Refrigeration and Air conditioning
Prof. M Ramgopal
Department of Mechanical Engineering
Indian Institute of Technology, Kharagpur

Lecture No. # 25
Refrigeration System Components Compressor (Contd.)

Welcome back. This lecture is the continuation of the earlier lecture wherein i introduced centrifugal compressors. In this lecture, I shall discuss the performance aspects of centrifugal compressors and briefly discuss the commercial centrifugal machines okay.

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The specific objectives of this particular lecture are to discuss refrigerant capacity of centrifugal compressors and how to control the refrigerant capacity and discuss of a phenomenon of surging in centrifugal compressors, discuss the effects of operating per temperatures and speed on performance comparison with reciprocating compressors and finally discuss briefly commercial systems with centrifugal compressors.

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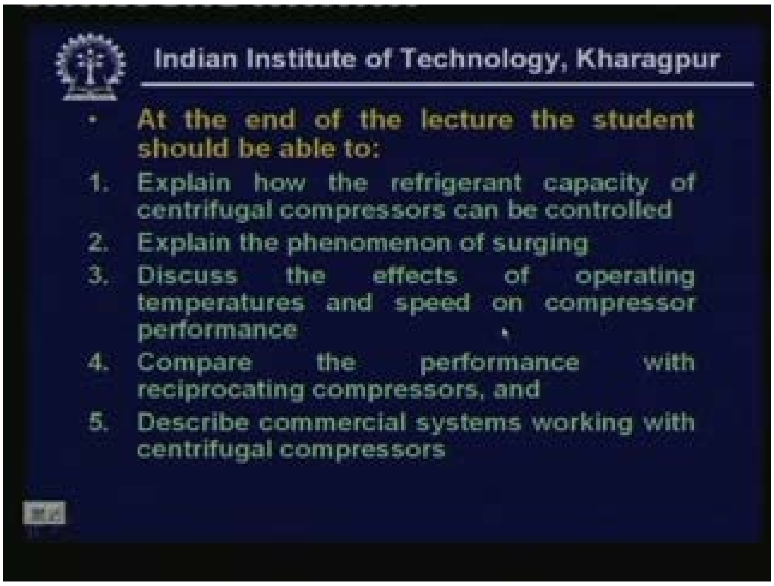

At the end of this lecture you should be able to explain how the refrigerant capacity of centrifugal compressors can be controlled explain the phenomenon of surging, discuss the effects of operating temperatures and speed on compressor performance, compare the performance with reciprocating compressors and finally describe commercial systems working with centrifugal compressors.

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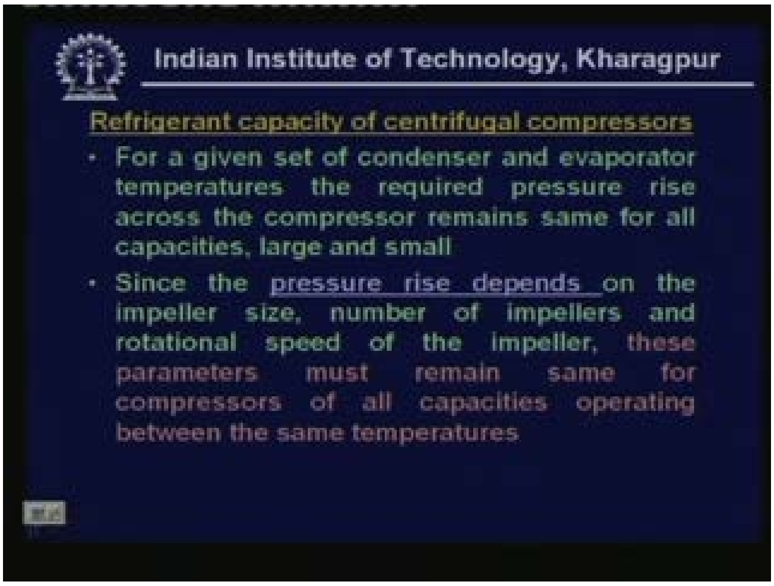

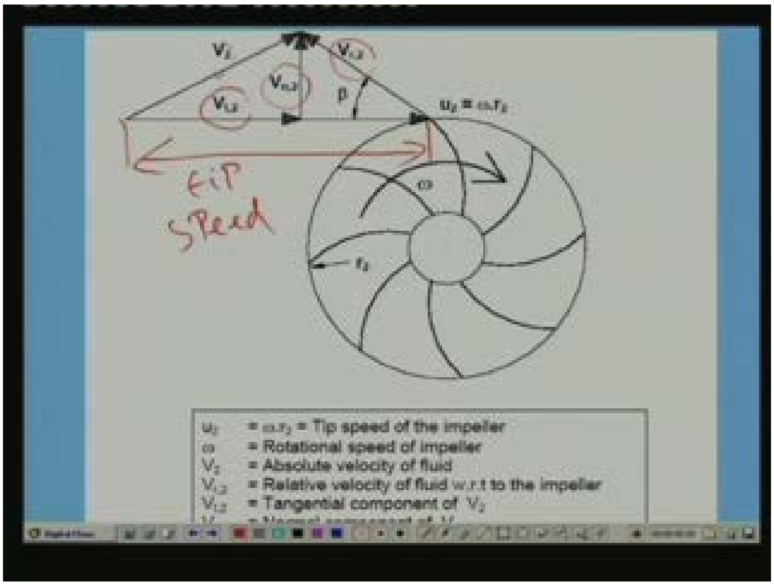

Now let us look at the refrigerant capacity of centrifugal compressors for a given set of condenser and evaporator temperatures. The required pressure rise across the compressor remains same for all capacities large and small. That means if you fix that evaporator and condenser temperatures obviously evaporator pressure and condenser pressure are fixed. So the pressure difference between the across the compressor also remains fixed. This is irrespective of the capacity. That means if you have a system working between zero degrees evaporator temperature and thirty degrees condenser temperature. So what ever be the capacity whether it is hundred watts or hundred kilo watts the pressure ratio and the pressure difference remains fixed irrespective of the capacity. This holds good for any type of compressor okay.

But there is a small problem when it comes to centrifugal compressors. Let us look at that we have seen that in centrifugal compressors the pressure rise depends on the impeller size number of impellers and rotational speed of the impeller. Since these parameters must remain same for compressors of all capacities operating between the same temperatures okay. Now let me explain this okay. So this is the picture I have shown you in the last lecture. We have the velocity diagram here and as I have explained you have the, this u two is your tip speed. This is the tip speed and i have defined all these things and in fact these things are again repeated here. Vt two is the tangential component of refrigerant at the exit of the compressor Vn two is the normal component and Vr two is the relative velocity and V two is the absolute velocity okay. This is what we have discussed in the last class and from by assuming that the inlet is axial.

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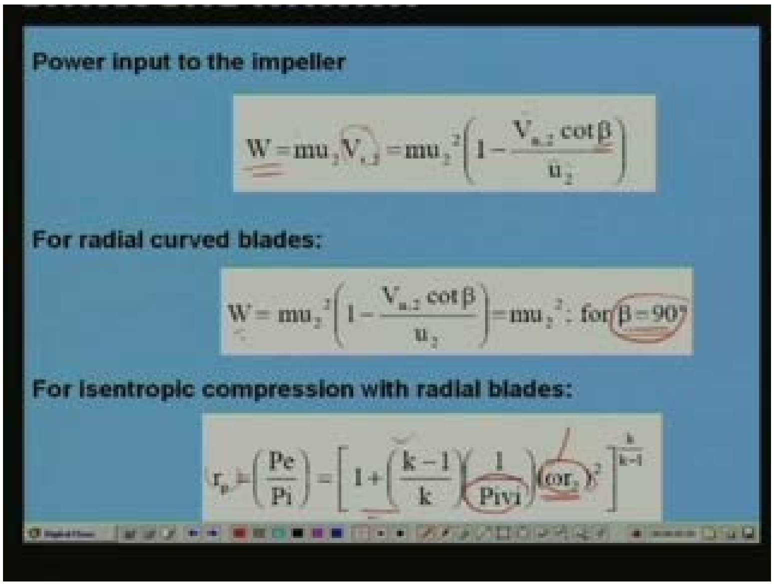

We have got these expressions, for example for an axial inlet. That means there is no tangential component at the inlet to the compressor. We found that the work input or power input to the compressor is given by this expression where m dot is the mass flow rate. u two is the tip speed. And Vt two is the tangential velocity and from the velocity diagram what we have done is, we have replaced this Vt two. And we have written this in terms of the tip speed u two and the normal component Vn two and the blade angle beta okay. And then we have extended this and we have derived the equation for an impeller having radial curved blades. And in fact in I have explained in the last class that for radial curved blades the angle beta is equal to ninety degrees okay.

So once the angle beta becomes ninety degrees you can see that the power input is equal to m dot into u two square or the specific power input or specific compressor work is simply equal to u tow square okay. And for isentropic compression with radial blades we have derived this equation which gives the ratio of this exit pressure to the inlet pressure. As a function of the inlet parameters pi vi isentropic coefficient k and the tip speed omega into r two square. And in fact in the last class I have explained that when you fix the inlet condition. That means Pi and vi are fixed and if we also assume that k is constant you can see that the pressure ratio is purely a function of your tip speed of the impeller. That means tips speed in turn depends upon the rotational speed omega and the outer radius r two okay. That means from this expression you see that once you fix the pressure difference or the pressure ratio. Then the required tip speed gets fixed irrespective of the capacity does not come into picture okay.

This is the typical characteristic of a centrifugal compressor. Let us see what is the consequence of this that's what is mentioned here. So if you want to design refrigerant system for a lower capacity what is that we have to do. First of all let us look at the expression for mass flow rate the mass flow rate of refrigerant is given by this expression where m m is the mass flow rate.

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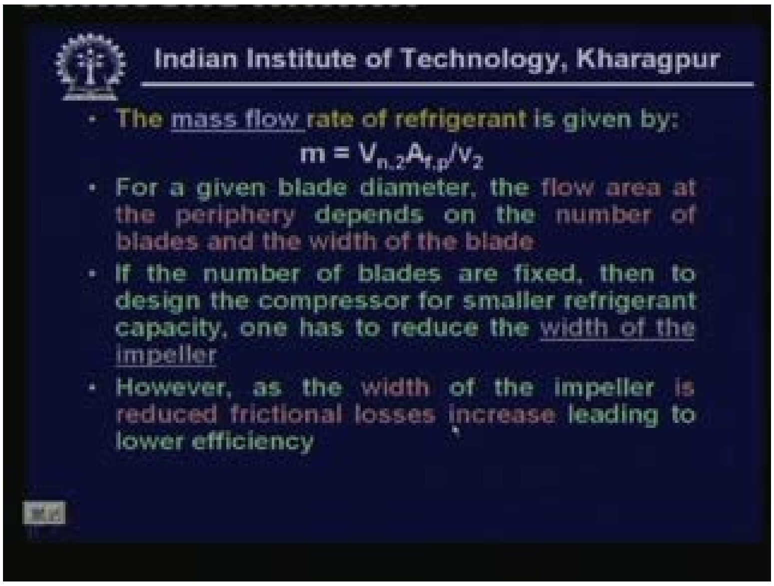

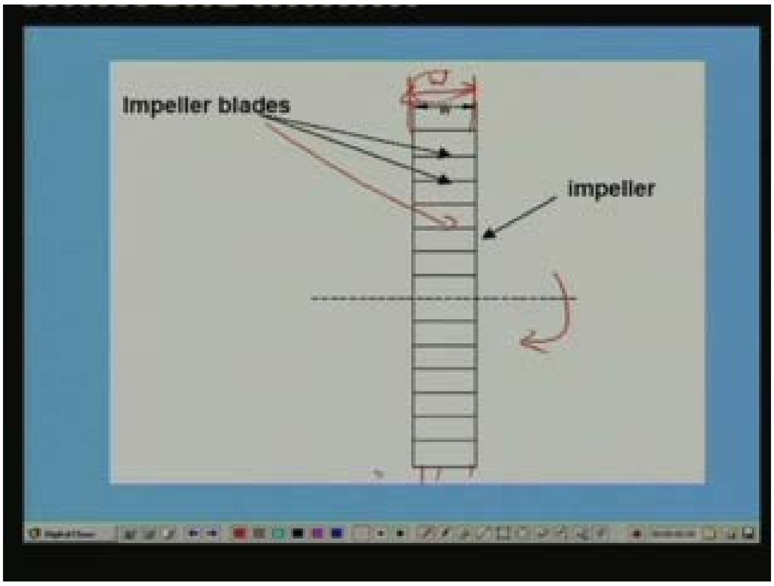

This is equal to Vn two into Afp divided by v two where Vn two. As I have already explained is a normal component of refrigerant velocity at the exit of the impeller. And Afp is the peripheral area peripheral flow area okay. And small v two is a specific volume of the refrigerant at the exit. So you can see that the mass flow rate depends upon the normal component area of cross section at the periphery and the specific volume. Suppose you are fixing the conditions at the exit then the specific volume v tow gets fixed okay. Then mass flow rate depends only on the normal velocity and also on the peripheral flow area okay. The ratio of the normal velocity two the tip speed is sometimes called as a flow coefficient okay. And this flow coefficient is normally obtained from measurements. Once you say that the flow coefficient is this and if you know the tip speed then you can calculate what is the normal component of velocity okay. And when if you know the flow area then you can calculate the mass flow rate right.

