Refrigeration & Air Conditioning
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Lecture No. # 27
Refrigeration Systems Component: Condensers

Welcome back the specific objectives of this lecture are to present expressions for overall heat transfer coefficient, present expressions for various heat transfer areas in plate fin type air cooled condensers, present typical correlations for heat transfer coefficients on air side water side and condensation heat transfer coefficient and discuss effects of non-condensable gases.

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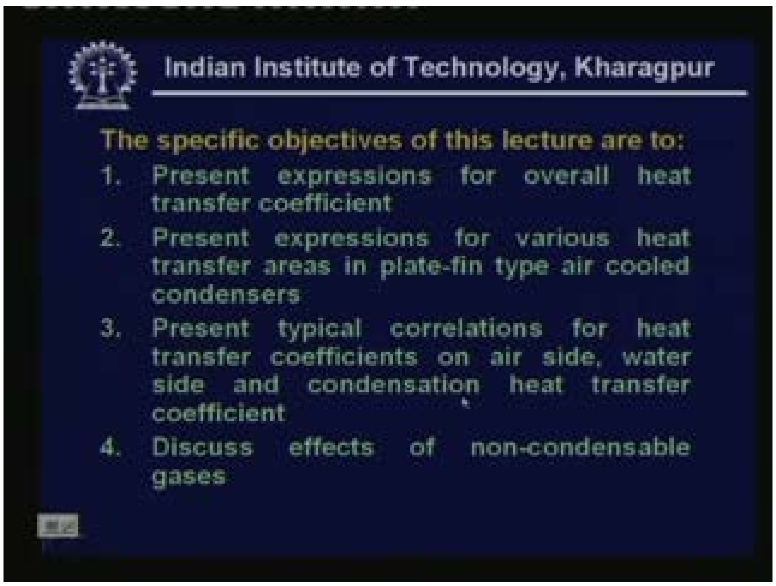

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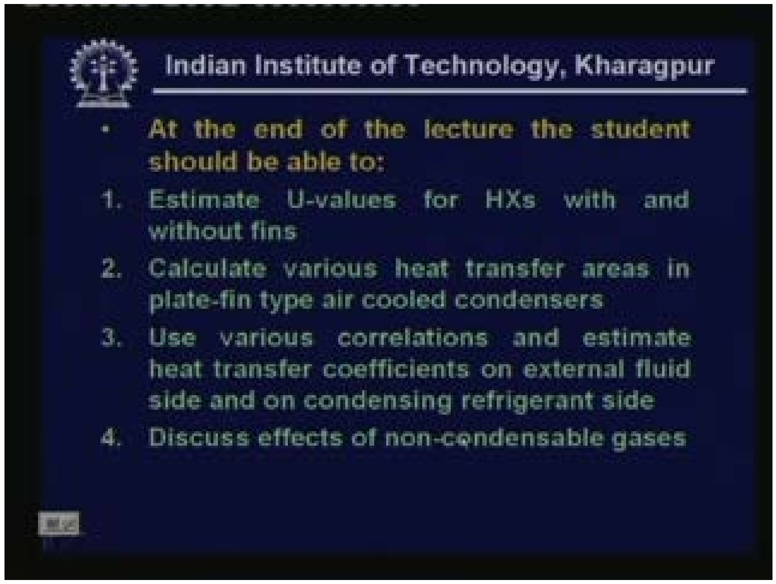

At the end of the lecture you should be able to estimate overall heat transfer coefficient values for heat exchangers with and without fins, calculate various heat transfer areas in plate-fin type air cooled condensers use various correlations and estimate heat transfer coefficients on external fluid side and on condensing refrigerant side and discuss effects of non-condensable gases.

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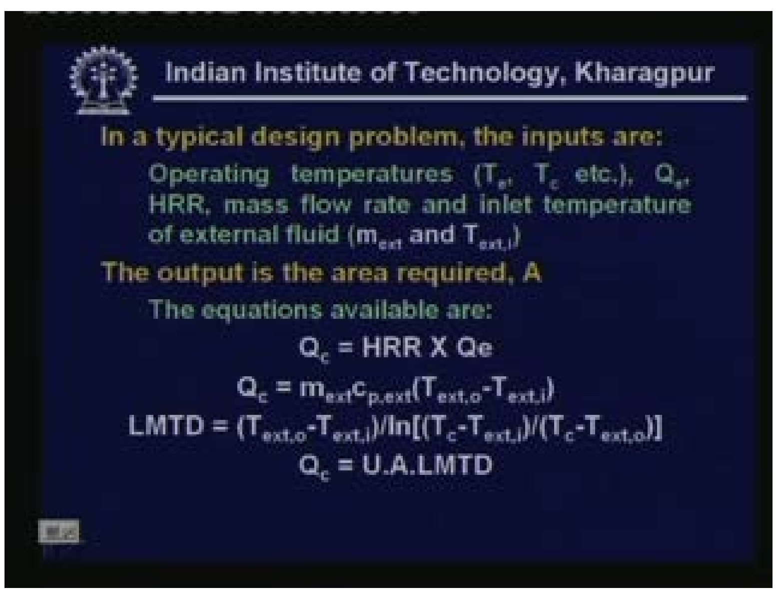

So let me give a let a let us look at a typical design problem normally in any design problem the inputs are the operating temperatures. That means the evaporated temperature condenser temperature etcetera refrigeration capacity Qe the heat recession ratio HRR mass flow rate and inlet temperature of external fluid that is m external and T subscript external I okay. Normally these values are available in any typical design problem and what is he objective of the design problem objective is to find out the area required. That is we have to find out A and the equations available are like this first equation is we have seen in the last class that the heat recession rate in the condenser Qc is given by Qc is equal to heat recession ratio HRR into refrigeration capacity Qe.

So we know both HRR as well as Qe. Because they are given as input so from this equation we can calculate what is the total heat transfer rate at the condenser okay. So this is the first equation one should use we can also write the heat recession rate at the condenser in terms of the external fluid. That means we can write Qc is equal to mcp of external fluid into T external out minus T external in right in this equation we you can see that we know that m external. That means the mass flow rate of the external fluid is known to us and the inlet temperature of the external fluid is known to us. So from this we can calculate what is the outlet temperature of the external fluid since Qc is known to us the third equation that will be using is the expression for log mean temperature difference.

This is defined in the last lecture this is defined in terms of the fluid inlet and outlet temperatures and a, and the condensing temperature since we know all this temperatures we can calculate what is the log mean temperature difference finally what we do is we use this equation that is Qc is equal to U into A into LMTD in this equation Qc is known to us LMTD is known to us and we have to find out A. So for what we have to do is we have somehow estimate the overall heat transfer coefficient U. So the using this one equation we can calculate what is the area required right so this is the general procedure to be followed in the design of any heat exchangers not necessarily the condenser okay. From the given input we have to find out the law mean temperature difference and then we have to find out what is the overall heat transfer coefficient okay.

Once we know the overall heat transfer coefficient then we can find out the area required this kind of problems are known as design problems okay. So the objective is to find or design the heat exchanger or to find out the area required there are other types of problems known as rating problems in which the area is given okay. And we have to find out what is the heat transfer rate okay. So in this particular lecture I am confining myself to the design problem typical design problem how to estimate the area okay.

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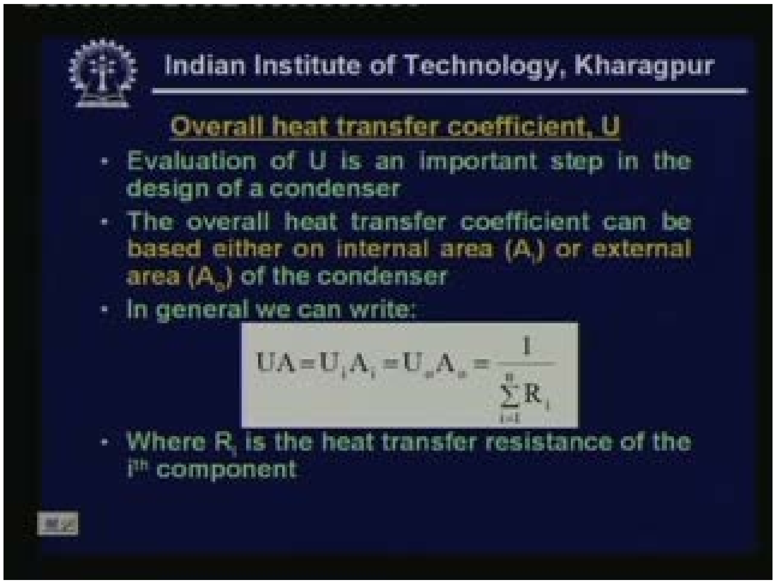

So first as I already mentioned in order to estimate the area we have to first find out what is the overall heat transfer coefficient U. So evaluation of U is an important step in the design of a condenser the overall heat transfer coefficient can be based either on internal area Ai or external area Ao of the condenser okay. And in general we can write it the product UA that is equal to UiAi which is equal to UoAo which is nothing but one by summation of Ri okay. From i is one to n where Ri is nothing but a heat transfer resistance of the ith component okay.

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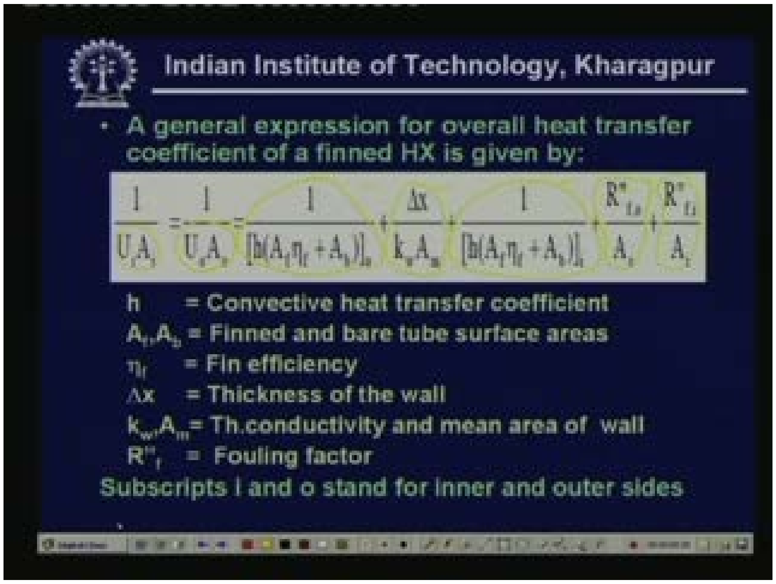

Now a general expression for overall heat transfer coefficient of the finned heat X is given by this expression. As I have already explained to you this is the overall heat transfer coefficient that means UA okay. This depends upon your different resistances which are in series so for any general heat exchanger there are five resistances here okay. For example the first resistance this resistance accounts for the convective resistance of the outer surface. That means that is why the subscript o is there okay of a finned heat X in there. So this is the convective heat transfer resistance of the outer finned surface. That is resistance one then you have second resistance this is nothing but the conductive resistance offered by the wall of the heat exchanger or wall of the condenser then the third resistance this one is nothing but the convective resistance offered by the inner finned surface okay.

