Refrigeration and Air Conditioning

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Lecture No. # 24
Refrigeration System Components: Compressor (Contd.)

Welcome back, in this lecture I shall discuss centrifugal compressors.

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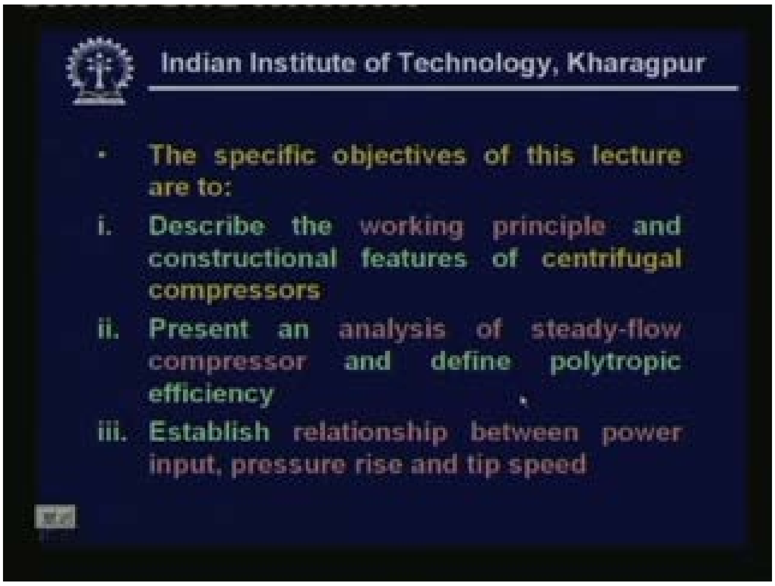

So specific objectives of this particular lecture are to describe the working principle and constructional features of centrifugal compressors, present an analysis of steady-flow compressors and define polytropic efficiency, establish relationship between power input pressure rise and tip speed.

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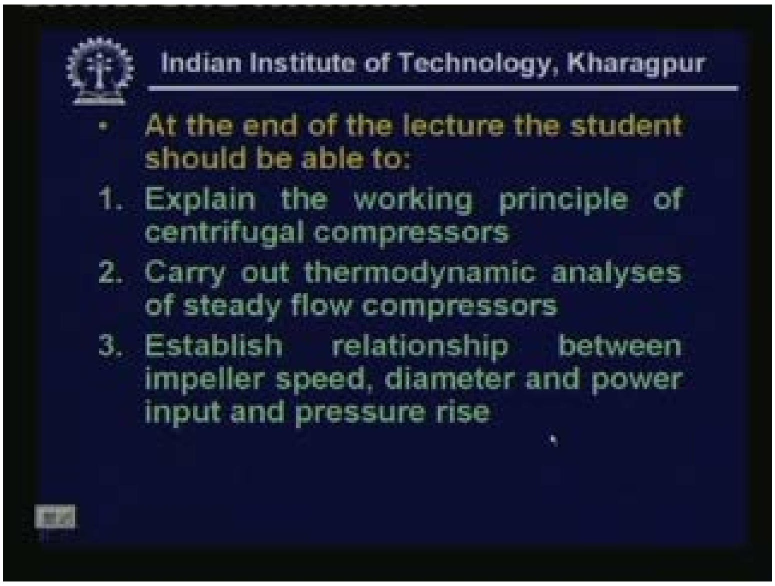

So at the end of this lesson you should be able to explain the working principle of centrifugal compressors carry out thermodynamic analyses of steady flow compressors and establish relationship between impeller speed diameter and power input and pressure rise.

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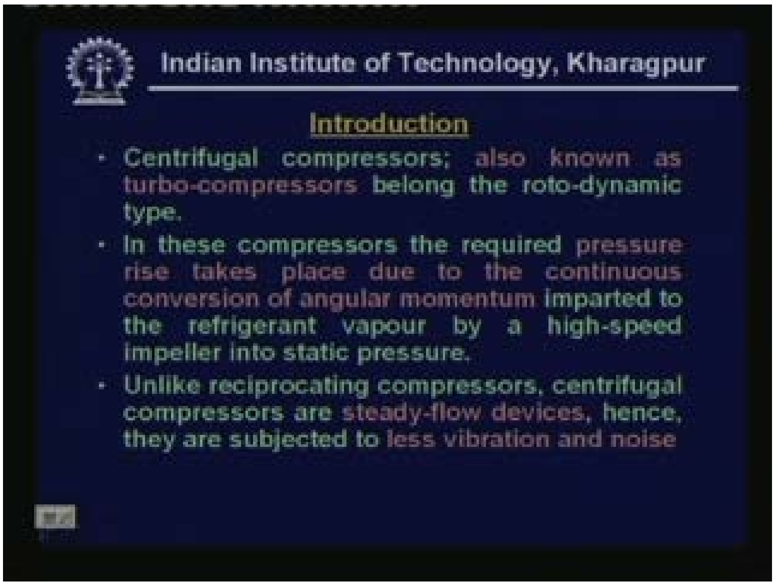

Let me give a brief introduction. As you know centrifugal compressors also known as turbo-compressors belong to the roto-dynamic type. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour by a high-speed impeller into static pressure. Unlike reciprocating compressors centrifugal compressors are steady-flow devices. Hence they are subjected to less vibration and noise.

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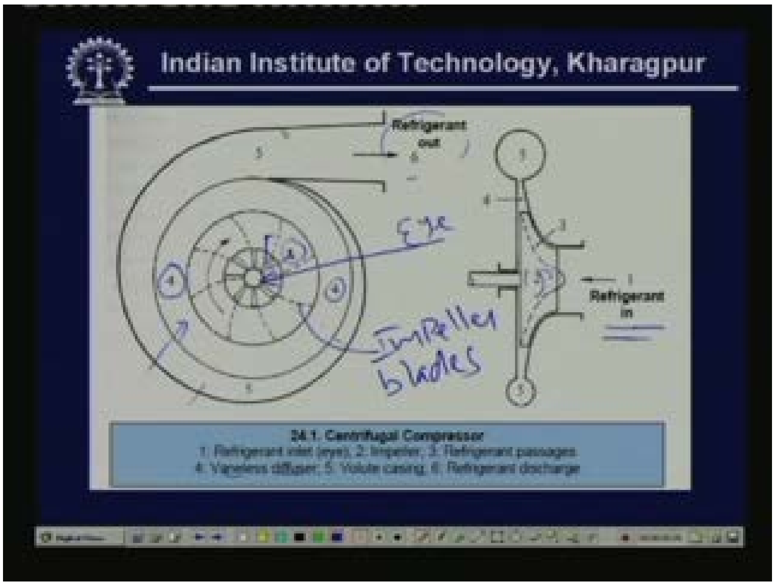

Now let me explain the construction features of a centrifugal compressor. You can see the side view and the front view of a centrifugal compressor refrigerant enters like this. And this portion, if you look at this is the inlet, refrigerant inlet. This is also known as eye of the impeller. So refrigerant enters like this and then takes a ninety degrees turn and enters into the blade passages of the impeller, these are the impeller blades. So the refrigerant flows through the passages provided by these impeller blades and these impeller blades will be rotating at very high speed.

As a result of which momentum will be imparted to the refrigerant when they come when it comes in contact with the impeller blades. So from the impeller it enters into a region called diffuser that is four okay. In the diffuser conversion of velocity pressure into static pressure takes place as a result of which its velocity reduces and the refrigerant pressure increases. And from the diffuser it enters into a volute casing you can see that the volute casing has uniformly a gradually increasing area. So as the refrigerant flows through the volute casing its velocity reduces gradually and velocity is converted into static pressure.

So finally the refrigerant comes out of the compressor at high pressure. So once again you can see the how the refrigerant flow is taking place and different parts. As I said one part one is the refrigerant inlet and part two is the impeller. Three is the refrigerant passages. These are the refrigerant passages and four is the vaneless diffuser. This is the vaneless diffuser and five is the volute casing and six is the refrigerant discharge.

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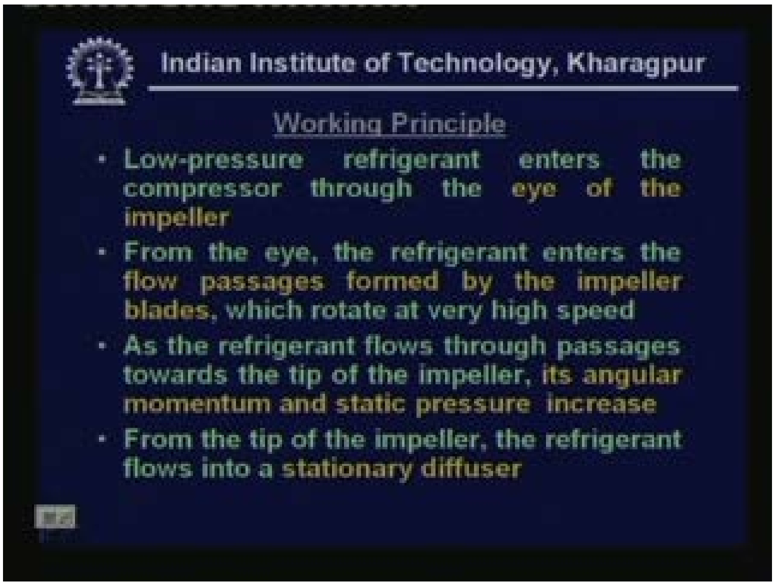

Now let me explain the working principle. Low pressure refrigerant enters the compressor through the eye of the impeller. Eye of the impeller is nothing but the inlet of the impeller from the eye the refrigerant enters the flow passages formed by the impeller blades which rotate at very high speed. That means the impeller will be rotating at very high speed. As the refrigerant flows through the passages towards the tip of the impeller its angular momentum and static pressure increase from the tip of the impeller the refrigerant flows into a stationary diffuser.

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In the diffuser the refrigerant is decelerated and as a result the velocity pressure drop is converted into static pressure rise. Thus increasing the static pressure further the vapour from the diffuser enters the volute casing where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally the pressurized refrigerant leaves the compressor from the volute casing. So this is the working principle of a centrifugal compressor this is different from your positive displacement type of compressor which are unsteady flow devices.

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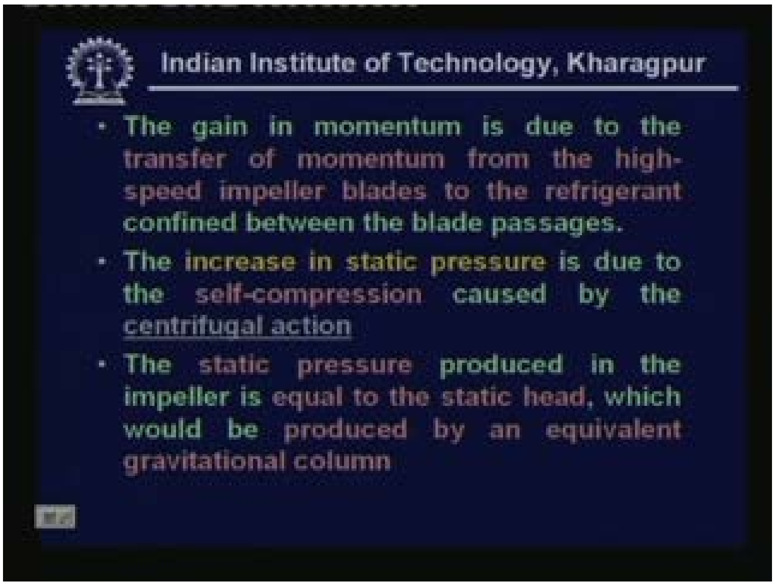

Now let me explain the physics the gain in momentum is due to the transfer of momentum from the high-speed impeller blades to the refrigerant confined between the blade passages. The increase in static pressure is due to the self-compression caused by the centrifugal action. I will explain what it is and the static pressure produced in the impeller is equal to the static head which would be produced by an equivalent gravitational column.

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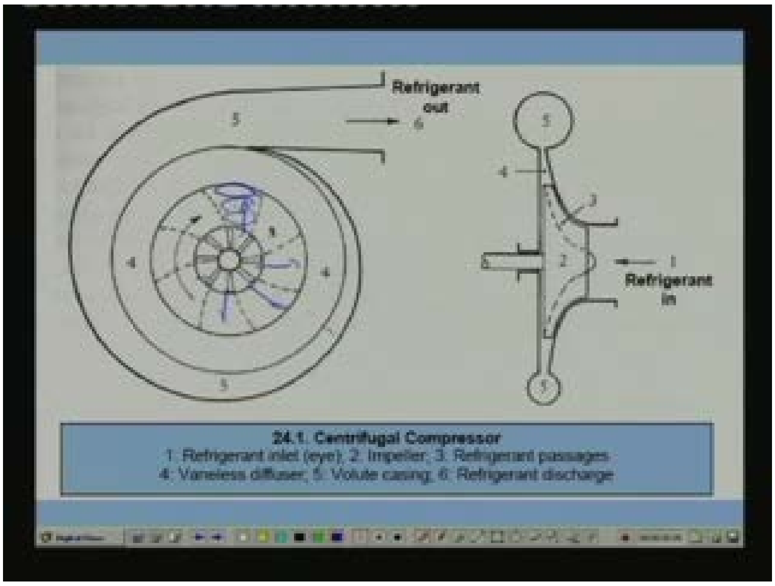

As you can see that, as the refrigerant flows through these passages and it comes in contact with the impeller blades. And since the impeller is rotating at very high speed refrigerant will be thrown towards the tip of the impeller because of the centrifugal action. So if you look at refrigerant at this portion it will be pressed by refrigerant at this portion because of the centrifugal action. Similarly refrigerant at this point will be compressed by the refrigerant at this point. So as a result as the refrigerant flows through the impeller flow passages it continuously gets compressed and its static pressure rises. That means the static pressure at this point will be less than the static pressure at this point.