Now let us see how the capacity is has to be reduced now for a given blade diameter the flow area at the periphery depends on the number of blades and the width of the blade oaky. And if the number of blades are fixed then to design the compressor for smaller refrigerant capacity one has to reduce the width of the impeller. What do we mean by designing the refrigeration system for smaller capacities when everything else is fixed the smaller capacity means smaller mass flow rate okay. So if you have to design the system for a smaller refrigeration capacity. You have to reduce the mass flow rate of refrigerant okay. And from this expression you see that if you have to reduce the mass flow rate of refrigerant. Either you have to reduce the normal velocity at the exit or you have to reduce the flow area okay.

Now let us look at the flow area flow area depends on what for a given impeller diameter the flow area depends upon the number of blades. It is less sense to the number of blades but it is it mainly depends on the width of the blade. So if you want to reduce the flow area you have to reduce the width of the blade okay. Let me show the blade from a side view. So this is the impeller and this is the shaft and it is rotating. Let us say in this direction. So this, the, is the width of the blade okay, width of the impeller W okay. And these are the impeller blades as I have shown all these things. So if you want to reduce the flow area you have to reduce this W that means these things become narrower okay. Now what is the problem if you reduce the width. You find that as the width of the impeller is reduced frictional losses increase leading to lower efficiency. That means when you are forcing the refrigerant to flow through a narrow passage obviously frictional losses increase okay. So if you are if you are trying to a design a system for the same for the small refrigerant capacity you have to keep the tip speed constant.

If you are keeping the rotational speed constant that means at the diameter of the impeller is fixed. So only option left for you is to reduce the of course is not the only option. But one option left for you is to reduce the width and we have seen that as you reduce the width the frictional losses increase. Once the frictional losses increase the efficiency of the compressor comes down okay. This is the one of the problems in reducing the capacity of a centrifugal compressor okay. Now let us look at other options.

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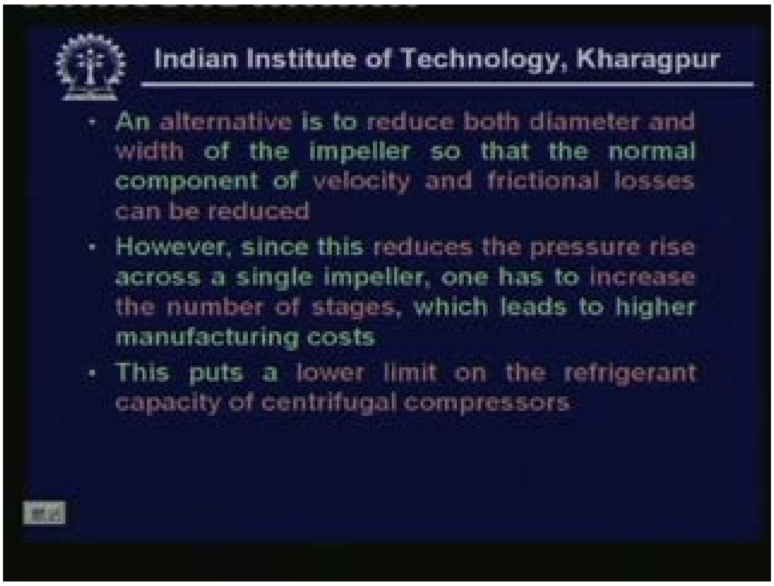

What are the other options an alternative is to reduce both diameter and width of the impeller simultaneously. So that the normal component of velocity and frictional losses can be reduced okay. If you want to reduce the frictional losses what you have to do is you have to reduce the length of travel okay. One way of reducing the frictional losses is to reduce the length of travel. That means you have to reduce the diameter of the impeller okay. That means you have to simultaneously reduce the diameter of the impeller and width of the impeller. So that the mass flow rate gets reduced okay but what is the problem with this. Once you reduce the diameter of the impeller and if you are keeping the rotational speed constant. Then your tip speed reduces once you once the tip speed is reduced the pressure rise across the impeller reduces okay.

Since the total required pressure rise remains constant and the pressure rise across a single impeller reduces. You have to go for multistage compressors okay. That means multi more number of impellers this is the problem with this method okay. So that is what is summarised here since this reduces the pressure rise across a single impeller one has to increases the number of stages what is the problem of increasing number of stages. If you increase the number of stages the manufacturing cost increases. That means the initial cost also goes up okay. So this is the disadvantage with this method thus you find that for a centrifugal compressor a lower limit is there on the refrigerant capacity of on the refrigerant capacity okay.

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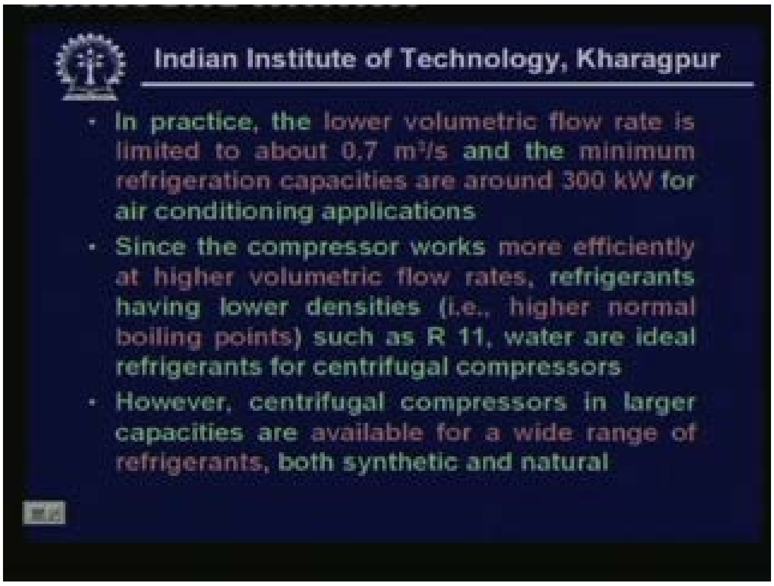

And in practice the lower volumetric flow rate is limited to about point seven metre cube per second and the minimum refrigeration capacities are around three hundred kilo watt for air conditioning application. That means you cannot really have a centrifugal compressor for let us say five kilo watt or ten kilo watt or something like that. Because that becomes to inefficient it not that you cannot design it but the compressor will be very inefficient okay. So if you want to have good efficiency you must maintain certain things at a minimum level then the capacity automatically the lower limit gets automatically fixed okay. So that is the reason why centrifugal compressors are normally used for high capacity applications okay. Since the compressor works more efficiently at higher volumetric flow rates refrigerants having lower densities. That means higher normal boiling points such as R eleven are ideal refrigerants for centrifugal compressors.

So if you want to have see the when you are fixing the tip speed or when you are fixing the by fixing the tip speed you are fixing the let us say the normal component okay. That means you are fixing the volumetric capacity okay. So the mass flow rate is nothing but the product of volumetric capacity and the density okay. If the density is less for the same volumetric capacity you can volumetric flow rate you can have less refrigerant flow rate okay. So for centrifugal compressors if the volumetric flow rate is large then you can go for wider impellers. That means you can reduce the friction losses okay. So ideally we should like to we would like to have as high volumetric flow rate as possible at the same time you would like to have refrigerant capacity small. This is possible only when the density of refrigerant at the impeller exit is small okay.

That means the pressure should be small this is typically possible with refrigerants having high normal boiling point okay. One typical example is R eleven R eleven has the boiling point of about twenty-four degree centigrade. That means it is density will be very low even at forty degrees or fifty degree centigrade okay. So it is an ideal refrigerant as far as centrifugal compressors are concerned. That is the reason why R eleven was very widely used in all centrifugal compressor based air conditioning systems okay. However unfortunately R eleven was banned due to its high ODP okay. So now you cannot have R eleven of course. It does not mean that you cannot have or you do not have centrifugal compressors. If you are working with a wide variety of refrigerant they are available okay. But of course some of them are better than the others okay. So as I said centrifugal compressors in larger capacities are available for a wide range of refrigerants both synthetic as well as natural.

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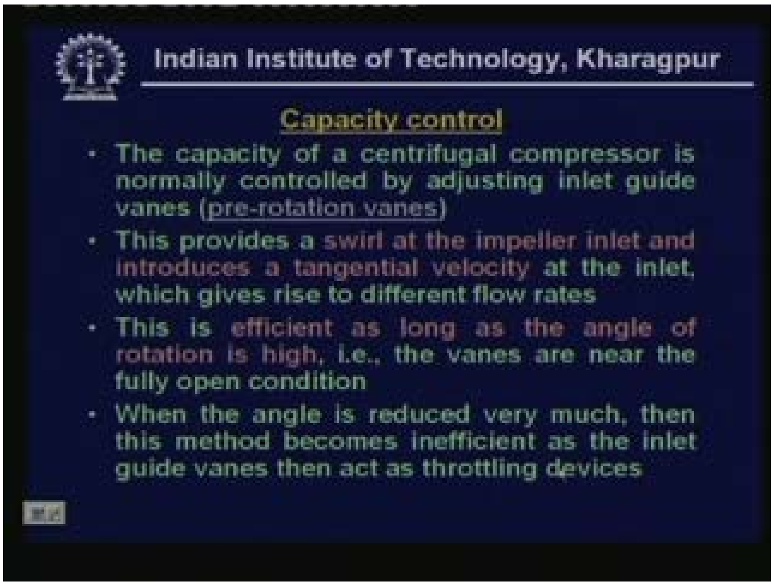

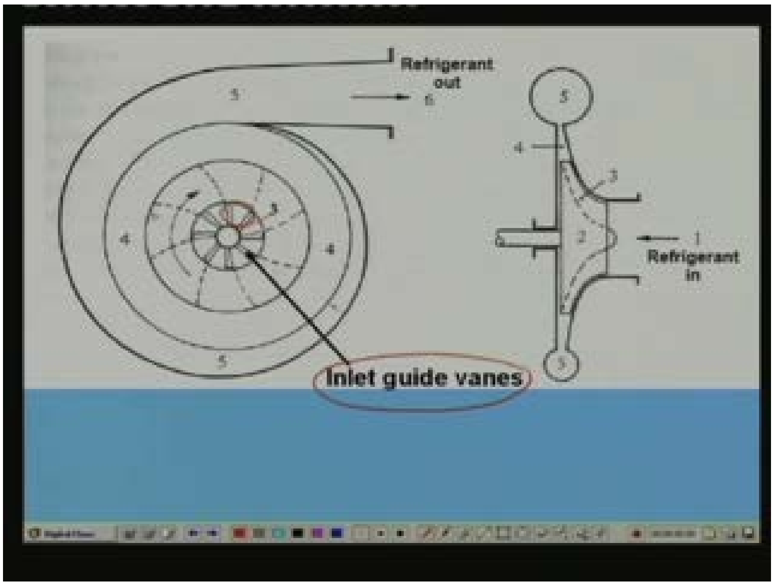

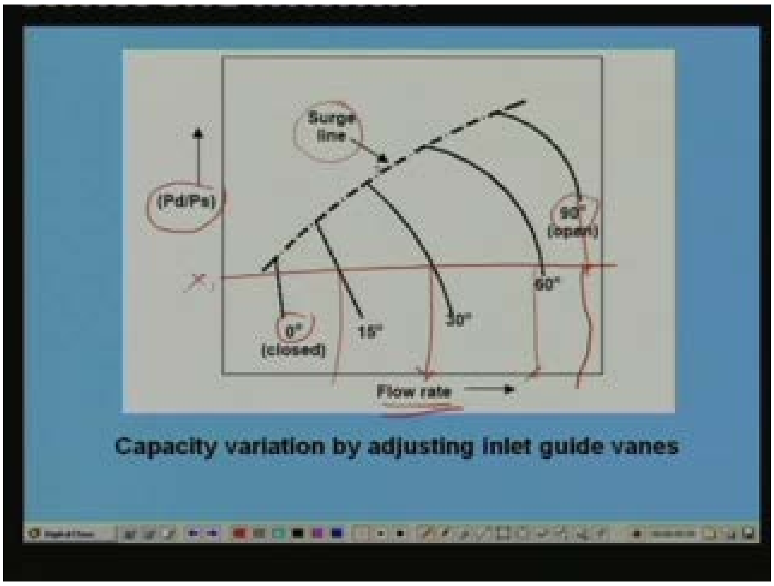

Now let us look at capacity control how do we control the capacity the capacity of a centrifugal compressor is normally controlled by adjusting inlet guide vanes. In fact in the last class I have explained what do we mean by inlet guide vanes. And I have also mentioned that these are mainly used for controlling the capacity so let us look at this okay. So this is the, this figure I have shown in the last lecture. So as I have already mentioned these are the inlet guide vanes inlet guide vanes and the through which the refrigerant enters okay. Then it takes a ninety degree turn and flows through the impeller okay. So when you have the guide vanes here by adjusting the guide vanes what you can do is you can introduce a tangential component at the inlet okay.