Then we also can have the resistance due to fouling on the outer surface and resistance due to fouling are scale deposition on the inner surface. So we have five resistances okay, one two three four five these five resistances are in series. So in order to calculate the overall heat transfer coefficient we have to add up the all these resistances and we have to inverse it okay. So in the above equation the nomenclature is given here h stands for the convective heat transfer coefficient A subscript f and A subscript b stand for finned and bare tube surface areas eta subscript f stands for the fin efficiency delta x is the thickness of the wall K subscript w and A subscript m stand for thermal conductivity. And mean area of the wall and R subscript f is the fouling factor and subscript i and o stand for inner and outer sides. Now let us look at some special cases where some simplification is possible.

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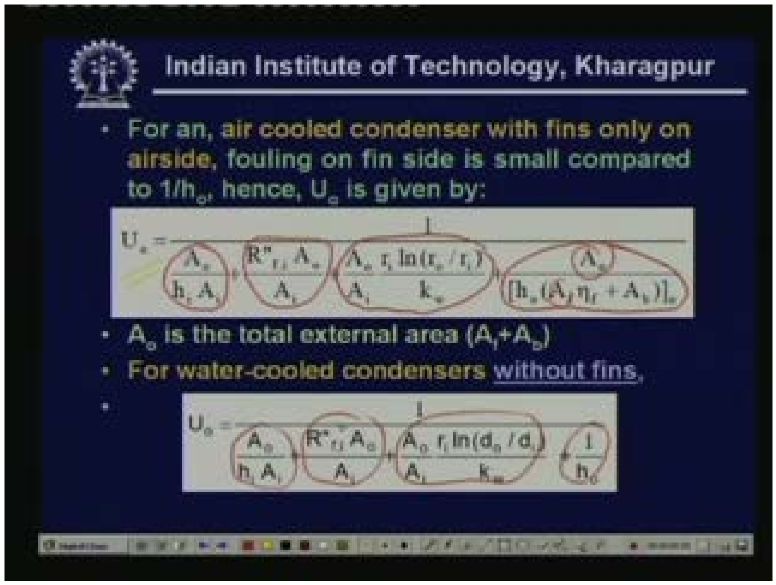

For example for an air cooled condenser with fins only on airside. That means we do not have any fins on the internal side okay. And for this kind of a condenser you find that the fouling on fin side is small compared to the convective heat transfer resistance on the fin side. So we can neglect the resistance due to fouling on the fin side if you neglect resistance due to fouling on the fin side you find that the expression for overall heat transfer coefficient is given by this equation okay. So this is obtained by simplifying the equation shown in the previous slide okay. So here we have the in a this term accounts for the internal convective resistance this term accounts for the resistance due to fouling on the internal side and this accounts for the conductive resistance of a cylindrical wall okay.

Here Kw is the thermal conductivity ro and ri are the outer and inner radii of the tube okay. And finally this one this accounts for the external or outer convective heat transfer resistance since there are fins on the outside we have to have the finned area and the fin efficiency okay. And here A naught is nothing but the total area which is equal to A fin plus A bare okay. And for water cooled condensers without fins or further simplifications are possible since there are no fins either on the outside or inside you do not find any fin efficiency or fin area okay. And for water cooled condensers in which the water flows through to the tubes the fouling on the outer side that means where the refrigerant flows is generally negligible okay. So you do not find any fouling resistance on the refrigerant side. So you have the internal convective resistance that means on the water side. And this is the fouling resistance on the water side and this is the resistance offered by the wall and this is the refrigerant side convective resistance okay. So the equation becomes simplified right.

So depending upon the particular case and starting with the general expression for the overall heat transfer coefficient you can simplify to particular case okay. Of course you can also do the design for heat exchanger where fins are there both outside as well as inside and fouling is there on both out as well as insides okay. And I was mentioning that for water cooled condensers without fins okay. Generally in a, water cooled condenser the water side heat transfer coefficient will be quite large okay. And if we are using ammonia as a refrigerant then you get very high heat transfer coefficient on the refrigerant side also.

So you find that for ammonia based water cooled condensers the heat transfer coefficient on refrigerant side as well as on the water side both are large. So you need not use fins okay but if you are using water cooled condensers with cfc refrigerants or fluorocarbon based refrigerants we find that the heat transfer coefficient on the refrigerant side is much smaller compared to the heat transfer coefficient on the water side okay. In such cases it may be necessary to use fins on the refrigerant side okay. So that is what is generally done in practice. So whenever fluorocarbons are used in water cooled condensers the tubes are tubes on the refrigerant side are made integrally finned okay. So that you can enhance the heat transfer on the refrigerant side okay. In such case you have to consider the fin effectiveness and fin area on the refrigerant side right.

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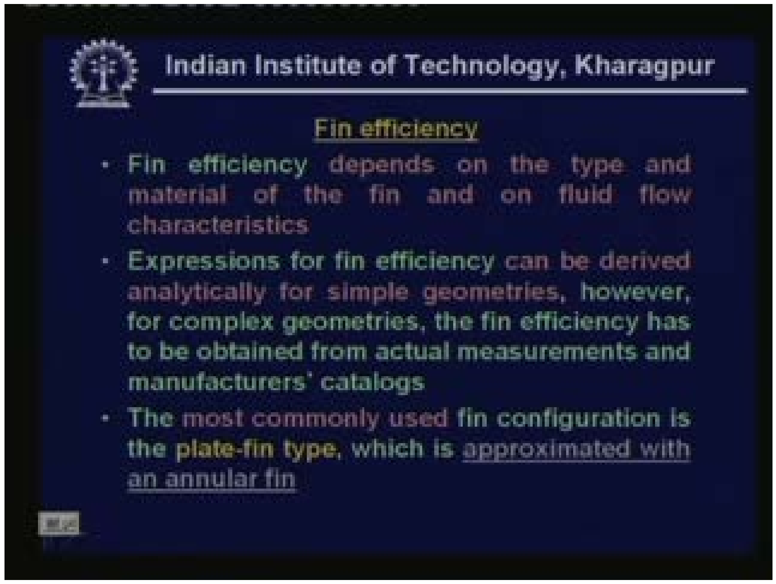

Now let us look at fin efficiency in finned heat exchanger we have to find fin efficiency fin efficiency depends on the type and material of the fin and on fluid flow characteristics expressions for fin efficiency can be derived analytically for simple geometries. However for complex geometries the fin efficiency has to be obtained from actual measurements and manufacturer's catalogs. The most commonly used fin configuration has have already discussed in the last class also is the plate-fin type which is approximated with an annular fin. So let us see how the approximation is done.

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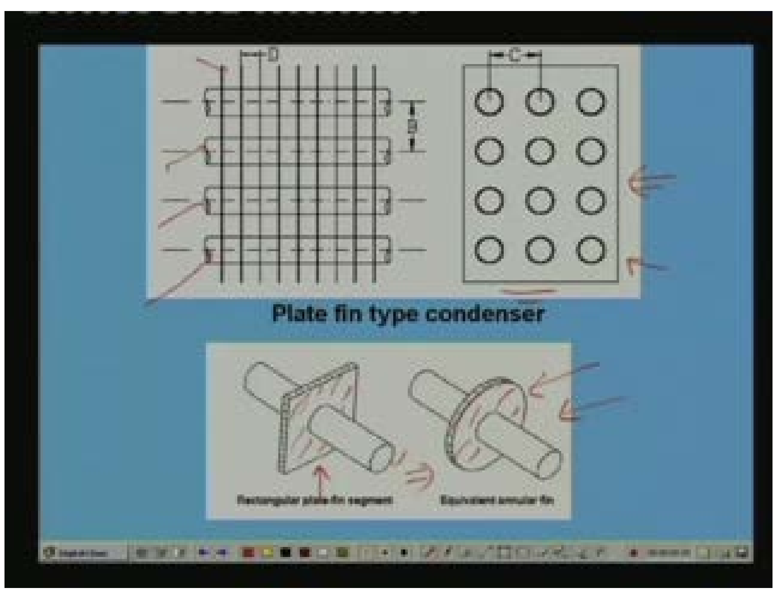

As you know this is the plate fin type of the condenser where you have the tubes okay. Through which the refrigerant flows and you have the plate fins on the outside are which the air flows and this is the side view of the heat exchanger okay. Now analytical expression for fin efficiency of this kind of heat exchanger is difficult okay. So what is done is this is equated to an annular fin. That means this is an annular fin for which analytical expressions are available. So what is done is first that entire fin area is equally distributed among all the tubes. For example if you look at this picture here there are twelve tubes okay. So the entire fin area is equally distributed among the twelve tubes then a one particular plate fin segment is considered that particular plate fin segment looks like this.

You have a single tube and the area associated with single tube okay. And this is equated to an annular fin and how the how this is done this is done by equating the fin areas okay. That means what is done is simply this area is equated to this area okay. And by equating these two areas the outer radius of this annular fin is obtained and then using the analytical expression for the annular fin the efficiency of the plate fin is obtained okay.

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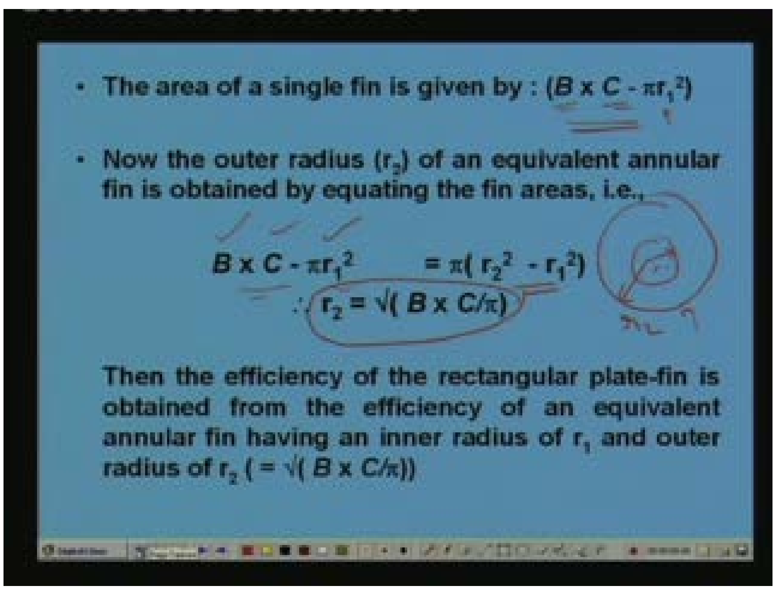

So let me equately show the equations. So the area of a single fin if you look at the layer picture is given by B into C minus pi r one square where B and C are the breadth and height of the rectangular fin and r one is the inner radius of the annular fin okay. So you have the annular fin like this so this is r one and this is r two okay r two is not known to us. Now what is done is this area is equated to the area of the annular fin okay. So area of the annular fin is nothing but phi into r two r two square minus r one square. So this should be equal to B into C minus phi r one square and B and C are known to us and r one is also known to us. So by equating these two expressions we get an expressions for r two and r two is simply equal to square root of B into C by phi okay. So we can find out the outer radius of an equivalent annular fin okay. Then the efficiency of the efficiency of the rectangular fin is obtained from the efficiency of an equivalent annular fin having an a inner radius of r one and outer radius of r two which is equal to square root of B into C by pi okay. So this is the procedure generally followed.