And this is equivalent the pressure rise between this is equivalent to the static head obtained by a column of refrigerant of this height okay. This is somewhat similar to for example if you have a column of fluid the fluid at upper layer will exert pressure on the fluid at the lower layer because of its weight okay. There the gravity is working here instead of gravity you have the centrifugal action. So because the centrifugal action the refrigerant near the inlet will be pressing the refrigerant which is away from the impeller. This is what is known as self compression. So ultimately when the refrigerant comes out of the impeller its static pressure rises because of the self compression. At the same time its angular momentum also rises because it comes in contact with the high speed impeller blades. So the impeller blades impart momentum to the refrigerant. So these two effects takes place as the refrigerant flows through the blade passages of the impeller.

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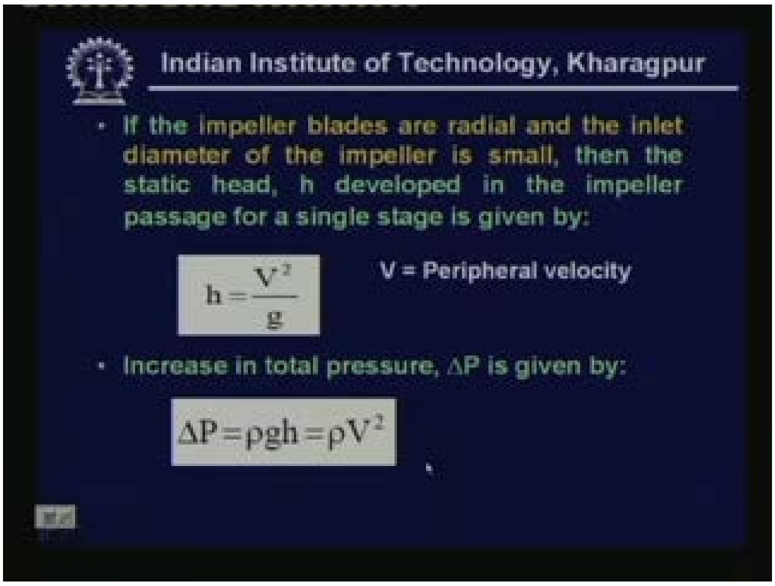

If the impeller blades are radial and the inlet diameter of the impeller is small then the static head h developed in the impeller passage for a single stage is given by that means we are assuming that we have radial blades and the inlet diameter of the impeller is small. Under these assumptions the static head developed in the impeller h is given by simply V square by g where V is the peripheral velocity or also known as tips speed and g is the acceleration due to gravity. So the increase in total pressure delta P is simply given by row gh that is equal to row into V square.

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So from this equation you can see that for a given refrigerant the pressure rise depends only on tip speed of the blade because delta P is equal to rho V square and if you are fixing the refrigerant and if you assume that row is constant then the pressure rise is simply proportional to V square. That is the tip speed of the blade the tip speed is proportional to the rotational speed and impeller diameter the maximum permissible tip speed is limited by the strength of the structural materials and the sonic velocity of the refrigerant. Hence the maximum achievable pressure rise which is nothing but which is related to your maximum achievable temperature lift of single stage centrifugal compressor is limited. So multi-staging is required for ah higher temperature lifts. So what it means is we have seen from this for a simple case of radial blades and negligible inlet diameter.

We found that the static pressure rise across the impeller is simply equal to row V square where V is the peripheral velocity or tip speed. So the tip speed again as I said is limited by your sonic velocity of your refrigerant. And it also limited by the structural limitations when the impeller is rotating at very high speed stresses will develop and the stresses will be maximum at the root of the impeller. So higher the velocity higher will be the stresses. So these the stresses developed and the sonic velocity impose an upper limit on the tip speed. So, that means there is a limit on V once there is a limit on V there is a limit on delta P if you are assuming that rho is constant okay. That means the pressure rise that one can achieve under these limitations is limited if you are using a single stage impeller or single stage centrifugal compressor.

Now what is this pressure rise this pressure rise is nothing but if you look at a refrigeration system it is nothing but the condenser pressure minus evaporator pressure okay, which again turn is related to the condenser temperature and evaporator temperature. So ultimately what it means is the temperature lift that is Tc that condenser temperature minus evaporator temperature of a single stage centrifugal compressor is limited by these considerations. So if the required temperature lift is high for example if you have a case where the evaporator temperature is very low or the condenser temperature is high. That means you have high temperature lift that means the required pressure rise across the compressor is very high okay. If that is the case then the required tip speed with a single stage compressor could be very high and it could be higher than the upper limit in such cases we have no option but to go for multi-stage compressors okay. Single stage compressor will not work in multi-stage centrifugal compressors.

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The discharge of the lower stage compressor is fed to the inlet of the next stage compressor. And so on in multistage centrifugal compressors the impeller diameter of all stages remains same. But the width of the impeller becomes progressively narrower in the direction of flow as density increases progressively. That means when you have a multi stage compressor. Let us say four stage centrifugal compressor all the four stages will have the same impeller diameter okay. But since the second stage the refrigerant pressure will be higher compared to the first stage. That means in the second stage refrigerant density will be higher. So if you from the continuity equation, so you know that the cross sectional area has got to be reduce okay. So only way of doing this is to reduce the width of the higher stage impeller that means the first stage impeller will have a higher width and the second stage will have slightly lower width and the third stage will have still lower width and so on okay. So that means the blades become narrower as you move in the direction of the refrigerant flow.

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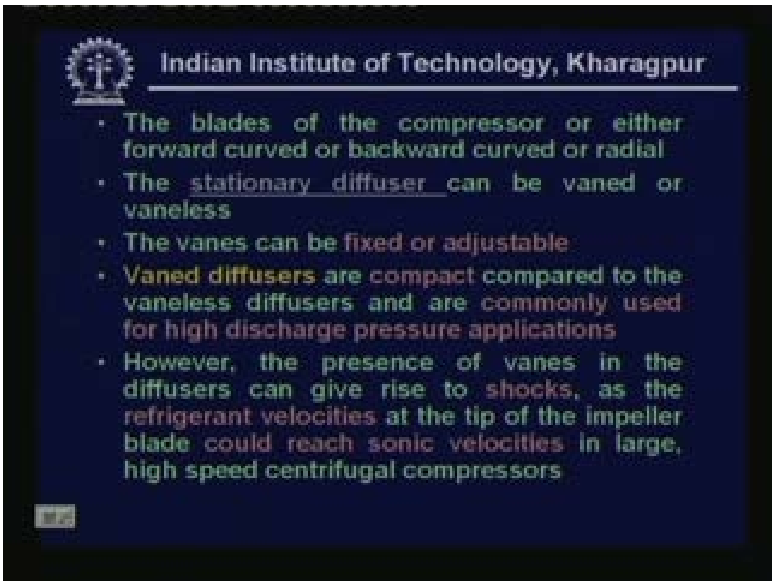

The blades of the compressor or either forward curved or backward curved or radial that mean in principle you can have either forward curved blades or radial curved blades or backward curved blades in the older machines mainly backward curved blades were used. But now-a-days the modern trend is to use radial curved blades the stationary diffuser can be vaned or vaneless I have explained that you have a component called diffuser.

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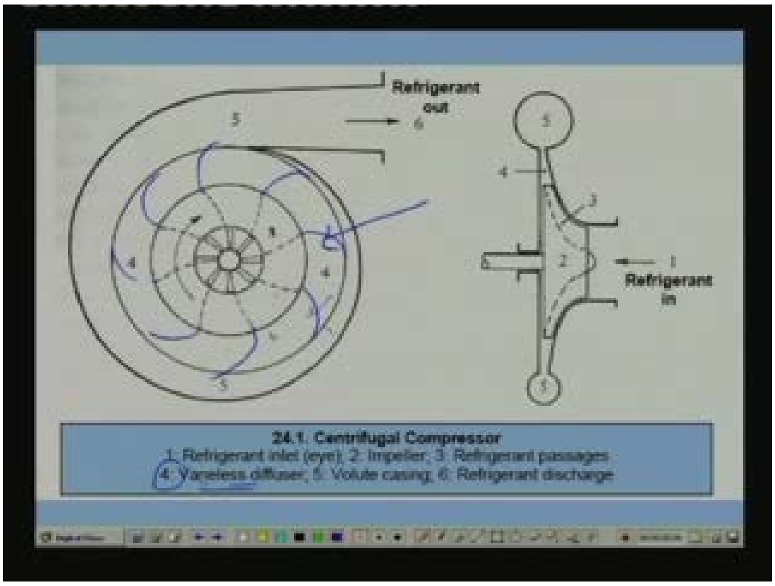

This is the, this space is called as the diffuser. As you can see four and here this is the vaneless diffuser. That means it is a simple empty hollow ring where you do not have any vanes okay. So this is the vaneless diffuser you can also have vane diffuser that means inside the diffuser you can have vanes for example you can have this kind of vanes okay. And these vanes can be fixed or adjustable that means in the fixed case you cannot change the position of the vanes. But in the adjustable case you can vary the position of the vanes in the diffuser okay.

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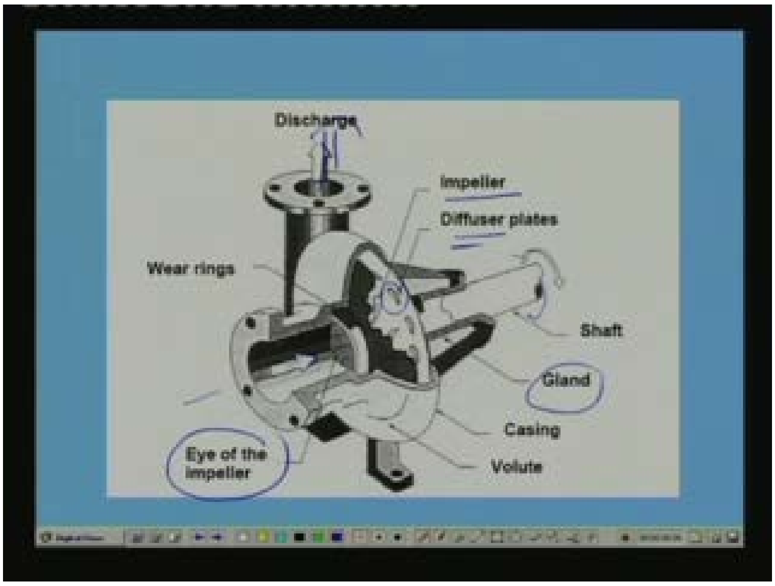

This is a cut view of an actual centrifugal compressor again you can see the different parts here refrigerant enters like this is the eye of the impeller okay. Then you can also see the impeller blades and you can see the diffuser plates and you have the vanes here okay. So this is the vane diffuser and finally you have the volute casing here and from the volute casing refrigerant high pressure refrigerant comes out like this. So this is the single stage centrifugal compressor and the diffuser is rotated by the shaft you can see the shaft and it is rotating in this manner okay. And of course you have to have some sealing and all. So gland is used for sealing the compressor okay.

So coming back to the diffusers as I said the diffuser can be vaned or vaneless and vanes can be fixed or adjustable vaned diffusers are compact compared to the vaneless diffusers and are commonly used for high discharge pressure applications the size of these compressors using vaned diffusers will be smaller and generally they are used for high discharge pressure applications okay. That is the advantage of vaned diffusers however vaned diffusers have one disadvantage the presence of vanes in the diffusers can give rise to shocks as the refrigerant velocities at the tip of the impeller blade could reach sonic velocities in large high speed centrifugal compressors. So what happens in large high speed centrifugal compressors is the tip speed will be very high okay.

Once the tip speed is high the refrigerant velocity at the tip of the impeller also will be high okay. And it may, so happen that it is higher than the sonic velocity okay. So when you have this kind of a situation very high velocity supersonic refrigerant when it enters into the vaned diffusers it produces a shock okay. So this is one of the problems of vaned diffusers of course there are measures to take care of the shock if it is not too high. But still this is a disadvantage especially when you have very high speed centrifugal compressors.

