Refrigeration and Airconditioning
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Lecture No. # 28
Refrigeration System Components: Condensers and Evaporators

How to the problems on condensers then I will introduce refrigerant and evaporators.

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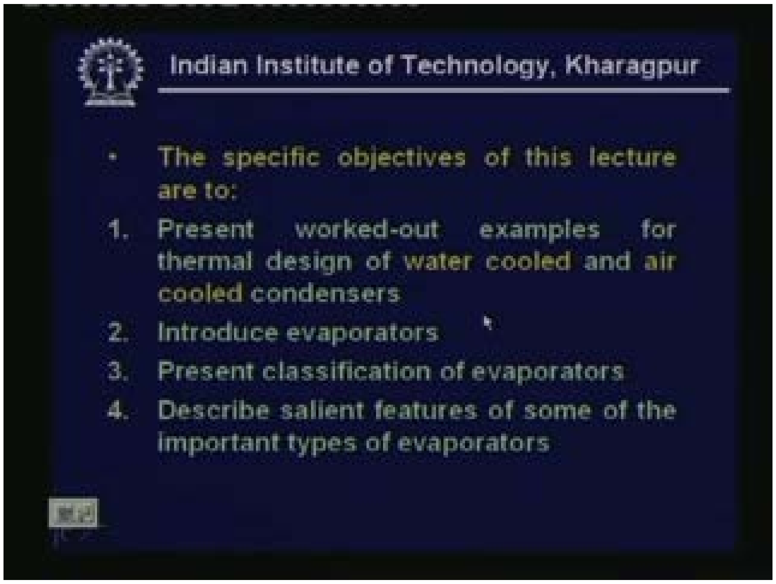

So the specific objective of this particular lecture are to present worked-out examples for thermal design of water cooled. And air cooled condensers introduce evaporators present classification of evaporators describe salient features of some of the important types of evaporators.

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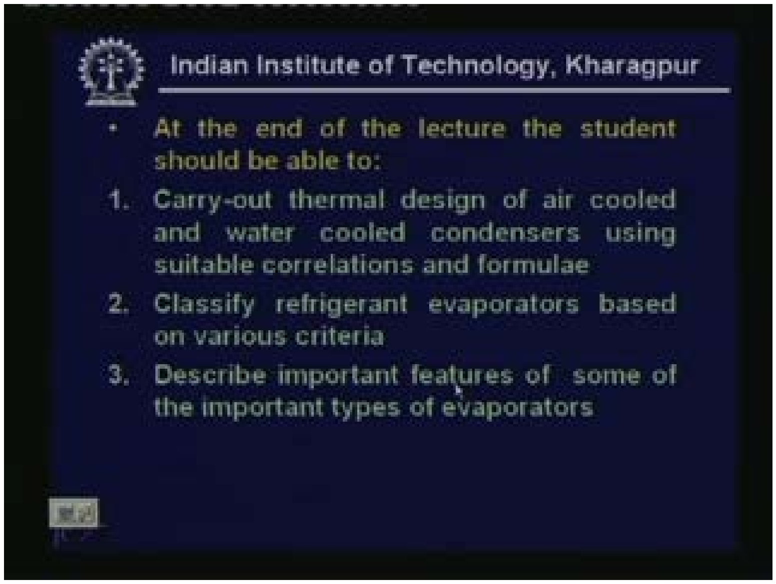

At the end of the lesson you be, should be able to carry out thermal design of air cooled and water cooled condensers using suitable correlations and formula classify refrigerant evaporators based on various criteria, describe important features of some of the important types of evaporators.

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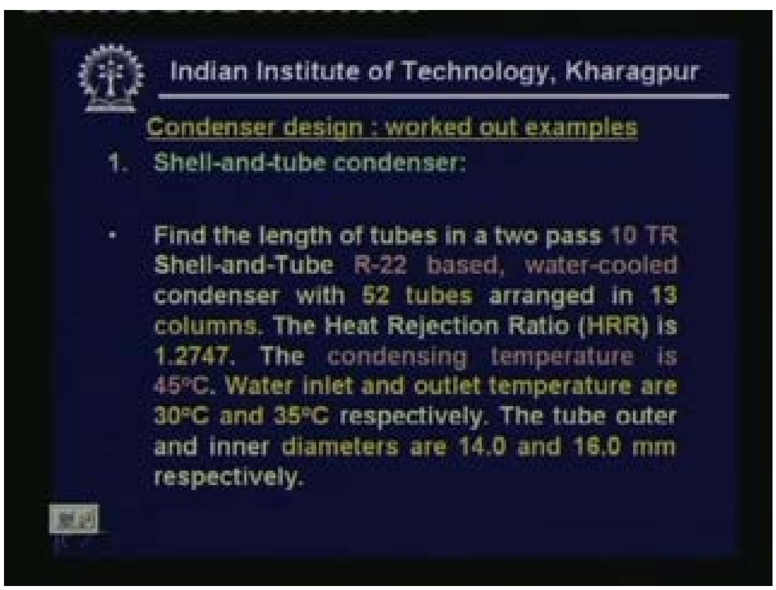

So let me begin with a worked out example. This an example on shell and tube condenser and the problem statement is like this. You have to find the length of tubes in a two pass ten ton capacity shell and tube R twenty-two based water cooled condenser with fifty-two tubes arranged in thirteen columns. The heat rejection ratio is given as one point two seven four seven the condensing temperature is forty-five degree centigrade water inlet and outlet temperatures are thirty degree centigrade and thirty-five degree centigrade respectively. The tube outer and inner diameters are fourteen and sixteen mm respectively. So uh the problem sure must be clear to you it’s an R twenty-two based condenser water cooled condenser it has fifty-two tubes and thirteen columns fifty tubes are arranged in thirteen columns and the other data is as per the problem. So now let as look at how to solve the problem.

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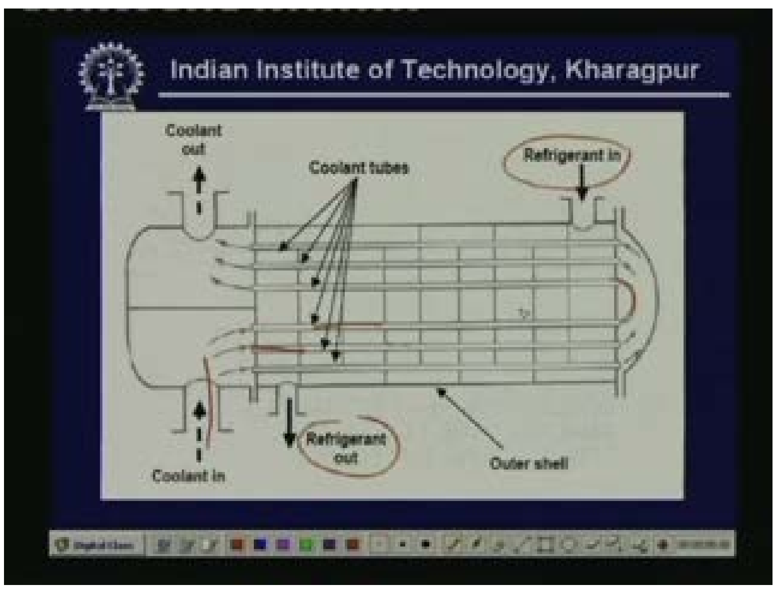

So this is the schematic in fact, I have last class, I have shown this schematic of a two pass shell and tube type of a condenser. And as I have explained to you here refrigerant is there on the shell side okay. You can see that refrigerant is entering at the top and it condensers as it comes in contact with the tubes. And the liquid refrigerant goes out from the bottom and water enters from the bottom like this coolant is nothing but the water. And it is flows through these tubes. Since it is a two pass the same water again will flow through the condenser tubes okay. It comes here and again its goes back to the condenser tubes that is why you call it as a two pass condenser. That means the water makes two passes as it flows to the condenser and in the problematic mention that there are fifty-two tubes okay. And they are arranged in thirteen columns.

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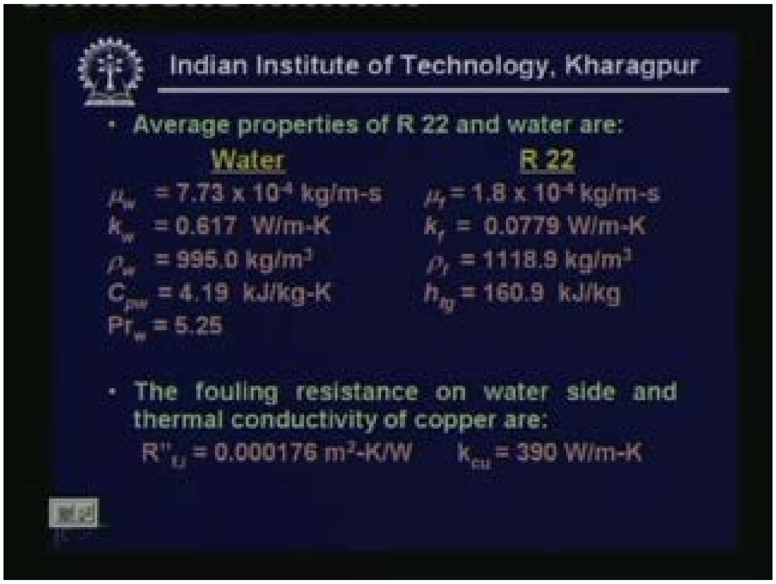

Now for the first thing you have to do you have to find out the properties of R twenty-two and water. Because we have to evaluate the heat transfer coefficient and all. So these are the properties at average temperature for water viscosity is given as seven point seven three into ten to the power minus four kg per meter second. Thermal conductivity is given as point six one seven watt per meter Kelvin and density is nine ninety-five kg per meter cube specific heat of water is given as four point one nine kilojoules per kg K and Prandtl number is given as five point two five. P r is Prandtl number and for R twenty-two the average properties are the viscosity is one point eight into ten to the power minus four kg per meter second thermal conductivity kf is point zero seven seven nine watt per meter Kelvin and density is eleven hm thousand one eighteen point nine kg per meter cube and latent heat of vaporization is one sixty point nine kilo joule per kg.

This is at a condensing temperature and is also given in the problem that the fouling resistance on water side and thermal conductivity of copper are fouling resistance is zero point zero zero zero one seven six meter squared Kelvin per watt. And the thermal conductivity of water is given as three ninety watt per meter Kelvin. So this is how the problem is stated so now we have to find out the required length of each tube okay.

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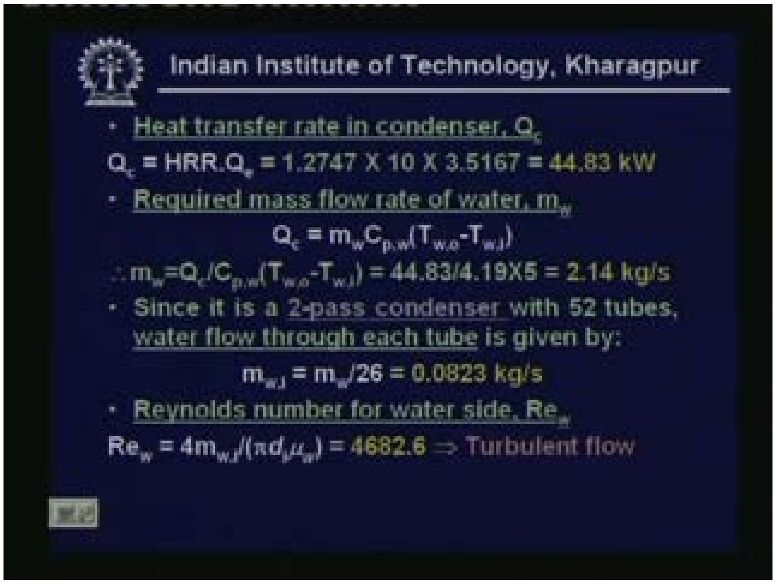

So first the step is to find out what is the required heat transfer rate in condenser you know that required heat transfer rate in condenser is nothing but Qc is equal to product of heat rejection ratio HRR into refrigeration capacity. And the heat rejection ratio is given as one point two seven four seven and the refrigeration capacity is given as ten tons. So we convert that into kilo watts. So this is ten into three point five one six seven kilo watt. So from this equation you find that the condenser heat transfer rate is forty-four point eight three kilo watt. Then we let us find the required mass flow rate of water.