In our analysis we assume that the tangential component is zero and the we have, that means the refrigerant enters axially at the inlet okay. And that is how we calculated the pressure rise and specific work input and all that okay. But it is possible to introduce a tangential component by adjusting the inlet guide vanes what happens when you adjust the inlet guide vanes. As I said it introduces a tangential component and once you introduce a tangential component the pressure rise across the impeller the mass flow rate everything changes okay. So you can control all these parameters by controlling the angle at which the guide vanes are kept inside the inlet okay. This is the one very popular way of controlling the capacity of centrifugal compressors. Now let me show a typical performance curves at different inlet guide vane angles okay.

This figure shows the pressure ratio okay, versus flow rate for different inlet guide vane angle. You can see that the angle is given as ninety degrees it varies from ninety degrees to zero degrees. When you say ninety degrees that means the inlet guide vanes are fully open okay. At this condition you get the maximum capacity for example if i am fixing the let us say the pressure ratio okay. Pressure ratio is fixed at his point let us say then you can see that as you are increasing the angle you are getting higher flow rate higher flow rate means higher capacity okay. So as you are reducing the angle that means as you are closing the guide vanes the flow rate is getting reduced while the pressure ratio is staying constant okay. So that is how you control the capacity.

And as I said this is the very popular method of controlling the capacity and here you can see that there is a surge line okay. I will explain what is surging in this class normally a performance is not shown beyond the surge line okay. So as I have already mentioned the adjusting of inlet guide vanes provides a pherl at the impeller inlet and introduces a tangential velocity at the inlet which gives rise to different flow rates. This method is efficient as long as the angle of rotation is high i the vanes are near the fully open condition okay. It is not that is a very efficient method under all conditions. The method of controlling the capacity by adjusting the inlet guide vanes is good as long as you are not closing the guide vanes too much.

That means it is staying near the fully open condition that means the angle is closed to ninety degrees not to zero degrees okay. And if you are trying to reduce the capacity too much by closing the inlet guide vanes too much. That means when your angle is closer to zero. That means the inlet guide vanes will be simply acting as throttling devices okay. They will be simply pressure deduction across the guide vanes and then the method becomes inefficient okay. So if you want to use this method you should use it within certain limits. So that you do not close the vanes too much okay.

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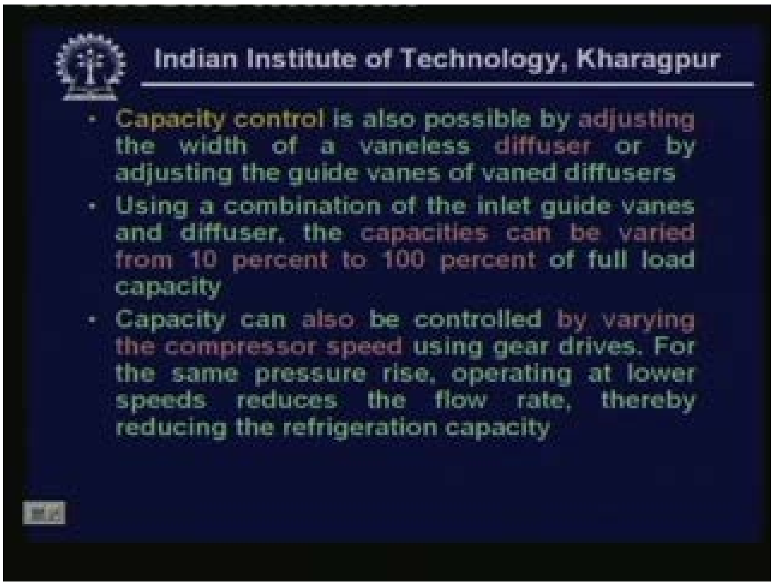

Capacity control is also possible by adjusting the width of the vaneless diffuser or by adjusting the guide vanes of vaned diffusers in the last class I have explained what is the function of a diffuser. And I have already I have also mentioned that you can have either a vaned diffuser or vaneless diffuser okay. And by adjusting the width of the vaneless diffuser or by the orientation of the vaned diffusers you can control the flow rate. Normally you do not use it alone. Normally the method the method of capacity control by the vaned diffuser is clubbed with the method of capacity control by the inlet guide vanes okay. So they are used in combination so using a combination of the inlet guide vanes. And diffusers the capacities can be varied smoothly from ten percent to hundred percent of the full load capacity okay. So you can get a very wide range of capacities by this methods capacity can also be controlled by varying the compressor speed using gear drives for the same pressure rise operating at lower speeds reduces the flow rate thereby reducing the refrigeration capacity. Of course keeping everything constant.

If you reduce the speed then obviously the all the velocities will get reduced. So the flow rate automatically gets reduced okay. So this is another good method of controlling the capacity. That means you are varying the speed okay. Of course when you are varying the speed it may also affect the pressure rise okay. So it is not that you can vary it over any range okay. Within certain range you can vary it right. So to sum up the most important methods of capacity control as for a centrifugal compressors are concerned are adjusting the inlet guide vanes and adjusting inlet guide vanes plus adjusting the diffuser or speed control okay.

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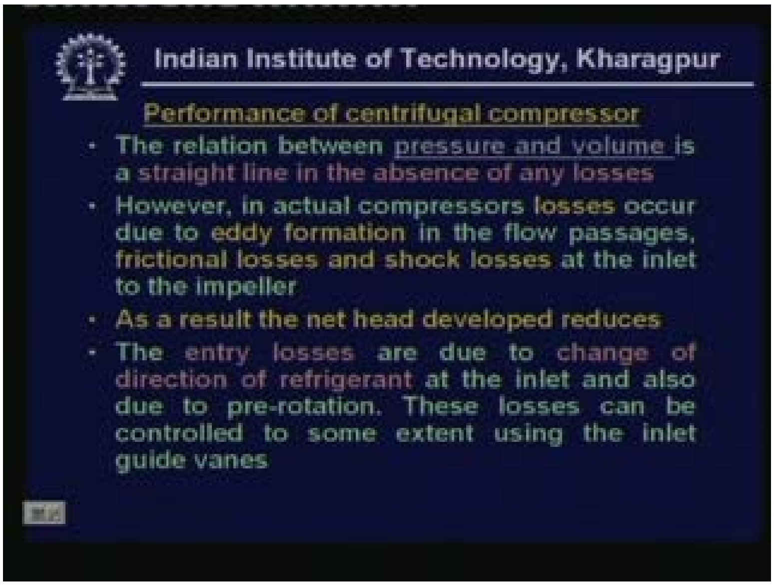

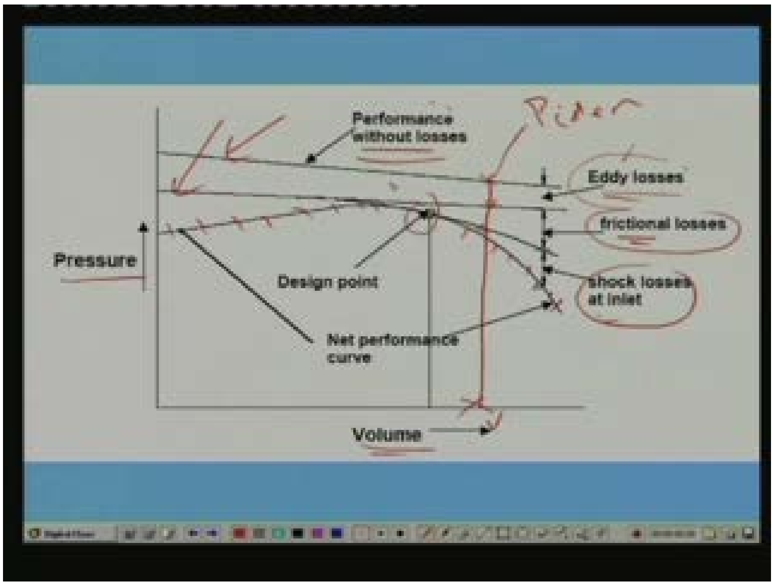

Now let us look at performance aspects of centrifugal compressors the relationship between pressure and volume is a straight line in the absence of any losses. That means for an ideal centrifugal compressor if you plot the pressure rise versus the flow rate you find that you get a straight line. That means I will show you that as the flow rate increases the pressure rise reduces or as the flow rate reduces the pressure rise increases this characteristic is typical of any dynamic machines such as centrifugal pumps or centrifugal compressors okay. Let me show the characteristics okay. You can see the, I have plotted pressure. That means the exit pressure versus the volume okay. So as I was mentioning this is the performance curve this curve without any losses. That means that of an ideal centrifugal compressor. Here you can see that the ideal centrifugal compressor the performance curve is the straight line okay. And you can see the as the volume is reducing pressure is as the volume is increasing, sorry, the pressure is reducing and as the volume increases the pressure increases okay. Now in actual compressors there are several losses.

So if you include the losses how does the performance characteristic look like for example what are the losses in a centrifugal compressor? I have explained in the last class one loss is the eddy loss okay. And I have explained in the last class that the eddy loss is because of your secondary flows okay. The coriolis acceleration component and due to the secondary flows you have eddy losses. And this eddy lossa also almost varies linearly. So when you include this eddy loss the head reduces okay. So this is the curve which includes the head losses I mean, I am sorry includes the frictional losses. So the only next loss is to be considered is the frictional loss. If you are considering the frictional loss the frictional losses do not vary linearly but they vary as a power of volume or as the power of velocity. So you can see that as the volumetric flow rate is increasing the frictional pressure losses are increasing. So when you include eddy loss as well as frictional loss you find that the head developed or the pressure gets reduced further.

So for example let me give a, let us say that this is the volume at which I am trying to find out the pressure. So without any losses this is the pressure rise okay. This is the ideal pressure rise. So if you are including eddy losses this will be the pressure developed. And if you are, including the frictional losses this will be the pressure developed. So the pressure developed gets reduced because of the losses. Finally if you also include the shock losses at the inlet this will be the resulting curve okay. So finally the curve resulting curve is this okay. So an ideal curve is a straight line like this. But the resulting curve the pressure increases initially with the volume and it reaches a peak and then again reduces okay. And the as I said the major three major losses are shock losses frictional losses and eddy losses okay. And normally this is the point called as the design point. How do you fix a design point? Design point is normally fixed in such a way that at that point the total losses are minimum okay. You would like to operate the you would like to operate the system always at the highest efficiency okay. That means where the losses are minimal that is how the design point is fixed okay. So as I have already mentioned if you in actual compressors losses occur due to eddy formation in the flow passages frictional losses and shock losses at the inlet to the impeller. And the shock losses or entry losses are due to change of direction of refrigerant at the inlet. And also due to pre-rotation theses losses can be controlled to some extent using the inlet guide vanes. So I have explained the meaning of eddy losses in the last class. And I have also explained the meaning of frictional losses okay, due to viscous shear.

I did not explain what is an inlet loss okay. The inlet loss mainly by happens because as soon as the refrigerant enters into the impeller through the inlet it has to take a ninety degree turn. That means there is a sudden change in the direction okay. Whenever there is a sudden change in the direction of the fluid as it is flowing there will be losses okay. This is one of the reason for the inlet losses. Apart from that when you have the inlet guide vanes they also introduce some losses okay. So mainly the inlet losses are as the result of the directional change and due to the presence of the guide vanes okay. Of course you can minimize them by adjusting the guide vanes properly okay, this I have already explained.