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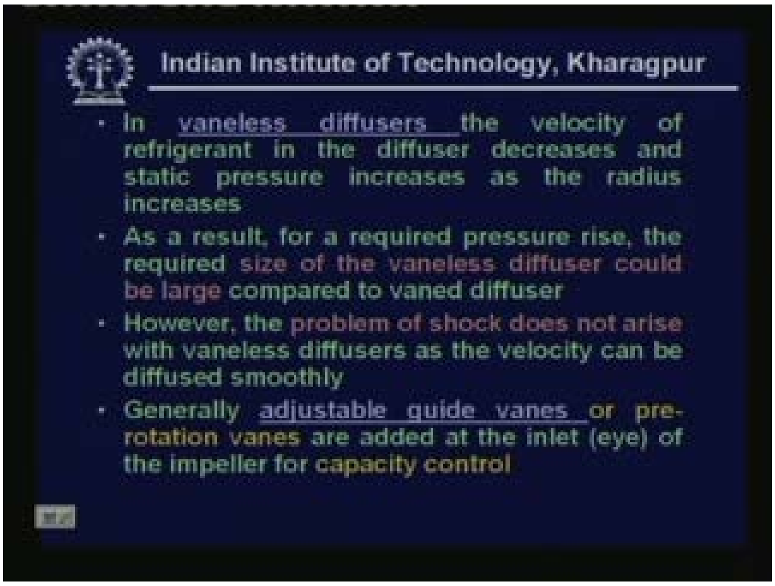

In vaneless diffusers the velocity of refrigerant in the diffuser decreases and static pressure increases as the radius increases. So remember that the principle of diffuser as I said the, is to convert the velocity pressure partly into static pressure okay. How do you do this how do you convert the velocity pressure into static pressure you have to reduce the velocity smoothly in such a way that the velocity gets converted into static pressure rise. So in a vaneless diffuser the velocity conversion takes place as the radius increases. That means the cross sectional area of a flow increases as a area increases velocity reduces and this this reduction velocity leads to increase in the static pressure okay. So this is the principle of vaneless diffuser since the pressure rise and the, for a given pressure rise or for a given drop in velocity you require a larger diameter. If you are using a vaneless diffuser as a result the size of the vaneless compressor using a vaneless diffuser will be larger compared to a vaned diffuser compressor okay.

So that is what is mentioned as a result for a required pressure rise the required size of the vaneless diffuser could be large compared to vaned diffuser. However the problem of shock does not arise with vaneless diffusers as the velocity can be diffused smoothly. So this is an advantage of vaneless diffusers. So you do not come across the problem of shock generally adjustable guide vanes or pre-rotation vanes are added at the inlet of the impeller for capacity control. So this is about the diffuser now if you look at the inlet can also can have vanes and these vanes are known as inlet guide vanes or pre-rotation vanes and what is the purpose of these pre-rotation vanes the purpose is to control the capacity.

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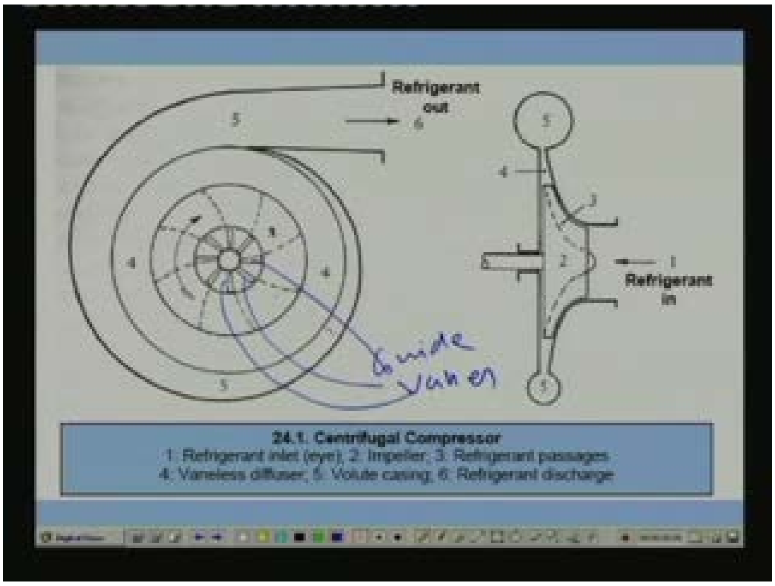

You can see that in this picture you have this is the inlet and you have the guide vanes here okay. So these are all the guide vanes all these okay. And the position of these guide vanes can be adjusted in such a way that the by adjusting the position you can change the flow rate and you can also introduce a tangential component of velocity at the inlet okay. So this will through this you can control the capacity of the compressor okay. So this is the purpose of guide vanes.

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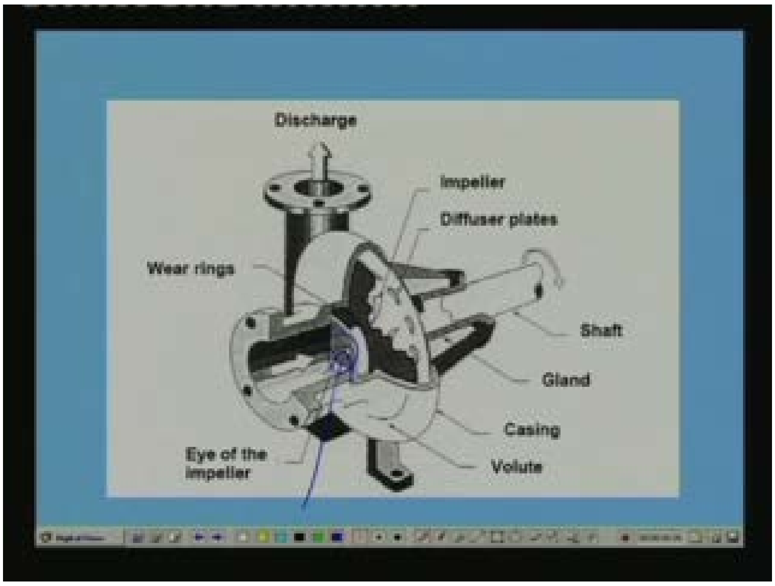

Of course you can also see the guide vanes here. So these are the guide vanes okay. So normally almost all centrifugal compressors use this guide vanes as a means of controlling the capacity okay, whereas the diffuser may or may not have vanes.

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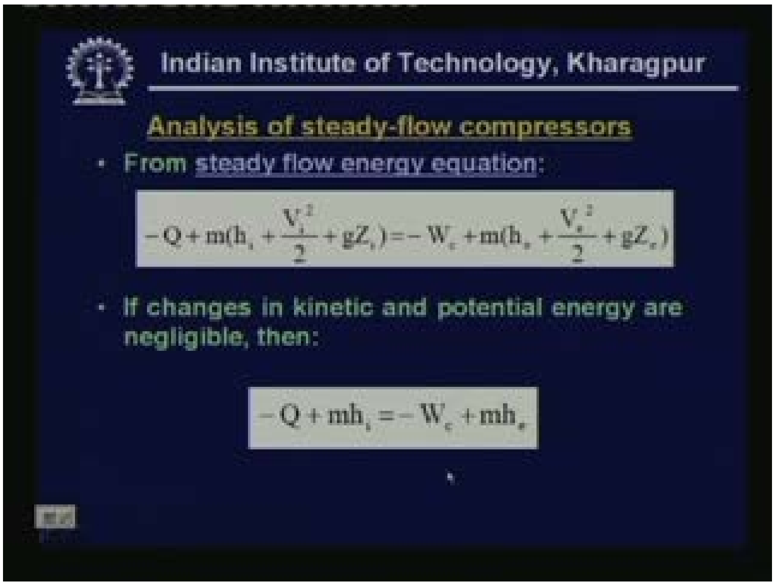

Now let us look at the analysis of steady-flow compressors this analysis is very simple analysis and it holds good for any steady-flow compressor not necessarily centrifugal compressor okay. So but we confine ourselves to centrifugal compressors in this particular lecture okay. And what we do in this analysis we simply apply the steady-flow energy equation to a steady-flow device that is nothing but the steady-flow compressor okay. So from steady-flow energy equation.

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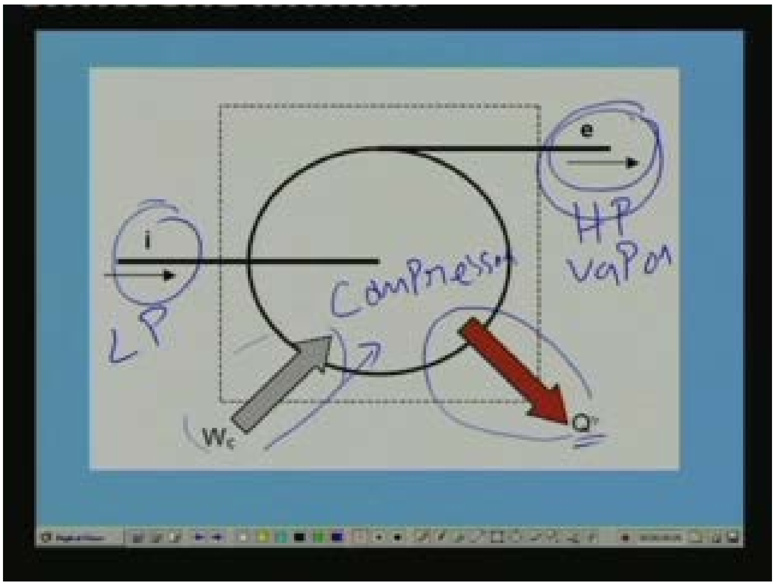

So you know very well what is the steady-flow energy equation is so we have the let us say that this is the compressor okay. So low pressure low pressure refrigerant at condition I enters into this compressor at a steady rate and work of compression is supplied to the compressor. And as a result of which the refrigerant gets compressed and it comes out as a high pressure vapour okay. So e stands for the exit that is high pressure vapour and I stands for the inlet that is low pressure refrigerant vapour depending upon the design depending upon the operation there can be some heat transfer from the compressor to the surroundings okay. So if you look at the energy flows you have the energy flow by means of mass flow here and mass flow over the inlet and mass flow at the outlet. And you can you also have energy input by means of the work input and energy interaction between the system and surroundings by means of the heat transfer rate okay. So this is the system and this is the control volume and to this we apply the steady-flow energy equation.

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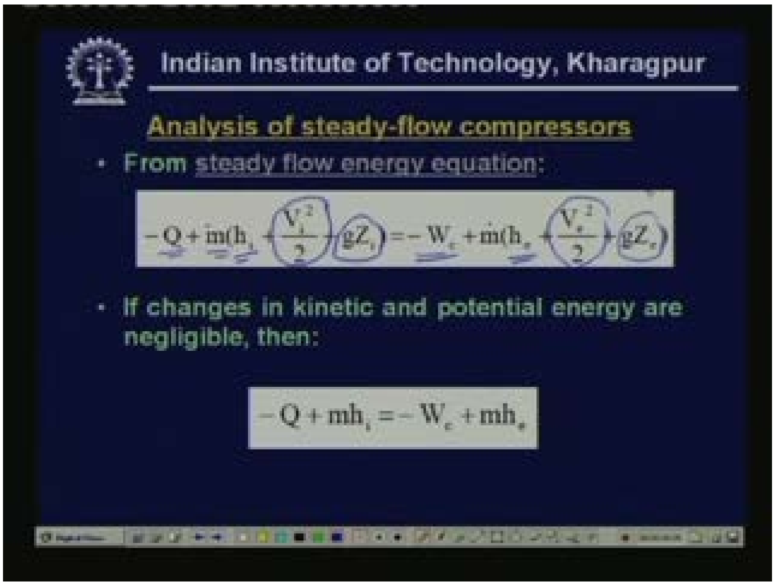

So as you know the steady-flow energy equation is something like this is the rate at which the heat transfer is taking place from the compressor to the surroundings. And m is the mass flow rate hi is the inlet enthalpy and this is the inlet specific kinetic energy at the inlet. This is the specific potential energy at the inlet this is the work input to the compressor and the power input to the compressors if you are writing in terms of mass flow rates. So Wc is the power input to the compressor he is the exit enthalpy and Ve square by two is the exit kinetic specific kinetic energy and gZ e is the exit specific potential energy. So this is the simple steady-flow energy equation which we have seen ah before okay.

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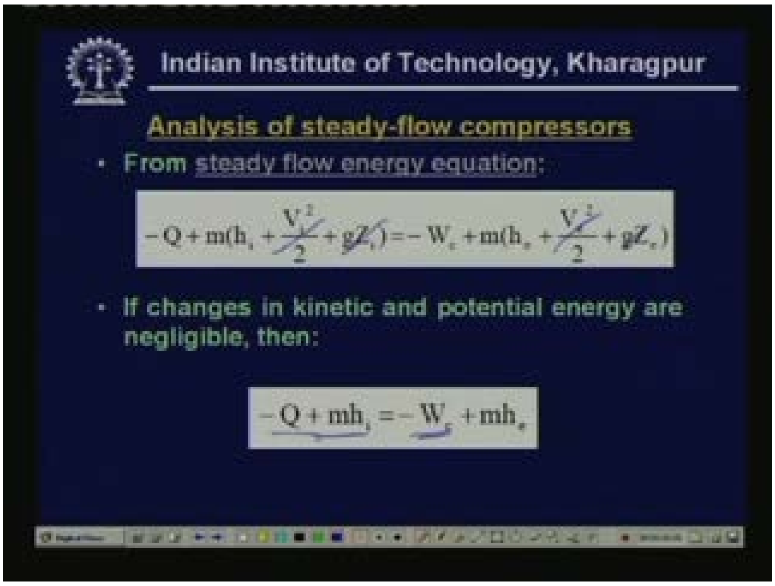

And if you assume that the potential and kinetic energy changes are negligible okay. That means this is almost same as this and this is almost same as this then we can drop this terms. And if you drop these terms this equation becomes simply like this Q as I said is the heat transfer rate from the compressor and Wc is the power input to the compressor okay.