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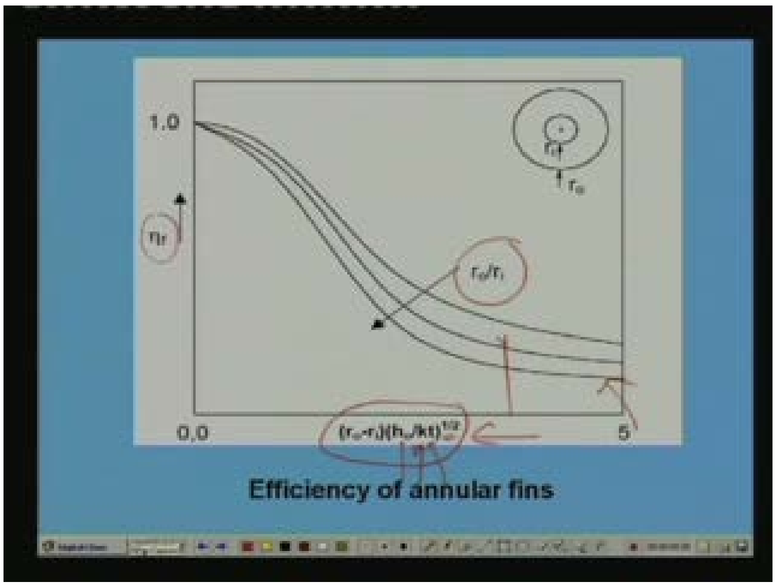

And this picture here shows the fin efficiency of an annular fin okay. So here we have on the y axis the efficiency on the x axi axis we have this parameter r naught minus ri into h naught divided by kt to the power of one by two okay. And this is plotted for different r naught by ri ratios and this is obtained analytically okay. And in this expression in the x axis the h naught is convert to heat transfer coefficient on the fin surface k is the thermal conductivity of the fin material and t is the thickness of the fin okay. So we can this kind of charts are available. So we can find out if you know r naught and ri and the thermal conductivity and thickness and heat transfer coefficient you can find out the fin efficiency okay. So this is how the fin efficiency of plate fin is obtained.

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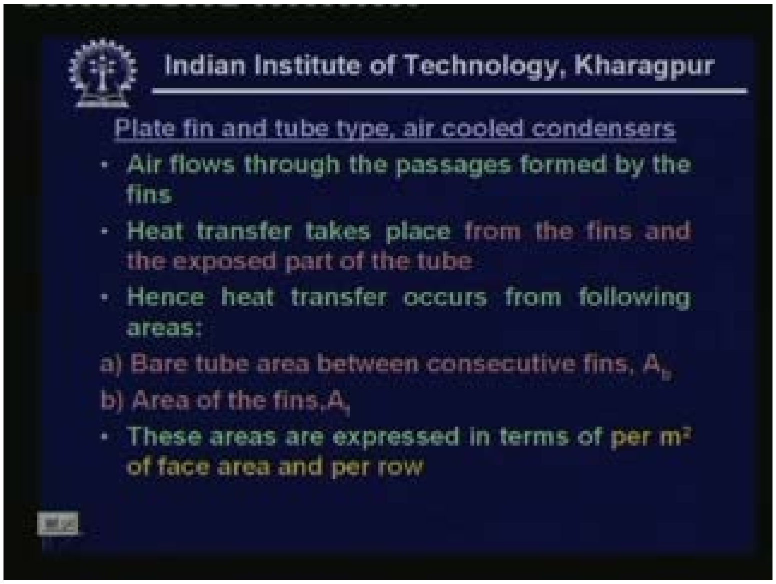

Now let us look at plate fin and tube type air cooled condensers. And let us look at various areas of heat transfer okay. First let us see what how this is constructed in this type of heat exchanger as we have already discussed air flows through the passages formed by the fins and heat transfer takes place from fins and the exposed part of the tube. Hence heat transfer occur from follow for from the following areas heat transfer takes place from the bare tube area between consecutive fins. And this area is called as A subscript b and heat transfer also takes place from the area of the fins A subscript f and these areas are expressed in terms of per meter square of face area and per row okay. So the units of for our example the units are meter square per meter square of face area per row.

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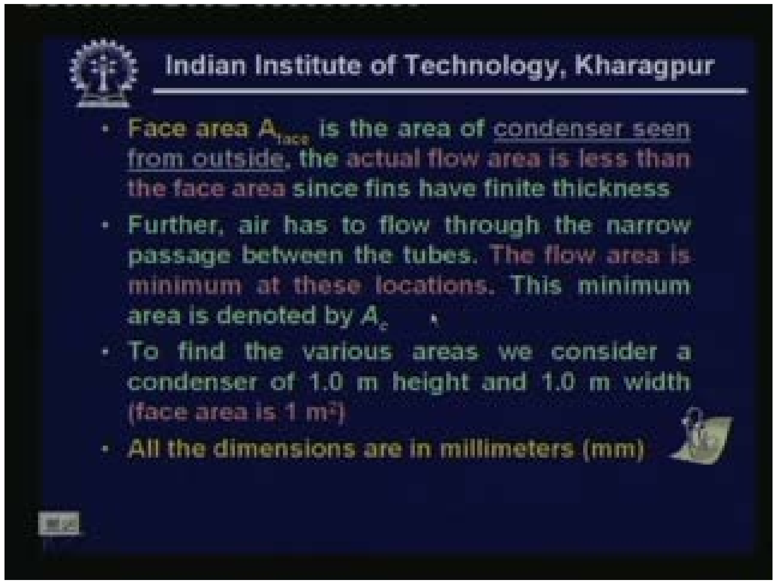

Now what is the face area face area is the area of condenser seen from outside okay.

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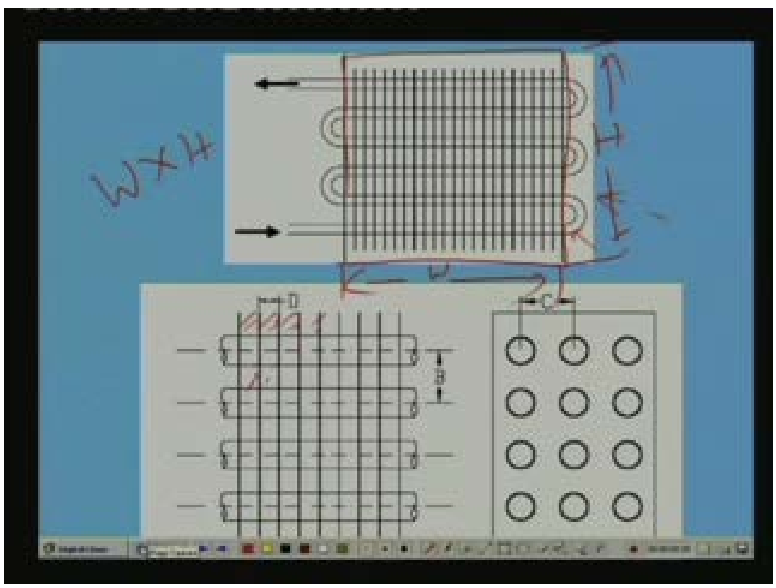

That means this is the plate fin type of condenser. So this is how you see it from outside okay so the face area is nothing but this area okay. If this is the width and this is the height okay. This entire thing is the height and this is the width then face area is W into H okay. But you will find from this figure that the actual area available for air flow is not equal to face area. Because actual area available if we look at this drawing is nothing but this okay. Because the face area consist of the area occupied by the fins as well as the area occupied by the tubes okay. So this area minus the area occupied by the fins minus area occupied by the tubes is what is available for the flow rate of the air okay. So the actual flow area is less than the face area since fins have finite thickness that have already explained further air has to flow through the narrow passages between the tubes and the flow area is minimum at these locations. And this minimum area is denoted by A subscript c to find the various areas we consider a condenser of one meter height and one meter width. That means we take a condenser having a face area of one meter square and all the dimensions are expressed in millimeters.

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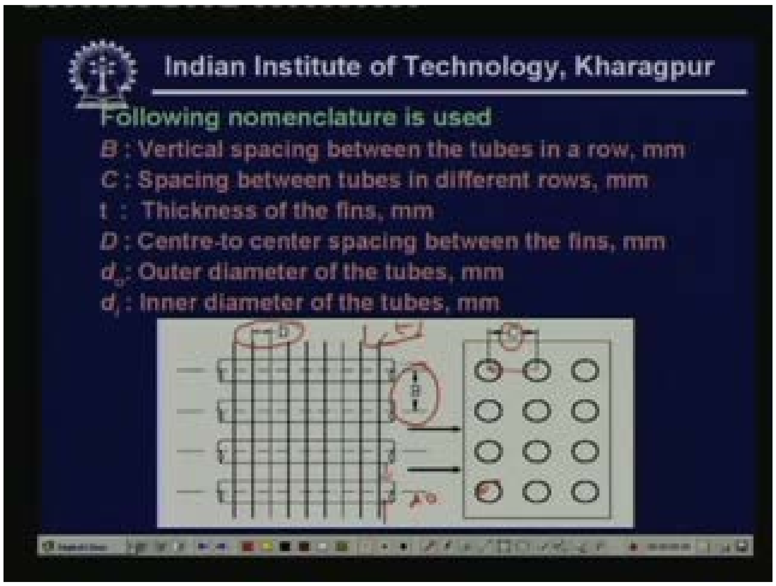

And we will use the following nomenclature B is vertical spacing between tubes in a row. That means this is this is one single row. So B is this okay, center to center distance between two adjacent tubes and C is the spacing between tubes in different rows. That means in the side view this is the center to this center distance between two adjacent rows okay. That is C and t is the thickness of the fin okay. This is t and D is the center to center spacing between the fins that is this and d naught is the outer diameter of the tube okay. That means this is the d naught and di is the inner diameter of the tube that mean this okay. So this is the nomenclature we use.

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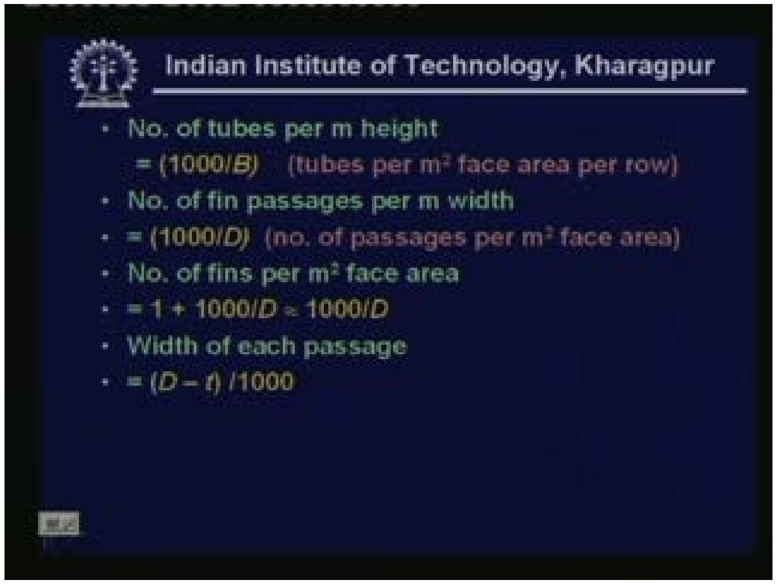

Now number of tubes per meter height is nothing but thousand divided by B we are using thousand because B is in millimeters okay. So number of tubes per meter height is thousand by B and these are these are the number of tubes per meter square face area per row next number of fin passages per meter width this is given by thousand by D and these are the number of pass fin passages per meter square face area per row and number of fins per meter square area is nothing but one plus thousand by D which can be approximated as thousand by D okay. Because the D is generally is small and width of each passage is nothing but capital D minus t by thousand. Because capital D is the center to centre distance between two adjacent fins and the small t is the thickness of a fin.