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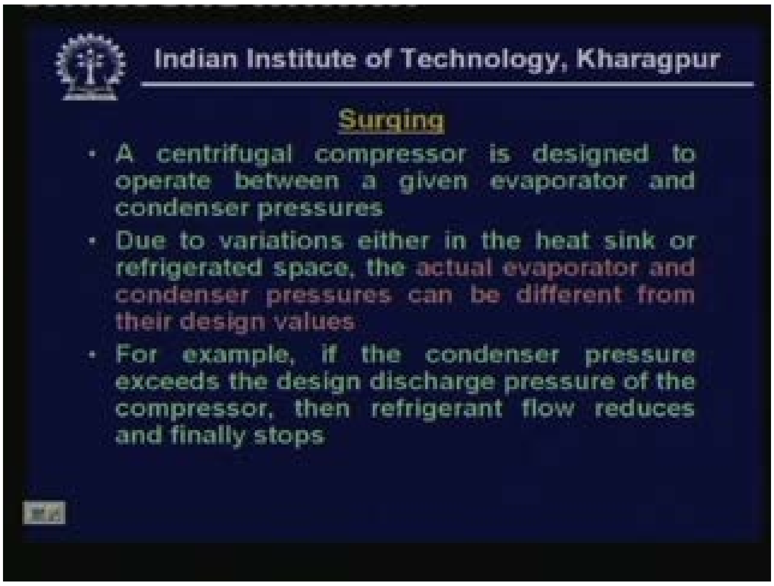

Now before we go to other performance characteristic curves I would like to explain an important phenomenon called surging okay. What do you mean by surging? This is very important in centrifugal compressors. So let me explain this. A centrifugal compressor is designed to operate between a given evaporator and condenser pressures due to variations either in the heat sink or refrigerated space. The actual evaporator and condenser pressures can be different from their design values. For example if the condenser pressure exceeds the design discharge pressure of the compressor. Then refrigerant flow reduces and finally stops okay. So let me explain this normally whenever you design a system you always design it for a design condenser and evaporator temperatures are pressures okay. You can fix either temperature then automatically pressure gets fixed okay, whether it is a reciprocating compressor or a centrifugal compressor.

So will I, since we are discussing centrifugal compressor let us specifically discuss what is the problem is centrifugal compressors okay. So when you design a centrifugal compressor for a fixed evaporator and condenser temperature and you choose the impeller speed, impeller diameter and the width of the impeller and all that right based on the design points. Now it is not necessary that the evaporator temperature and condenser temperature have to stay always at the same designed design conditions okay. Either evaporator temperature can reduce or increase or the condenser temperature can also increase or reduce depending upon the conditions okay. The evaporator temperature can reduce when the load is falling. That means the refrigeration load is falling or it can increase when the refrigeration load is increasing. Similarly the condenser temperature can increase when the heat sink temperature is increasing or the heat transfer rate on the in the condenser is reducing okay. So due to these reasons u there are many I mean there are many possibilities where the system will be working at off design conditions okay.

That means that pressure ratio or the pressure difference will be different from the design pressure difference okay. When the system is operating at off design conditions you will find that the centrifugal compressor behaves in a different manner compared to reciprocating compressor okay. Let me give an example of the variation of condenser temperature. Let us say that the heat sink temperature has increased okay. And as you know the heat sink normally depends upon the ambient conditions okay. If the ambient temperature goes up, let us say in summer then heat sink temperature increases, once heat sink temperature increases.

Since the condenser is fixed the condenser temperature increases condenser pressure also increases okay. So the condenser actual condenser pressure will be higher than the design condenser pressure.

Once this is higher than the design condenser pressure you find that the centrifugal compressor you will find it increasingly difficult to maintain the same flow rate okay. So as soon as the condenser pressure starts going beyond the design condenser pressure for a given evaporator pressure the flow rates starts reducing okay. And a point will come at which the flow rate completely stops and any further rise in condenser the flow rate i the flow actually reverses. That means instead of flowing from the evaporator to the condenser the flow will take place from the condenser to the evaporator okay. So reversal of flow takes place when the condenser goes beyond a certain value okay. Let us see what happens because of this. So as I said further increase in condenser pressure causes a reverse flow of refrigerant.

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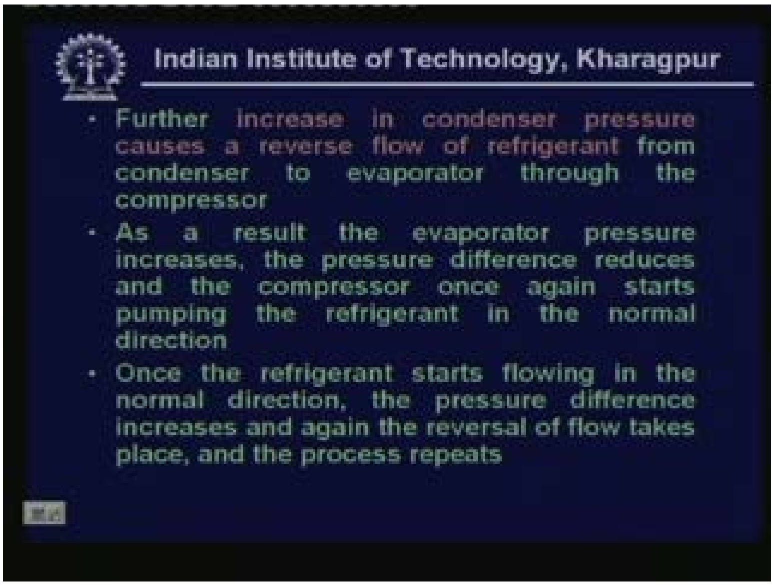

From condenser to evaporator through the compressor understand clearly that why this happens this happens because unlike reciprocating compressor in a centrifugal compressor the pressure rise is almost fixed when you are fixing the tip speed okay. That is what we have discussed and that’s what we have seen from the equations okay. So as long as you are running the compressor at the same speed since the diameter is already fixed. That means the pressure developed across the impeller remains same. So it cannot increase just because the condenser pressure is increasing okay. So the pressure developed remains mainly a function of the tip speed. So if you are keeping the tip speed constant pressure developed also remains constant okay. Since it cannot develop a pressure higher than the condenser pressure then obviously due to the pressure radiance the flow will take place in the reverse direction. That means a hot refrigerant from the from the condenser enters the evaporator through the compressor okay.

Since you do not have any valves or anything it can freely flow through the compressor and it can go to the evaporator okay. Now if you compare this with what happens in reciprocating compressor. What happens in a reciprocating compressor when the condenser pressure is increased. So if you remember the working principle of a reciprocating compressor in reciprocating compressor we have suction and discharge valves okay. So when the condenser pressure is increasing the discharge valve will not open okay. So what happens is the pressure inside the cylinder also has to increase okay. That means that a opening of the discharge valve is delayed due to higher condenser pressure okay. And only when the pressure inside the cylinder goes beyond the condenser pressure. Then the discharge valve opens and flow takes it takes place in the normal direction there is no way flow reversal can take place. Because as long as the discharge valve is closed flow cannot take place from the condenser to the evaporator okay.

Because you have the valves there whereas in centrifugal compressor we do not have any valves okay. So reversal of flow can always take place and since this is not a positive displacement type of machine. It cannot go on pumping even though the pressure is increasing okay. So to put it simply a reciprocating compressor can work or can go on compressing the refrigerant and go on pumping the refrigerant at a lower efficiencies but it can still pump a finite amount of refrigerant okay. In the normal direction whereas the centrifugal compressor beyond a certain point it cannot simply pump the refrigerant okay. So it will stop pumping and in fact the reversal of flow takes place okay. So this is a typical characteristic of a dynamic machine now what is the problem because of this. As a result of this the evaporator pressure increases. Now what happens when reversal of flow takes place and hot refrigerant from the condenser enters into the evaporator. That means you are adding load to the evaporator artificially right.

That means the load on the evaporator increases once the load on the evaporator increases evaporator temperature increases. Once the evaporator temperature increases evaporator pressure increases. Once the evaporator pressure increases the pressure difference across the evaporator and condenser reduces okay. Once it reduces the centrifugal compressor can now again pump in the normal direction okay. So pumping starts to take place again in the normal direction okay. So again refrigerant flows from the evaporator to the condenser. Once refrigerant starts flowing from the evaporator to condenser again the evaporator pressure drops okay, again the pressure difference increases. Once the pressure difference increases again the compressor will find it difficult to pump in the normal direction, reversal of flow takes place and again this cycle continues okay. This phenomenon is known as surging right okay. So as I said once the refrigerant starts flowing in the normal direction the pressure difference increases and again the reversal of flow takes place and the process repeats.

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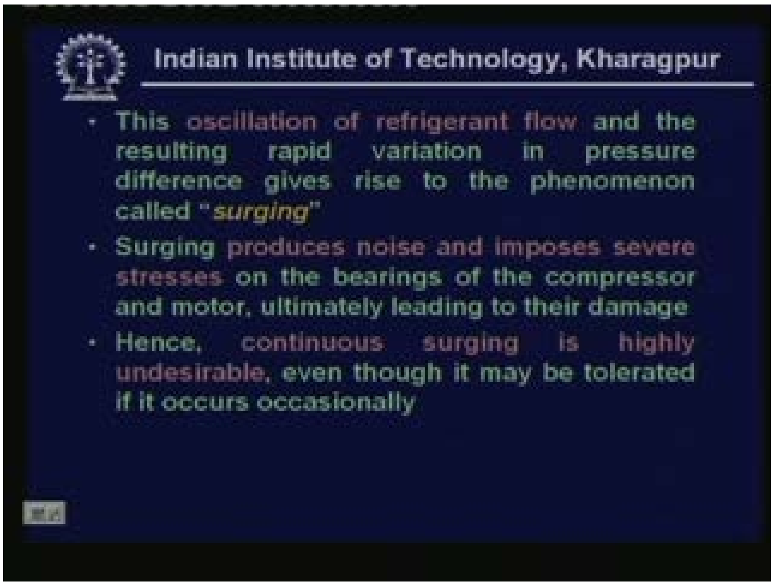

This oscillation of refrigerant flow and the resulting rapid variation in pressure difference gives rise to the phenomenon called surging produces noise and imposes severe stresses on the bearings of the compressor and motor ultimately leading to their damage okay. So because surging, you find that there is an oscillation of refrigerant flow from evaporator to condenser and then condenser to evaporator. So there is a rapid oscillation and the, there is rapid variation of pressures okay. Due to these rapid changes very objectionable noise is produced okay. That is not the only effect since the pressure variation is very rapid they will be severe stresses on the bearings of the motor and on the compressor. Ultimately if nothing is done the motor and compressor get damaged okay. So this is a very important problem which has to be taken care in actual systems. So continuous surging is highly undesirable even though it may be tolerated if it occurs occasionally.

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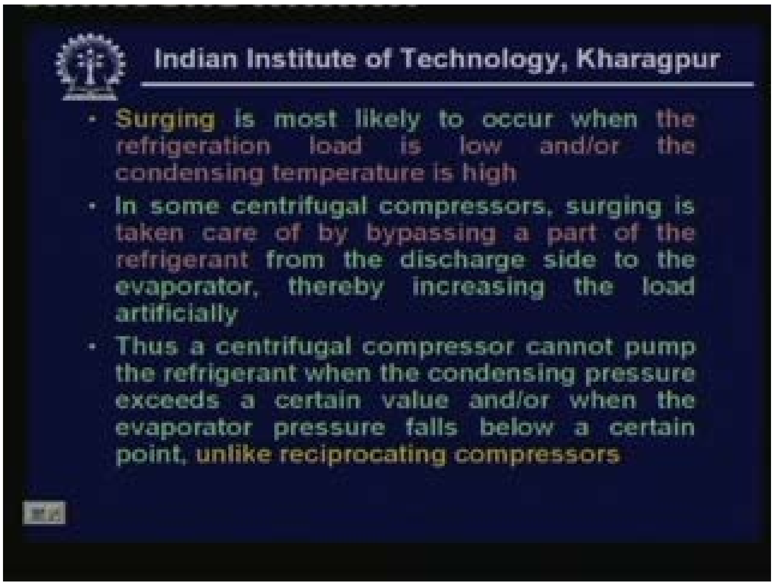

Now as I have already explained it must be clear to you that surging is most likely to occur when the refrigeration load is low and or the condensing temperature is high okay. So surging takes place when the pressure difference across the compressor is high. So when the pressure difference will be high either when either the condensing temperature pressure is high or the evaporator pressure is low okay. Evaporator pressure will be low when the load is low okay. So you find that surging will normally occur when the heat sink temperature is very high and refrigerant refrigeration load is low okay. So this is the most favourable condition for surging in some centrifugal compressors surging is taken care of by bypassing a part of the refrigerant from the discharge side to the evaporator thereby increasing the load artificially okay. So something hot gas bypass is done so that you artificially increase the evaporator capacity. That means you increase the evaporator pressure and reduce the pressure difference and avoids surging thus a centrifugal compressor cannot pump the refrigerant when the condensing pressure exceeds a certain value. And or when the evaporator pressure falls below a certain point unlike reciprocating compressors, I have explained this already.