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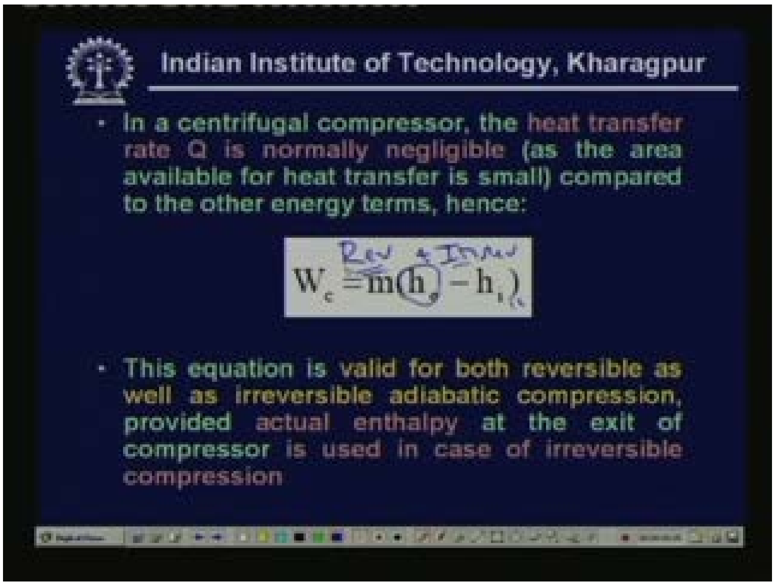

So in a centrifugal compressor the heat transfer rate Q is normally negligible as the area available for heat transfer is small okay. That means heat transfer rate is negligible compared to other energy terms so you can neglect heat transfer rate. So once you neglect the heat transfer rate the power input to the compressor is simply given by mass flow rate into enthalpy rise across the compressor that is he minus hi okay. So this equation is valid both reversible as well as irreversible adiabatic compression provided actual enthalpy at the exit of compressor is used in case of irreversible compression okay. That means this is for both reversible as well as irreversible okay. If you are using reversible then this will be isenthalpic enthalpy isenthalpic enthalpy rise if it is irreversible then you have to use the actual enthalpy at the exit of the compressor okay, that is the difference.

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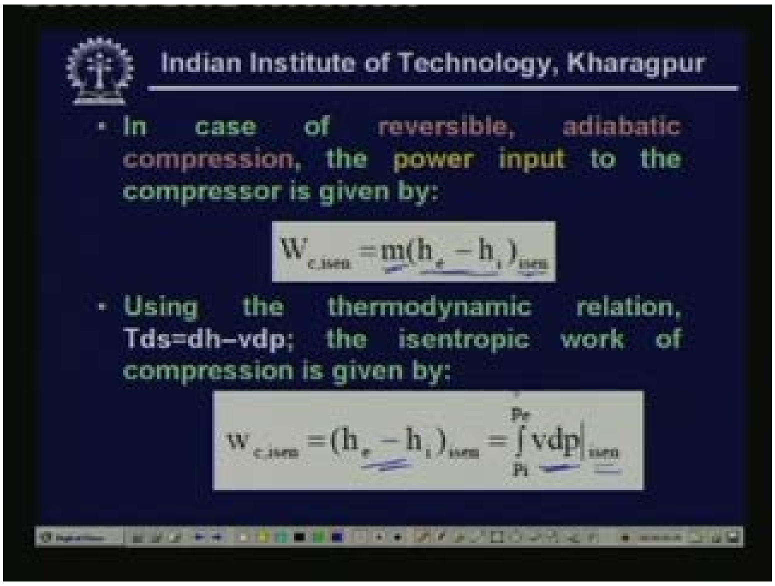

In case of reversible adiabatic compression the power input to the compressor is given by as I have just now I have explained it is nothing but mass flow rate into a specific enthalpy rise for an isentropic process. So this is en subscript stands for isentropic process. he as I said is the exit enthalpy and hi is the inlet enthalpy okay. So power input is simply mass flow rate into he minus hi okay. And using the thermodynamic relation TDs is equal to dh minus vdp the isentropic work of compression is given by this how do we get this since this process is isentropic process ds is zero okay. So from this thermodynamic relation Tds is equal to dh minus vdp we get that dh is equal to vdp. So work of compression specific work of compression is nothing but integral dh which is equal to this and that is equal to integral vdp this integration has to be performed along the isentropic process from the inlet pressure Pi to the exit pressure Pe.

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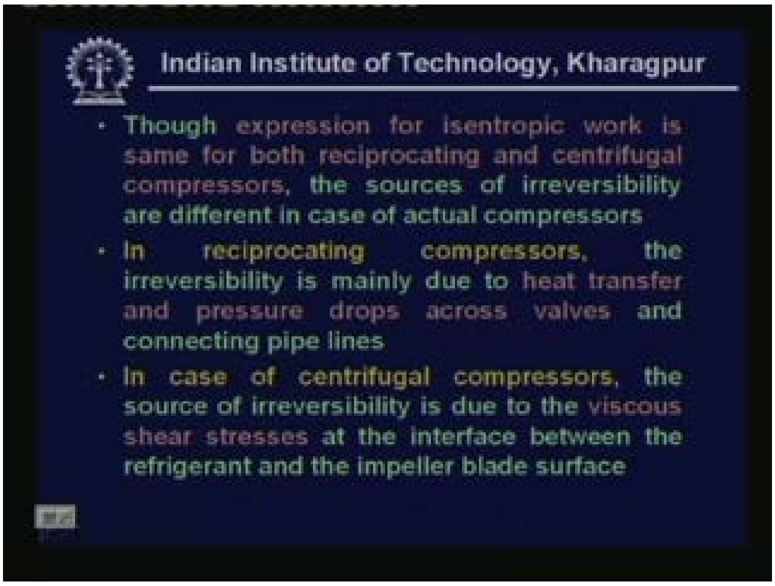

Though expression for isentropic work is same for both reciprocating and centrifugal compressors the sources of irreversibility are different in case of actual compressors. If you look at the expression and if you compare this expression with the specific work expression that we obtained in case of reciprocating compressor you find that the expression is same okay. In both cases we got the specific work for an isentropic compression integral vdp along the isentropic process okay. So as far as the isentropic ideal compressors are concerned the expression is same. But when you talk about the actual compressors the expression will be different because the phenomena will be different in case of an actual centrifugal compressor and actual reciprocating compressor. Let us see what are the differences in reciprocating compressors the irreversibility is mainly due to heat transfer and pressure drops across valves and connecting pipe lines.

So what is the difference between an ideal isentropic compressor and actual isentropic compressor the difference lies in the irreversibilities. And what are the irreversibilities if you remember in case of a reciprocating compressors the irreversibilities are mainly due to heat transfer from the compressor to the surroundings. And from the compressor to the refrigerant or from the refrigerant to the surroundings okay. So this is an irreversible process and this causes irreversibility in addition to that we also had irreversibilities due to pressure drops across the suction and discharge valves and also across the connecting pipe lines okay. So these are the major source of irreversibilities in case of a reciprocating compressor. Now let us look at what are the sources of irreversibility in case of centrifugal compressors in case of centrifugal compressors. The source of irreversibility is due to the viscous shear stresses at the interface between the refrigerant and the impeller blade surface okay. So if you look at the working principle of a centrifugal compressor.

You find that between the impeller blade passages refrigerant flows at very high speed okay when it flows at a high speed and normally impeller diameter also large. So it flows for a considerable distance at very high velocities okay. And as it flows at very high velocities there will be shear stresses at the interface between the refrigerant and the impeller blade okay. This is because of the fluid friction okay. So this causes irreversibility and this is the major source of irreversibility in case of centrifugal compressor. As I have mentioned before the irreversibility due to heat transfer is generally small in case of centrifugal compressors because given the size and given the capacity the heat transfer is almost negligible okay. So the irreversibility due to heat transfer is negligible. So major source of irreversibility is because of the viscous shear stresses okay. So this is the major difference between an actual centrifugal compressor and an actual reciprocating compressor and in reciprocating compressors the work is required to overcome the normal.

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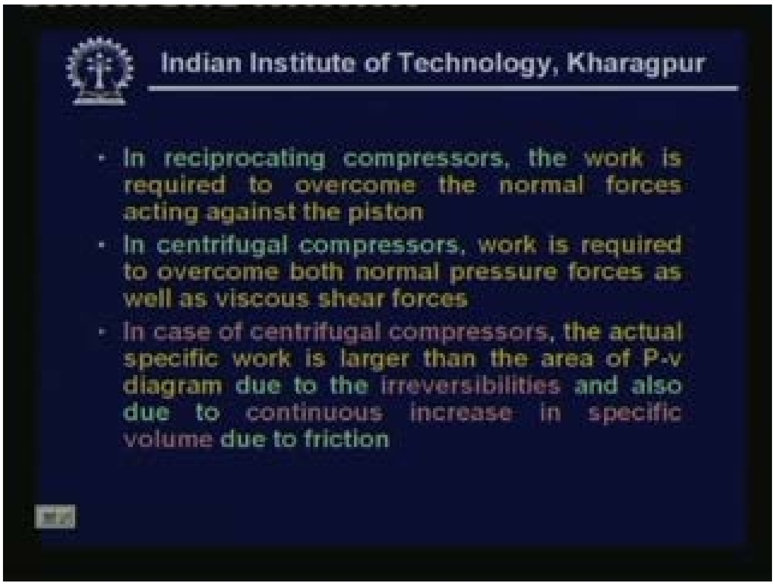

Forces acting again as the piston so why do we require work input in reciprocating compressors if you think about the working principle of a reciprocating compressor when you compress. That means when the piston is compressing the refrigerant it encounters a resistance by way of a normal force and this normal force is due to the pressure acting against the piston. And what is the maximum amount of this force this maximum amount of this force is equal to the area of the piston multiplied by the maximum pressure developed inside the compressor. And this maximum pressure is almost equal to the condenser pressure okay. So the work is required to be done in overcoming this normal force which is equal to P condenser into the area of the piston okay. Where as in centrifugal compressors work is required to overcome both normal pressure forces as well as viscous shear forces okay. So here the refrigerant has to flow against a increasing pressure so you have a normal pressure acting on the refrigerant in addition to that it also has to overcome viscous shear forces okay. So the work input is required for these two purposes.

In case of centrifugal compressors the actual specific work is larger than the area of P v diagram due to the irreversibilities. And also due to continuous increase in specific volume due to friction okay. So if you have a P v diagram of a centrifugal compressor you cannot find out the work input from the P v diagram. Because the area of the P v diagram does not give you the actual work input to the centrifugal compressor. You will find that the actual work input is always higher than the area of the P v diagram in case of a centrifugal compressor. And this difference is due to the irreversibilities and its also due to the fact that the specific volume continuously increases because of the fluid friction okay, and to account for the irreversibilities.

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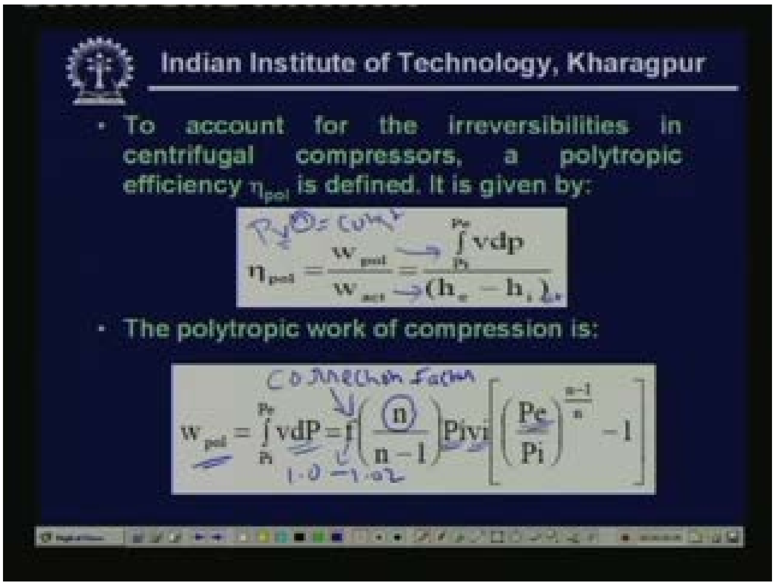

In centrifugal compressors a polytropic efficiency eta pol is defined it is given by the polytropic efficiency is a ratio of polytropic work divided by the actual work. So if you are writing in terms of specific work this is nothing but integral vdp integration has to be performed from Pi to Pe divided by he minus hi. So this is your polytropic work and this is your actual work okay. So this is got to be actual enthalpy rise ah and this process as you know the polytropic process can be written as something like this Pv to the power of n is constant.