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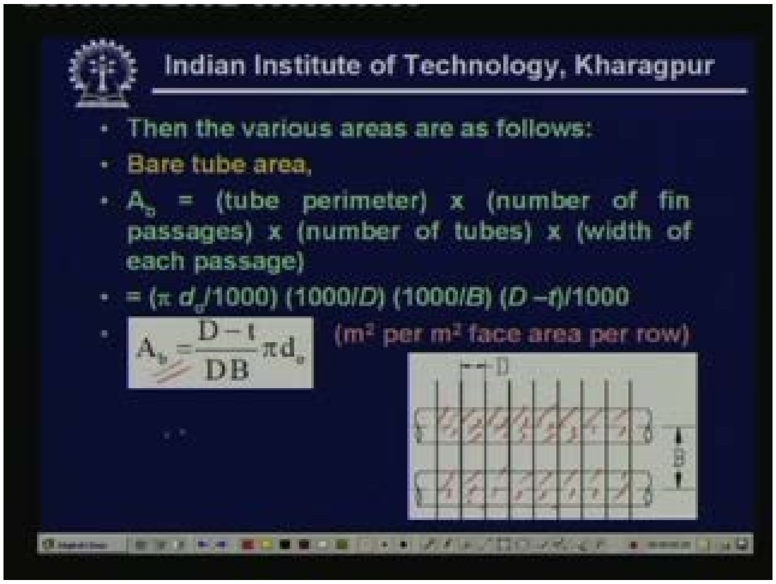

Now then the various areas are as follows the bare tube area okay. A subscript b is given tube perimeter into number of fin passages into number of tubes into width of each passage okay. So bare tube area is basically nothing but this area okay. So excluding the fins like this you have to configure all the tubes in a row per meter square of face area okay. So if you substitute the expressions are tube perimeter which is nothing but phi d phi d o divided by thousand and number of fin passages are thousand per divided by D number of tubes thousand divided by B and width of each passage is D minus t by thousand. So if you substitute all this finally you find that the bare tube area is given by D minus t divided by DB into phi d and this if a meter square per meter square in face area is per row okay.

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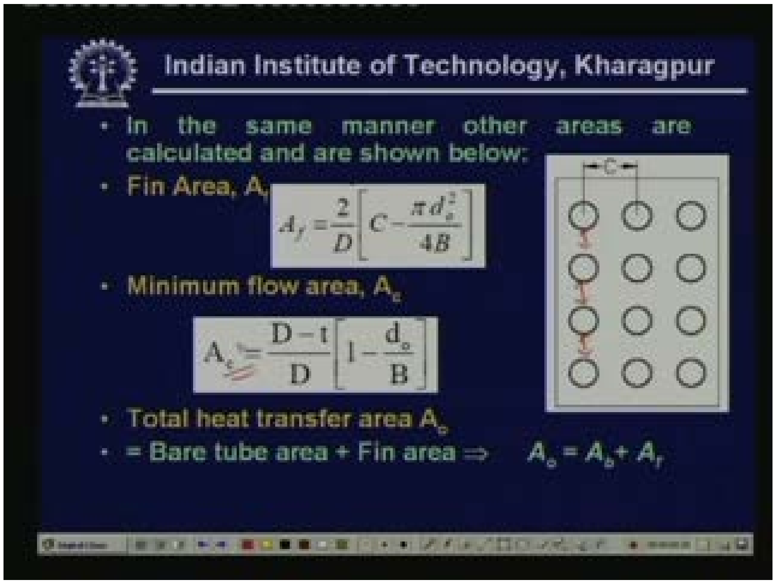

Likewise we can calculate other areas for example the fin area okay. Fin area means for example you look at one fin single fin for a single fin the fin area is nothing but this area excluding the area occupied by the tubes okay. You have to multiply this by two because heat transfer taking place from both sides of the fin this is the area of a single fin. So like that we have to find out how many fins are there per meter square of face area okay. So if you substitute the expressions you will find that the fin area is given by A subscript fin is equal to two by capital D into C minus phi d o phi d o square divided by four B similarly the minimum flow area minimum flow area is actually between the tubes and the minimum flow area is somewhere here okay. So this is where the passage becomes narrow okay.

On this distance is the minimum distance so we can find out what is the flow area at this point okay. So the expression for that is given by this A subscript c is equal to capital D minus t by D into one minus d o by B okay. And the total heat transfer area A subscript o is bare tube area plus fin area that is Ab plus Af. So if you know all these parameters like C capital C capital D small do B etcetera. You can calculate all these areas per meter square face area per row okay.

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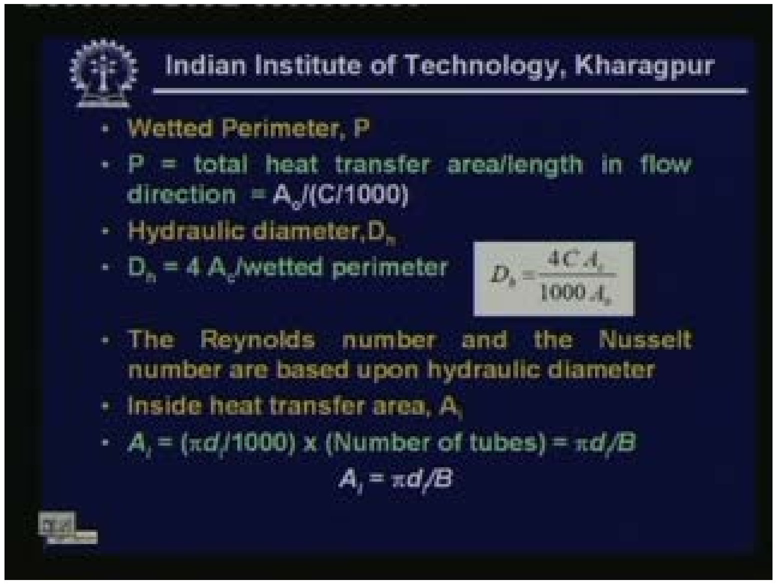

Next we find wetted perimeter wetted perimeter P is defined as total heat transfer area divided by length in flow direction length in flow direction is nothing but per row it is nothing but C divided by thousand. So wetted perimeter is A subscript o that is a total area divided by C by thousand okay why do we need wetted perimeter. Because the hydraulic diameter is defined as four into minimum flow area that is A subscript c divided by wetted perimeter okay, hydraulic diameter is defined in terms of wetted perimeter and minimum flow area. So f you substitute the expressions we find that the hydraulic diameter is given by four into C into A subscript c divided by thousand A subscript o okay.

And why do we need hydraulic diameter because the Reynolds number and the Nusslet number are based upon hydraulic diameter. So if you want to find out the Reynolds and Nusslet number because the heat transfer coefficient depend on these number we have to find out the wet hydraulic diameter okay. And the inside heat transfer area A subscript i is nothing but phi d i by thousand into number of tubes. So this becomes simply becomes phi di divided by capital B okay.

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So this is how we can estimate if we know the center to center spacing between the tubes fins fin thickness etcetera per meter square per row we can estimate different areas okay. Heat transfer areas and we can also estimate the hydraulic diameter which will be used in estimating the heat transfer coefficients okay. Now let us look at how to estimate heat transfer coefficients on the external fluid side as well as on the refrigerant side okay.

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First let us look at the heat transfer coefficient on the external fluid side. First let us look at air cooled condensers. So air side heat transfer coefficient in air cooled condensers flow over finned surfaces okay. So that means we will be obtaining heat transfer coefficient for air flow or finned surface. So correlations have been obtained by Kays and London for various fin and tube configurations for both inline as well as staggered arrangements okay. So what is inline and staggered arrangements.

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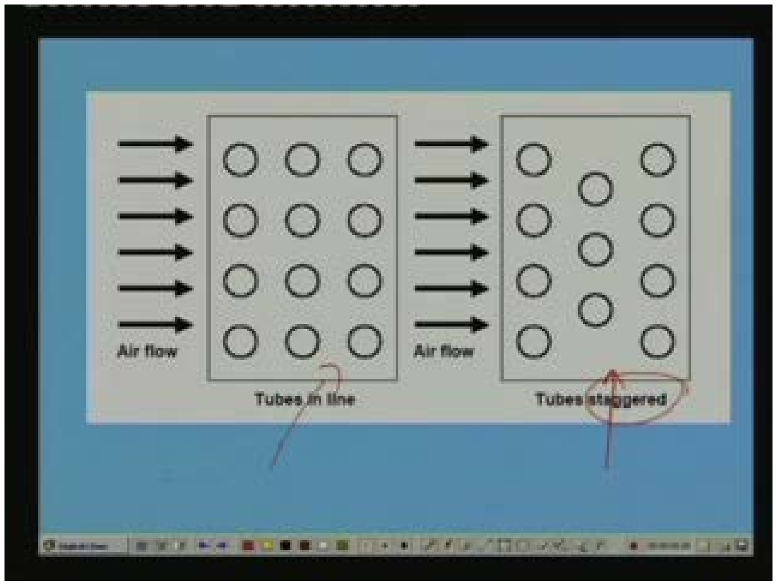

This is tubes inline okay, so you can that tubes in adjacent rows there in line okay. So this inline arrangement whereas here the adjacent tubes are not inline okay, so we call this as staggered arrangement okay, so Kays and London have obtained expressions for heat transfer coefficient for both these arrangements. That means for inline as well as staggered arrangements and they have found a general correlation that is given by Nusselt number is equal to point one one seven Reynolds number to the power of point six five prank number to the power of one by three and remember that here the Nusselt and Reynolds number are based on hydraulic diameter. So once you know the hydraulic diameter and velocities etcetera. You calculate the Reynolds number prank number from that you calculate the Nusselt number. And from the Nusselt number you calculate the heat transfer coefficient.

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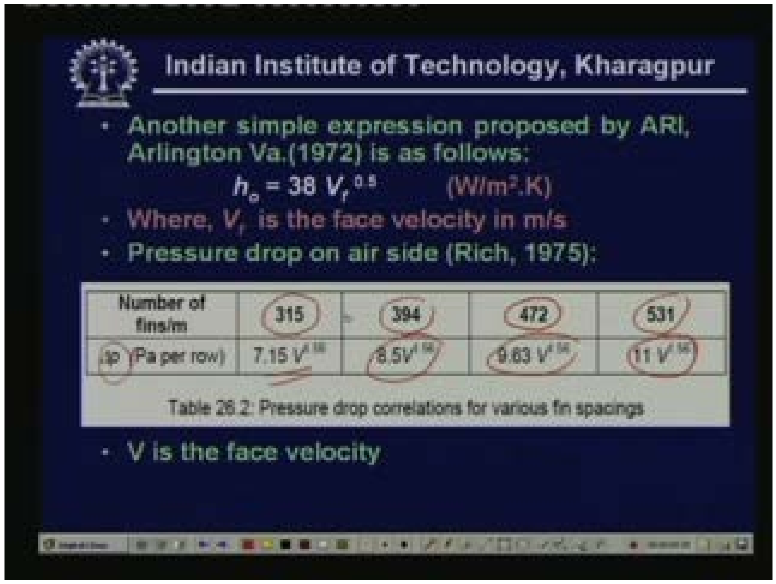

Now there is another simple expression proposed by air conditioning and refrigeration institute Arlington the USA that is simply it relates heat transfer coefficient to the face velocity okay. So that is given by h naught is equal to thirty-eight into V to the power of point five where V subscript f is the phase velocity in meter per second and h naught is the heat transfer coefficient in watt per meter square Kelvin okay. So in the absence of detailed data you can use this simple expression for rough estimation of heat transfer coefficient on finned surfaces okay. Then pressure drop on air side for fin surfaces Rich has proposed the following correlation. Here the correlation is it correlates pressure drop in Pascal's per row in terms of number of fins per meter and the face velocity V okay.