So from energy balance we can also write Qc is equal to mw Cpw into Two minus Twi where as you know m w is the mass flow rate of water C p w is the specific heat of water Two and Twi are exit and inlet temperatures of water. And we know here, we know the values of Qc. We also know the value of Cp. We also know the temperatures. So if you substitute those values you find that the total mass flow rate of water is two point one four k g per second. Now this two point one four kg per second is distributed in many tubes okay. Since it is a two pass condenser it is fifty-two tubes water flow through each tube is given by this formula. This is m substitute w i is the water flow through each tube that is equal to mw by twenty-six. Why do we get twenty-six there are fifty-two tubes.

But it is a two pass two pass condenser that means in one hm in one pass the water flows through twenty-six tubes.

So the total flow rate is given by two point one four kg per second. So through individual tube a flow rate is two point one four divided by twenty-six so that is zero point zero eight two three kg per second. Now once we know the water flow rate and once we know the water properties we can calculate the Reynolds number for water side as, so you know Reynolds number for water side is Row v d by Mew. This can also be written in terms of the mass flow rate. So in terms of mass flow rate its four into m w i divided by pi d i mew w where m w i. As you have seen, is the mass flow rate through each tube and d i is the internal diameter of the tube and mew w is the viscosity of water. So if we substitute these values before everything is known to you if we substitute these values you find that the Reynolds number for water side is four six eight eight eight two point six. Since this is greater than two thousand three hundred we can this is turbulent flow. So we have to use the turbulent flow correlation for finding the heat transfer.

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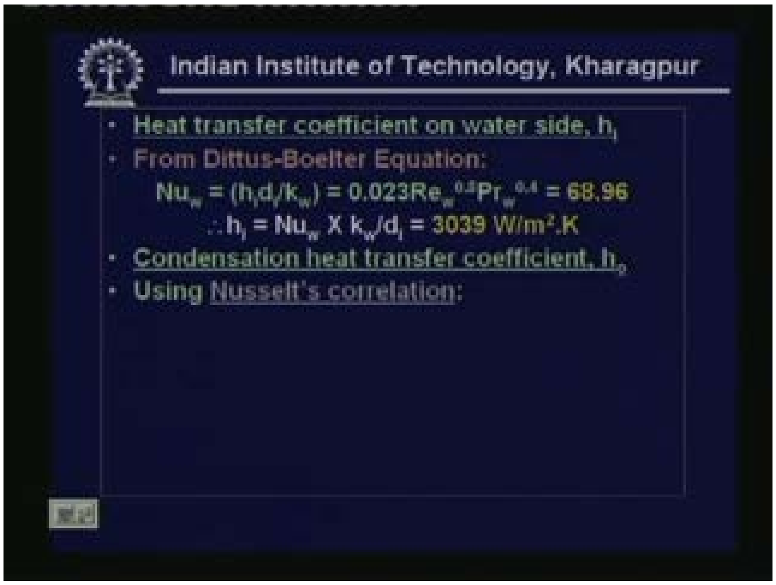

So let us find the heat transfer coefficient on water side. In fact in last class I have mentioned that if it is turbulent flow we can use Dittus Boelter equation or equation. So in this problem let us use Dittus Boelter equation Dittus Boelter equation as you know is nothing but Nusselt number is equal to point zero two three Reynolds number to the power of point eight Prandtl number to the power of point four. So we know Reynolds number we also know the Prandtl number. So if we substitute these values you find that the Reynolds or Nusselt number for water side is sixty-eight point nine six. So once you know the Nusselt number you can find out the heat transfer coefficient. Because heat transfer coefficient is nothing but Nusselt number into thermal conductivity of water divided by the diameter. So we substitute these values you find that heat transfer coefficient on water side is three thousand thirty-nine watt per meter squared Kelvin. Now let us find the hm heat transfer coefficient on the condensation side. So condensation heat transfer coefficient.

Since condensation is a, this is a shell and tube type of condenser. So obviously water is flowing through the tubes. So condensation is taking place outside the tubes and this is a, since nothing is mentioned we let us takes these as a horizontal shell and tube type of condenser. So let us use the correlation for horizontal condensational horizontal tubes. So in fact in the last class I have mentioned that we can use the classical correlation given by Nusselt in fact Nusselt's correlation is valid for laminar flow. For here this is valid in fact you can cross check that you will find that is valid here. So let us apply the Nusselt's correlation for finding the condensation heat transfer coefficient. So what is Nusselt's correlation.

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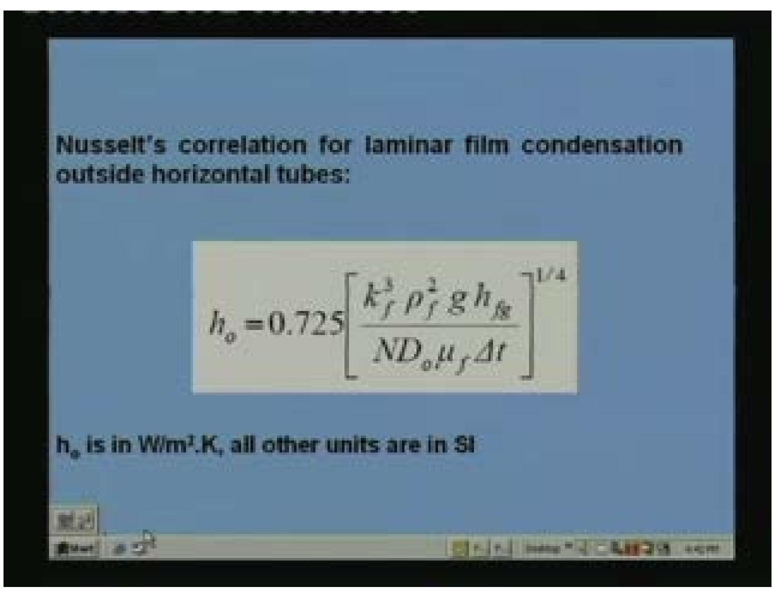

So this is the Nusselt's correlation for laminar film condensation outside horizontal tube I have explained this. In the last lecture okay, here as you know this is the thermal conductivity of saturated refrigerant. This the density of saturated refrigerant at condenser temperature and this is the acceleration due to gravity is the latent heat of vaporization at that temperature and pressure N is the total number of tubes in a row and D not is the outer diameter of the tube on which condensation is taking place mew f is the viscosity of the saturated liquid. And delta t is nothing but the temperature different between the refrigerant and the wall okay, surface, let us say okay. So this the Nusselt's correlation and here h not is in watt per meter squared Kelvin and all other units are in S I. So you have to use the S I units, this you have to be careful while using the units. So we know from the problem we know everything except delta t.

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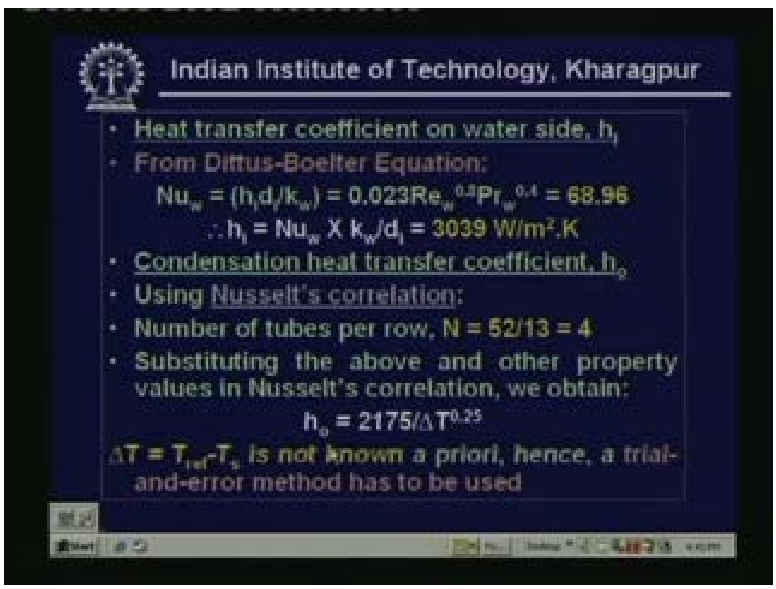

So let us find h not okay. So if you substitute number of tubes per row we have to find out that is capital N since it is mentioned that there are total number of there are fifty-two tubes and their distributed in thirteen rows. So number of tubes per row is nothing but fifty-two by thirteen that is four. So if we substitute this four and other property values we will find that the Nusselt correlation. From the Nusselt's correlation h not is given as two one seven five divided by delta T to the power of point two five okay.

So here delta T as i had already mentioned is nothing but the temperature difference between the condensing refrigerant and the surface this is not known to us. So what we have to do is, we have to use the trial and error method okay. Trial and error method means initially we use the guess value of delta T find out h not find out u not and all. And finally you have to cross check whether the guess value is correct value or not I will show you that the procedure now.

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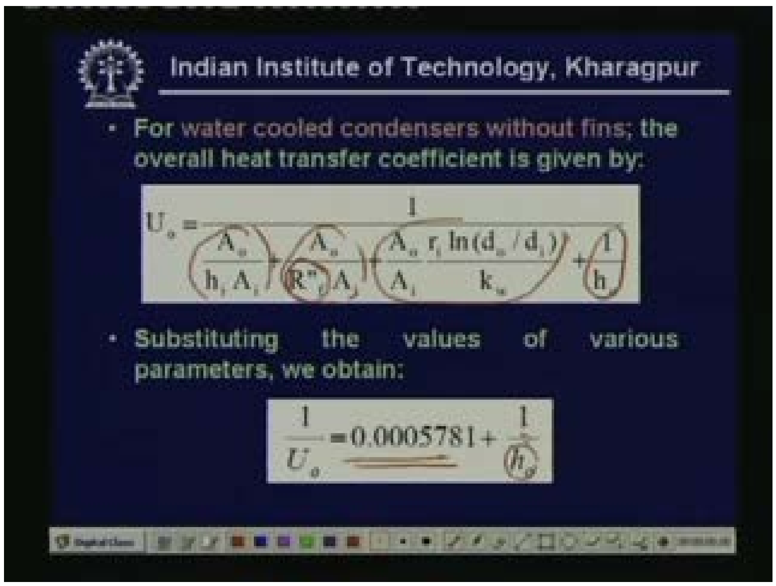

Before that for water cooled condensers without fins the overall heat transfer coefficient is given by this formula, this, all I have explained in the last class are we know this the convert to resistance the inside this the fouling resistance on water side. And this is the resistance offered by the wall and this is the X null resistance okay. Now everything is known to us because A not and i Ai and all can be express in terms of diameters. So diameters are given to us properties are all, so given and these fouling resistance is also mentioned in the problem statement. So we know everything okay. So if we substitute everything you find that the overall heat transfer coefficient is like this one by U not is equal to point zero zero zero five seven eight one plus one by h not h not is not fully known to us. Because we do not know what is the delta T.