So all polytropic processes can be represented by this equation and if you substitute this equation here you can find out the expression for polytropic work here okay. So I am making use of this equation here so. You find that the expression for polytropic work is given by f into n by n minus one into Pi vi into Pe by Pi to the power of n minus one by n minus one

Okay. Now let me explain the terms this f is a correction factor this is a correction factor this takes care of the variation of the index n during the compression okay. And this factor is actually negligible it is found that its value varies from one to one point zero two okay. That means the error will be a maximum error will be about two percent. So most of the times you can neglect this but if you are very particular then you can include this term and this term has to be obtain from experiments and this takes care of the variation of the index n. Where n is as you can see is the index of the equation Pv to the power of n is constant okay.

And Pi is the inlet pressure that is the suction pressure and vi is the specific volume of the refrigerant at the suction condition and Pe is the exit pressure or the discharge pressure okay. So these are the different terms of this equation and as I said this equation is obtained by simply integrating integral vdp from Pi to Pe.

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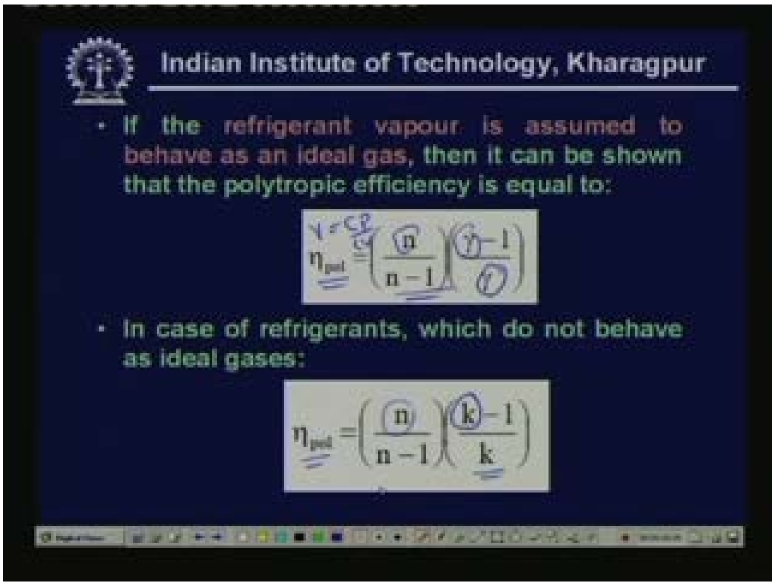

And if the refrigerant vapour is assumed to behave as an ideal gas then it can be shown that the polytropic efficiency is equal to this expression. So under the assumption that refrigerant is behaving as an ideal gas we can get a very simple expression for the polytropic efficiency in terms of the index n and gamma and gamma as you know is the specific heat ratio CP by CV okay. So this equation is valid only for the ideal gases okay. So in case of refrigerants as you know which do not really behaves as a ideal gases. What is normally done is this gamma is replaced by the isentropic index of compression k okay. That means the polytropic efficiency is simply given by n by n minus one into k minus one by k where n is the index of polytropic compression k is the index of isentropic compression okay. So this is a simplification actual analysis is very complicated but this is the generally practised. So that you can have a rough idea of the polytropic efficiencies okay.

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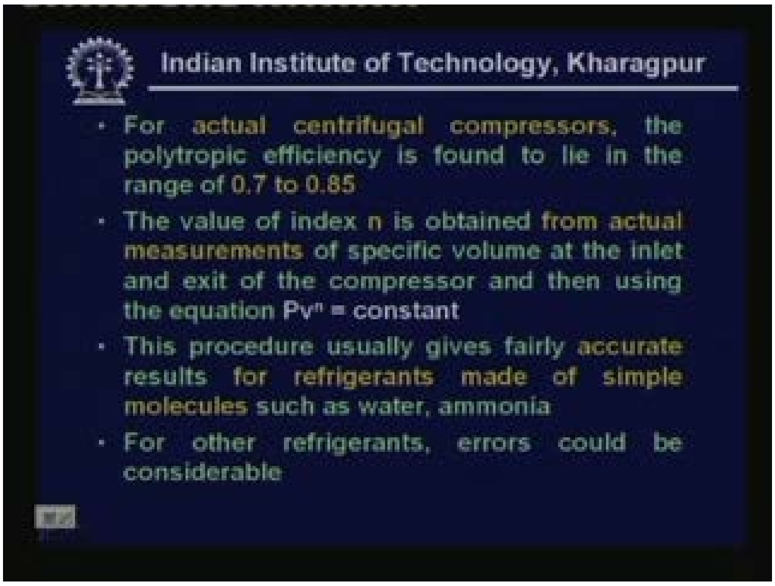

For actual centrifugal compressors the polytropic efficiency is found to lie in the range of point seven to point eight five. As I said this is obtained from experiments and from experimental value show that the efficiency lies between seventy to eighty- five percent the value of index n is obtained from actual measurements of specific volume at the inlet and exit of the compressor. And then using the equation Pv to the power n is constant okay. So the actual process will be irreversible. So you really do not know what is the path of the process. So in order to find out this n what is done is experiments are conducted and you find the suction pressure and suction specific volume you measure the suction pressure and suction specific volume.

And you also measure the exit pressure and exit specific volume okay. So you know Pi vi and Pe ve then use the equation Pv to the power of n is constant. That means Pi vi to the power of n is equal to Pe ve to the power of n and pi vi and Pe ve are known. So from this you can find out the value of n okay. So these how the index n is obtained in practise this procedure usually gives fairly accurate results for refrigerants made of simple molecules such as water or ammonia for other refrigerants errors could be considerable okay. So this procedure and the expression for polytropic efficiency and all is not a hundred percent correct method or it does not give you hundred percent correct value okay. And it is obtained, it is found that it generally gives reasonably good results if you are talking about simple refrigerants made of simple molecules such as water or ammonia okay. Then this procedure is pretty good whereas for other refrigerants for example for other heavy synthetic refrigerants this may give rise to errors okay.

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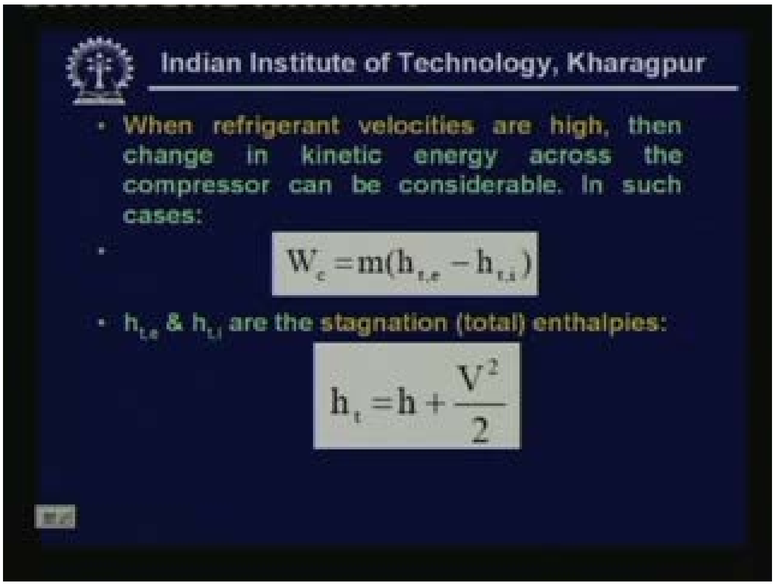

When refrigerant velocities are high then change in kinetic energy across the compressor can be considerable. So far we have been we assume that the kinetic energy change across the compressor is negligible okay. But in centrifugal compressor you find that velocities can be quite high and the kinetic energy change could be quite considerable okay. So in some cases you have to consider the kinetic energy change you cannot neglect it okay. So if you consider the kinetic energy change how do you find the power input the power input is simply obtained by this expression.

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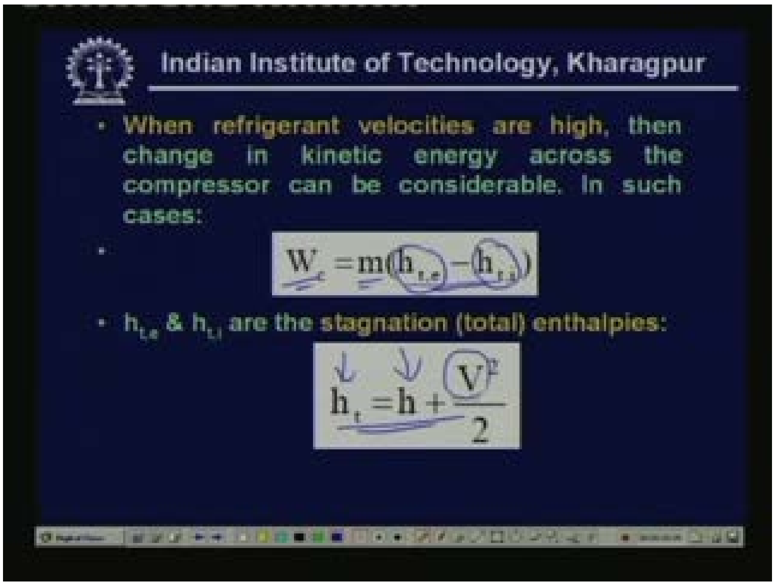

Where Wc is the power input this is again the mass flow rate into the h t e minus h t i and where this h t e and h t i are not the specific enthalpies. But they are what is known as stagnation or total enthalpies at the exit and inlet of the compressor okay. So when the kinetic energy is not negligible you have to use static you have to use stagnation enthalpies instead of static enthalpies okay. And these two are related by this expression ht is the stagnation or total enthalpy and h is the static enthalpy and where V is the velocity of the refrigerant okay. As you can see here when the velocity is negligible ht will be almost same as h that means the static enthalpy will be close to the stagnation enthalpy.

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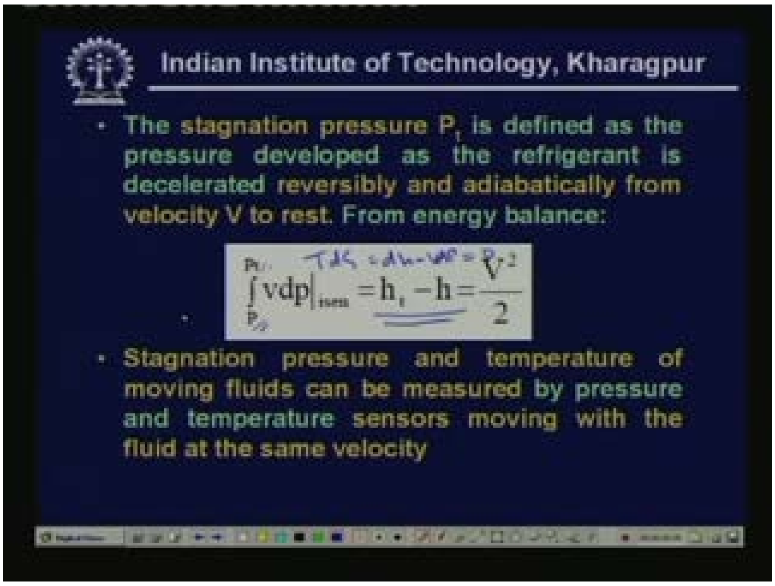

The similar to stagnation enthalpy or total enthalpy you can also defined what is known as stagnation pressure and stagnation temperature. The stagnation pressure pt is defined as the pressure developed as the refrigerant is decelerated reversibly and adiabatically from velocity V to rest okay. And from energy balance you can show that so when you adiabatically reversibly and adiabatically bring down the velocity from value V to zero you find that you get this expression ht minus h is equal to V square by two and you get this expression again to get this expression you have to use Tds is equal to dh minus vdp and this is a reversible adiabatic process so this is equal to zero.

So dh is equal to vdp okay so dh is nothing but ht minus h. So ht minus h is nothing but integral vdp along the isentropic process and the integration has to be performed from the static pressure P to the stagnation pressure pt okay. And this is equal to V square by two stagnation pressure and temperature of moving fluids can be measured by pressure and temperature sensors moving along with the fluid at the same velocity. So we want to find out the stagnation pressure or stagnation temperature of a moving fluid only moving fluids have stagnation properties so if you want to find out the stagnation pressure or temperature you have to have the sensors which move along with the fluid at the same velocity as that of the fluid okay. So the measurements recorded by these sensors moving at the same velocity will give you the stagnation pressure and stagnation temperature for an ideal gas you can have assumptions for an ideal gas you can write.