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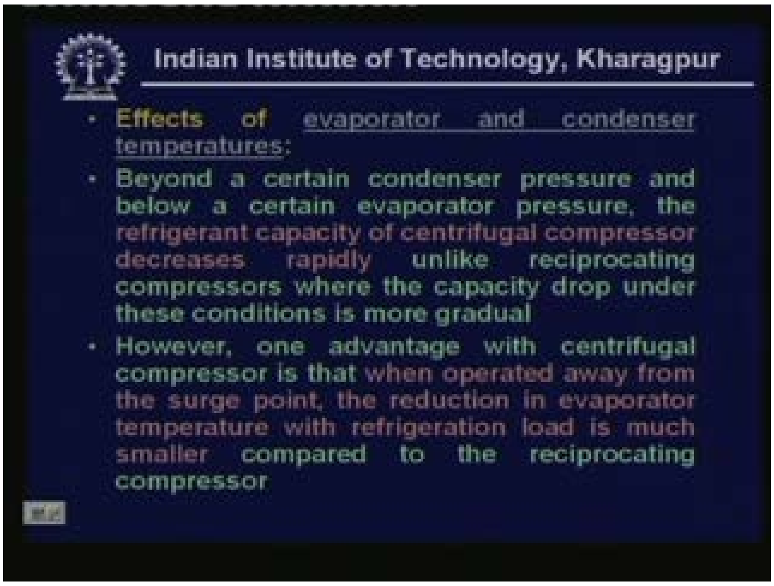

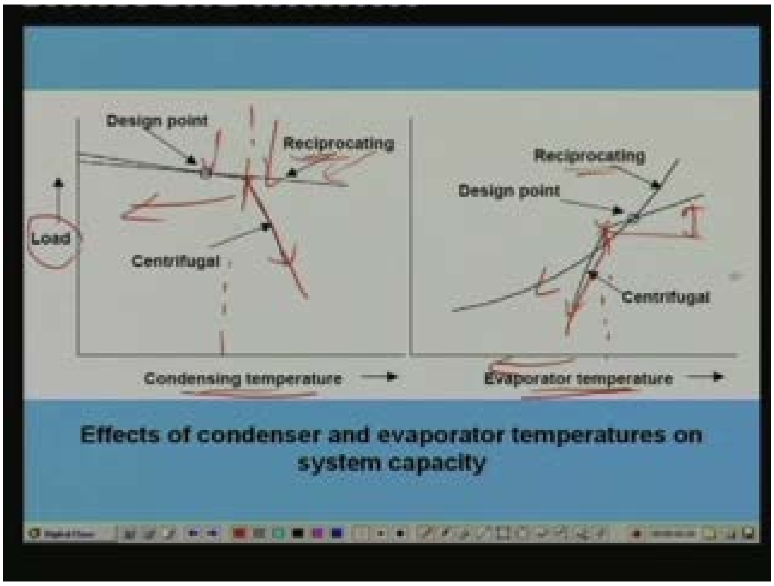

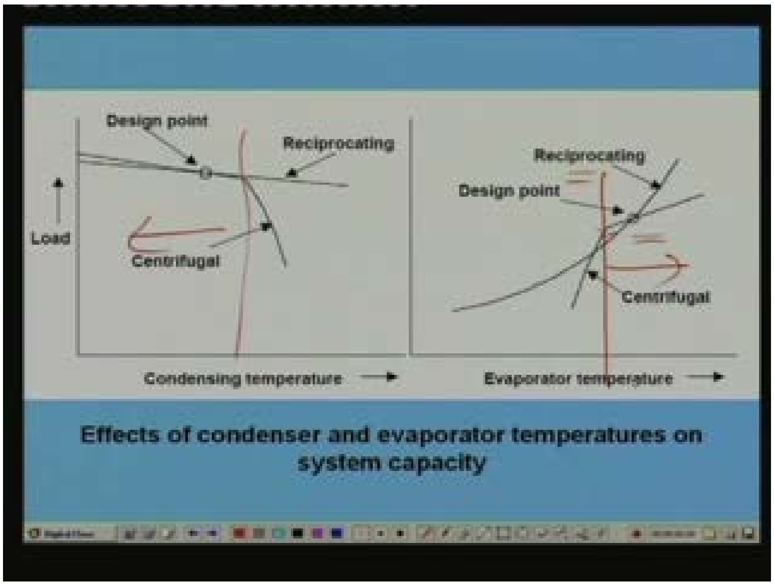

Now let us look at the effects of evaporator and condenser temperatures. Let us see how when you vary them what happens to the compressor performance okay. So here I have plotted condensing temperature in this curve as on x-axis and load or the refrigeration capacity on the y-axis. And for comparison I have shown the performance of a reciprocating compressor also here. If you look at the reciprocating compressor we have seen that as the condensing temperature increases the refrigeration capacity of reciprocating compressor reduces okay. Because this will reduce the volumetric efficiency. So mass flow rate gets reduced and the refrigeration effect also gets reduced slightly as the result the capacity reduces. So this curve is for the reciprocating compressor. Now what happens to the centrifugal compressor? You find that up to a certain point just like reciprocating compressor for this centrifugal compressor.

Also refrigeration capacity reduces as condensing temperature is increase up to a certain point. And this is more or less linear and in fact the variations is not too much okay. So this is almost similar to reciprocating compressor. But you find that beyond certain point there is sudden drop in the capacity of the centrifugal compressor. This sudden drop takes place because of the problem that we have discussed just now as because of the surging and the related phenomenon okay. So up to a certain point centrifugal compressor works normally but beyond that point the capacity reduces drastically okay. So normally the centrifugal compressors are operated towards the left of this point where the sudden drop takes place and normally this will be the some kind of a design point okay.

And this figure shows the effect of evaporator temperature on the load okay. Here again for comparison I have shown the reciprocating compressor performance and the centrifugal compressor. If you look at the reciprocating compressor you find that as the evaporator temperature is reduced the refrigeration capacity reduces rapidly okay. You know that this is because of the reason that as you are reducing the evaporator temperature for a given condensing temperature the volumetric efficiency reduces. And the refrigeration effect also reduces as the result the refrigeration capacity reduces okay. But it is more or less monotonous, it reduces monotonously okay. But if you look at centrifugal compressor you find that up to this point as you reduce the evaporator temperature the reduction in capacity okay. There is a reduction in capacity with reduction evaporate temperature. But this reduction capacity is not much okay.

But beyond this point you find that when the evaporator temperature is reduce. Further there is a sudden drop in the capacity again this is also linked to your problem of surging and the related phenomenon okay. So the, from this you find that as long as you operate the system to the left of the condensing temperature curve or to the right of the evaporator temperature. This thing your performance of the centrifugal compressor is good. But once you exceed the once you go beyond these points you find that there is a sudden drop in the capacity okay. So you should never operate the systems at under these conditions okay. One more thing you notice here is for example look at the effect of the evaporator temperature. When you look at the effect of the evaporator temperature you find that compared to the centrifugal, compared to the reciprocating compressor the effect of evaporator temperature is more severe on reciprocating compressors okay.

That means for a given change in the refrigeration capacity the change in evaporator temperature is less for the centrifugal compressors. So that is what is mentioned here beyond a certain condenser pressure and below a certain evaporator pressure the refrigerant capacity of centrifugal compressor decreases rapidly unlike reciprocating compressors where the capacity drop under these conditions is more gradual. However one advantage with centrifugal compressor is that when operated away from the surge point the reduction in evaporator temperature with refrigeration load is much smaller compared to the reciprocating compressor. That means you find that centrifugal compressors when they are operated in the design range even if the capacity varies the evaporator temperature does not vary too much this is an advantage okay.

For example when you are the capacity is reduced and if the evaporator temperature is also reduced very much then you can have a problems of freezing of water. For example if you are using it for water cooling then evaporator temperature can become so low that the external fluid may freeze or you may have very low humidity on the external fluid side okay. Because the large variation in the evaporator temperature but in case of centrifugal compressor you find that the variation evaporator temperature is less for a given variation in the capacity okay.

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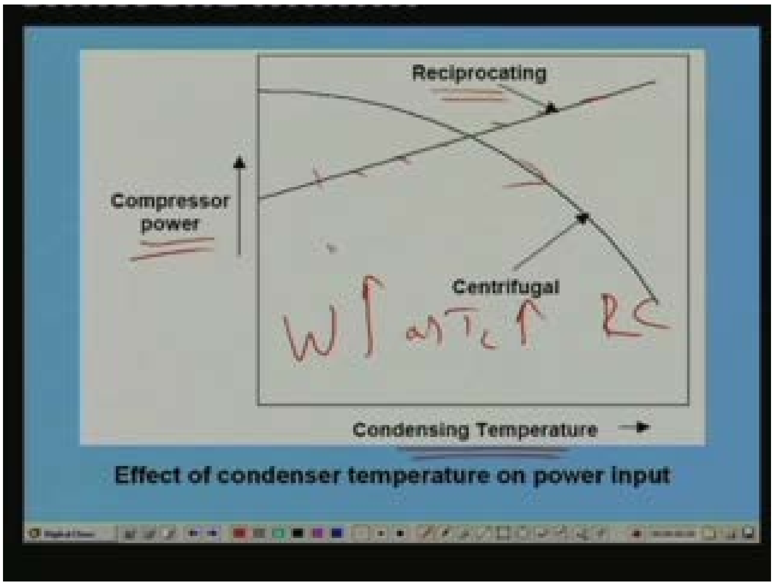

Now let us look at the effect of power input effect on power input what happens when you increase the condensing temperature to power input okay. Again you can see here that I have plotted condensing temperature versus compressor power and we have seen that for reciprocating compressor the power input increases okay. You can see that this is for the reciprocating compressor. The power input increases as the condensing temperature is increasing okay. That means W increases as Tc is increasing for reciprocating compressor but you find that for the centrifugal compressor as the condensing temperature is increasing the power input is reducing. Why the power input reduces as the condensing temperature is increasing? As the condensing temperature increases the refrigerant capacity reduces. Because the mass flow rate reduces okay.

So the reduction in mass flow rate is much larger than the increase in the specific work of compression. As a result the required power input reduces as the condensing temperature increases. In case of centrifugal compressors of course this is an advantage. Because you do not have the problem of motor overloading at under high condensing temperatures okay. Whereas this possibility is there in reciprocating compressor when the condensing temperature is very high the compressor may get overloaded okay. But you do not have the overloading problem in centrifugal compressors. As I said this is due to the rapid drop in refrigeration capacity of centrifugal compressor with condensing temperatures. This characteristics implies that the problem of compressor overloading at high condensing temperatures does not exist in case of centrifugal compressors.