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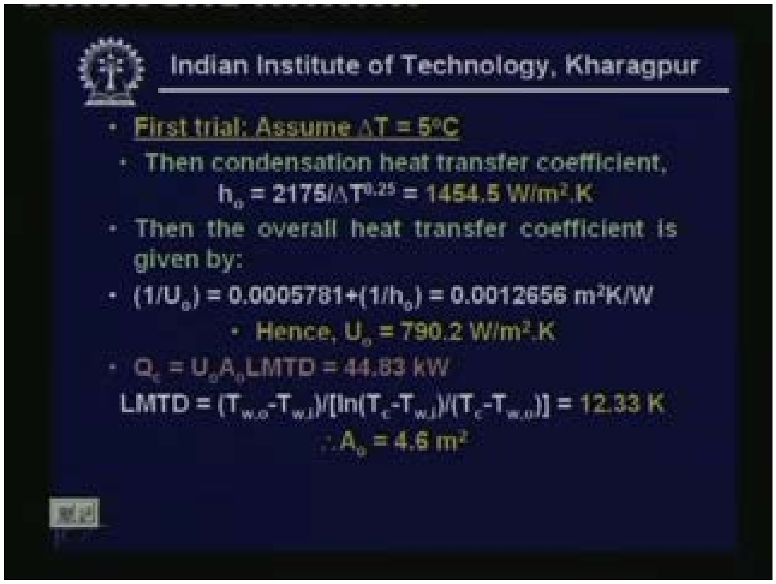

Okay, so as I said we have to go for a trial and error method. So first let us take a initial guess value five degree centigrade. So once you take a guess value of five degree of centigrade condensation heat transfer coefficient h not is equal to two one seven five divided by delta T to the power of point two five so delta T is five degrees. So from this we will find that heat transfer coefficient is fourteen fifty-four point five watt per meter squared Kelvin on the refrigerant side once you know this you can substitute this in the expression for overall heat transfer coefficient. And you find that one by U not is equal to point zero zero one two six five six meter squared Kelvin per watt. That means the overall heat transfer coefficient U not is equal to seven ninety point two watt per meter squared Kelvin.

So we have found the U not and we know that for the condenser we can write this equation Q c is equal to U not A not into LMTD which is equal to forty-four point eight three kilo watt Q c is forty-four point eight three kilo watt in order to find out A not w have to find LMTD because U not is known to us. So LMTD as you know is for a condensation process LMTD we have can be written like this T w o minus T w i divided by natural log of T c minus T w i divided by T c minus T w o where as you know T c with the temperature of the refrigerant T w i and T w o are the inlet and outlet temperatures of water okay. For all these things are known to us. So if we substitute these values you find that the LMTD is twelve point three three Kelvin okay. So if we substitute the values of LMTD and U not in the expression for Q c then you find that the area the required area that is it outer area is four point six meter squared.

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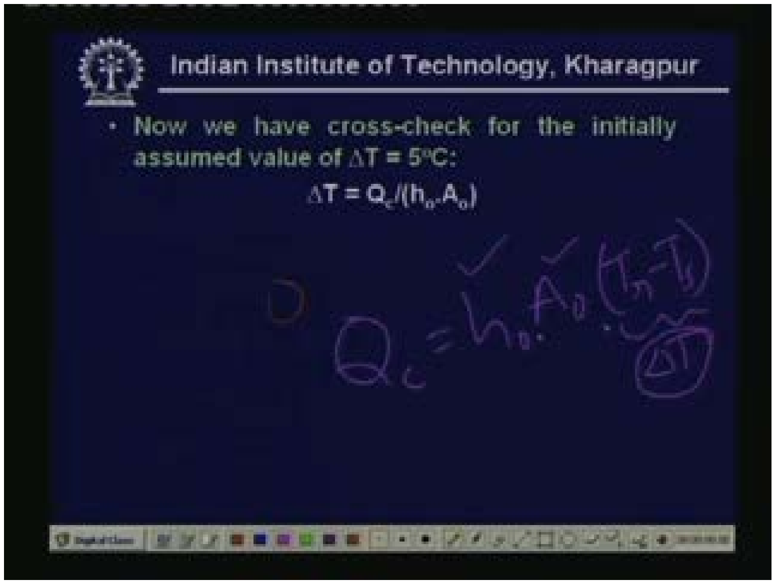

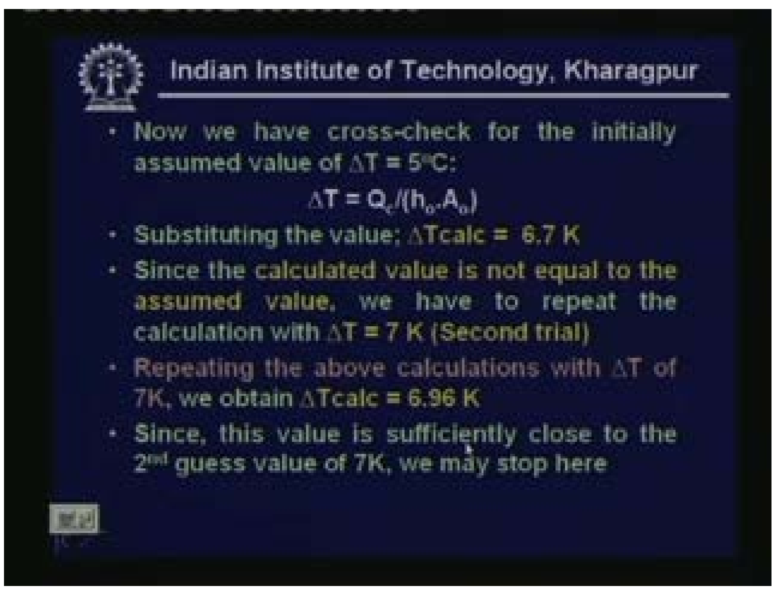

Now as they, I have already mentioned you should not stop the, this thing here. Because this value we got by taking an initial guess value of five degree Kelvin per for delta T okay. Now we have to see whether the delta t is really five degree Kelvin or not if it's not five Kelvin we have to go for the next trial okay. So let us calculate what is the delta T is now delta T can also be written. In this manner delta T is equal to Q c divided by h not into A not because you can write hm Q c in terms of Qc can be written in terms of outer transfer coefficient outer area into refrigerant temperature minus surface temperature okay, is, it nothing but delta T right. So h not is nothing but condensation heat transfer coefficient which is known to us A not just we have computed okay. So if we substitute everything you can find out what is the calculated value of delta T that's what we are doing now okay.

So if we substitute the values for Qc h not and A not we find that delta T calculate the six point seven Kelvin. So you find that the guess value is we have started the solution taking the initial guess value of five degree Kelvin. But when at the end when you calculate delta T you find the degree six point seven Kelvin since there is a difference of two degree two degree is between the guess value and the calculated value we have to go for a next trail okay. So in the next trail what we do is let us assume the delta T of seven Kelvin okay and repeat the problem repeat the calculations. So what we do is as I said since the calculated value is not equal to the assume value we have to repeat the calculation with delta T is equal to seven Kelvin. This is the second trial once you assume delta t as seven Kelvin again you have to find out h not because h not is expressed in terms of delta T.

Once you find the h not you find the overall heat transfer coefficient. Once you find the overall heat transfer coefficient you find out area and the, from the area again you have find out what is the delta T calculated and you compare delta T calculated with again the guess value okay. For this process has to be repeated till you get the converge values right. So from the, if you repeat the calculation with seven Kelvin guess value you find that the delta T calculated will be six point nine six Kelvin okay. This almost equal to seven Kelvin. So you can stop here but you want more accuracy of course you can again repeat the calculation by taking it is third trial value right. But since it is a difference is very less we need not go for third trial value okay. Since this value is as is said sufficiently close to the second guess value of seven k we may stop here okay.

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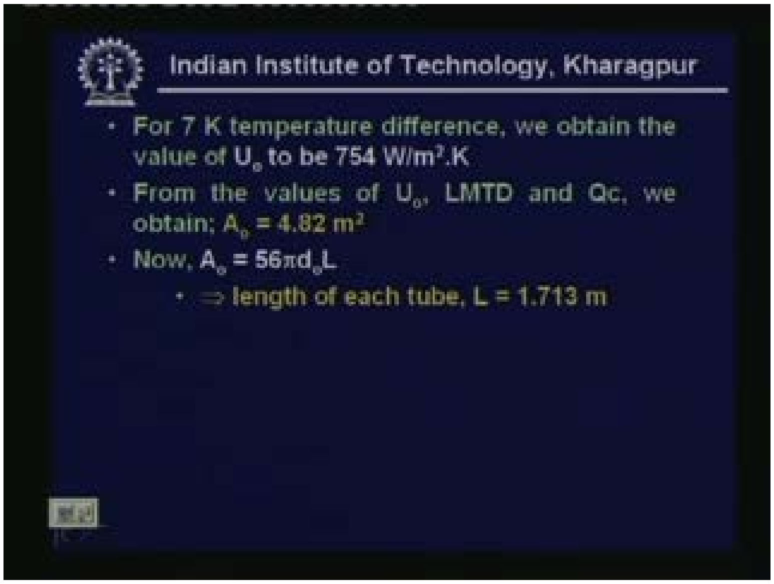

For seven K temperature difference we obtain the value of U not to be seven fifty-four watt per meter squared Kelvin. So once you know the value of U not and LMTD and Q c we find the value of A not is equal to four point eight two meter squared okay. Ultimately we have to find the length of the cube okay. So A not is nothing but there are fifty-six tubes okay. So total area is fifty-six into pi d O into L okay where D o is the outer diameter of the tube right there are fifty-six tubes. So we use fifty-six L is the length of the, this thing which is unknown to us but D not is the outer diameter which is known to us sixteen mm okay. So if we substitute those values you find that the required length is one point seven one three meter okay. This is how you have to do the design of a shell and tube condenser of course this is not a complete design this is only a thermal design of shell and tube condensers okay.

So you have to proceed in a systematic manner first by getting the required properties right and then using the correct formulae and then calculate the various quantities. And then you have to go as I said you have to go for trial and error method okay. And when you are using the trial and error method you have to take use intelligent guess values okay. You should not use unrealistic guess values then hm it may take a long time you have to do the hm calculus for many trails you have to take many trails okay. When you use the values judiciously then it will converts within two three steps right okay.

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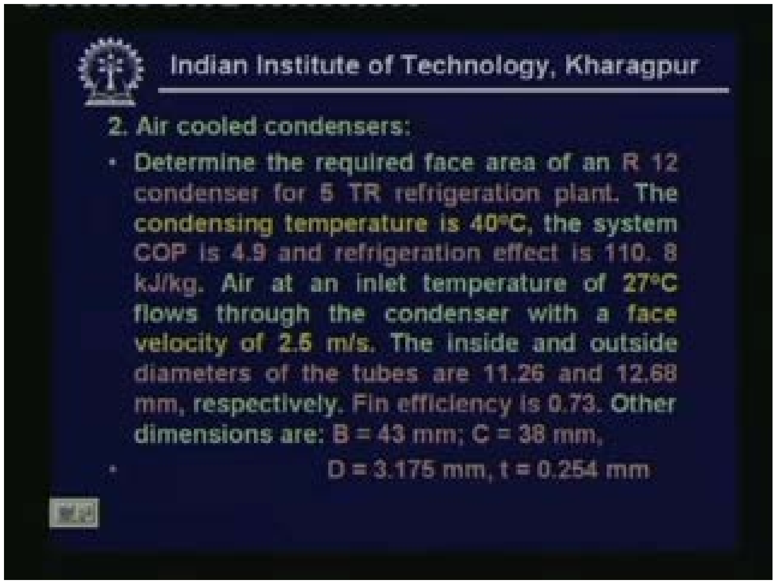

Now let me explain the design of an air cooled condenser again let me read the problem you have to determine the required face area of an R twelve condenser for five ton refrigeration plant. The condensing temperature is forty degree centigrade the system COP is four point nine and refrigerant effect is one ten point eight kilo joule per kg. Air at an inlet temperature of twenty-seven degrees centigrade flows through the condenser with a face velocity of two point five meter per second the inside and outside diameters of the tubes is eleven point two six and twelve point six eight millimeter respectively.

Fin efficiency is given as point seven three and other dimensions these are the facings and thickness of a fins okay the facing. Spacing between two cubes in a row would B is given by forty-three mm and C is thirty-eight mm D is three point one seven five mm and the thickness of the fin is point two five four mm okay. So what is given here capacity is given and condensing temperature is given COP is given and refrigeration effect is given okay. And then air inlet temperature given outlet temperature is not given and face velocity of air is given as two point five meter per second and inner and outer diameters of the tubes are given and other heat exchanger dimensions are given okay.