So this is the correlation delta p in Pascal's per row is an empirical correlation. So this is equal to seven point one five V to the power of one point five six is the number of fin per three fifteen per meter. Similarly this is the expression for number of fins of three ninety-four per meter four seventy-two fins per meter. This is the expression five thirty-one fins per meter. This is the expression and here again V is the face velocity okay. So this is again a simple empirical correlation based on experimental observations.

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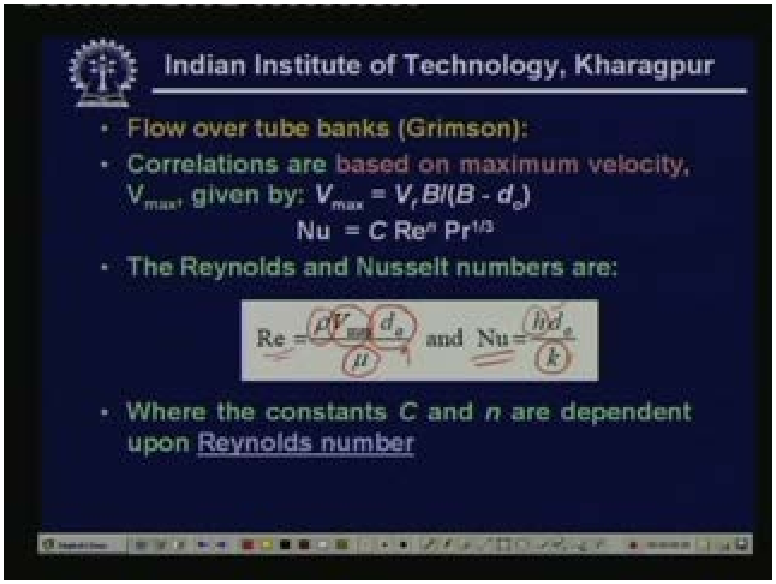

So let us look at flow over tube banks for flow over tube banks Grimson has proposed correlation the correlation are based on maximum velocity V max V max is where the area is minimum okay. So V max is nothing but Vf into B divided by B minus d naught where V subscript is the face velocity B as you have seen is nothing but the center to center distance between tubes in a particular row and d naught is the outer diameter of the tube. So the expression is like this Nu into C into Re to the power of n into Pr to the power of one by three where R is the Reynolds number and pr is the prank number and the Reynolds and Nusselt numbers are defined like this row into V max okay.

So here the, you they you have to be very careful how the Reynolds number and Nusselt numbers are defined okay. So here it is defined in a slightly different manner here we are not using hydraulic diameter anything but simply we are using the outer diameter of the tube. Because we are talking about tube banks okay. So d naught is the outer diameter of the tube mu is the viscosity of air rho is the density of air and V subscript max is the maximum velocity given by the above expression. Similarly Nusselt number is given by h into d naught divided by k where k is the thermal conductivity of air h is the convective transfer coefficient. So first find out Reynolds number and prankle number. And then find our Nusselt number and from that find out the heat transfer coefficient and the constant c and n are dependent on Reynolds number.

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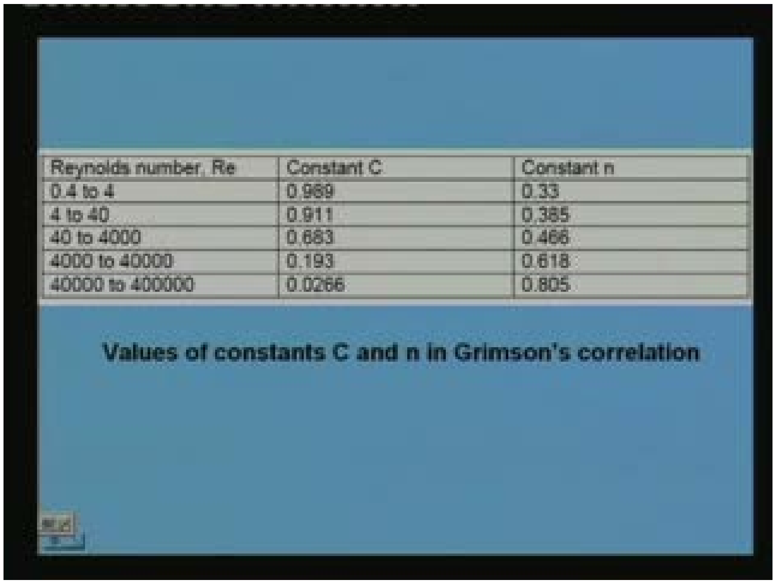

Reynolds number, Re	Constant C	Constant n
0.4 to 4	0.989	0.33
4 to 40	0.911	0.385
40 to 4000	0.683	0.466
4000 to 40000	0.193	0.618
40000 to 400000	0.0266	0.805

Values of constants C and n in Grimson's correlation

For example these constants are given in this table if the Reynolds number is between point four to four then the constant c point nine eight nine and constant n is point three three. And if it is between four to forty the constant c is point nine one one and constant n is point three eight five like that for different Reynolds number covering a wide range. Now the different values of constant c and n are given okay. So we can use this table and get the values of c and n then use a correlation and find out the heat transfer coefficient.

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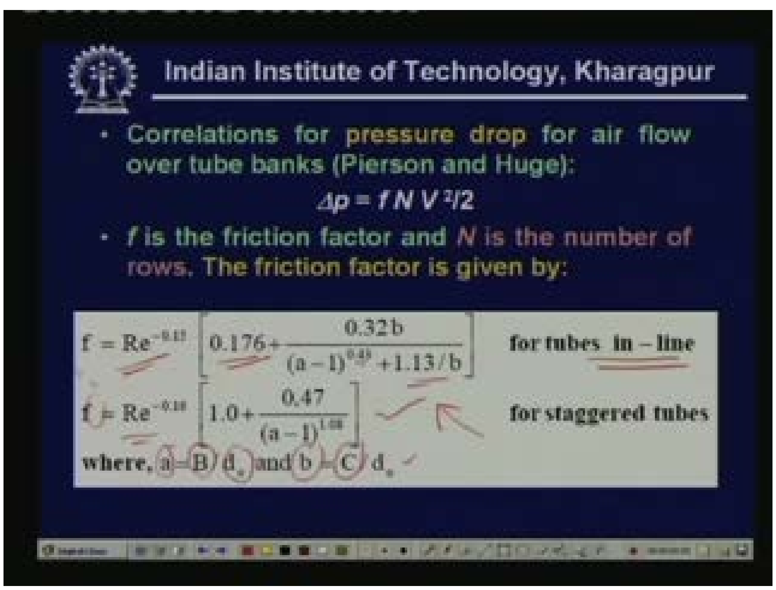

And correlations for pressure drop for air flow over tube banks okay. We have seen the pressure drop correlation for fin surfaces. So now the pressure drop correlation for tube banks is given by this expression it is given as suggested by Pierson and Huge and here delta p is given by fNV square by two where f is the friction factor and N is the number of rows okay. You see can the pressure drop depends upon the number of rows and V is the velocity and the friction factor f is given by this expression f is equal to Reynolds number to the power of, for tubes in line okay, inline arrangement.

This is the expression f is equal to Re to the power of minus point one five multiplied by point one seven six plus point three two b divided by a minus one to the power of point four three plus one point one three by b. Similarly for staggered tube arrangement this is the expression for friction factor okay. So if you know the number of rows and if you know the velocity and if you then you can calculate the friction factor and delta p and in this friction factor expression. The constants a and b are defined in terms of the tube to tube spacing capital B and the outer diameter of the tube d naught and row to row spacing C okay. If you know these details we can find out a and b then from that we can find out the friction factor okay.

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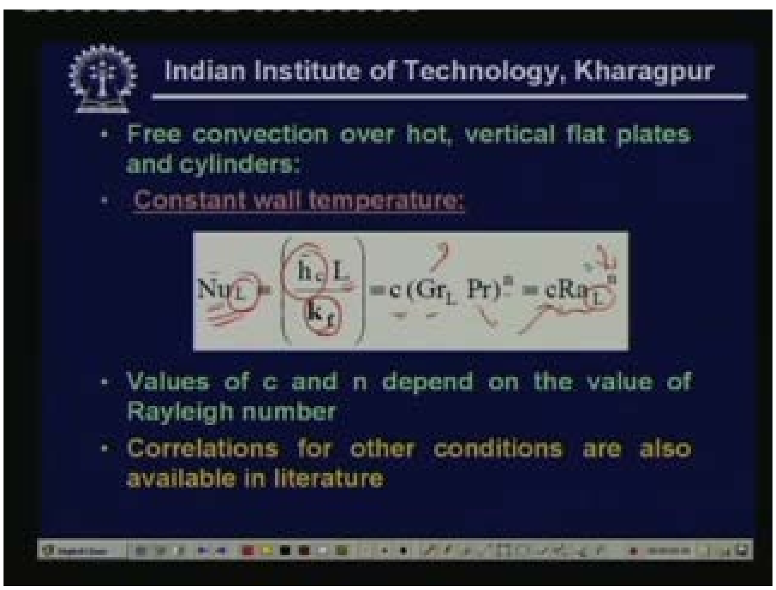

Now for this is what are the expressions we have shown. So for whole good for force convection type air cooled condensers okay. So in fact in the last lecture I have i have also mentioned natural convection type condensers okay. One thing I would like to emphasize here is at the correlations. I am showing are not the only correlations okay. If you look at heat transfer fluid mechanics literature large number of correlations have been developed over the years okay. The correlations varying their applicable for some particular ranges and some correlations are more accurate than other correlations right. So huge amount of literature is available and one has to choose the right correlation. And use the correlation and obtain the heat transfer coefficient and pressure drop values okay.

So for example for free convection over hot vertical flat plates and cylinders a typical correlation is given here the average Nusselt number NuL okay. Here L is the height of the plate or height of the cylinder and Nu is the average Nusselt number this is defined as average heat transfer coefficient multiplied by the height of the plate or cylinder divided by the thermal conductivity of air okay. That is equal to c into G r subscript L into Pr to the power of n where Gr is the Grashof number Pr is the Prankel number and the product of Grash of Prankel number is called the another is rally number okay, Ra and here the subscript L means the characteristic um length should be L that is the height of the plate right. So here again you can see that there are constants c and n and the values of c and n depend on the value of rally number. So just like the earlier expression where we had values for c and n for different Reynolds number for this particular case also the values of c and n for different values of rally number have been obtained and they are tabulated.