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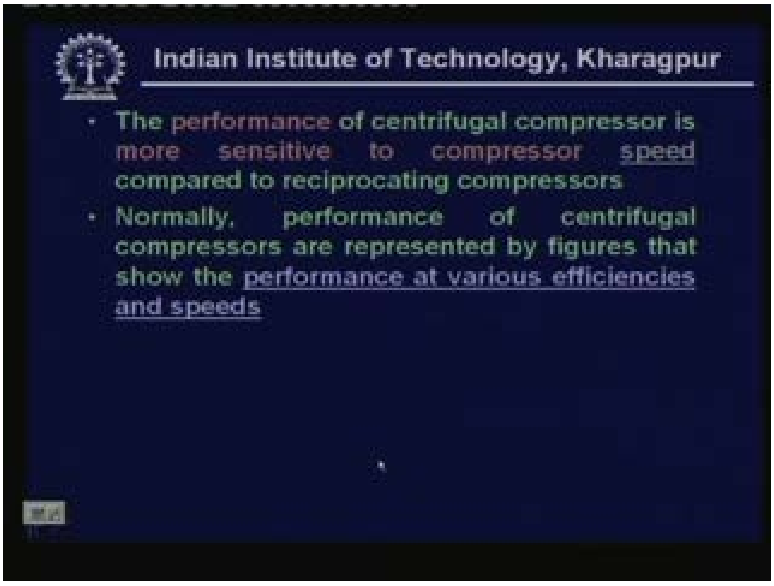

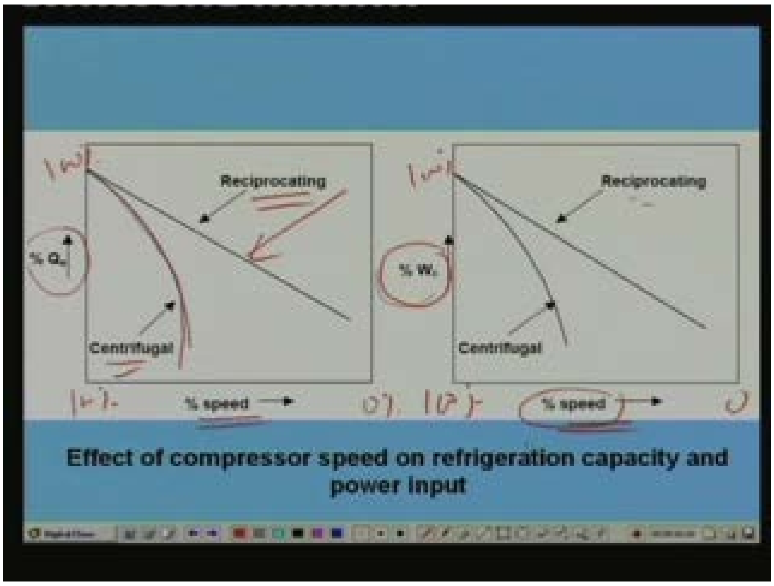

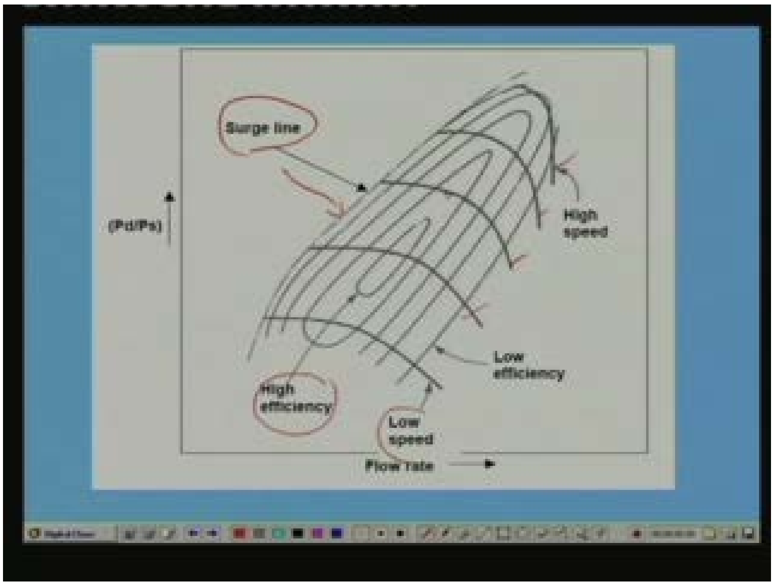

The performance of centrifugal compressor is more sensitive to compressor speed compared to reciprocating compressors you find that again. This is I have varied speed okay, x-axis is speed and in this particular curve the y-axis is the percentage of refrigeration capacity and this is the percentage of power input. And this curve is for the reciprocating compressor as you have, as I have written here and this curve for the centrifugal compressor. You can see that the reduction in the refrigeration capacity with reduction percentage speed is more gradual in case of reciprocating compressor okay. So this is the hundred percent speed and let us say that this is zero percent speed and this is hundred percent capacity okay. So similarly this is hundred percent speed this is hundred percent capacity okay. So as you are reducing the speed the refrigeration capacity is reducing almost like a straight line. But for centrifugal compressor you find that the reduction is much rapid okay.

For a given change in the percentage speed drop in refrigeration capacity for centrifugal compressor is much higher compared to reciprocating compressor. And the characteristics of the centrifugal compressor and reciprocating compressor in terms of power input can be seen from this figure. Again you can seen that the performance of centrifugal compressor is more sensitive to reduction in speed okay. This should be percentage reduction in speed right. Normally performance of centrifugal compressors are represented by figures that show the performance at various efficiencies and speed. Let me show a typical performance curve which is very useful this is the typical performance curve of a centrifugal a typical centrifugal compressor. And normally the flow rate is shown on the x-axis and the pressure ratio that is the discharge pressure to the suction pressure is shown on the y-axis.

And here the performance is shown for different iso efficiencies lines okay. So these lines are constant efficiency lines this one this one okay. All these lines are constant efficiencies lines and the inner most one as you can see here is the high efficiency line this is the high efficiency line okay. So the then this is for the low efficiency okay. As you move away the efficiency reduces and on the same curve we can also see the constant speed lines okay. For example these are low speed line and these are all the constant speed lines okay. So using this figure and normally the performance is not shown beyond the surge line okay. So you can see the surge line here because you do not want to operate the system in the surging zone okay. So normally performance is not shown in the surge region. So you, this figure is very useful for example you want to find out at a particular, let us say that at a particular flow rate okay. And you know the pressure ratio then you can find out what will be the efficiency that you can expect and what is the required speed okay. Right, that means given any two parameters you can find out the other parameters using this curves okay. They are very useful performance curves okay. So this I have already explained.

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Now let us briefly look at commercial refrigeration systems with centrifugal compressors. Centrifugal compressors are available for a wide variety of applications using a wide variety of refrigerants okay. And let me show some of the specifications of commercial machines. Normally systems are available for evaporator temperatures varying from minus hundred degrees to plus ten degree centigrade very large variation. And the systems are available for evaporator pressures varying from fourteen kPa kilo Pascal to about seven hundred kilo Pascal. And the discharge pressures can vary up to two thousand kilo Pascal's and the rotational speeds can vary anywhere between eighteen hundred rpm to ninety thousand rpm. Normally very high speeds are obtained by using gear drives okay. And the refrigerant capacity can vary from three hundred kilo watts to thirty thousand kilo watts okay. You can see that normally they are used for high capacity applications.

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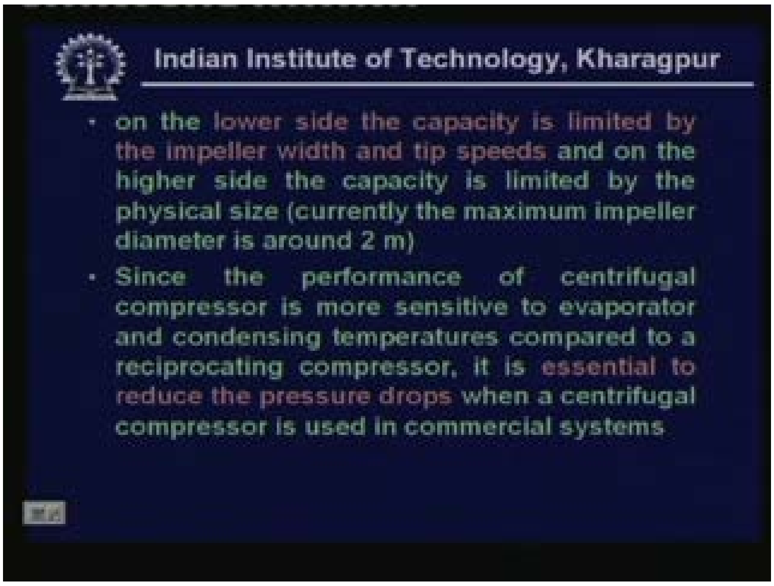

And what is the limitation on the, this I have already explained. But let me once again repeat what is the limitation on the lower side of the capacity and what is the limitation on the higher side of the capacity. On the lower side the capacity is limited by the impeller width and tip speeds and on the higher side the capacity is limited by the physical size okay. These two constraints put a limit on the available capacities of the commercial systems okay. Right, now as I said for air conditioning application the minimum capacity available is about three hundred kilo watts okay. As well the maximum capacities is about thirty mega watt okay. And normally the maximum impeller diameter is limited to about two metres. Because I have already explained in the last class as the impeller size becomes large the tip speed becomes large. So there will be severe stresses at the root of the impeller okay. So structural it becomes very difficult to design it.

Since the performance of centrifugal compressor is more sensitive to evaporator and condensing temperatures compared to a reciprocating compressor it is essential to reduce the pressure drops when a centrifugal compressor is used in commercial systems okay. So the design strategy has got to be different for the refrigerant system when you are using a centrifugal compressor and when you are using a reciprocating compressor centrifugal compressor is more sensitive to pressure drops. We have seen just now it cannot handle when the pressure rise goes beyond the design pressure rise. So you have to minimize the pressure drops okay. So you have to design the system in such a way that the pressure drops in the lines or across any valves or anything is minimal okay.

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And commercial refrigeration systems using centrifugal compressors normally incorporate flash intercoolers to improve the system performance. I have explained this in when we were discussing multistage refrigeration system. What is the meaning of a flash intercooler and what is the purpose of that? Normally when we use centrifugal compressors in commercial systems a flash intercoolers are used for improving the efficiency of the overall efficiency of the system. And incorporation of flash intercooler is much easier in centrifugal compressors. Because normally most of the centrifugal compressors are multistage that means more number of a large number of impellers will be there and the pressure across each impeller will be gradually increasing right. So when you have a flash intercooler all that you have to do is the exit of the flash intercooler can be connected to an intermediate point right. You do not have to have a separate compressor unlike in reciprocating compressors okay.

That is the reason why this is quite popular in centrifugal compressors. Since the compressor is normally multi-staged use of flash intercooler is relatively easy in case of centrifugal compressors. Commercially both hermetic as well as open type centrifugal compressors are available open type compressors are driven by electric motors internal combustion engines using a wide variety of fuels or even steam turbines okay. Normally the smaller capacity systems are hermetic systems whereas the large capacity centrifugal systems are open type

Okay. So when you have an open type of compressor you can have use any kind of a drive okay. You can use an internal combustion engine or you can use an electrical motor or you can even use a steam turbine okay.

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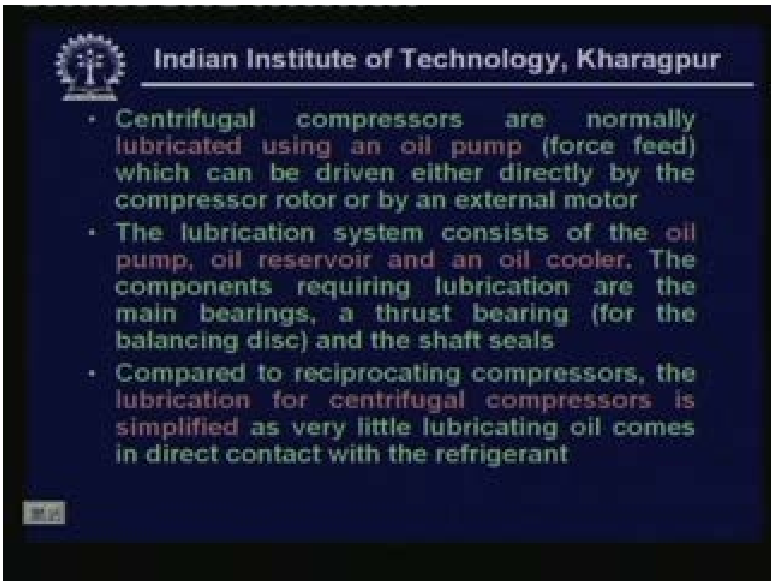

Centrifugal compressors are normally lubricate during an oil pump you also require lubrication here. And normally in large systems an oil pump is used that means you have force feed lubrication. And this oil pump can be driven directly by the compressor rotor or by an external motor. The lubrication system consists of the oil pump oil reservoir and an oil cooler the components requiring lubrication are the main bearings a thrust bearing and the shaft seals. These are the components which require lubrication okay. A thrust bearing is used because if you look at a multistage centrifugal compressor you find that with the number of the stage the pressure will be increasing. Right, that means there is a imbalance the final stage the pressure will be much high compared to the first stage okay. So there is pressure force acting in a particular direction.

So if you want to balance it normally they use what is known as the balancing disks okay, to statically and dynamically balance the compressor. So you require lubrication where you put the balancing disk also. However compared to reciprocating compressors the lubrication for centrifugal compressors is simplified as very little lubricating oil comes in direct contact with the refrigerant. You find that the design of lubricant system is not as complicated as in case of reciprocating compressor. Because normally they do not get mixed in case of centrifugal compressors.