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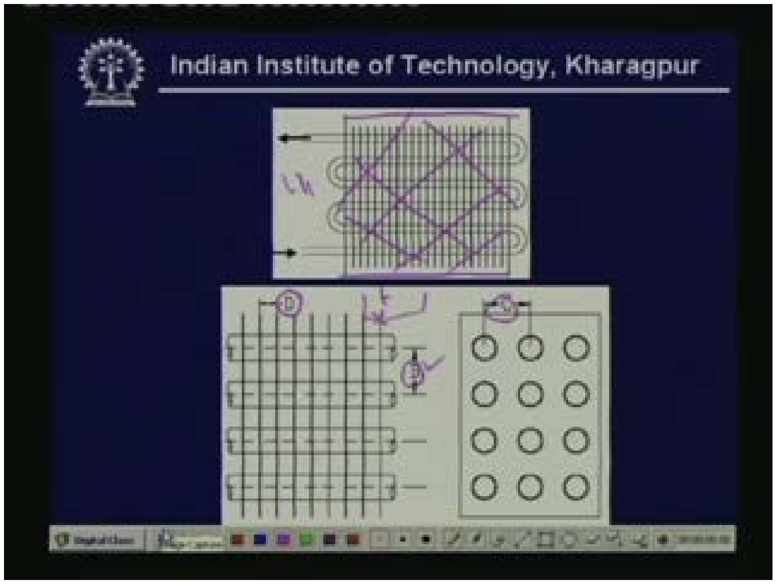

So as you know this we have discussed in the last class. I have shown you a typical, I start these is the typical fin and tube type of condenser and as I said this is the face area okay. This is the face area we have to find out what is the face area right and we have also defined certain parameters. For example as I had telling it just now B is nothing but the center to center distance between two consecutive tubes in a row right this value is given C is this is the side view okay. C is a center to center distance between two rows right also that is C and D is a center to center distance between two consecutive fins. And t as I said is the thickness of the fin right. So the all these parameters are given to us. So we have to ultimately find out the face area and we also know the face velocity okay. Face velocity is mentioned as two point seven meter per second.

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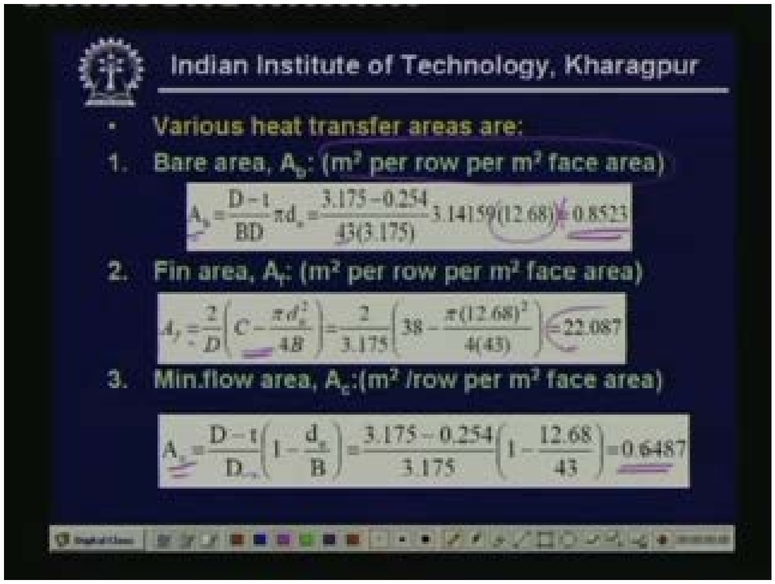

Now first thing we do here is let us find out various heat transfer areas all these areas have been defined in last lecture. So you can refer to the last lecture for these formulae okay. I am just giving the formula here. The first is the bare tube area A b. A b is given by D minus t by B D into pi d not and D is nothing but the fin to fin facing that is three point one seven five millimeter minus point two five four is the thickness of the fin and B is the spacing the center to center facing between two cubes in a row that is given as forty-three mm. And, so if you substitute these values D not is the outer diameter of the tube. S o if you substitute all these values you get these values for A b and remember that this is defined as meter squared per row per meter squared face area this is very important okay, the unit.

So everything all the areas we are expressing as meter square per row per meter squared face area okay. Then, let us calculate the fin area A f fin area A f this is the formula again everything is known to us D is known C is known D not is known B is known. So if we substitute everything you get the fin area to be twenty-two point zero eight seven meter square per row per meter square per face area okay. Now let us find out what is the minimum flow area minimum flow area is as I said is nothing but the area between the two cubes okay. The where the velocity becomes maximum and the flow area becomes minimum okay. So that is defined as this D minus t by D into one minus D not by B and again everything is known. So if we substitute that you find the minimum flow area is point six four eight seven meter square per row per meter square face area.

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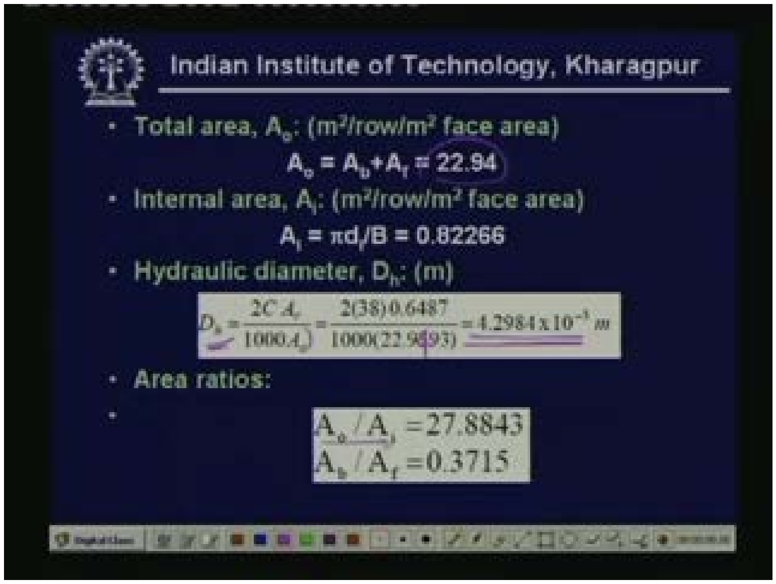

Then total area A not is nothing but A b plus A f that is bare tube area plus fin tube area. So that is found to be twenty-two point nine four meter squared per row per meter square face area okay. And the internal area internal area is A i and the formula A i is for A i is pi d i by B and this works out to be point eight two two six six meter squared per row per meter squared face area okay. Then we have to find out the hydraulic diameter because we want to find out the Nusselt number and Reynold's numbers hydraulic diameter is defined if I remember as the four into minimum flow area divided by the vertex perimeter okay. And the formula for that is given like this okay this is the formula for minimum flow area. I mean hydraulic diameter D h and C is known to us A c is the minimum flow area and A not is the A not is also known we these are the total area okay that is twenty-two point nine four.

So if we substitute everything here okay. This here should be twenty-two point nine four I had rounded of it okay. It is actually twenty-two point nine three nine three but I rounded it of to twenty-two point nine four. So you find that the hydraulic diameter is given as four point two nine eight four into ten to the power of minus three meters. And we also need these area ratios for calculating the overall heat transfer coefficient okay.

This area ratio A not by A i is the total area divided the internal area that works out to be twenty-seven point eight eight four three and A b by A f that is the bare tube area divided by fin area is point three seven one five. So now we have found all the required areas.

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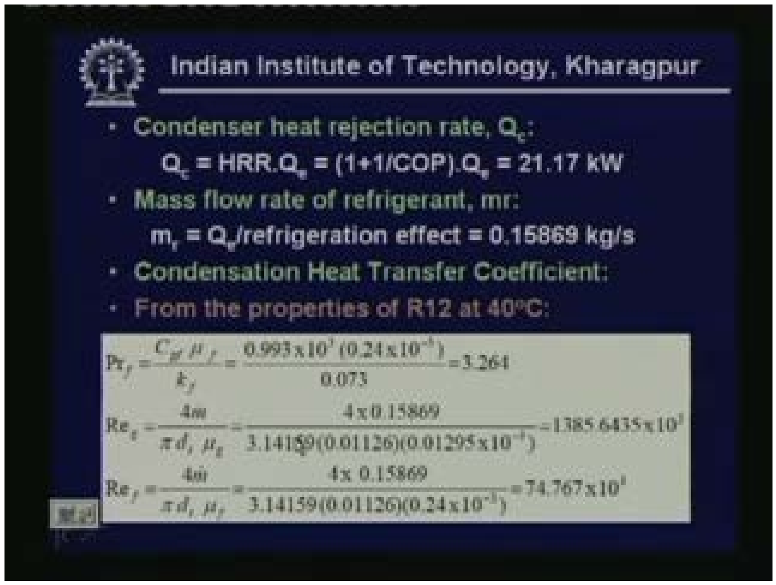

Now let us find out the condenser heat rejection rate see in the last problem for water cooled condenser the heat rejection ratio is given directly okay. But in these problems it is not given directly but the COP value given as specified. And we know that the heat rejection ratio is nothing but one plus one by COP okay. So we have to find out the heat rejection ratio first and from the heat rejection ratio we can find out the high transfer rate at the condenser okay. So that is what I have done here. The condenser heat rejection rate Qc is equal to HRR into Q e where Q e is the refrigerant capacity. That is five tons and HRR is one plus one by COP and COP is given as four point nine and Qe.

We know five ton convert that into kilo watts multiplying into three point five one six seven. If we multiply that and substitute the values you find that the required heat rejection rate, its condenser that is Qc is equal to twenty-one point one seven kilo watts. Now we have to find out what is the mass flow rate of refrigerant to find out the mass flow rate of refrigerant. We know the refrigerant capacity and also the refrigeration effect is specified okay. So we know that if we do an energy balance for the evaporator neglecting the hm the kinetic and potential energy changes. We know that the refrigerant capacity is nothing but mass flow rate of refrigerant into refrigeration effect okay. The refrigeration effect is specified. So substitute the value of refrigeration effect and refrigeration capacity you find that the mass flow rate of refrigerant is equal to point one five eight six nine kg per second okay.

So now first let us find out the condensation heat transfer coefficient for that we need the properties of R twelve. So the properties of R twelve I have not mentioned here but the properties have to evaluate at condensing temperature and condensing temperature is specified as forty degree centigrade. So you have to find out the saturated the properties of R twelve liquid and vapor at forty degree centigrade okay. I have use those values and using those values I had found these non dimensional numbers okay. First I have found the prandtl number of refrigerant that is C p f Cp mew by k and this is the prandtl number for the saturated liquid okay, f stands for liquid okay. So you have to use all the liquid values right. So if you use these values you find that prandtl number for the liquid refrigerant is three point two six four and the Reynolds number for the gas okay. g is for the vapour or the gas okay. That is again can be written in terms of the mass flow rates four into m dot divided by pi d i mew c and this Reynolds number is calculated when all the mass is in vapour form okay.

So you have to use the total mass flow rate here right so this Reynolds number is the Reynolds number when all the refrigerant is in vapor form. Similarly Reynolds number of the fluid is calculated assuming that all the refrigerant is in liquid form okay that is R e f. So if we substitute the values you find the, this is the Reynolds number of the refrigerant vapor. Similarly Reynolds number of the refrigerant liquid is this okay. So got the Reynolds numbers and prandtl number.

