simple analysis of the flow through a diffuser. We will restrict our discussion to diffuser, the vane type of diffuser because it is much more simpler in understanding and analysis and we will restrict our discussion to these conventional type of vane diffusers.

So, this is the impeller vane and the flow exiting the impeller, first enters into a certain region, known as the vaneless space and then it is guided out of the compressor into the volute through the diffuser vanes.

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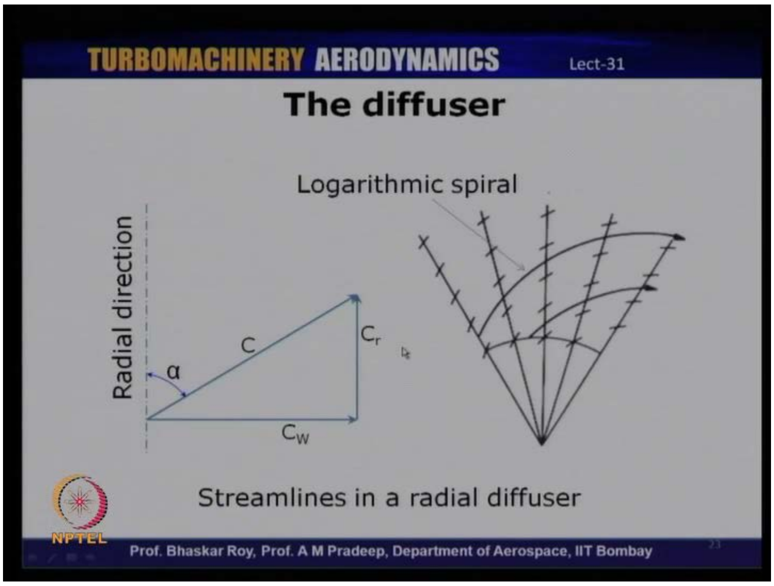

So, as the flow leaves the impeller, it basically leaves the impeller with a radial velocity and this is the radial direction as you can see. And so, the absolute velocity that actually leaves the impeller is at an angle of c, absolute velocity c. And so, as the flow leaves the impeller and it moves towards the vaned diffuser because of the velocity triangle, as you see here, the flow actually follows, what is known as logarithmic spiral because of the fact, that the flow leaves the impeller in a radial direction and then it suddenly encounters vaneless space.

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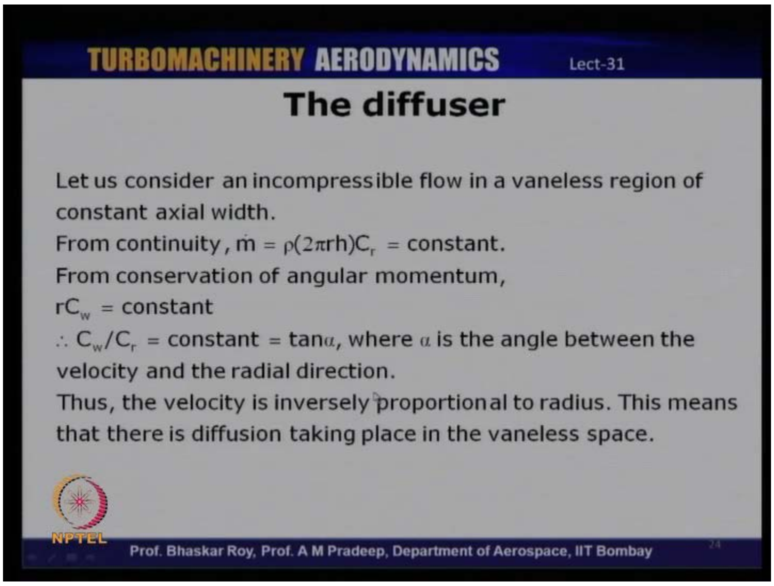

So, if you consider an incompressible, let us for the moment consider an incompressible flow in the vaneless region, which has constant axial width. The mass flow through the, through the vaneless space, let us represent that by m dot. So, from the continuity equation we have m dot is equal to rho times the angular area, that is 2 phi r h, where h is the width of the vaneless space times the radial velocity C r. So, this is equal to a constant.