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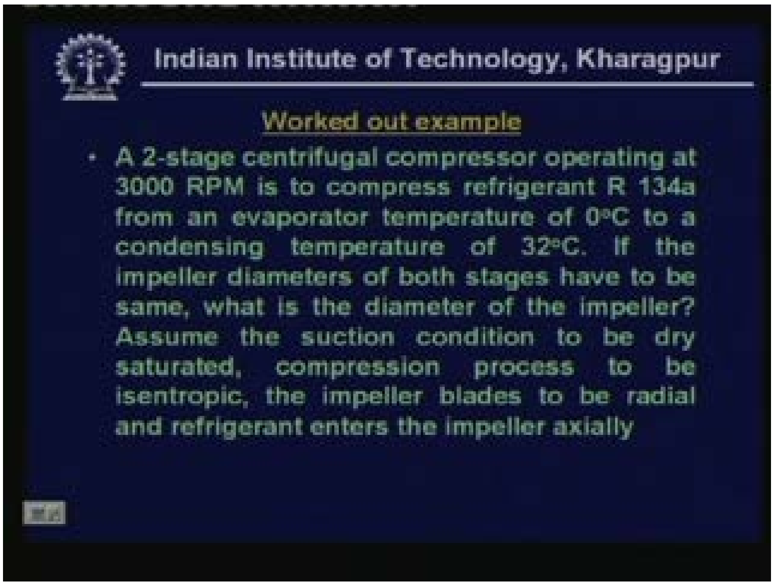

Okay, so this with this I complete my lecture on centrifugal compressors. So let me work out a simple problem on centrifugal compressors okay. So the problem is like this a two stage centrifugal compressor operating at three thousand rpm is to compressor, is to compress refrigerant R one thirty four a from an evaporator temperature of zero degree centigrade to a condensing temperature of thirty-two degree centigrade okay. If the impeller diameters of both stages have to be same what is the diameter of the impeller assume the suction condition to be dry saturated compression process to be isentropic the impeller blades to be radial. And refrigerant enters the impeller axially so here the let us look at a, the given information okay.

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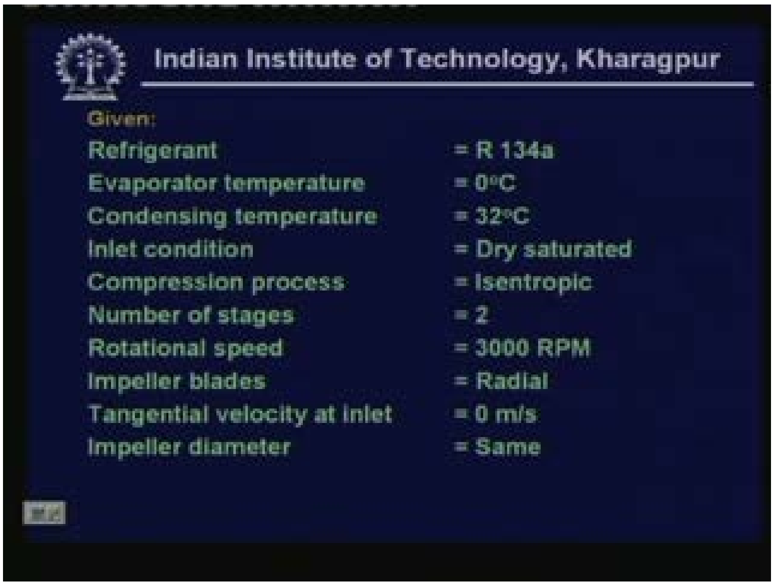

The information given is like this refrigerant is R one thirty-four an evaporator temperature is zero degree centigrade condensing temperature is thirty-two degree centigrade. Inlet condition to the compressor is dry saturated compression process is isentropic number of stages are two rotational speed is three thousand rpm impeller blades are radial tangential velocity at inlet is zero metre per second. That means the refrigerant enters axially and the impeller diameter is same for both the stages. So this is the information given and from this information we have to find out what is the required diameter of the impeller.

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This as I said is a simple problem. So first let us look at the refrigerant property data then you have to find out the property data from R one thirty-four a properties and it is mentioned that evaporator temperature is zero degree centigrade and it is dry saturated inlet is dry saturated okay. So you can find out what is he enthalpy of saturated vapour at zero degree centigrade and from the property data you find the this is three ninety-eight point six kilo joule per kg okay. And the enthalpy at compressor exit we know the compressor condenser temperature. That means we know what is the exit pressure and we also know the exit entropy because this is an isentropic process.

So at the exit condition we know the pressure and entropy and as you know the exit condition lies in the super heated region. So from the super heated property tables from the known values of pressure and entropy you can find out the other properties okay. And you find that for pressure corresponding to thirty-two degree centigrade condensing temperature and entropy corresponding to dry refrigerant vapour at zero degree centigrade the enthalpy at the exit of the compressor is found to be four hundred nineteen point eight kilo joule per kg. And since the blades are radial with no tangential velocity component at inlet the enthalpy rise across each stage. Delta h one is the enthalpy rise across stage one this is equal to an enthalpy rise across stage two delta h two which is equal to u two square okay. As I have already mentioned both the impeller stages are mounted have same diameter and they also have the same rotational speed that means the tip speed for both the stages are same.

Once the tip speed is same we have seen from our equations that the work of compression for each stage should remain same okay. That means delta h one that means the work of compression or enthalpy rise across impeller. One should be same as enthalpy rise across impeller two because both of them are equal to u two square where u two is the tip speed tip speed is same for both the stages okay. But since the total enthalpy rise that means h exit minus hi is equal to enthalpy rise across impeller one plus enthalpy rise across impeller two. You find that the total enthalpy rise across the compressor is equal to two into enthalpy rise across each stage okay that is what is mentioned here. Enthalpy rise across the compressor he minus h one is equal to h delta h one plus delta h two which is equal to two into delta h stage.

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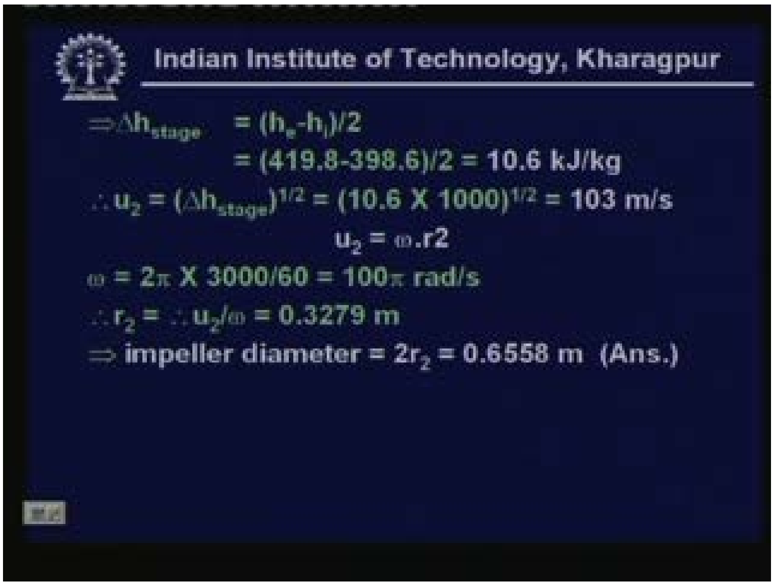

So delta h stage is nothing but he minus hi by two and from the given input data delta h stage is equal to four hundred nineteen point eight minus three ninety-eight point six divided by two that is equal to ten point six kilo joule per kg okay. And we know that this is equal to u two square. That means u two is equal to square root of delta h stage. Since this delta h is in kilo joule per kg and if you want to find out velocity in metre per second you have to multiply this into thousand. So u two is equal to square root of ten point six into thousand that is equal to hundred three point hundred three metre per second, So this is the tip speed of both the stages and we know that tip speed u two is equal to rotational speed omega in radiance per second into the outer radius okay. So and we know what is omega okay, omega is nothing but two pi into rpm by sixty.

So rpm is given as three thousand. So you have to convert that into revolutions per second and multiply that into two pi to convert this into radiance per second. So three thousand rpm works out to be hundred pi radiance per second. So if you substitute this value you find that the required outer radius is point three two seven nine metres. And the required impeller diameter is equal to two into r two that is equal to point six five eight metre okay. So this has to be same for both the stages this is a simple problem okay. But it basically outlines the procedure using which you can calculate the pressure rise or if you given the other pressure rise and all how to calculate the impeller diameter etcetera okay.

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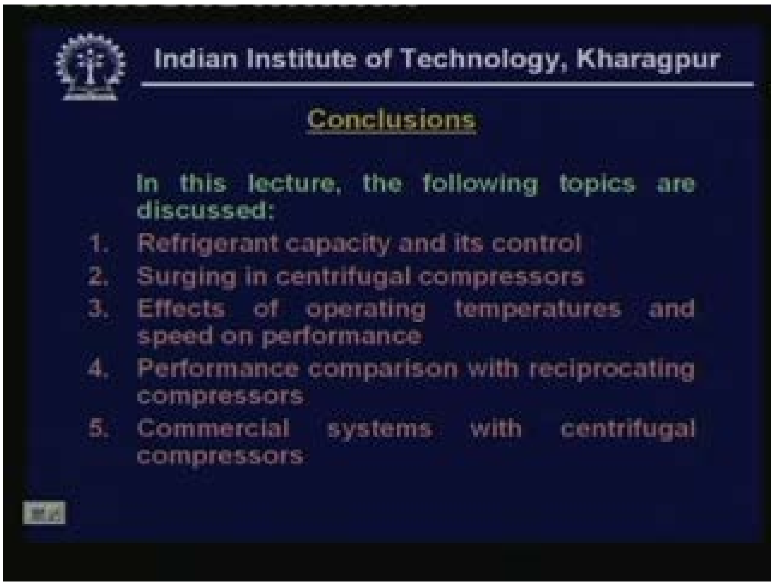

Let me give a home work problem which you can try in the, at home. Before that let me conclude what we have learned in this lesson. In this lecture the following topics are discussed refrigerant capacity and how to control it surging in centrifugal compressors effects of operating temperatures and speed on performance comparison with reciprocating compressors and commercial systems with centrifugal compressor these are the aspects covered in this lecture. And as I have already mentioned this with this lecture i complete my talk on compressors okay.

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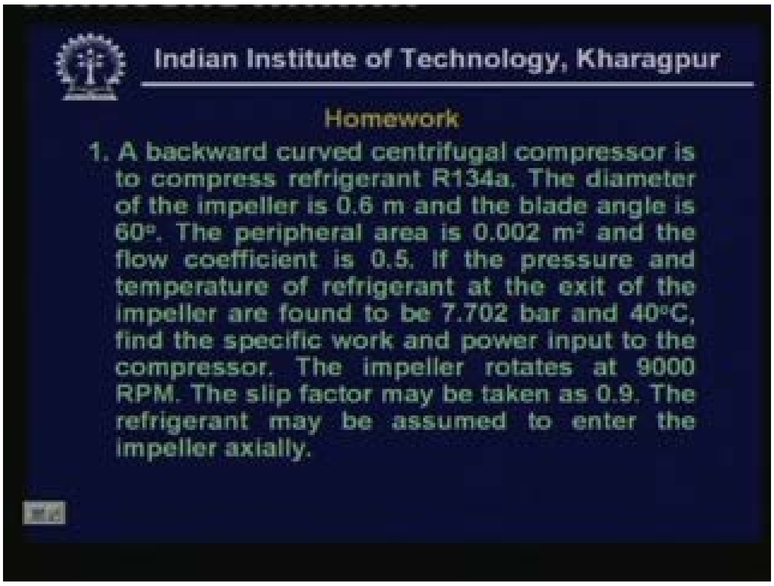

Before I leave off let me give a home work problem. The problem is like this. A backward curved centrifugal compressor is to compress refrigerant R one thirty-four a diameter of the impeller is point six metre and the blade angle is sixty degrees. So this is not a radial curved blade. This you have to keep that in mind the peripheral area is given to be point zero zero two metre square and the flow coefficient is point five what is flow coefficient flow coefficient is nothing but the ratio as I have already mentioned is the ratio of the normal component of velocity at the impeller exit to the tip speed of the impeller okay. So this is the definition of flow coefficient. That is the, as per the velocity diagram, this is the ratio of Vn two to u two okay. The flow coefficient is given as point five if the pressure and temperature of refrigerant at the exit of the impeller are found to be seven point seven zero two bar and forty degree centigrade respectively.