So we can find these values if you know the rally number right and correlations for other conditions are also available in literature. For example for horizontal flat plate or horizontal cylinders etcetera okay. So what I am showing just an example of a typical example of heat transfer coefficient correlations.

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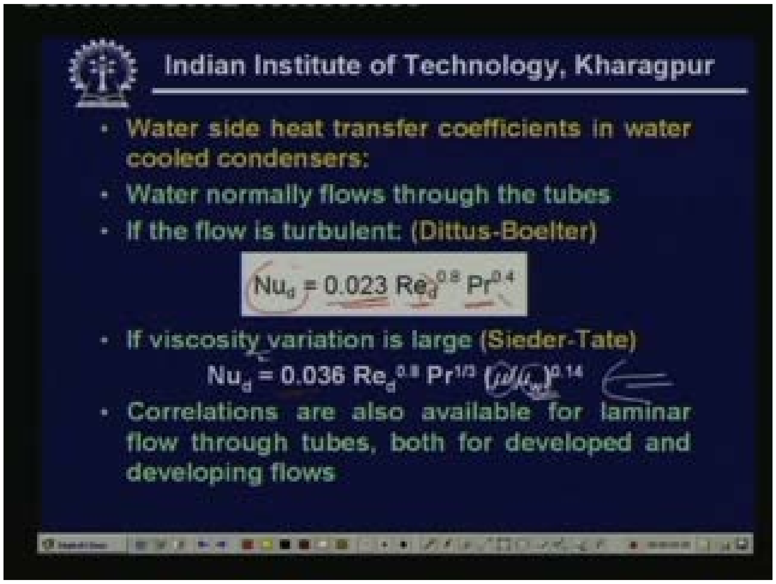

Now let us look at water cooled condensers and see how to calculate water side heat transfer coefficients in water cooled condensers. Normally we have seen that water flows through the tubes and generally that flow is turbulent. So if the flow is turbulent and if the flow is through the tubes then we can use the simple Dittus-Boelter correlation which was discussed earlier the Dittus-Boelter correlation is very popular. And it is given by this expressions Nu subscript d this is the Nusselt number based on the inner diameter of the tube. This is equal to point zero two three into Reynolds number to the power of point eight and Frankel number to the power of point four okay. This is for turbulent flow inside the tubes right and as is said Re is Reynolds number and Pr is the Prankel numbe rand this Re that is Reynolds number and Nusselt number are based on the internal diameter of the tube.

This can be used most of the time but if viscosity variation is large then one can use Sieder-Tate equation which gives better accuracy okay. So this Sieder-Tate correlation is given by this Nu subscript d is equal to point zero three six Re to the power of point eight Pr to the power of one by three multiplied by mu by mu w to the power of pint one four. Here mu w is nothing but the viscosity of water at wall temperature and this is the viscosity of the water at bulk temperature okay. So this expression can be used if the temperature variation between the wall bulk fluid is null or the if the viscosity variation is large okay. And correlations are also available for laminar flow through tubes both for developed and developing flows.

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Now let us look at condensation heat transfer coefficient condensation can be in general filmwise or dropwise for a okay, as a name implies filmwise condensation means the entire surface will be covered with a film of refrigerant okay. The, that means the liquid film of refrigerant covers the entire surface of the condenser tube okay. So is a or a plate so that kind of condensation is known as filmwise condensation dropwise condensation means the liquid does not cover the entire surface. But there are areas where liquid is present and there are areas where liquid is not present basically condensation takes place in the form of droplets okay. So that is known as dropwise condensation then only we find that whether the condensation is filmwise or dropwise depends upon several factors okay.

One of the important factors which decides the whether the condensation is filmwise or dropwise is the nature of the surface on which condensation is taking place okay. And generally the dropwise condensation gives much higher heat transfer coefficients compared to filmwise condensation okay. So surfaces can be treated. So that we can have dropwise condensation on the surfaces okay. But in heat transfer literature and in the design of condensers normally we assume that the condensation is filmwise condensation okay. This is because even though you have dropwise condensation in the beginning after prolonged use the surface may use its characteristics. And slowly the dropwise condensation may give rise to filmwise condensation. So if you have designed it the condenser based on dropwise condensation then you find that after some time the condensation becomes filmwise condensation and the heat transfer coefficient will be less okay. The, that means after prolonged use the capacity of the condenser reduces because the heat transfer coefficient reduces okay. So to avoid this one all the condensers based are designed based on assuming that the condensation is filmwise condensation okay. So this is the conservative approach. Because the area that you have obtained using this assumption is conservative. That means you under predicting the heat transfer coefficient okay.

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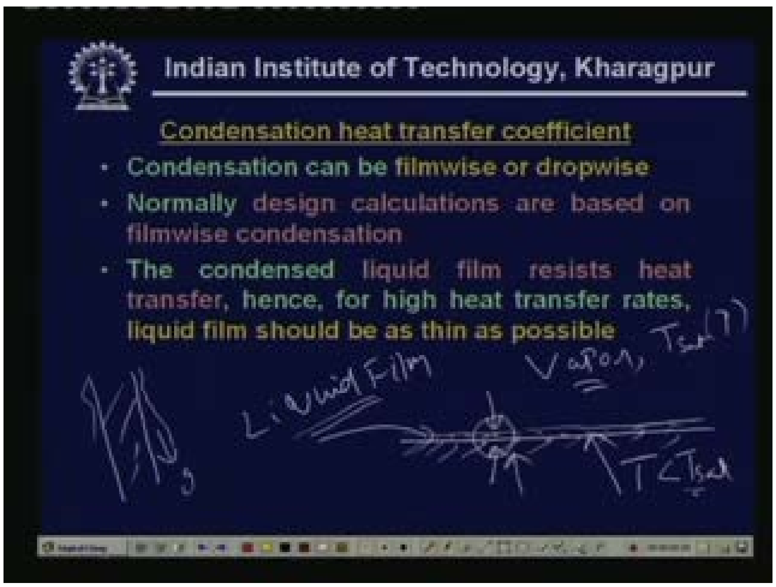

As I said normally design calculations are based on filmwise condensation. The condensed liquid film resists heat transfer hence for high heat transfer rates liquid film should be as thin as possible. Let me quickly explain this with a very simple example for example we have a plate let us say okay. This is plate this plate let's say for the time being the horizontal plate this is in contact with vapour. Let us say this vapour has a saturation temperature of T sat okay. At the, that particular pressure p and let the surface be at a temperature T which is less than T sat right. So when this vapour comes in contact with surface this surface whose temperature is less than the saturation temperature of the vapour then this vapour condenses on the surface and rejects the heat of condensation to the, to this particular surface okay. That means you will find that this surface will be gradually covered with layer okay. If you are assuming filmwise condensation you will find that this surface is covered by a liquid film okay. Once the liquid film is formed on the surface these liquids will add resistance for heat transfer. That means this itself prevents or reduces the rate at heat which heat is being transferred from vapour to the surface okay. That means additional resistance is created.

Because of the presence of the liquid film. Now how effective is this condensation process depends upon what is the thickness of the liquid. Now you have a horizontal surface then this liquid film may grow in thickness. So the, you will find the condensation rate reduces progressively. But suppose you have a vertical surface let us say or vertical cylinder and if condensation is taking place on the outside then because of the gravity the liquid film will continuously drain out. So you will find that the condensation rate will be high okay. So the basically the condensation heat transfer coefficient depends upon how you oriented the surface and how fast the condensate is being drained out okay. So this is an important factor to be considered in the design of condensers. So as I said for a good design the continuous draining of condenser liquid is required and this depends mainly on the geometry of the condensing surface.

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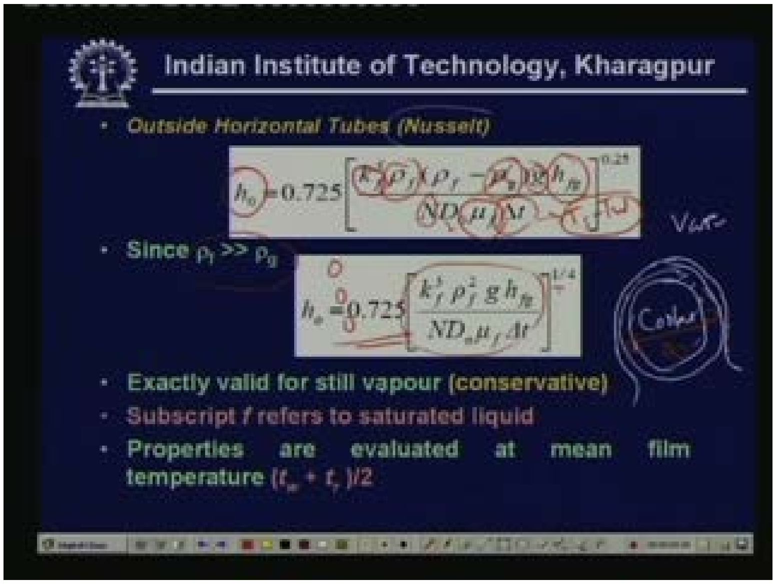

Now let us look at simple case where condensation is taking place outside horizontal tubes okay. That means you have a okay. So you have a tube like this let us say and inside the tube the coolant is flowing okay. This is coolant and outside you have the refrigerant vapour okay. So when the vapour comes in contact with this tube outside the tube condensation occur. So a film of a liquid film of refrigerant forms on the outside of the horizontal tube okay, for such cases Nusselt has given a correlation this is a classical correlation suggel suggested by Nusselt here. In this particular correlation h naught is the condensation heat transfer coefficient okay. And here kf this is the thermal conductivity of the saturated liquid refrigerant. And rho f is the saturated density of the liquid refrigerant and rho g is the satu saturated density of the vapour g is the acceleration due to gravity hfg is latent heat of vaporization N is the number of tubes in a row okay.

In a vertical row that means in a vertical row if there are three tubes then N becomes three and D naught is the outer diameter of the tube mu f is the saturated viscosity of the liquid and delta t is the temperature difference between the surface and the saturated refrigerant okay. So delta t is t refrigerant minus okay. So this is t saturated minus t wall right since the density of the liquid is much larger than the density of the vapour so we can neglect this term okay. So then this Nusselt correlation simplifies to this form okay h naught is point seven two five multiplied by this factor to the power of one by four okay.

This expression has been obtained analytically assuming that the vapour is still okay that means the vapour is not moving you have a condensing surface in a still vapour okay.

But in a actual case you find that there will be some vapour movement okay. That means the correlation suggested by Nusselt will always result in is a conservative correlation. Because the heat transfer coefficient obtained by using this will be slightly smaller than the actual condensation heat transfer coefficient. Because of the movement of the vapour okay. And here all the properties that is the thermal conductivities densities latent heat viscosity etcetera are evaluated at mean film temperature okay, mean film temperature is defined as the saturation temperature of the vapour plus wall temperature divided by two okay. So this is the condensation heat transfer coefficient outside horizontal tube.