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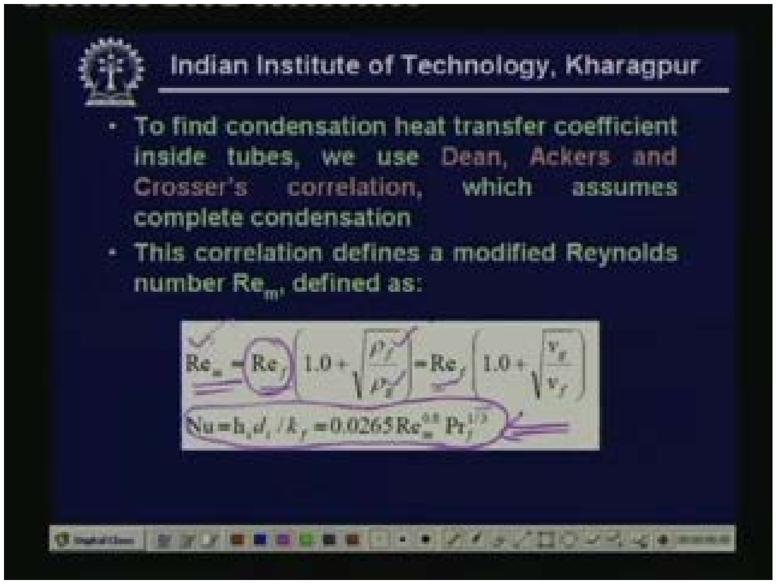

Then to find the condensation heat transfer coefficient inside tubes we use Dean Ackers and Crosser's correlation which assumes complete condensation. In fact this is a new correlation. In the last lecture I have shown two other correlation and other correlation okay. But here I am using different correlation okay. This is because I want to present as many correlation as possible right. So let us calculate the condensing heat transfer coefficient using this dean Ackers correlation and this correlation is valid under the assumption that the condensation is complete okay. So this correlation defines a modified Reynolds number R e m and this modified Reynolds number is defined like this okay. This correlation if you notice correlates Nusselt number in terms of Reynolds number and prandtl number.

So this is the basic correlation Dean Ackers and Crossers correlation. So we can see that this is look like Deters Bolts equation okay. You have a constant here R e to the power of point eight and P r to the power of one by three okay. Only difference is the Reynolds number that i am using here is a modified Reynolds number okay. It is neither Reynolds number of the vapor nor the Reynolds number of gas it is a modified Reynolds number and modified Reynolds number formula is like this. It is written in terms of the Reynolds number of the refrigerant liquid and the density of the saturated liquid and density of the saturated vapor row f and row g. So densities can be written in terms of specific volumes. And we know these specific volumes from the property data. That is at forty degree centigrade and Ref also we have found. So you can find out what is the modified Reynolds number and prandtl number is also known to us. So if we substitute all this values we can find out what is the Nusselt number.

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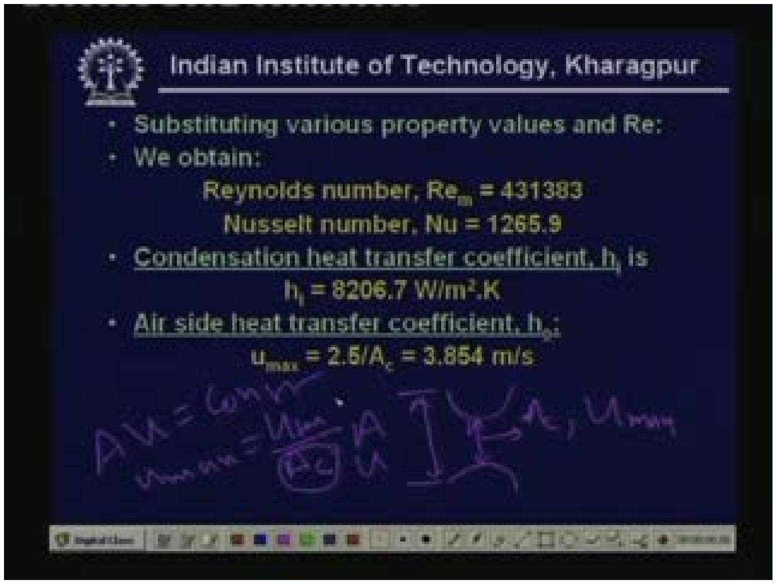

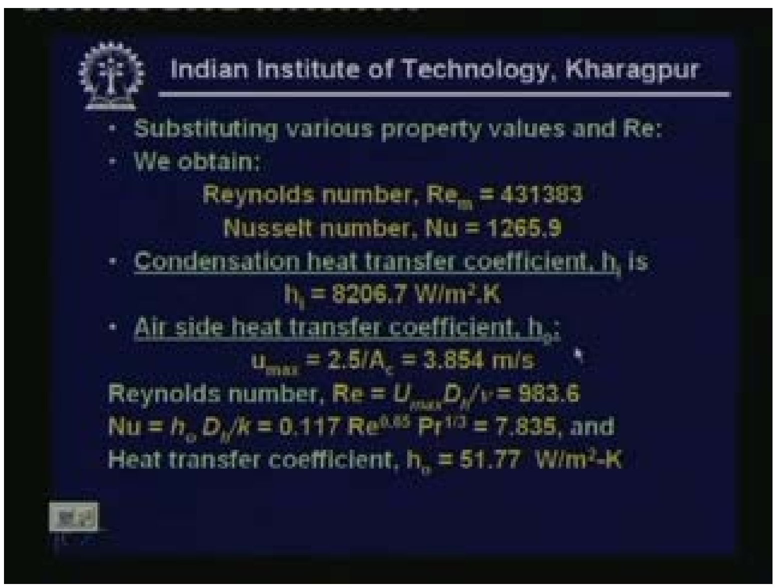

So substituting various property values and Reynolds number we find that Reynolds number modified Reynolds number is four thirty-one three eighty-three. And Nusselt number is twelve sixty-five point nine and condensation heat transfer coefficient h i is eight thousand two hundred and six point seven watt per meter squared Kelvin okay. So that the way we have found the condensation heat transfer coefficient. Now let us find the air side heat transfer coefficient because for calculating the overall heat transfer coefficient you require heat transfer coefficient on the refrigerant side heat transfer coefficient on the air side okay.

So let us find out the heat transfer coefficient on the air side remember that this is a fined plate fin type of a condensers we have to use suitable correlations for plate fined type of heat exchanger oaky. In fact if we remember last time I was mentioning that case in London have given several correlations for different types of condensers. So we will be using the general correlations suggested by case in London okay. In fact if we remember the case in London correlation is the Reynolds number is defined in terms of the maximum velocity and maximum velocity takes place where the flow area is minimum okay. So maximum velocity is nothing but the face area you can show that it is nothing but the face area divided by I am sorry where face velocity divided by minimum flow area this comes from your continuity equation okay. How did you get this?

Let us here two tubes here okay. And this is the minimum flow area A c okay. And at this point the velocity is U max right and let us say this is the, at the face at the face you have the face area okay. Or area between two tubes and you also know the velocity, face velocity okay. And from mass balance we know that A u is constant okay, U max is you will find that face velocity divided by A c. Because we are writing A c in terms of per meter squared face area okay. So that is why the face area termed as in come here. Because we are doing all the calculation per meter squared face area okay. So you find that using that we find that the maximum velocity is three point eight five four meter per second.

So the face velocity is two point five meter per second but by the time the air comes between the two tubes it its area of cross section gets reduced. So its velocity gets increases to three point eight five okay. Then we find the Reynolds number. Reynolds number is U max into hydraulic diameter divided by mew. So whatever we are calculated the hydraulic diameter and all will be used now okay. So we know the hydraulic diameter value computed. This is the kinematic viscosity and U max is the maximum velocity. That is three point eight five four meter per second. So we substitute that you find that Reynolds number is nine eighty-three point six. Then as I said we use the general correlations suggested by case in London which is given as Nusselt number is h not d h by k that is equal to point one one seven Reynolds number to the power of point six five prandtl number to the power of one by three.

So we Reynolds number we know prandtl number. So we can substitute this. So we find that the Nusselt number is seven point eight three five and heat transfer coefficient h not is equal to fifty one point seven seven watt per meter squared Kelvin.

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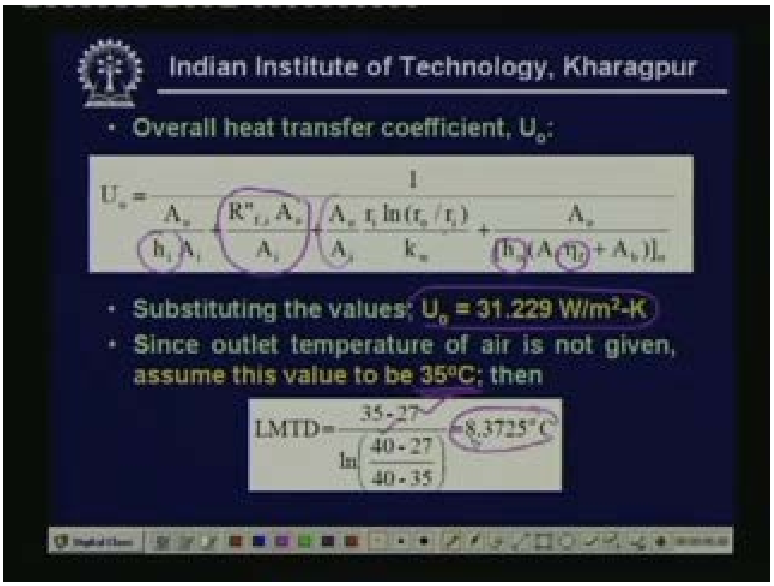

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Now overall heat transfer coefficients for plate fin type heat extinguisher formula is like this formula is mentioned in the last class also. So here you have the fouling resistance on the refrigerant side. We do not have any fouling resistance on the air side. That mean outside fouling resistance is not there okay. And you know various this thing A not by i and these are the area ratios. R i R i and R not are the inner and outer of the tubes and beta f is the fin efficiency which is known to us h not is the external heat transfer coefficient h i is the internal heat transfer coefficient okay.

If all these things are known to us we have already completed all this things. So substitute those values you find that overall heat transfer coefficient is thirty-one point two two nine watt per meter squared Kelvin okay. And now we have to find out the LMTD because we have to find out the area. So for that we have to find out the LMTD okay. Again we have to do some trial and error method here. Because we do not know what is the outlet temperature of air. Neither we know the outlet temperature of air nor we know the mass flow rate of air okay.

Since we do not know these two parameters. We cannot calculate the LMTD directory. So what we have to do is we have to assume either mass flow rate or the outlet temperature. So it is easier to okay. Is it, is all the same but let us assume the outlet temperature of air okay. The outlet temperature of the air you have to assume in such a way it should not be greater than the condensing temperature. Obviously because heat transfer as you takes place from refrigerant to air okay. So in this problem let us assume value of thirty-five degrees for the outlet temperature of air. So once you assume the outlet temperature of the air inlet temperature is given as twenty-seven degrees condensing temperature as given as forty. So we can calculate what is the LMTD the LMTD is eight point three seven two five degree centigrade okay.

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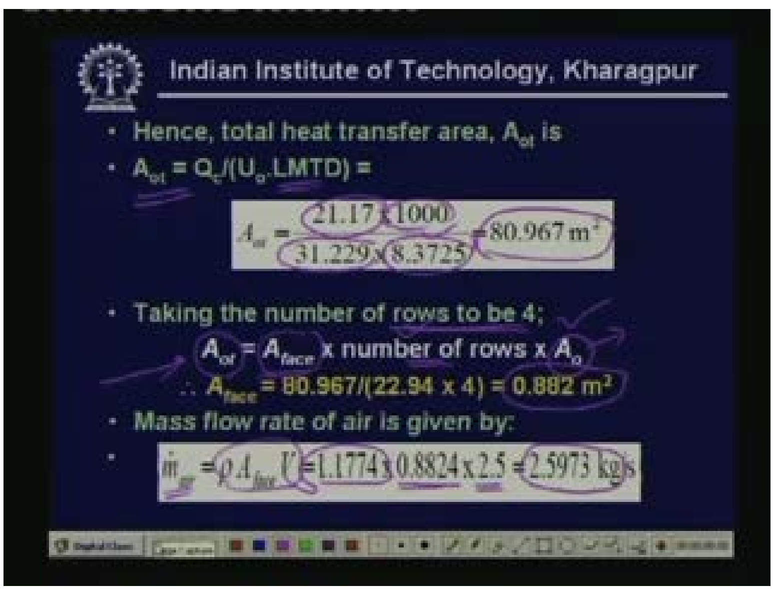

So once you know the LMTD we can find out what is the total external area okay. A total how do you know this because Qc is nothing but U not into total external area multiplied by LMTD so LMTD is computed eight point three seven two five and Q c is twenty-one point one seven and U not is thirty-one point two two nine okay. I am multiplying here. Because this twenty-one point one seven is kilo watts. So I am converting everything into watts because U not is in watt per meter squared Kelvin. So if you do that you find that the total external area is eighty point nine seven six meter square okay. So this is the total area and remember that this is a, this can have many rows right. So total area means, so total area of the condenser. That means area of all the rows right. And so for nothing has been mentioned about the rows right and in the last class I have mentioned that the number of rows can vary anywhere between two to eight. Since this is not a very large capacity system it is a five ton capacity system. Let us take the number of rows to be four okay.