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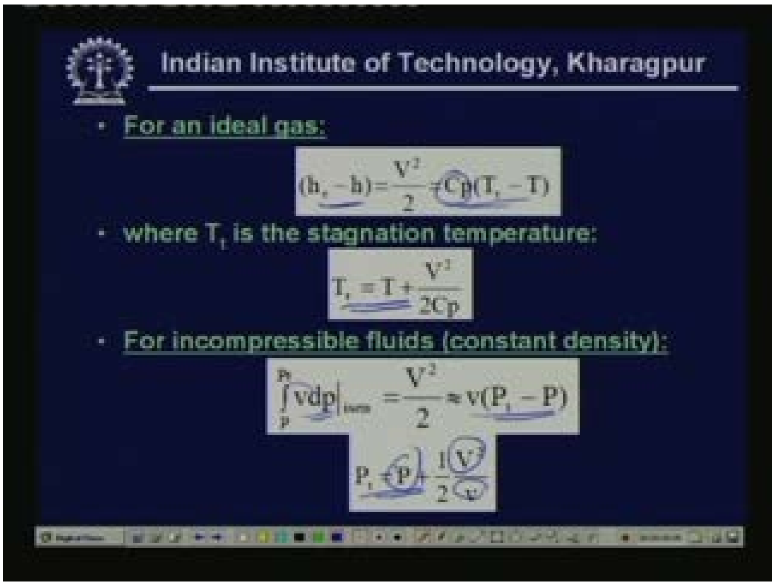

ht minus h as you know the enthalpies of an ideal gas depend only on temperatures. So ht minus h can be written as Cp into Tt minus T where Tt is the stagnation temperature and T is the static temperature here we are assuming that the Cp is constant or you are using a constant average specific heat okay. So you can write ht minus h as Cp into Tt minus T and this as you know is equal to V square by two.

So from this expression you get an equation for the stagnation temperature Tt that is equal to T static temperature T plus V square by two Cp okay. And you have to be careful about the units here when you are using Cp as kilo joule per kg Kelvin and if you are using V as metre per second then you have to divide this expression by thousand okay, because the only then the units will tally. So you can see here that when the velocity is small static pressure will be almost same as the static temperature is almost same as the static stagnation temperature okay. And for incompressible fluids that means for fluids which whose density remains almost constant you know that the integral vdp along the isentropic process V is the specific volume which is almost constant.

So you can write integral vdp as V into Pt minus P where Pt is the stagnation pressure and P is the static pressure this is equal to V square by two. So from this equation you can get an expression for the stagnation pressure in terms of the static pressure P velocity V and the specific volume small v okay. So this is how you can find out the stagnation properties right and as I said the stagnation properties become important when the velocities are very high okay.

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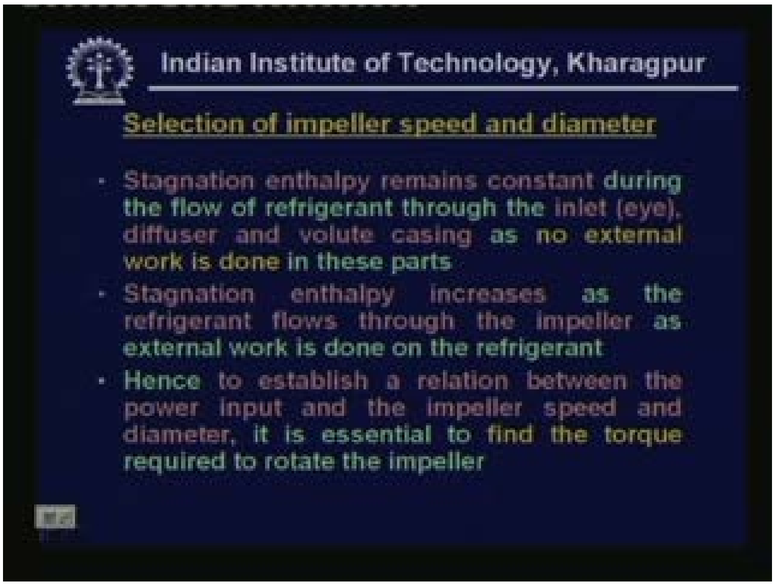

Now let us look at the selection of impeller speed and diameter which is very important in the design of centrifugal compressors. How do we select these and what is the relationship between these parameters and the performance of the compressor. Now let us once again let us look at the working principle and from the working principle. We know that the stagnation enthalpy remains constant during the flow of refrigerant through the inlet through the diffuser and through the volute casing as no external work is done in these parts. So if you look at the refrigerant flow through the inlet refrigerant flow through the diffuser and refrigerant flow through the volute casing in these three parts we are not adding any energy or you are not removing any energy okay. That means stagnation enthalpy remains same okay. Only thing that happens here is velocity will be converted into static pressure otherwise the total enthalpy remains constant okay.

Whereas in the impeller you are adding work input okay. So in the across the impeller you find that the stagnation enthalpy is increasing okay. So stagnation enthalpy increases as the refrigerant flows through the impeller as external work is done on the refrigerant. Hence to establish a relation between the power input and the impeller speed and diameter it is essential to find the torque required to rotate the impeller. So the basic principle as I have mentioned at the very beginning is that you impart angular momentum to the refrigerant. And then you convert the momentum into static pressure okay. So work is required to impart the angular momentum to the refrigerant. So we would like to find out what is the power input or what is the work input to the compressor. So in order to find that first we want to find out what is the torque to be imparted to the refrigerant. So we first find the torque and from the torque we find the power input okay.

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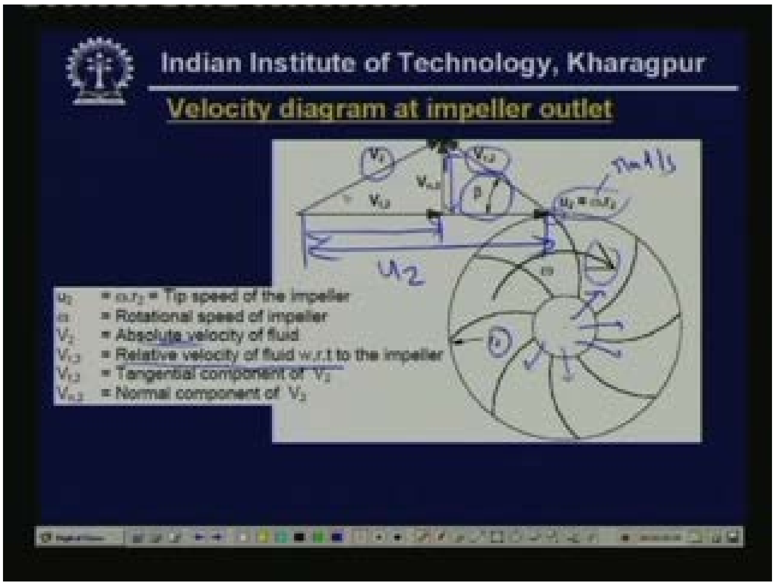

To find the torque we have to look at the velocity diagram at the impeller outlet okay. So this picture shows the velocity drawing velocity diagram at the impeller outlet. Of course this is, I am showing the velocity diagram at the outlet and here I am assuming that the refrigerant enters axially at the inlet okay. So at that means at the inlet it doesn’t have any tangential component it enters axially okay. So this is one of the assumptions so you can see here that the impeller is rotating in this direction okay. So this is the direction of rotation right and if you are drawing the velocity diagram at the tip of the impeller. That means at this point you find there are different velocities and what are the velocities first you have what is known as the velocity peripheral velocity or tip speed that is given by u two. And that is nothing but this okay.

So this is your u two and what is this is nothing but the product of the rotational speed of the impeller in radiance per second this is in radiance per second multiplied by the impeller radius r two okay. So if you multiply the rotational speed with the impeller radius you'll get the tip speed u two okay. And you also have Vr two okay. The subscript r stands for relative and two is the exit. So V r two is nothing but the relative velocity as you can see here relative velocity of the refrigerant with respect to the impeller okay. And if this is the relative velocity of the refrigerant and if this is the u two is the peripheral velocity of the impeller. Then the difference between these two is nothing but your absolute velocity of the refrigerant that is given by capital V two okay. So capital V two is the absolute of the refrigerant this absolute velocity can be split into tangential component Vt two this is the tangential component Vt two and normal component Vn two okay. So this is the typical velocity diagram at the impeller outlet.

So we have four velocities that is the let me repeat once again you have the tip speed small u two and absolute refrigerant velocity capital V two. And the relative velocity of the refrigerant V subscript r two and then you have the tangential component of the absolute velocity V subscript t two and the normal component V subscript n two. And this angle beta is known as the blade angle. If you are assuming that there is no slip I will explain what is the slip okay, little later. So this is the velocity diagram at the impeller outlet. Now how do we use this velocity diagram to find the torque and from that how do you find the power input.

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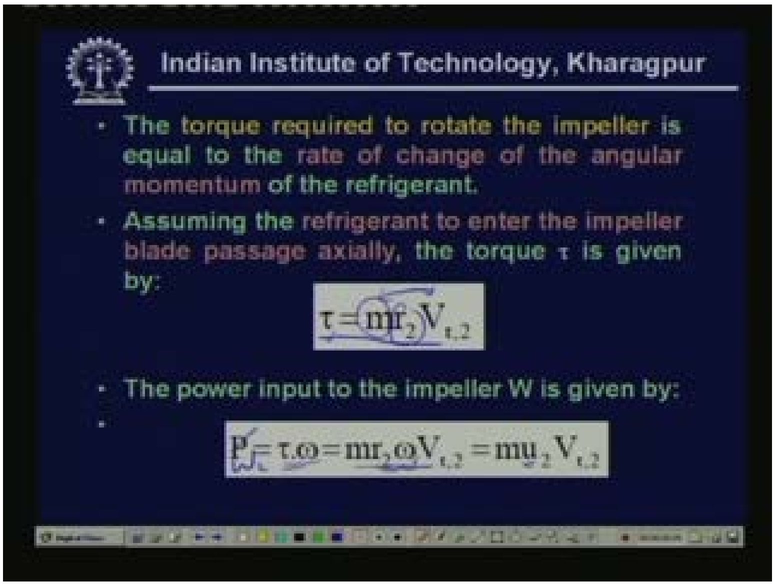

The torque required as you know is equal to the rate of change of the angular momentum of the refrigerant okay. This is the definition of the torque and assuming the refrigerant to enter the impeller blade passage axially. That means it does not have any tangential component at the inlet the torque tau is given by this expression. Where m is the mass flow rate r two is the radius of the impeller and Vt two is the tangential velocity of the refrigerant okay. So basically mass flow rate into velocity will give you the linear momentum multiplied by multiply that into the radius r two will give you the torque okay. This is the rate of, since we are talking about the mass flow this will give you the rate of change of angular momentum which is nothing but the torque tau okay.

Now the power input to the impeller is simply equal to tau into rotational speed omega okay. So power input P or i am using actually the capital W okay, let me stick to that. So capital Wc is nothing but the torque multiplied by the rotational speed that is tau into omega. And you substitute the expression for tau here tau is nothing but m r two into V t two. So power input becomes m r two into omega into V t two. And as you know r two into omega is nothing but u two which is nothing but the tip speed. So finally we get that the power input for this particular case is given by mass flow rate into the tangential velocity component Vt two into tip speed of the impeller u two okay.

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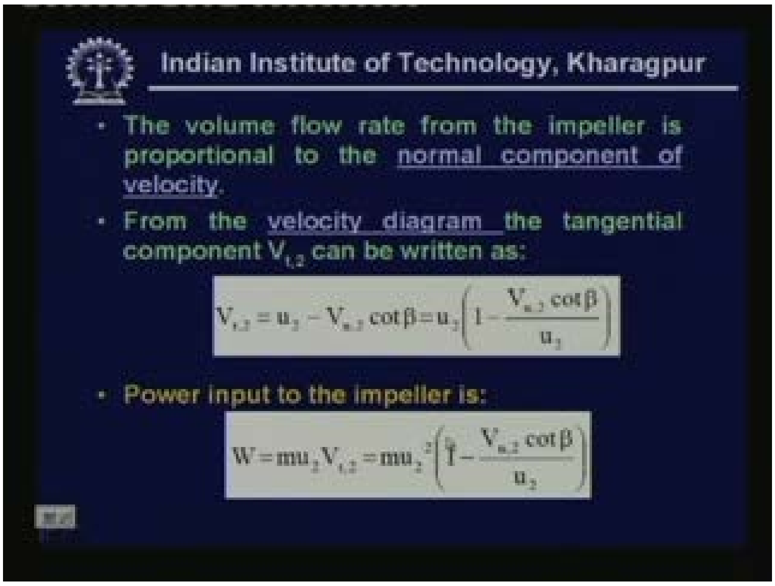

Now the volume flow rate from the impeller is proportional to the normal component of velocity okay.

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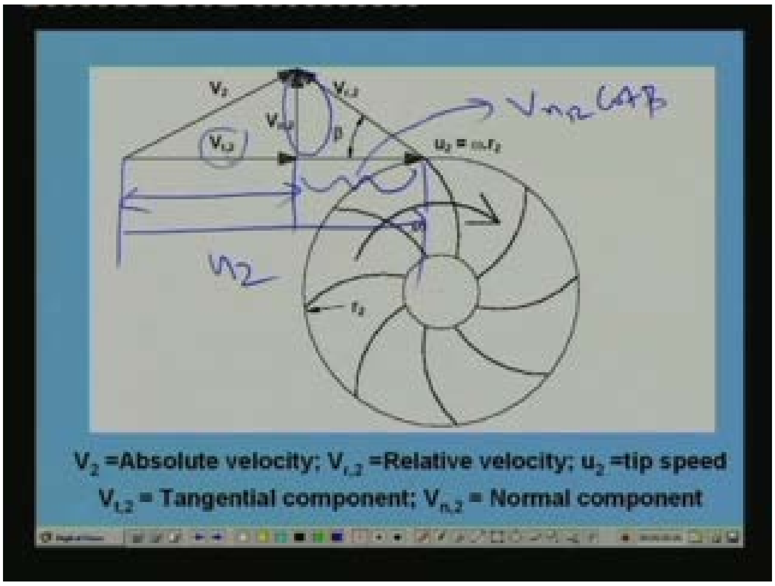

So the volume flow rate basically depends upon this okay. This is the normal component of velocity and we have just now shown that the power input and torque depends upon this component. Now this component can be written in terms of the tip speed u two and the normal component. How, this is nothing but the peripheral velocity u two minus this okay. And this is nothing but Vn two into cott beta okay. So this distance is Vn two into cotangent beta. So u two minus Vn two into cotangent beta will give you Vt two okay.