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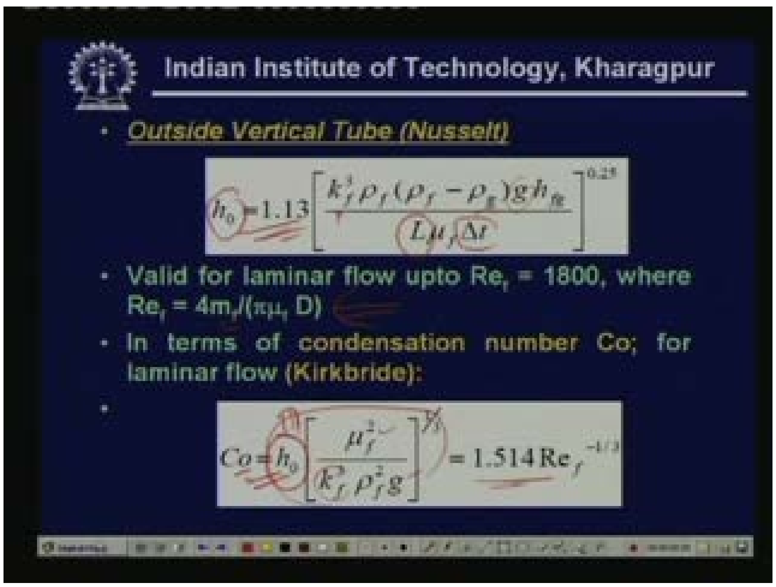

Again Nusselt has suggested an analytical expression for condensation outside vertical tubes okay. So this also looks similar in form so the earlier expression okay. Again the nomenclature is same this is the thermal conductivity of the saturated liquid here s stands for saturated liquid and g stands for saturated vapour okay. And here L is the height of the vertical tube and again delta t is the temperature difference g is the acceleration due to gravity hf is the latent heat of vaporization. If you compare a horizontal and vertical tube and the expressions you find that the heat transfer coefficient for a vertical tube if everything else is remaining constant will be larger compared to horizontal tube because the condensate can drain out better in case of a vertical surface okay. So you get a higher heat transfer coefficient here and this above expression is valid for laminar flow.

That means up to Reynolds number of eighteen hundred and the Reynolds number here is defined as Ref this is the Reynolds number expression Ref is four into m subscript f divided by pi mu f into D where m subscript f is the condensation rate. And mu f is the viscosity of the saturated liquid and D is the outer diameter of the tube and in terms of condensation number Co Kirkbride has suggested this correlation. This is Kirkbride's correlation for condensation on a vertical tubes outside the vertical tubes. To here this h naught is the heat transfer coefficient obtained from Nusselt's expression okay. And again mu f is the viscosity and rho f is the density g is the acceleration due to gravity and kf is the thermal conductivity.

So here this entire quantity is known as condensation number condensation number is equated to Reynolds number right Ref so condensation number is one point five one four into Reynolds number to the power of minus one by three. So this is the Kirkbride's correlation.

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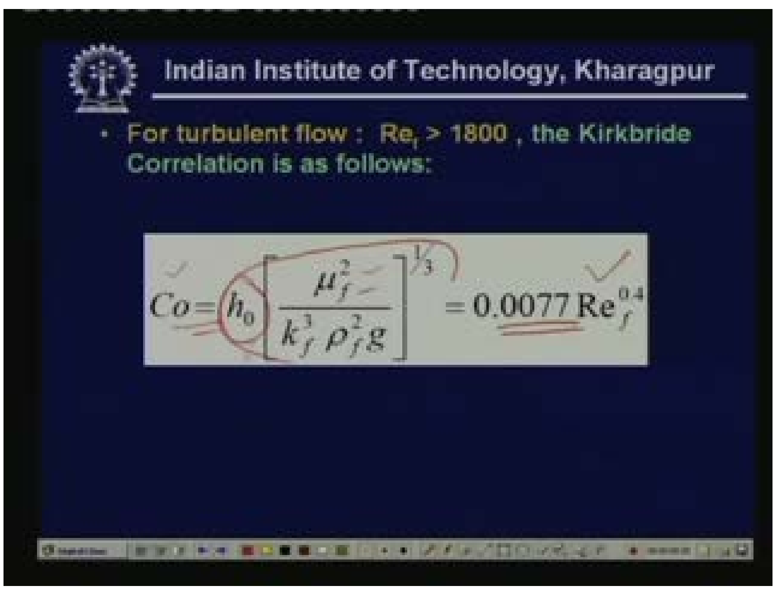

Kirkbride has also suggested a correlation for turbulent flow. That means when the film Reynolds number is greater than eighteen hundred okay. So if the film Reynolds number is greater than eighteen hundred then the condensation number defined by this parameter is equal to point zero seven seven into Ref to the power of point four okay. Where h naught is the condensation heat transfer coefficient mu f is the thermal conductivity and all. So if you know the film Reynolds number and if you know the properties then you calculate the condensation number and from the condensation number you can calculate what is the heat transfer coefficient okay.

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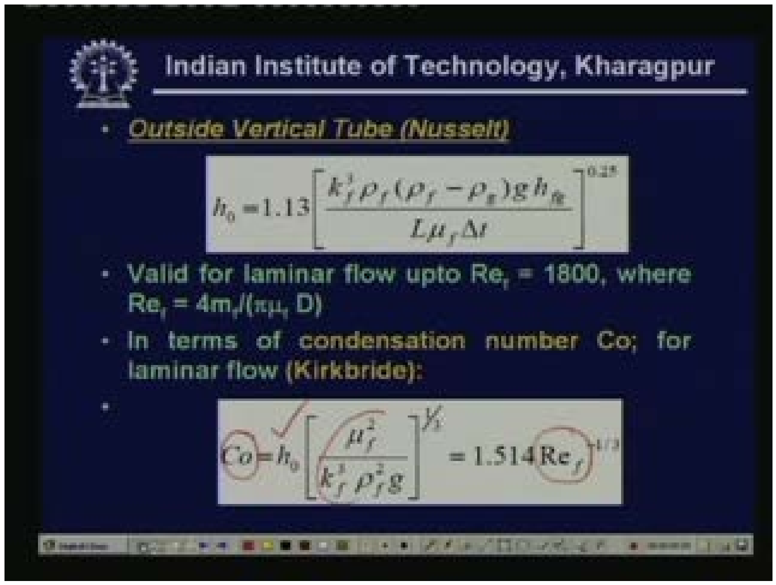

Similarly in the previous this thing in the previous expression. So laminar flow also you can find out the film Reynolds number then you calculate the condensation number and from the condensation number. And the properties you can calculate what is the external heat transfer coefficient.

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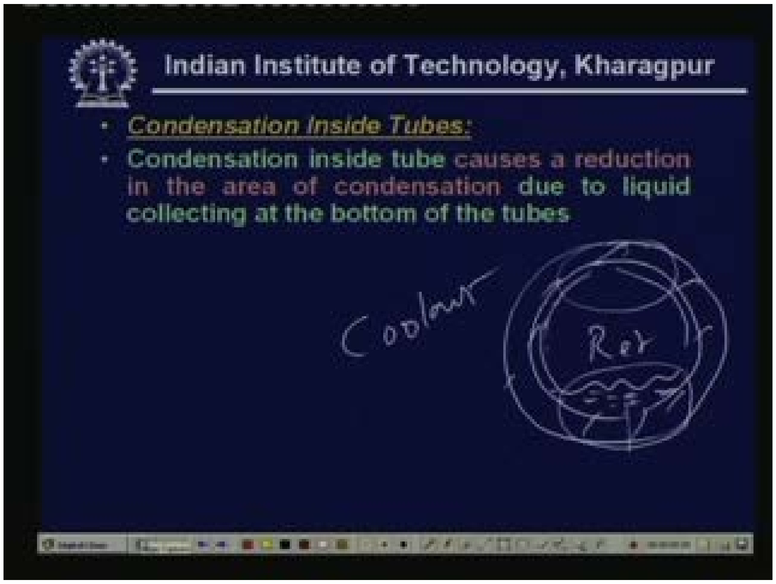

Now let us look at condensation inside tubes this is more complicated actually okay. Compared to condensation outside the tubes or outside the surface this is because condensation inside tube causes a reduction in the area of condensation due to liquid collecting at the bottom of the tubes. That means let us say that refrigerant is vapour refrigerant vapour flowing through the tube. Let us take a large tube okay.

This is a wall of a tube and outside you have coolant and refrigerant is flowing through the tube. When refrigerant is flowing through the tube since the coolant is flowing outside the refrigerant starts condensing on the surface of the inner surface of the tube. When it condenses in the inner surface of the tube the liquid. Because the gravity will come down and you may find that liquid is settling down at the bottom okay. When liquid is settling down at the bottom the resistance at the bottom is different from the resistance at the top right and the heat transfer area effective heat transfer area available for heat transfer area also gets reduced. Because of the liquid collection at the bottom of the tube okay.

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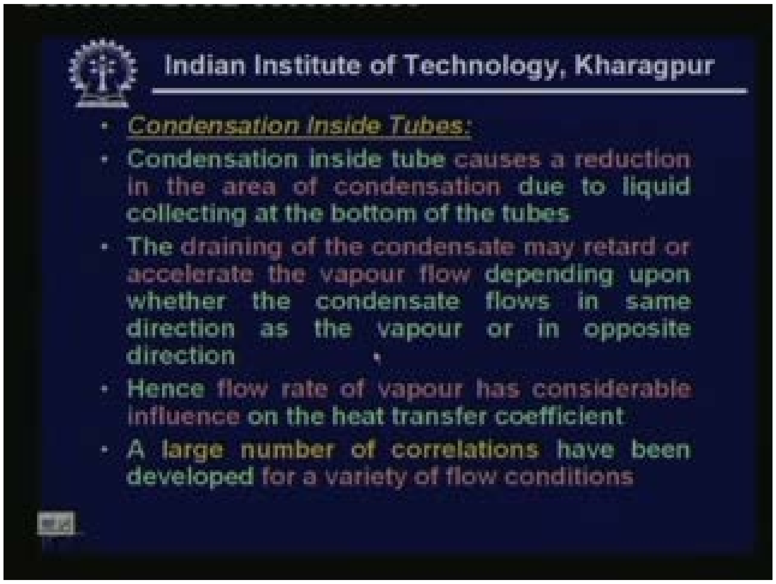

So this fact has got to be considered while estimating the condensation heat transfer coefficient inside the tubes. And the draining of the condensate may retard or accelerate the vapour flow depending upon whether the condensate flows in the same direction as the vapour or in opposite direction. That means when the condensate and vapour are flowing in the same directions direction then the vapour flow gets accelerated where as if they are flowing in the opposite direction the vapour flow gets retarded. And this also has an effect on the heat transfer coefficient hence flow rate of vapour has considerable influence on the heat transfer coefficient a large number of correlations have been developed for a variety of flow conditions.