So I am assuming the number of rows to be four once you assuming the number of rows to be four total area is nothing but A face area into number of rows into A not why is we are writing like this. Because A not is nothing but the total area per meter squared face area per number of rows okay. That is why you get the this kind of formula and A not is known to us number of rows are assume to be four and A face area is not known to us but A total is just now we are calculated eighty point nine seven six. So if we substitute all this value you find that face area is point eight eight two meter squared. But this is this may not be the correct face area you have to check. Because you have assume that the outlet temperature to be thirty-five degree centigrade okay whether that is right or wrong we have to check.

Now how do we check that we calculate the mass flow rate of, air mass flow rate of the air is nothing but, if you apply the continuity equation is nothing but row into A face into V where V is the face velocity that is two point five meter per second and A face is completed to be point eight eight two four meter squared. And row is the density of air which is one point one seven seven four kg per meter cube this is the mean density right. So if we substitute these values you find that this is the mass flow rate of air right.

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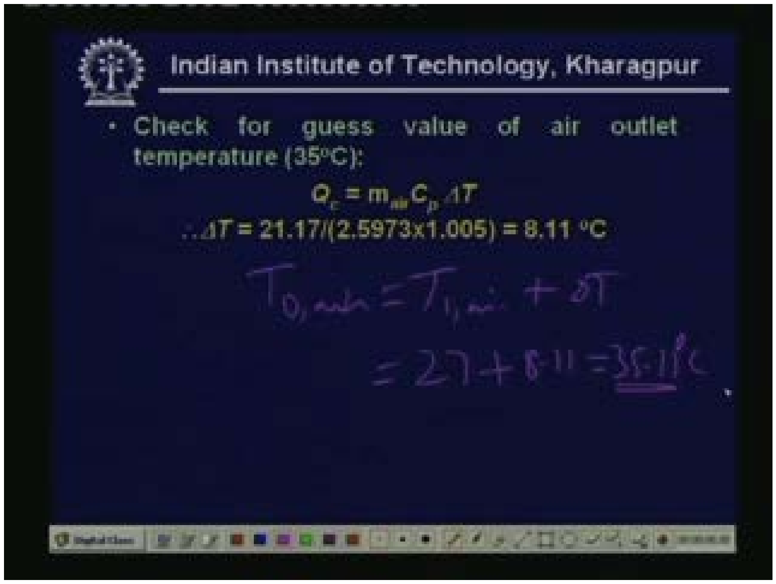

Once you know the mass flow rate of air you can easily calculate what is the outlet temperature. How can you calculate that, we know that for the condenser you can also right Q c as mass flow rate of air into C p into delta T so delta T is nothing but Q c divided by mass flow rate of air into C p C p I have taken here as one point zero zero five okay. So and the mass flow rate of air we have computed and this is Q c right. Here you need not multiply into thousand because this is in kilo joule per kg Kelvin. So you find that delta T we have obtained to be eight point one one degree centigrade right.

Now what is the delta T is eight point one one degree centigrade mean what is the outlet temperature. Outlet temperature is nothing but T outlet for A of air is T inlet of air plus delta T okay. T inlet is given as twenty-seven plus eight point one one is the thirty-five point one one degree centigrade. So you find that coincidentally the water you have calculated is very close to water you have guessed and this guess value and calculated values are coming very close okay. So if this accuracy is sufficient you can stop the trial and error procedure. At this point and you can take this face area is the required face area. But if we want very high recession then you go for a second trial by taking the outlet temperature to be let us say thirty-five point one okay. So you can continue this but sometimes, if you get accuracies of this kind of accuracy there is no point in really going for more and more number of trials. Because remember that there is always an element of uncertainty in the calculation of heat transfer coefficients okay. The they themselves may give uncertainties could be as highest about plus or minus twenty-five percent okay. So there is no point in really looking for very close accuracy's in areas right. So that is what I have mentioned here.

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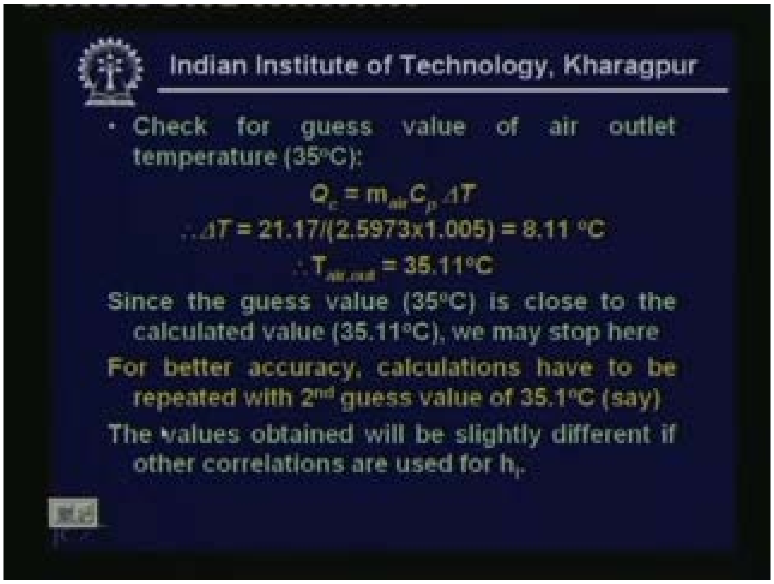

So T air out as calculate is thirty-five point one one degree centigrade. This is, the guess value is close to the calculated value we may stop here for better accuracy calculations have to be repeated with second guess value of thirty-five point one degree centigrade okay. And there is one thing you must keep in mind the values obtained will be slightly different if other correlations are used for h i as I said large number correlations are available for estimating the heat transfer coefficient. For example on the condenser side or on the air side had used some other heat transfer correlation you might of got slightly different results okay. That is what I mean by saying there is lot of uncertainty right.

If you want you can check by using other correlations and see what value you are getting okay. This is the procedure for estimating of a design thermal design of condensers okay. If you proceed in a systematic manner the problem is very simple okay. So at this point I stop my lecture on condensers and let us go to the next important component that is evaporators. So I will give you brief introduction and classification of evaporators in this lecture okay.

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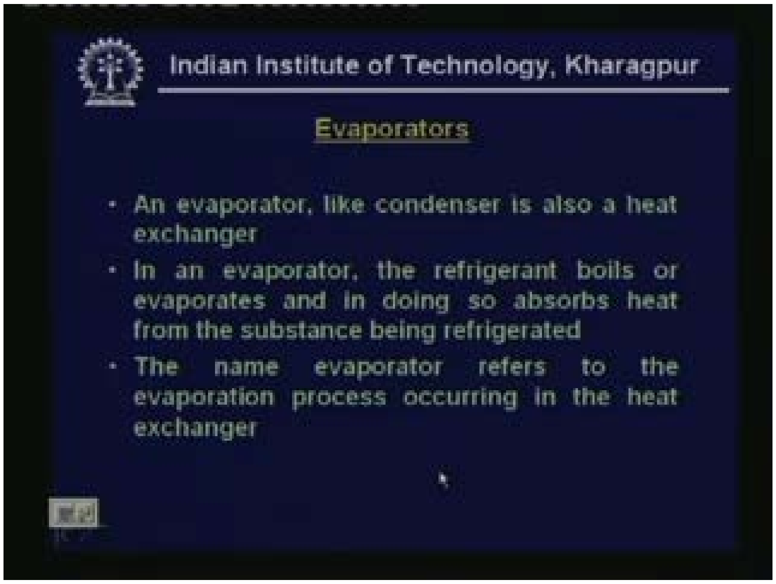

Okay, as you know evaporator like condenser is also a heat exchanger in an evaporator the refrigerant boils or evaporate and in doing. So absorbs heat from the substance being refrigerated the name evaporator refers to the evaporation process occurring in the heat exchanger.

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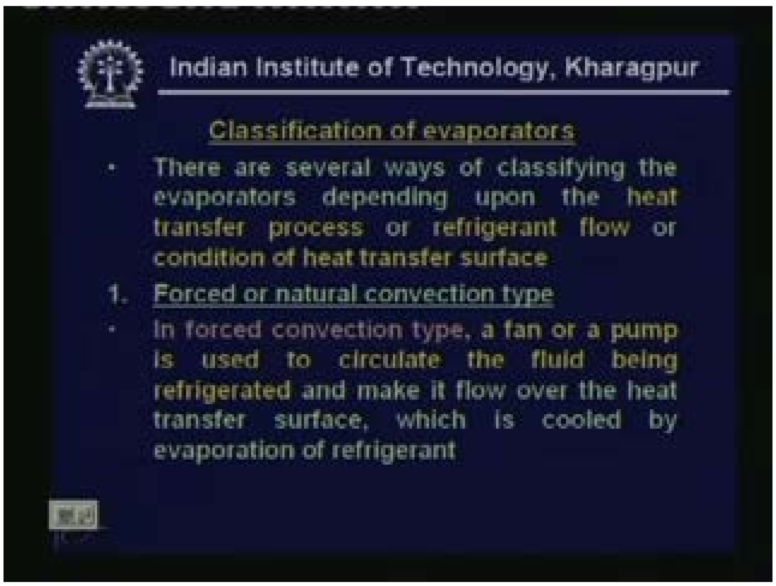

Now let us look at the classification of evaporators in fact you can classify the evaporators in many ways. There are several ways of classifying the evaporators depending upon the heat transfer process or depending upon the refrigerant flow type or depending upon the condition of heat transfer surface extras okay. For example you have, you can classify them as either forced convection type or natural convection type in the, as you know in, what is the forced convection type evaporator in forced convection type evaporator. Just like force convection type condenser a fan or pump is used to circulate the external fluid external fluid could be water or air or any other media okay. So you need a pump or fan for circulating this external fluid okay. And makes the, make it flow over the heat transfer surface which is cooled by evaporation of refrigerant this is the forced convection type evaporator.

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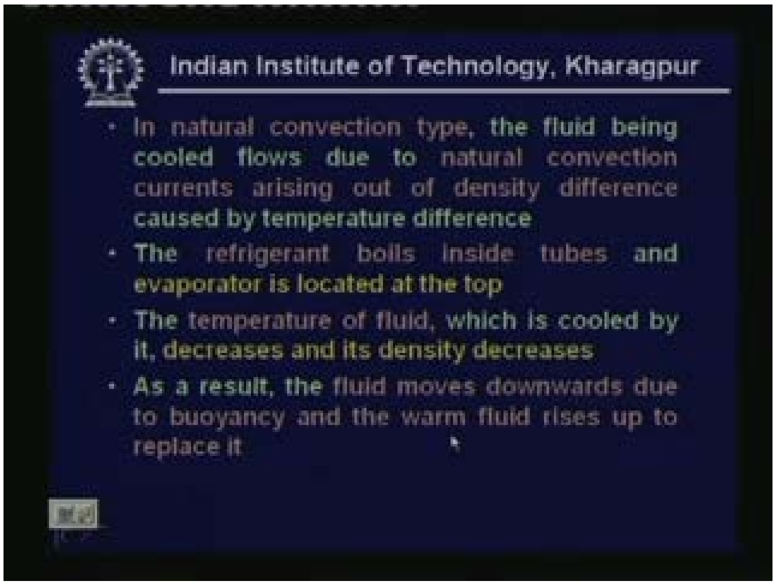

In natural convection type as you know in natural convection type we do not use either a fan or pump. And the circulation takes place because of buoyancy effects. In the buoyancy effect are induced due to density differences which are caused by temperature difference okay. This is nothing but the natural convection type evaporator just like natural convection type condenser okay. Then natural convection type evaporator refrigerant always boiled inside tube and evaporator is located at the top.