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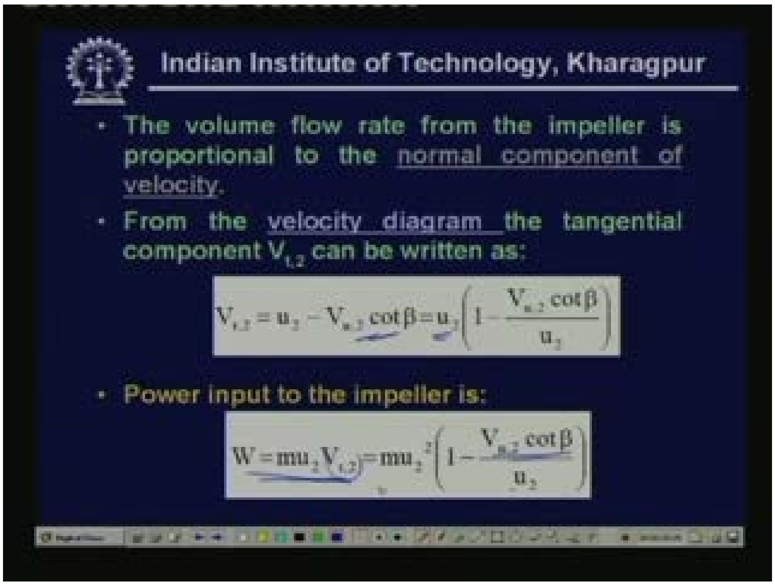

So that is what is done here from the velocity diagram the tangential component Vt two can be written as Vt two is equal to u two minus Vn two cot beta. And here you take the u two as the common okay. So this can be written as u two into one minus Vn two cot beta by u two okay. So now you substitute this expression in the expression for power input we had the expression for input like this. So we are writing power input as m into u two into Vt two Vt two we are substituting in this expression. So finally this becomes m into u two u two square multiplied by one minus Vn two cot beta divided by u two okay. The objective of this is to finally correlate the power input and the pressure rise to the geometry of the impeller and the impeller speed okay. That is why we are doing this manipulations right.

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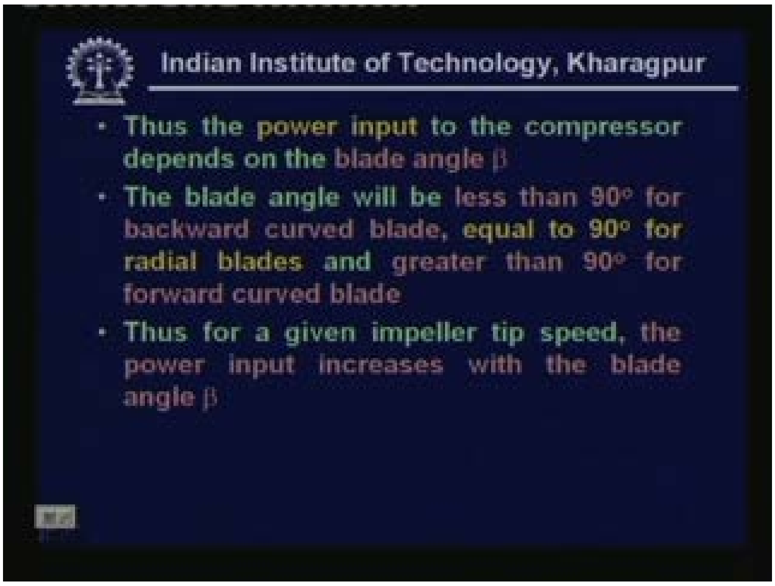

Thus the power input to the compressor depends upon the blade angle beta. As I said the blade angle will be less than ninety degree for backward curved blade. This is important to remember equal to ninety degree for radial blades and greater than ninety degrees for forward curved blades thus for a given impeller speed the power input increases with the blade angle beta. Because from the expression for power input you find that you have one minus cot one minus Vn two into cot beta by u two okay. So as beta increases your cot beta reduces okay. Once the cot beta reduces the power input increases okay.

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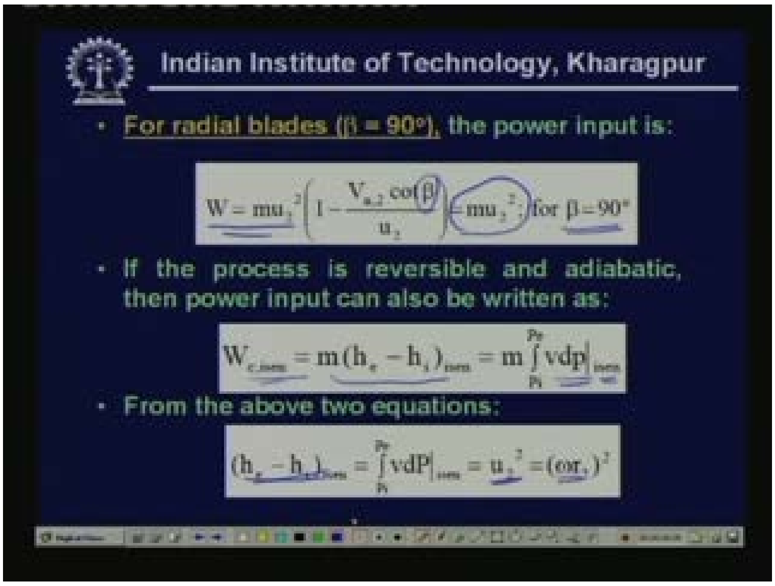

Now let us take a simple case as I said the actual analysis of centrifugal compressor is very complicated okay. So what we are i am presenting here is a simple analysis okay. Analysis become simple if you assume that there is no tangential component at the inlet and if you also assume that the blade is radial okay. Once the blade is radial beta becomes ninety degree ninety degrees okay. So for radial blades when beta is ninety degrees the power input is given by this expression. So here beta is ninety so cot beta becomes zero so power input is simply equal to m into u two square okay. So you get a very simple expression which directly relates the power input to the tip speed okay. Remember that this is only valid for beta is ninety degrees and with note tangential velocity at the inlet right.

And if you are assuming that the process is reversible and adiabatic you can also write the power input in terms of the enthalpy rise of the isentropic process and the mass flow rate okay. That we have seen earlier so power input is also equal to mass flow rate into isenthalpic isentropic enthalpy rise. So this as we have seen is nothing but m into integral vdp from Pi to Pe along the isentropic process okay. So if from if you equate this two equations you finally find that the enthalpy rise for the isentropic process is linked to your tip speed u two square and tip speed is nothing but the product of rotational speed into the radius of the impeller okay. So that is how we link the enthalpy rise to the blade parameters.

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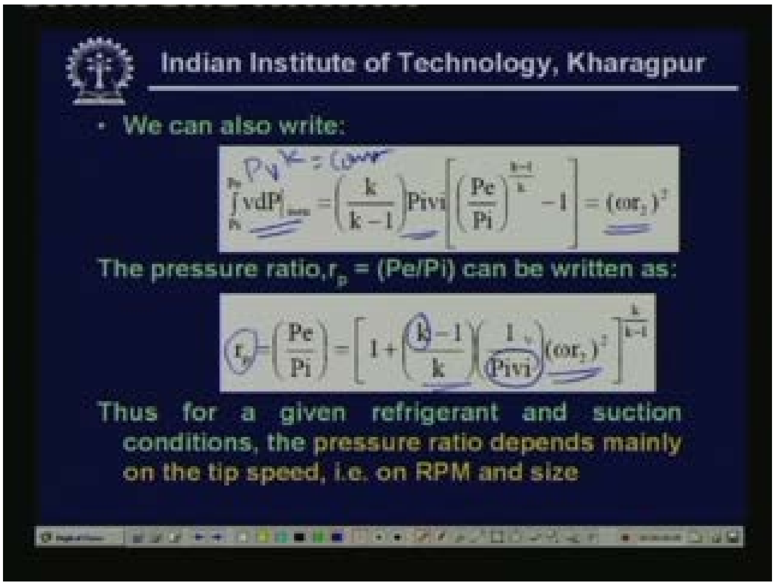

So we can also write this. We can also perform the integration and we can if you perform the integration you get this expression. Because for isentropic process as you know Pv to the power of k is constant where k is isentropic index of compression. So if you substitute that here you get this expression okay. And this whole thing is equal to omega into r two square from this expression you do mathematical manipulation a simple manipulation. You can show that the pressure rise pressure ratio. I am sorry pressure ratio rp that is discharge pressure divided by the inlet pressure is given by this expression okay. And you can see that for a inlet conditions Pi and vi and for a given refrigerant let, that means, let assume that k is constant you find that the pressure ratio depends purely on the rotational speed of the impeller and the impeller diameter okay. So that is how we established relationship between the power and the pressure ratio with the impeller parameters okay. That is the whole objective of this analysis.

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Generally from material strength considerations the tip speed is limited to about three hundred metre per second okay. This industry standard normally in industry they do not exceed this speeds. So this puts an upper limit as I have already explained on the temperature lift with a single stage centrifugal compressor hence for larger temperature lifts required we require multi-stage compression. For a given impeller rotational speed and impeller diameter the pressure rise also depends on the type of the refrigerant used you have seen that in the pressure ratio expression you have Pi vi and you also k okay. All these parameters depend upon the type of the refrigerant used okay. Let me give a small example.

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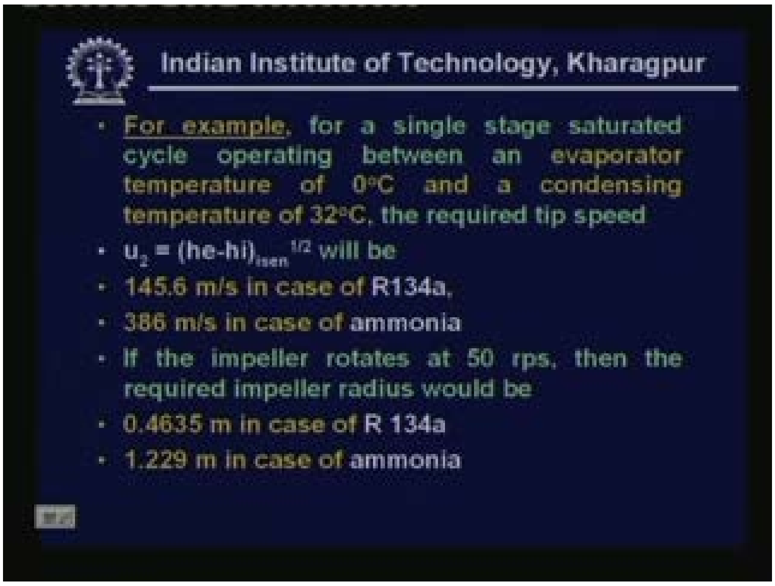

For example for a single stage saturated cycle operating between an evaporator temperature of zero degree centigrade. And a condensing temperature of thirty-two degree centigrade the required tip speed to the calculations required tip speed from the expression is nothing but square root of he minus hi along the isentropes where he is the enthalpy at the exit of the compressor and hi is the enthalpy at the inlet to the compressor. And in this case am assuming saturated cycle. So hi is nothing but the saturated enthalpy at zero degree centigrade and he is the enthalpy of the super heated refrigerant at condenser pressure. And entropy same as the inlet entropy okay. So this you can find out from the refrigerant property tables and if you find the properties from the refrigerant property table. And substitute these values you find that the required tip speed will be one forty five point six metre per second in case of one thirty four a refrigerant R one thirty four a and the required tip speed will be three eighty six metre per second in case of ammonia.

So you can see that if you are using ammonia the required tip speed becomes very large okay. So everything else remaining constant you find that for ammonia you require very high tip speed in fact this tip speed exceeds the safe limit or the maximum upper limit okay. So if you are using ammonia for these conditions and if you want to use a centrifugal compressor you may have to go for multi-stage compression okay. And if you can also calculate what is the required impeller radius if you are assuming that the impeller is rotating at fifty r p s and from the tip speed and the r p s you can find out what will be the required radius. You find that the required radius in case of R one thirty four a will be point four six three five metres. And in case of ammonia it will be one point two nine metres. That means for a given rotational speed you find that ammonia compressor if you are using a single stage will be very large compared to R one thirty four a okay. In fact we will see later for centrifugal compressor we find that normally the high boiling point refrigerants are more suitable okay, compared to low boiling point refrigerants.