Now, from the conservation of the angular momentum we also know, that r times C w is also a constant. Therefore this means that this ratio C w by c r is also a constant, that is, velocity ratio, the tangential velocity to the radial velocity. As we have seen in the velocity triangle, this is also equal to a constant and the angle is given by tan alpha. So, here, the angle alpha is basically the angle between the velocity and the radial direction, the absolute velocity leaving the impeller to the radial direction.

So, this means that this velocity is basically inversely proportional to radius and so as we increase the radius, the velocity would kind of reduce. So, in the vaneless space we have velocity, which is inversely proportional to radius and therefore, with increase in the radius, the velocity, absolute velocity actually reduces, which means, that there is a diffusion taking place even in the vaneless space. And once the flow leaves the vaneless space and enters the diffuser vanes, since the flow is guided through a diffusing passage, which has an increase in area in the radial direction, the flow continues to decelerate and there is further static pressure rise taking place even in the vaned section of the diffuser.

So, what we can see here is the fact, that the vaneless space also contributes to the diffusion overall diffusion process, which begins right from the inducer. As we have seen, that diffusion also takes places in the inducer vanes, it continuous in the impeller. Of course, in the impeller there is another mechanism, which contributes towards the pressure rise, which is the centripetal forces, which are coming there.

Once the flow leaves the impeller and enters the vaneless space, the flow continuous deceleration and that is because the velocity is inversely proportional to r and it follows a logarithmic spiral. From there, from the vaneless space, the flows enters the vaned diffuser space, where the flow is further decelerated and the flow exiting the diffuser vanes, then goes into the volute or the collector, and then it is exhausted from the centrifugal compressor. So, this is the overall working of a Centrifugal compressor.

So, what I will do is quickly take a recap of what we have discussed in today's class. We had discussed about 2 distinct aspects of centrifugal compressors. We started of our discussion today with the thermodynamics of a centrifugal compressor, where we looked at how diffusion is indeed achieved in a centrifugal compressor from a thermodynamic perspective in terms of temperature and entropy.

We have seen, that in a centrifugal compressor energy is actually added in the impeller, where, where we have an increase in the stagnation temperature, which also contributes to, which also gets converted into the stagnation pressure rise taking place in the centrifugal compressor.

After the impeller we have the vaneless space and the diffuser, where there is no energy addition taking place, but static pressure rise continuous to take place through these components as well.

So, after the temperature entropy diagram, we also wrote down the governing equations for centrifugal compressor and how one can analyze the flow through a centrifugal compressor. After this we took up the individual components of a centrifugal compressor and how one can analyze the flow through these different components.

We have seen, that impeller, there are different configurations of an impeller: the straight radial, the backward face, backward leaning blade and the forward leaning blade. I made a passing remark, that forward leaning blades are unstable. We will discuss details of that in the next class, where we take up the instabilities, etcetera in centrifugal compressor.

And then, we discussed about the flow through an impeller and the velocity triangles at the inlet of an impeller, as well as the exit, as the flow exits the impeller, and how the velocity triangle gets modified as we change the configuration of the impeller from straight radial to backward leaning blades.

We then discussed about inducer and the flow analysis of the, as the flow passes through the inducer, we have seen, that it undergoes a deceleration. And as the flow leaves the impeller, it enters into a vaneless space, diffusion continues in the vaneless space as well as in the diffuser or the vaned space of which follows the vaneless space. So, this was in a nutshell about what we had discussed in the various slides we had for discussion today.

And we will continue with discussion of different aspects of centrifugal compressors in the next class as well. We will take up important aspect of centrifugal compressors, that is to do with Coriolis acceleration first and then we will define, what is known as the slip factor associated with the centrifugal compressor.

After that, we will take up the performance characteristics of centrifugal compressors and we will also talk about details regarding stall, surge and choking associated with centrifugal compressors.

So, we will take up these, some of these topics for detailed discussion in the next class on centrifugal compressors. Subsequent to that of course, we will have one session, which would be a tutorial on centrifugal compressors. We will solve some problems, which are related to centrifugal compressors and we will also have some exercise problems for you, which you can solve based on our discussions.

So, in the next class we will basically have these topics for discussion. We will talk about Coriolis acceleration, the slip factor, the performance characteristics and stall, surge and choking associated with centrifugal compressors. So, we will take up these topics for discussion in the next class, which would be lecture number 32.