You have to locate the evaporator at the top because you are relaying on the natural convection. Natural convection means, what happens is, when you are keeping at the top warm air comes in contact with the evaporator it becomes cold once. It becomes cold, its density increases once. Its density increases because of the buoyancy effect it tries to settle down. When it settles down the warm air from the bottom raises up and warm air goes to the evaporator. It gets cooled and again its come down. So this cycle is into repeated okay. So to continue maintain this cycle you have to keep the evaporator at a height. The temperature of the fluid which is cooled by decreases am just explain what is the mechanism temperature of fluid which is cooled by it decreaser and decreases. And its density decreases as a result the fluid moves downwards due to buoyancy and the warm fluid rises up to replace it.

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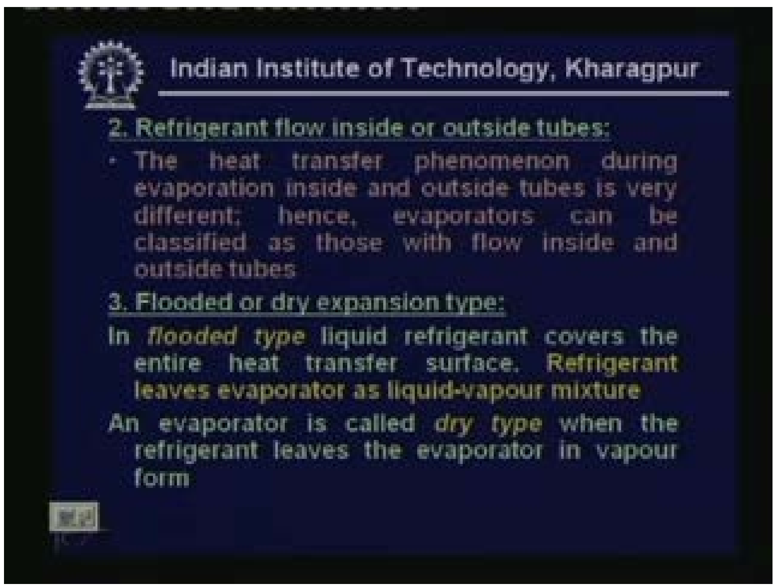

Okay, you can also classify the evaporators based on the refrigerant flow whether it is taking place inside the tubes or outside tubes. The heat transfer, the, this is very important because the heat transfer phenomenon okay. Just like condensation is entirely different if it is evaporation boiling is taking place inside the tubes or if it is taking outside the tubes okay. The phenomenon is different the correlations will be different and the values of high transfer coefficient also will be different. So it is very important to keep this in mind and use the suitable correlations okay. So this is another way of classifying the third way of classifying is by classifying them either as flooded type of evaporators or dry expansion type of evaporators.

In flooded type evaporators liquid refrigerant covers the entire heat transfer surface okay. This is known as a flooded type evaporator and the refrigerant leaves evaporator as liquid vapor mixture okay. And what is the dry expansion type in a dry expansion type the refrigerant leaves the evaporator in vapor form and not the entire heat transfer ratio of surface area is covered with liquid okay. There are some area is covered with the vapor okay. So this is known as dry type.

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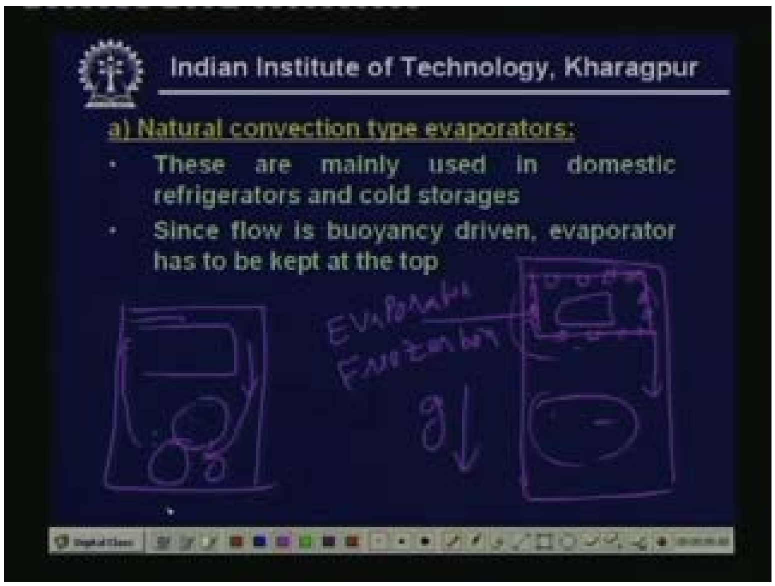

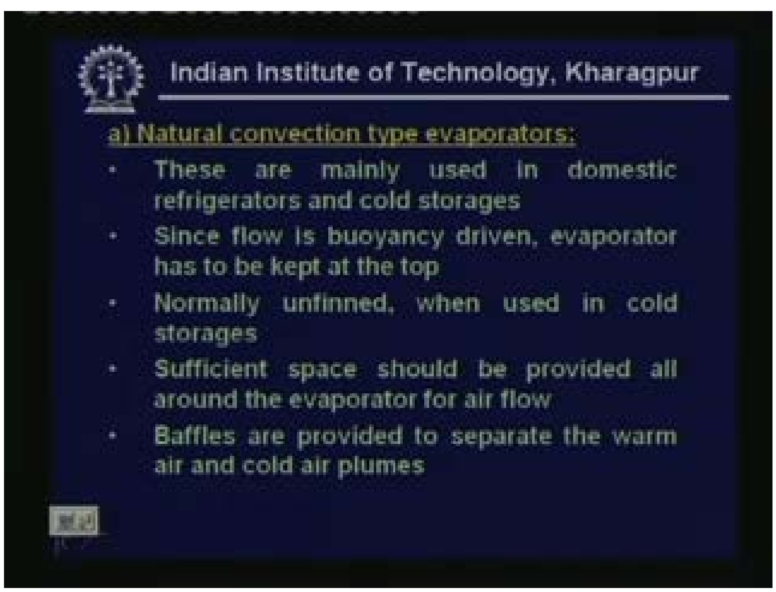

Now let me explain the salient features of some of the important types of evaporators. Let me begin with natural convection type of evaporator okay. Natural convection type of evaporators are mainly used in domestic refrigerators and cold storages. Since flow is buoyancy driven evaporator has to be kept at the top. You might have for a example let me everybody must of seen it in old type refrigerators not fastly type in convection type refrigerators the evaporator is kept at the top okay.

In fact this is nothing, but evaporator comes freezer box okay evaporator and it also acts as a freezer compartment or freezer box okay. So evaporator tubes are kept here it can be a roll bond type or cube and plate type and you store the food products everywhere fruits and food is kept here and other food stuffs are kept here vegetables fruits extra okay. So the gravity since in this air is the top. So since you are keeping the evaporator at the top the air close to the evaporator becomes cold. And since it its density increases the air will come down okay. If we see from of the side the evaporator will be something like this okay. So cooled air comes down right. And as it is comes down it comes in contact with the food products kept at the bottom. And it takes the heat from the food products and it is becomes warm.

Once it becomes warm again it rises up okay. Once it rises up again it comes in contact with the evaporator surface it becomes cooled and again it is comes down. So this is a natural circulation maintain by keeping the evaporator at the top okay. Normally these are unfinned when used in cold storages. The, as I said these are mainly used in domestic refrigerators in cold storages. In domestic refrigerators Finns are added. But in cold storages Finns are not used okay. A sufficient space should be provided all around the evaporators for air flow. This is very important just like your condensers the flat back condensers are wire and tubed type of condensers. You are realing on buoyancy effect for the air flow okay.

So the delta T s are not normally very high. So the potential for the air flow is generally small. So if there is large resistances then the air flow gets affected adversely okay. So if you want have good air flow you have to provide sufficient space all around the evaporator. So that air can flow with minimum resistance okay. That is why you might of seen in the domestic refrigerator. They do not put the evaporator right at the top. That means there be some space all around the evaporator. So that air can flow all around the evaporator okay and baffles are provided to separate the warm air and cold air plumes.

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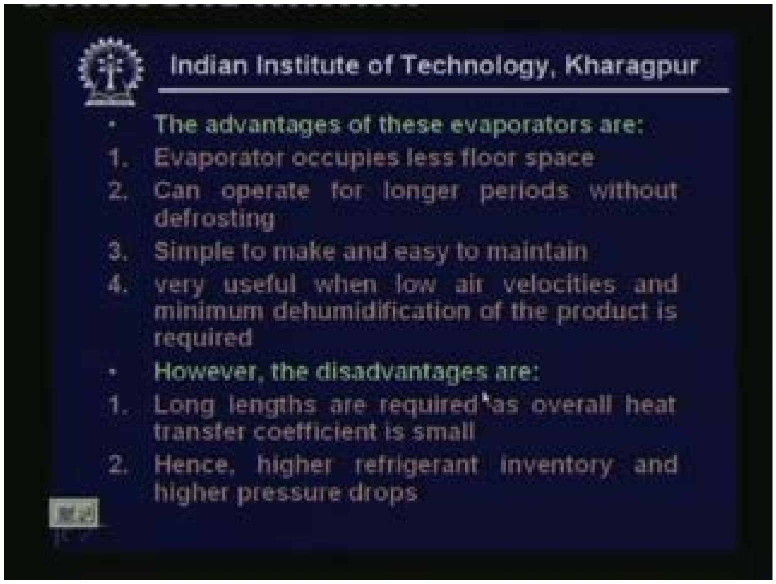

Now what are the advantages of natural circulation evaporators? Evaporator occupies less floor space. This is very important especially in cold storage condenser. Because floor space if the system occupies lot of floor space. Then the valuable floor space is loss because you could have stored some products in that floor space. And by storing more products you could have earn more money okay so floor space is very valuable. So when you are using natural convection type of evaporators in cold storages. They are kept at the right near the ceiling okay. So it does not occupy any floor space right this is the one of the advantage of natural convection type evaporators.

Second advantage is they can operate for longer periods without defrosting and they are simple to make and easy to maintain. Because especially in cold storages and all they are plain tubes okay normally they are unfinned. So they are they will be simply welded at the side. So they are no fins, nothing just you have to take the pipes and just weld the pipes at the side okay. And the maintenance is also easy and very useful when low air velocities and minimum dehumidification of the product is required. Since here relying on an natural convection and at the delta T available for natural convection obviously. We will not find any air blast or anything okay. So the air velocity will be very small once the air velocity is small there is no danger of products getting dried up to much okay. This is the one typical problem with force convection type of evaporators because the air velocity is high. High air velocity means high heat and mass transfer rate. So drying of products takes place where as in natural convection type this problem is not there okay. However there are certain disadvantages. Disadvantages is obviously is, that you required very long lengths. Because the overall heat transfer coefficient is typically small. This is small because you are realing on natural convection.

Hence higher refrigerant inventory and higher pressure drops. Once the required tube length becomes large. You have to put large amount of refrigerant inside the system okay. Once you put the large amount of refrigerant inside system there are other problems. The cost will be more okay. And if the refrigerant is toxic and flammable it will half safety problem okay. In addition to that there are other problems like the pressure equalization takes a long time and defrosting also takes long time okay. These are some of the disadvantages of having long lengths okay. In addition to this if you have long refrigerant tubing pressure drop also will be large. So if you want to minimize the pressure drop you may have to go for parellel circuits okay.