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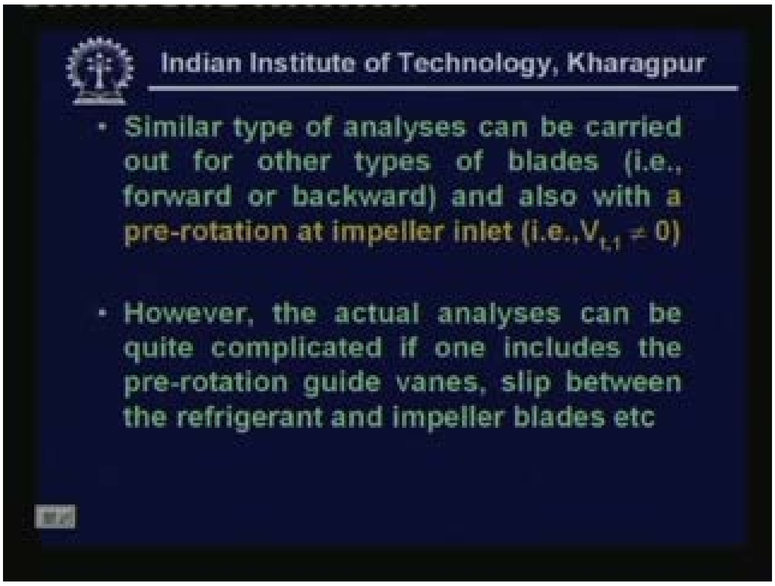

Similar type of analyses can be carried out for other types of blades that means forward or backward curved and also with pre-rotation at impeller inlet okay. So you can extend this analysis by including beta okay. Here we assume radial blades so beta become cot beta becomes zero. So beta does not appear in the expression for pressure ratio or the power input. But you can include the beta for other types of you have to include the beta for other types of blade. That means forward or backward curved blades and if you have any tangential velocity at the inlet you also have to include that in the expression for the torque okay. However the actual analyses can be quite complicated if one includes the pre-rotation guide vanes slip between the refrigerant and impeller blades.

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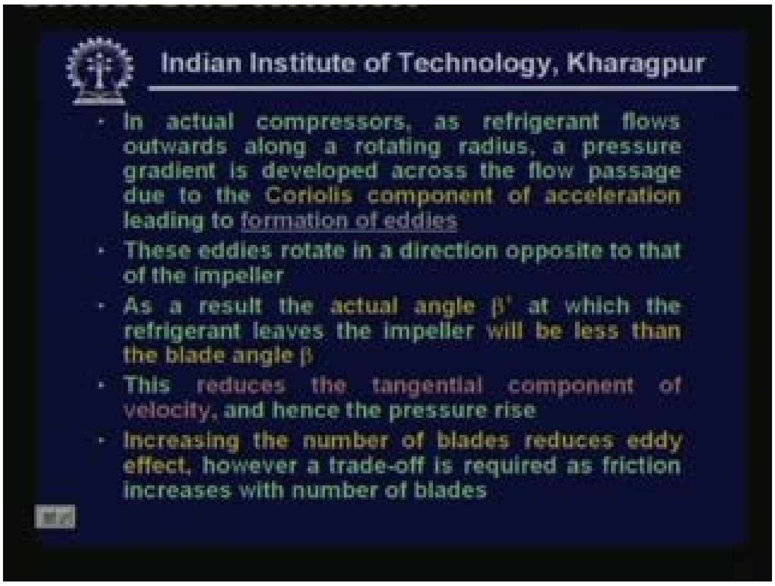

In actual compressors as refrigerant flows outwards along a rotating radius a pressure gradient is developed across the flow passage due to the coriolis component of acceleration leading to formation of eddies okay. I was mentioning slip if there is no slip then the angle beta is nothing but the blade angle that is what I said okay. But what is this slip? This slips comes into picture because the refrigerant has to flow along a radius which is rotating at high speeds okay. This introduces what is known as a coriolis component of acceleration okay. And this also introduces pressure gradient across the refrigerant passage because of this pressure radiant a secondary flows are induced in the flow passages. And this gives rise to what is known as eddies okay, for example you can see here.

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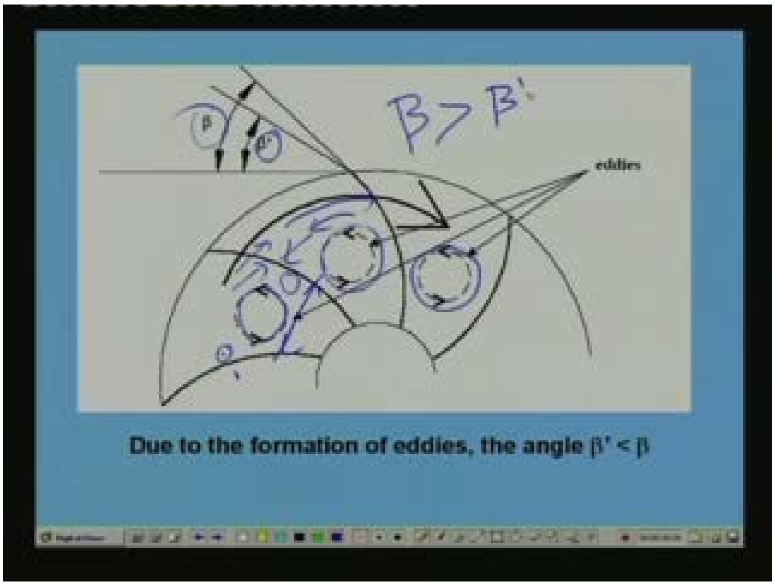

These are the eddies formed in the flow passages okay. And if you look at the pressure you find that pressure on this side of the impeller will be greater than the pressure on this side okay. Only then this blade can rotate in this direction. So here the pressure will be greater and if you look at this side of the channel you find that this pressure will be lower than this pressure because this side pressure will be again higher than this pressure. That means there is a pressure radiant across the channel and this pressure radiant gives rise to eddies of this form okay. These are nothing but the secondary flows inside the flow passages and you can see that the eddies rotate in an opposite direction when the impeller is rotating in this direction eddies are rotating in this direction okay.

And because of the presence of the eddies you find that the velocity at which the refrigerant leaves the impeller will be beta dash okay, whereas beta is the blade angle okay. So because of the eddy formation you find that beta is greater than beta dash okay. These eddies rotate as I said in a direction opposite to that of the impeller. As a result the actual angle beta dash at which the refrigerant leaves the impeller will be less than the blade angle beta this reduces the tangential component of velocity and hence the pressure rise. So what is the effect of these eddies. Obviously these eddies induce some losses and because of these eddies you find that the actual tangential component at the tip of the impeller is less than the ideal tangential velocity. In fact the ratio of the actual tangential velocity divided by the ideal tangential velocity is known as slip okay. Because of this slip the pressure rise will be reduced the pressure rise can be this effect can be taken care of by increasing the number of blades. That means by making the channels narrower.

However as you make the channels narrower you find that the frictional pressure drop increases. So you have to make it trade-off between the eddy formation and the frictional pressure drops and decide the number of the blades okay. So this at this point I stop this lecture and we will continue this lecture in the next class where we discuss the performance aspects of the centrifugal compressors okay.

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Now let me conclude briefly in this lecture the working principle and constructional features of centrifugal compressors are explained and steady-flow analysis of compressors is presented and a simple analysis to establish a relationship between power input pressure rise impeller diameter and impeller speed is presented okay. And as I said we will continue this lecture in the next class.

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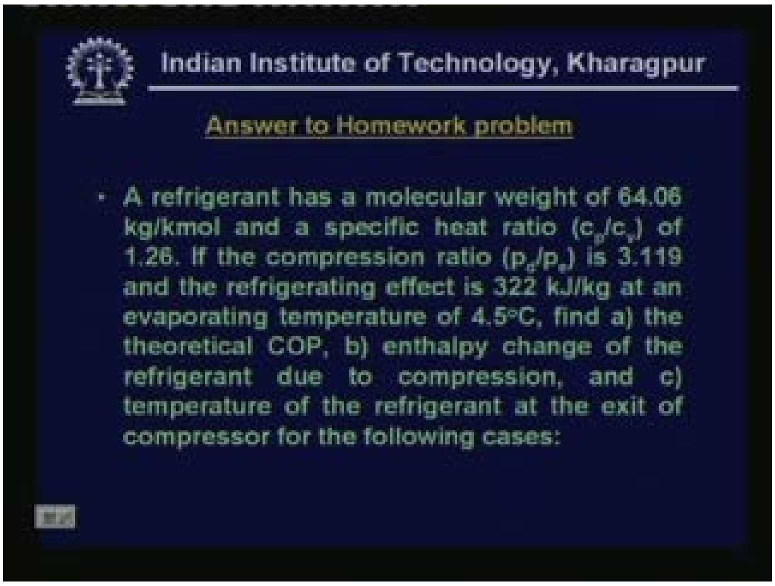

And before I leave let me give the answers to the homework problem given in the last class okay. This is the homework problem given in the last class and I have asked you to calculate the COP enthalpy change and the temperature of the refrigerant for the following three cases.

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First is the reversible adiabatic compression the answers are COP six point nine eight enthalpy change is forty-six point one five kilo joule per kg. And the temperature at the exit is seventy-eight degree centigrade and for reversible polytropic compression with n is one point

two two the answers are like this COP is seven point zero nine enthalpy change is thirty-nine point seven two kilo joule per kg and the exit temperature is sixty-seven point seven degree centigrade. And for the third case of irreversible adiabatic compression with index n is equal to one point three the COP is six point one five enthalpy change is fifty-two point three six kilo joule per kg and the exit temperature is eighty seven point nine degree centigrade okay.
Thank you.

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Welcome back this lecture is the continuation of the earlier lecture where in 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 phenomena of surging in centrifugal compressors, discuss the effects of operating temperatures and speed on performance comparison with reciprocating compressors. And finally discuss briefly commercial systems with centrifugal compressors.

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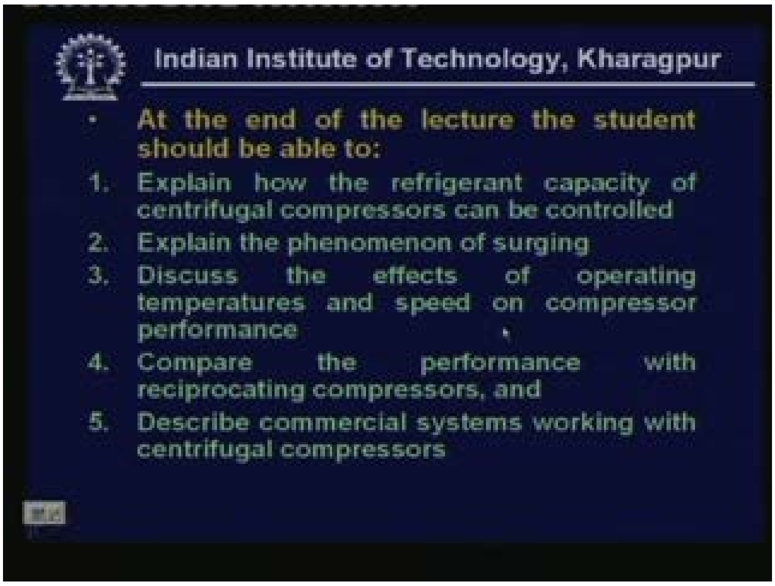

At the end of this lecture you should be able to explain how the refrigerant capacity of centrifugal compressors can be controlled, explain the phenomenon of surging discuss the effects of operating temperatures and speed on compressor performance, compare the performance with reciprocating compressors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

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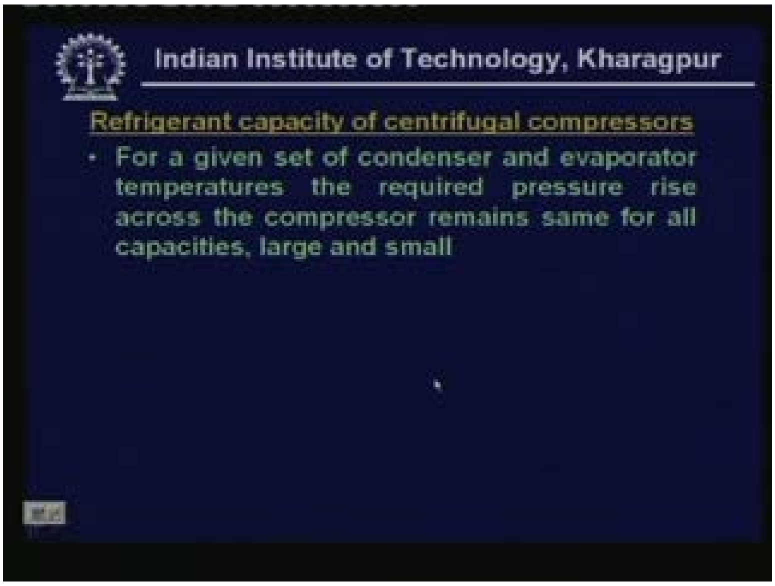

Now let us look at the refrigerant capacity of centrifugal compressors for a given set of condenser and evaporator temperature the required pressure rise across the compressor remains same for all capacities large and small. That means if you fix the evaporator and condenser temperatures obviously evaporator 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 are 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.