Find the specific work and power input to the compressor the impeller rotates at nine thousand rpm the slip factor may be taken as point nine the refrigerant may be assumed to enter the impeller axially okay. The slip factor, if you remember I have defined as the actual tangential component of velocity at the exit to the tangential velocity component in the absence of eddy losses or in the absence of any slip okay. So this is the definition of slip factor. That means Vt two actual divided by Vt two in the absence of eddy losses okay. So this is the information given from this information you have to find out what is the power input and the specific work to the compressor.

What you have to do is you have to, from the given information first. You have to find out what is the tip speed and from the tip speed you have to find out what is the tangential component okay. Because once you know the tip speed you know the normal component. And since the beta is given you can find out what is the tangential component when there is no slip okay. And once you find this you can find out what is the tangential component with slip. Because the slip factor is given once you know this one you can easily find out what is the work of compression. And what is the power input I will give you the answers to this problem in the next class. Let me also give you few objective type questions.

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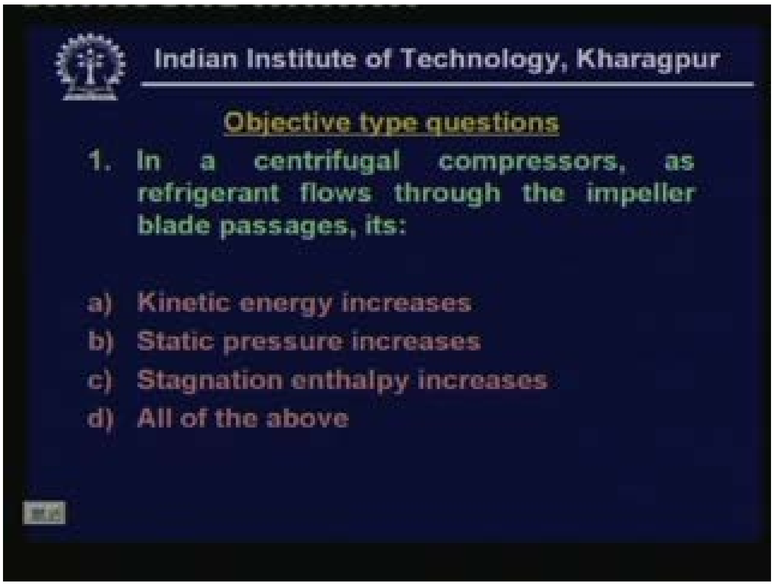

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The first question is in a centrifugal compressors as refrigerant flows through the impeller blade passages. What happens a, it is kinetic energy increases b it is static pressure increases c stagnation enthalpy increases and d all of the above.

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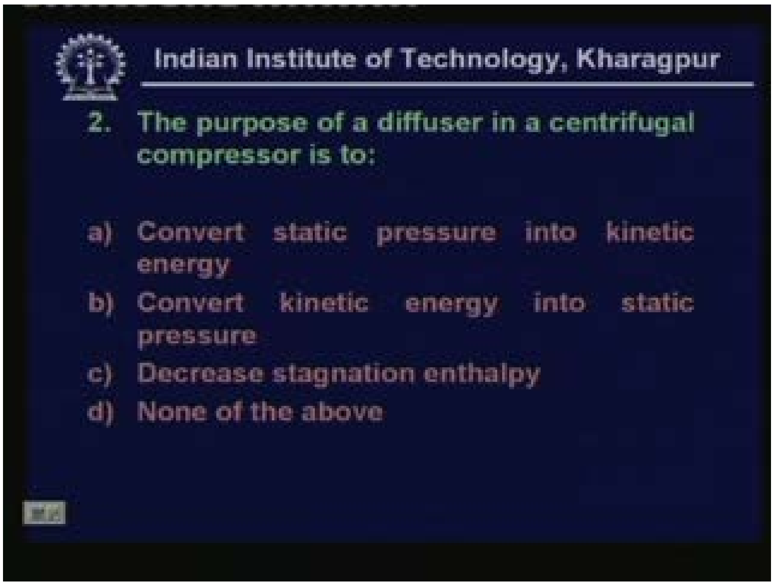

And second question is the purpose of a diffuser in a centrifugal compressor is to convert static pressure into kinetic energy, convert kinetic energy into static pressure or decrease stagnation enthalpy or d none of the above okay.

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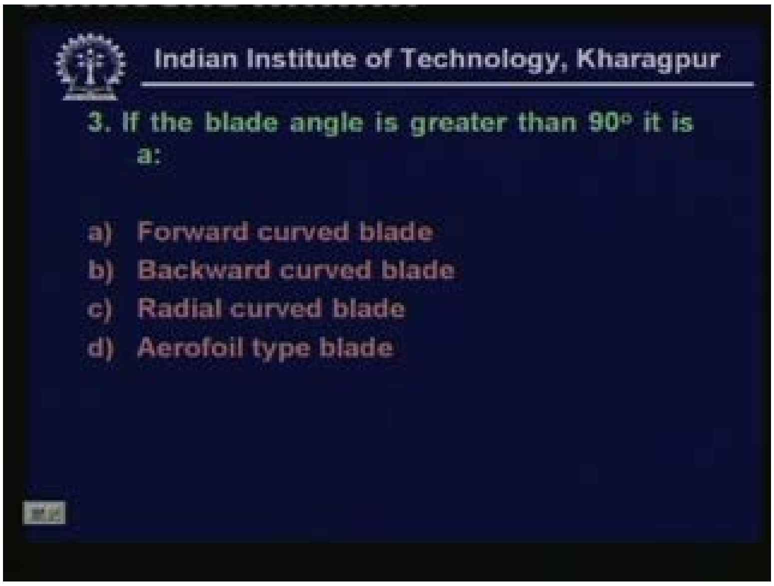

And third question is, if the blade angle beta is greater than ninety degrees it is a forward curved blade b backward curved blade c radial curved blade d aerofoil type blade.

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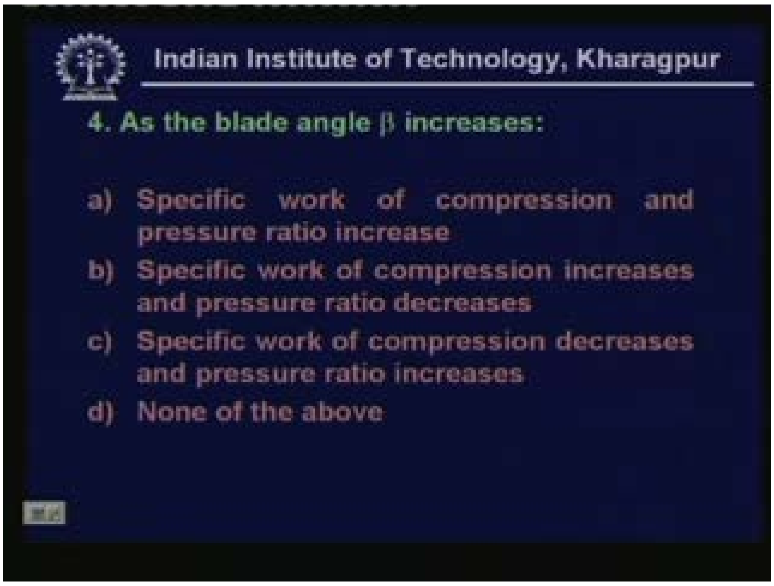

Four as the blade angle beta increases a specific work of compression and pressure ratio increases b specific work of compression increases and pressure ratio decreases c specific work of compression decreases and pressure ratio increases and d none of the above.

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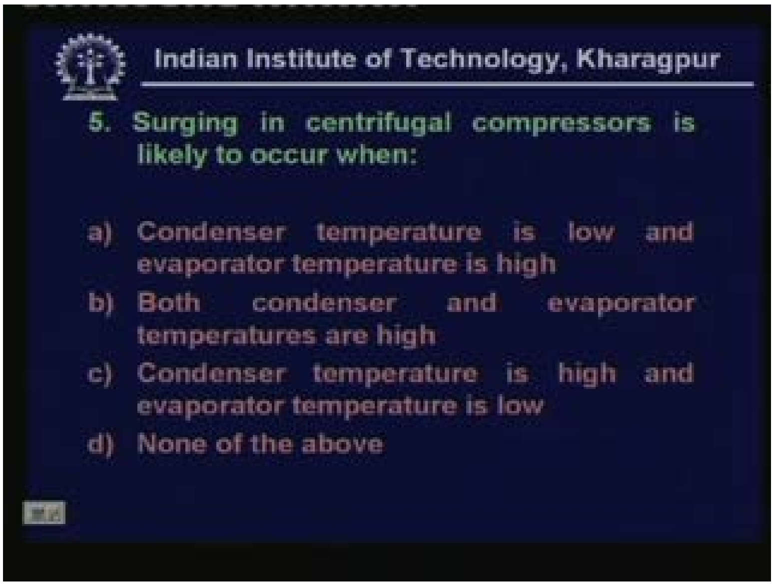

So question number five surging in centrifugal compressors is likely to occur when condenser temperature is low. And evaporator temperature is high b both condenser and evaporator temperatures are high c condenser temperature is high and evaporator temperature is low d none of the above.

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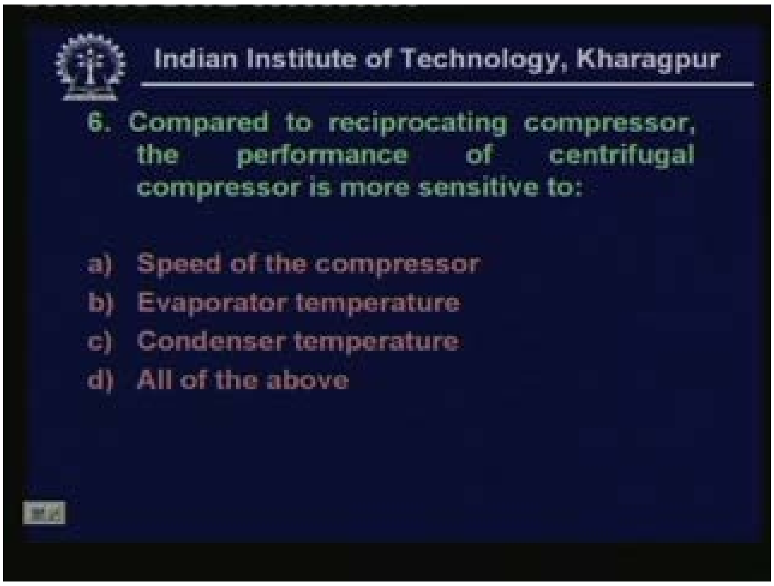

Question number six compared to reciprocating compressor. The performance of centrifugal compressor is more sensitive to speed of the compressor b evaporator temperature c condenser temperature and d all of the above.

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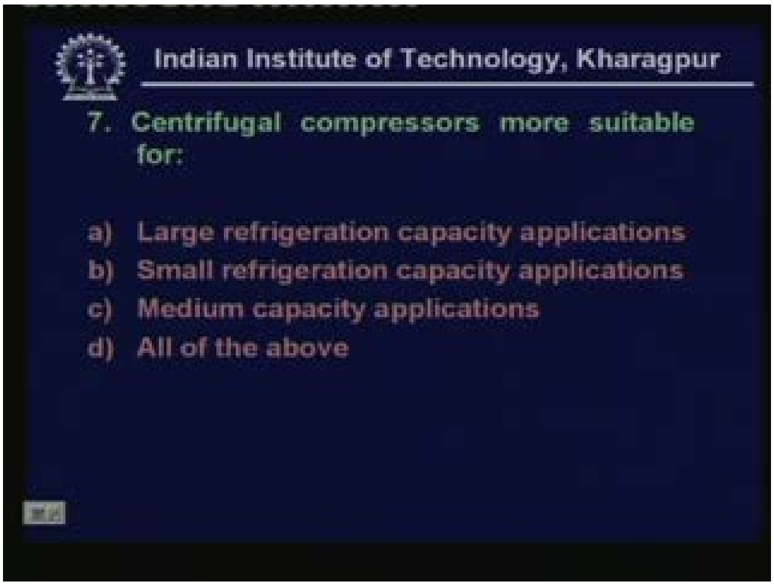

And question number seven centrifugal compressors are more suitable for large refrigeration capacity applications small refrigeration capacity applications c medium capacity applications and d all of the above.

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And the last question irreversibilities in centrifugal compressors are mainly due to heat transfer effects viscous shear forces pressure drops across valves and all o f the above okay. I will give answers to these questions in the next class. Thank you.