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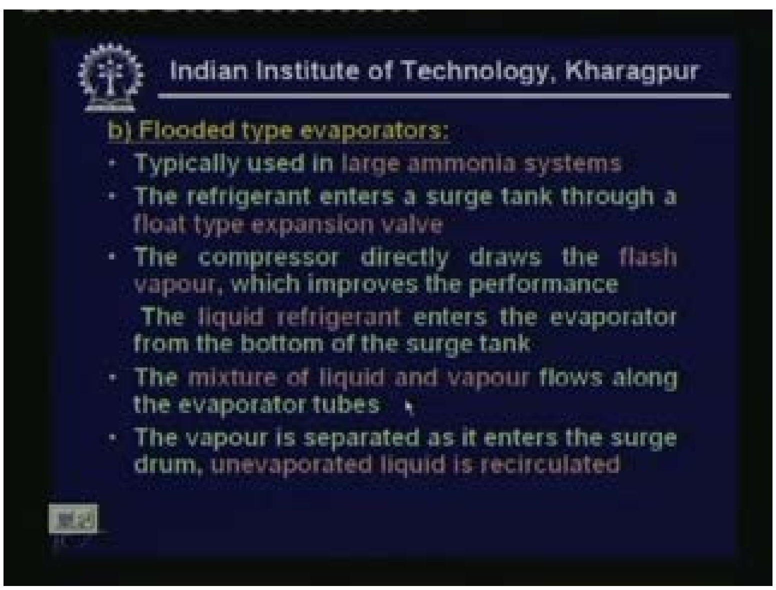

Now let me quickly explain the second type that is the flooded type evaporators of they are typically used in large ammonia systems. The refrigerant enters a surge tank through a float type expansion valve the compressor directly draws the flash vapor which improves the performance. As you know that once you flash vapor is not allowed to go to the evaporator performance improves we have seen this in multi state systems. And the liquid refrigerant enters the evaporator from the bottom of the surge tank the mixture of liquid and vapor flows along the evaporator tubes the vapor is separated as it enters the surge drum or surge tank okay, evaporated liquid is re circulated.

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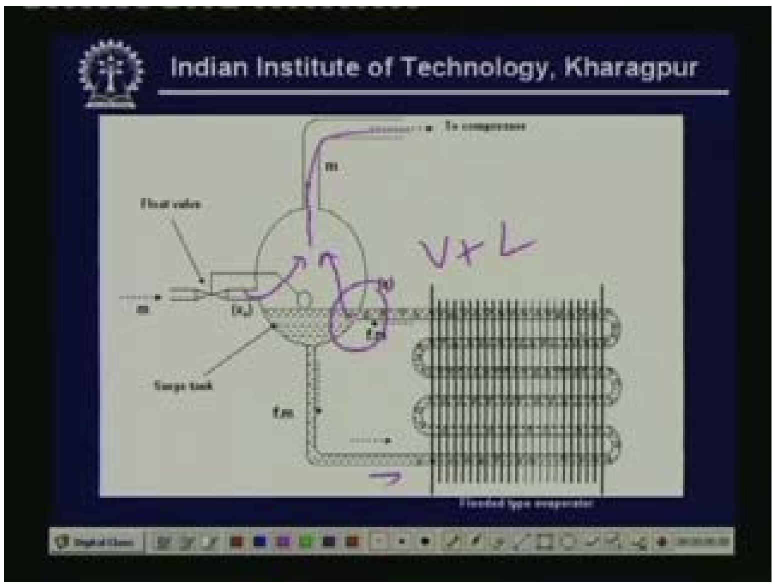

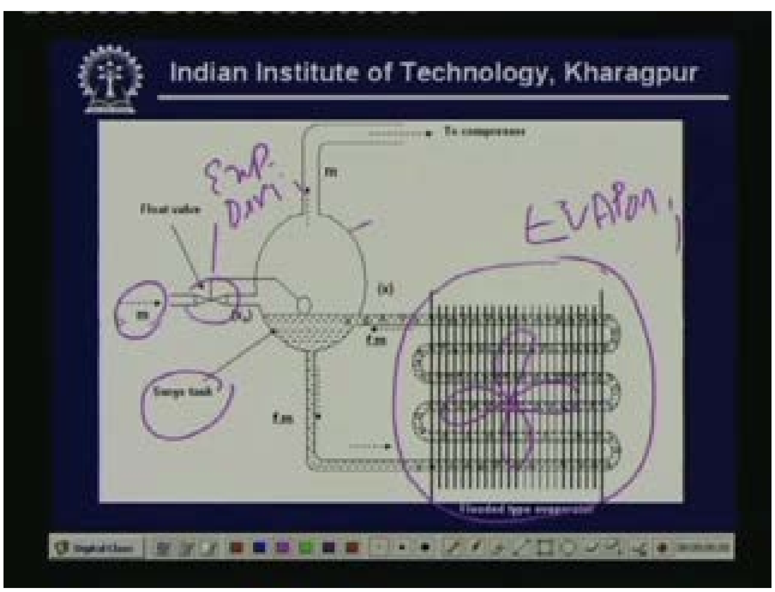

So let me quickly explain this. So this is the flooded type of evaporator okay this is the evaporator portion right let us for the timing. Let us assume that this is use for, let us say cooling air. Let us you have a fan and finns and all air is blowing over this. So we have a, the component surge tank here and refrigerant from the condenser enters the surge tank through the float valve. This float valve acts as an expansion device here okay. This expansion device used here and what is the purpose of this float type of wall it always maintains a required level of refrigerant in the surge tank okay.

When the level falls the wall will open more refrigerant will come here. So normally this is connect to a condenser or receiver condenser okay. And you can see that when the refrigerant enters at this point it is vapor plus liquid. Because due to flashing across that expansion device some vapor would have been generated. So you have both vapor and liquid but what you are doing is by separating vapor and liquid in the surge tank you are allowing only liquid to go to the evaporator and vapor. Instead of going to the evaporator it directly goes to the compressor okay. That is why you can improve the efficiency of the evaporator. So you can see the, from the picture that only liquid refrigerant goes to the evaporator. And as they it flows through the evaporator it takes heat from the surroundings and vapor is generated okay. So you can see the vapor bubbles.

So you find that at the outlet of the evaporator again you have vapor liquid mixture. This is the feature of flooded type of evaporator not all that refrigerant that goes to the evaporator will evaporate okay. There will be a large amount which is evaporated. The evaporated amount will simply re circulate. So whatever vapor is there again that vapor plus water vapor is generated here both will be compressed by the compressor okay. This is the flooded type evaporator. You can see that there are liquid in the evaporator. And the evaporator surface is there always wet that will give you higher heat transfer coefficient in the refrigeration side. So this is the advantage of flooded type of evaporator okay.

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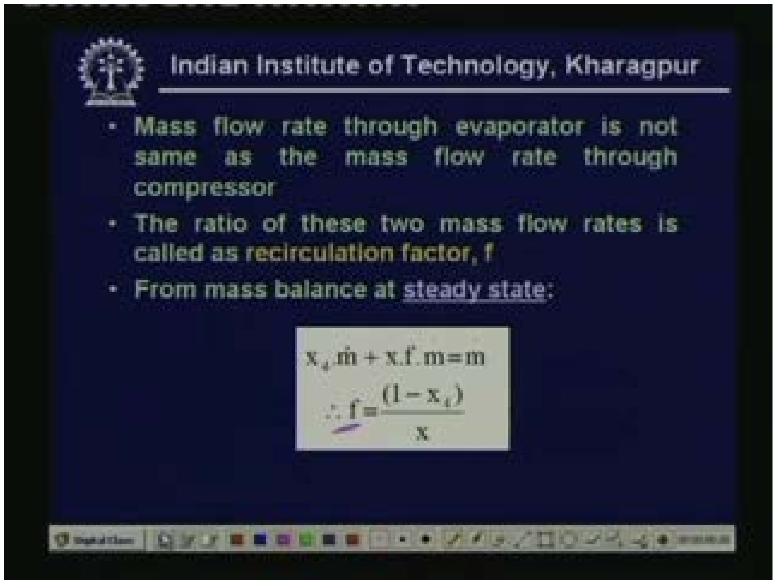

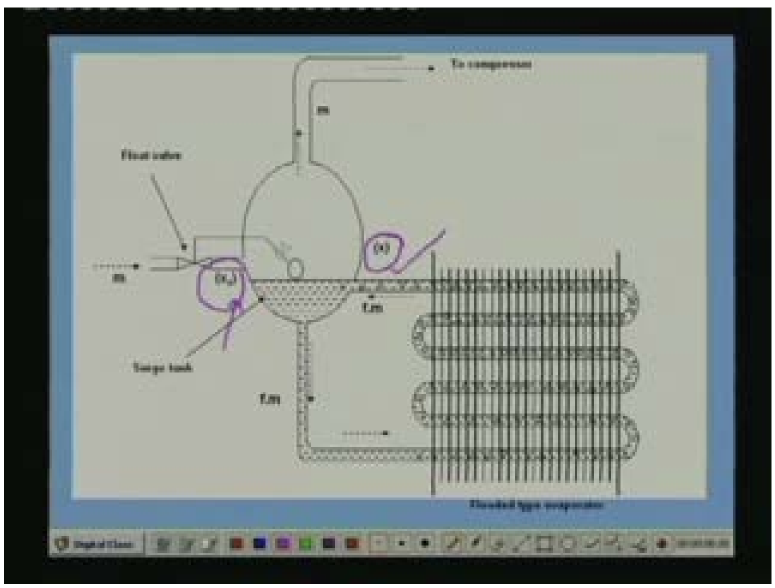

And mass flow rate through evaporator is not same as the mass flow rate. Through compressor and the ratio of these two mass flow rates is called as recirculation factor f and if we apply the mass balance for steady state. Whatever mass is leaving the surge tank must enter the surge tank okay. Then the, that mean what you have to do is you have to apply a mass balance across the surge tank okay. From that you can easily show that the recirculation factor f is one minus x four divided by x what are x four and x. x four is the quality of refrigerant at the inlet to the surge tank and x is the quality of the refrigerant at the outlet of the evaporator okay. So this is mass balance of the surge tank okay. So once you know the qualities you can calculate what is the circulation factors circulation factor will be greater than one. That mean more refrigerants circulates through the evaporator than it is evaporatored okay.

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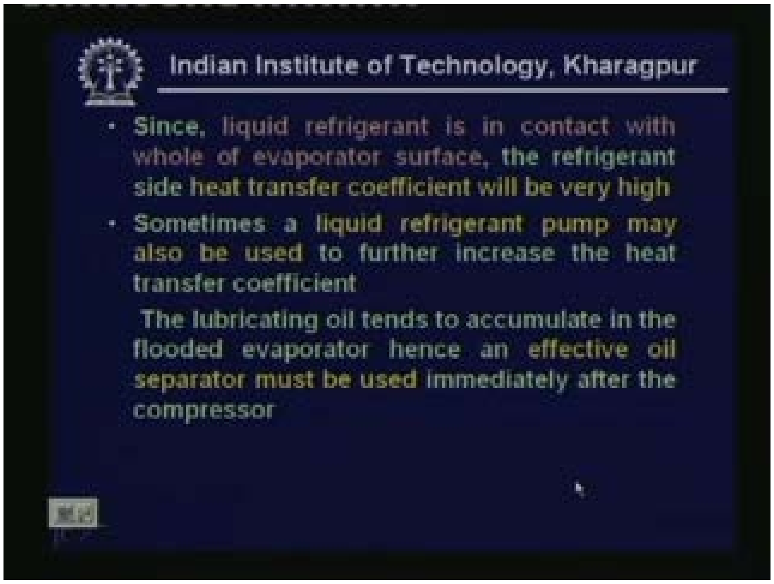

So since liquid what are the advantages since liquid refrigerant is in contact with whole of evaporator surface. The refrigerant side heat transfer coefficient will be very high. Sometimes a liquid refrigerant pump may also be used to further increase the heat transfer coefficient. The lubricating oil tends to accumulate in the flooded evaporator hence an effective oil separator must be used immediately after the compressor okay. Right, I will stop the lecture here. And I will continue this lecture in the next class okay. In the next class I will give you formula for calculating the high transfer coefficient in the area of evaporators okay.

Thank you.