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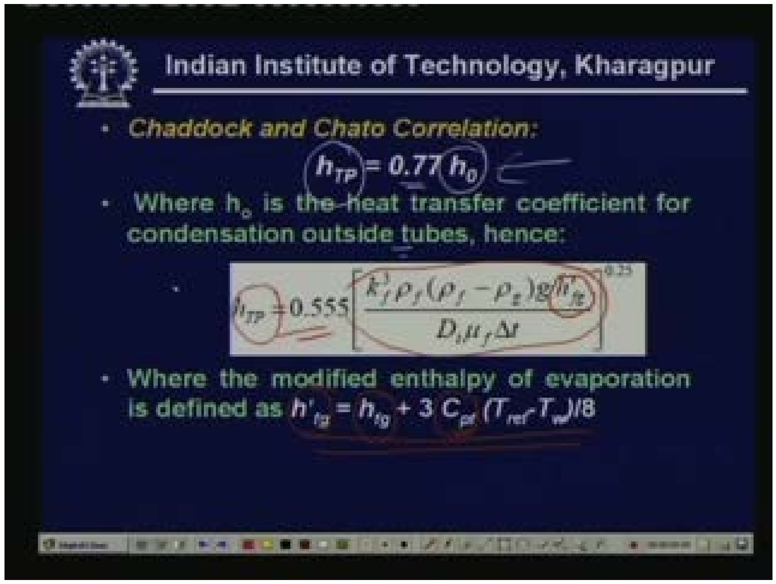

And I will show here one of the simplest correlation this is suggested by Chaddock and Chato okay. This is known as the Chaddock and Chato correlation here the correlation is like this h subscript tp is nothing but the condensation heat transfer coefficient inside the tube okay. This one this is equated two h naught where h naught is the heat transfer coefficient for condensation outside the tubes okay. That means you can use Nusselt's correlation or Kirkbride's correlation and estimate h naught which is nothing but the condensation heat transfer coefficient outside the tubes.

Then condensation heat transfer coefficient inside the tube is equal to point seven seven multiplied by h naught okay. And if you substitute the expression for h naught the Chaddock and Chato correlation is like this heat transfer coefficient inside the tubes okay. So is point five five five multiplied by this factor. So this is similar to your Nusselt's correlation with the difference that instead of latent heat of vaporization here what is used is a modified enthalpy of evaporation is used in Chaddock and Chato correlation. That is defined by this the modified enthalpy of evaporation h subscript fg okay h prime subscript as g is equal to hsg which is nothing but the latent heat of vaporization plus three Cpf where Cpf is the saturated specific heat of the liquid multiplied by T refrigerant minus T wall divided by eight okay. So you have to first find out this then you substitute this in this then find out the heat transfer condensation heat transfer coefficient inside the tubes okay. So this is one of the simplest correlation available for heat transfer coefficient inside the tubes.

As I said this is a very complicated case and the heat transfer coefficient value will be varying depending upon the flow resume or flow pattern. For example is the tube is vertical you have one heat transfer coefficient if it is horizontal you will have another heat transfer coefficient, if it is inclined you will have different heat transfer coefficient. So depending upon the geometry depending upon the fluorides and all the heat transfer coefficient will be different okay. And large amount of literature is available for estimating condensation heat transfer coefficient for different types of geometries for different fluorides etcetera okay.

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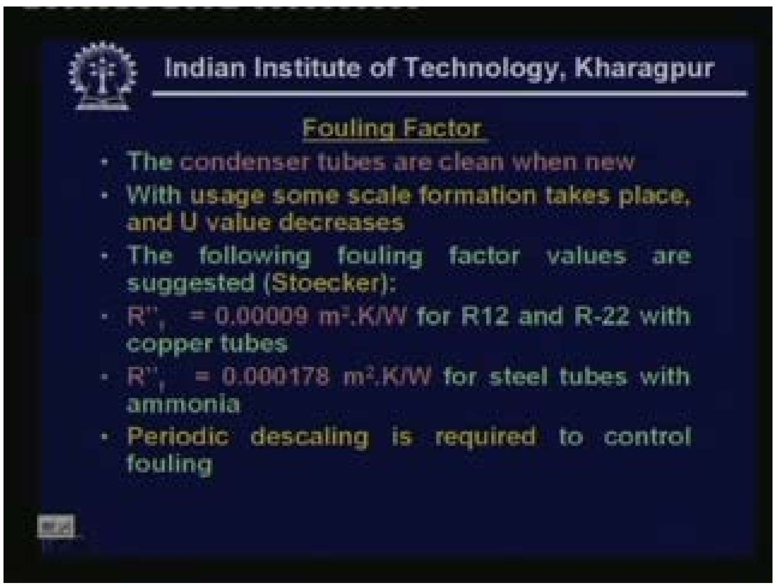

Now let us look at the other factors I have, I was discussing about the fouling factor okay. The generally the fouling takes place when we are using water cooled condensers. Because on air side the fouling is generally less when whenever we are using the water cooled condensers normally fouling occurs on the water side okay. And the condenser tubes are cleaned when they are new okay. The overall heat transfer coefficient is high when it is new but with usage some scale formation takes place scale formation can take place both on water side as well as on refrigerant side. But most of the time the scale formation is mainly on the water side and as a result of the scale formation additional resistance to heat transfer is created and because of this the value of overall heat transfer coefficient decreases.

So the we have to know what is this fouling factor and what is done generally is at the time of design itself even though new tubes cannot have any scale formation we take account of the fouling factor. And use some fouling factor and estimate the overall heat transfer coefficient and Stoecker has suggested the following values for fouling factors. For example the fouling factor is point zero zero zero zero nine meter square Kelvin per watt for fluorocarbon based refrigerant with copper tubes. And the fouling factor is point zero zero zero one seven eight meter square Kelvin per watt for steel tubes with ammonia okay. And periodic descaling is required to control fouling okay. So from time to time you have to shut down the plant and clean the tubes.

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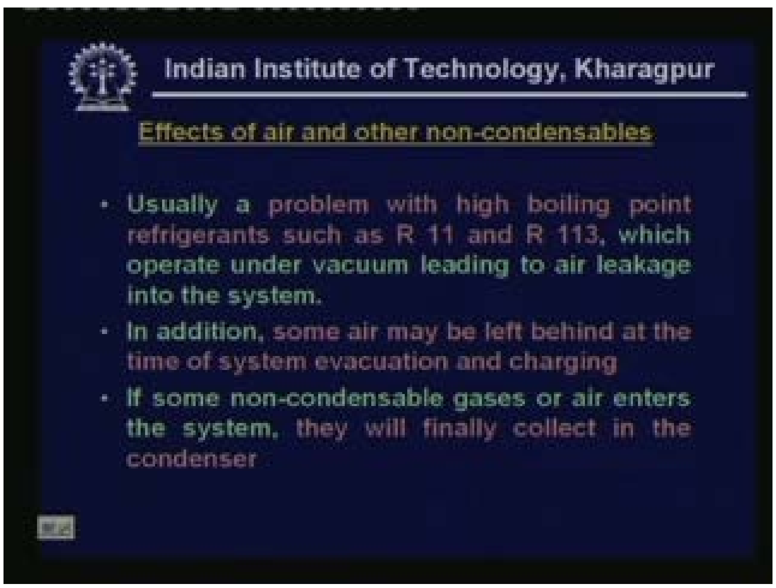

Now let us look at effects of air and other non-condensables this is usually problem with high boiling point refrigerants such as R eleven and R one one three which operate under vacuum leading to air leakage into the system. In addition some air may be left behind the behind at the time of system evacuation and charging okay. So because of these two reasons in addition to refrigerant you will also find some air or other non condensable gases inside the system. And if some non-condensable gases or air enters the system they will finally collect in the condenser.

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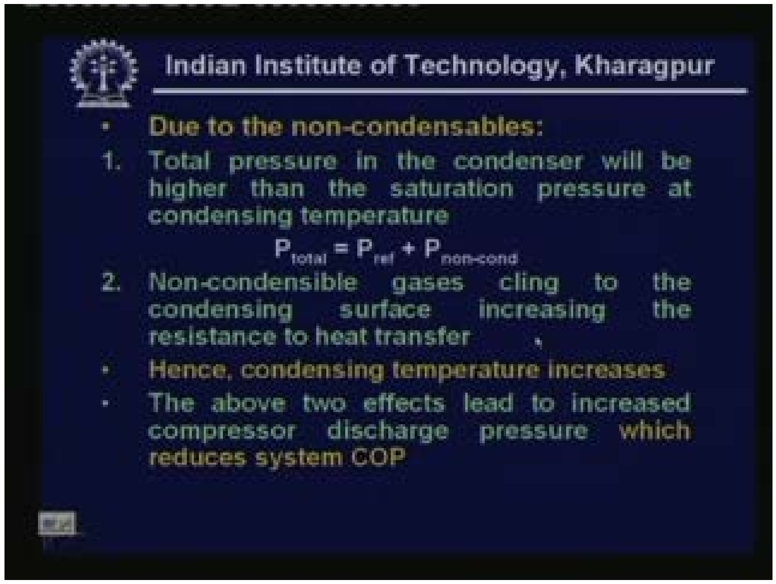

And what is the effect, these non condensable gases which get collected in the condensers. The first effect is that the total pressure in the condenser will be higher than the saturation pressure at condensing temperature. That means the total pressure is equal to P saturated at the condensing temperature that is P ref plus P non condense. That means here in this expression this is the total pressure and this is the partial pressure of the refrigerant. And this is the partial pressure of the non-condensable gases see you will find the total pressure increases total. Pressure increases means the discharge pressure of the compressor has to increase not only this you also have a second effect.

That is the non condensable gases cling to the condensing surface increasing the resistance to heat transfer okay. Hence condensing temperature increases right. So because of these two factors we find that there is an increased compressor discharged pressure. That means whenever there are non condensable gases the condensing temperature increases. And the discharge pressure also increases once the discharge power increases the compressor power input has to increase for the same mass flow rate okay. So ultimately the cop of the system reduces. So we have to make sure of we have to maintain the concentration of the non condensable gases below a certain acceptable level okay. If it increases you may have to stop the plant and purge the plant of all this non condensable gases and air okay.

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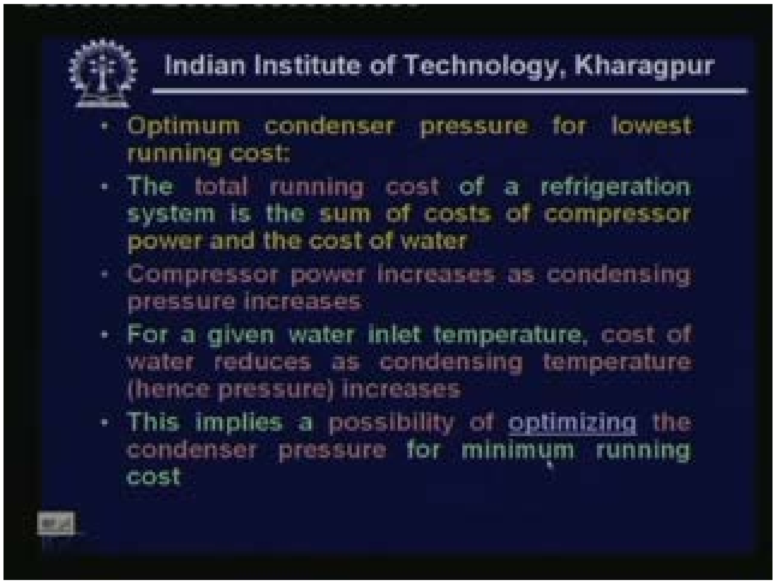

Now let us quickly look at the existence of an optimum condenser pressure for lowest running cost this is for water cooled condenser based systems. For any water cooled condenser based refrigerant system the total running cost of a refrigerant system is the sum of cost of compressor power. And the cost of water and compressor power increases as condensing pressure increases okay. This you have seen in the earlier lectures and for a given water inlet temperature cost of water and the water flow rate etcetera reduces as condensing temperature or condensing pressure increases okay. So this implies a possibility of optimizing the condenser pressure for minimum running cost okay.

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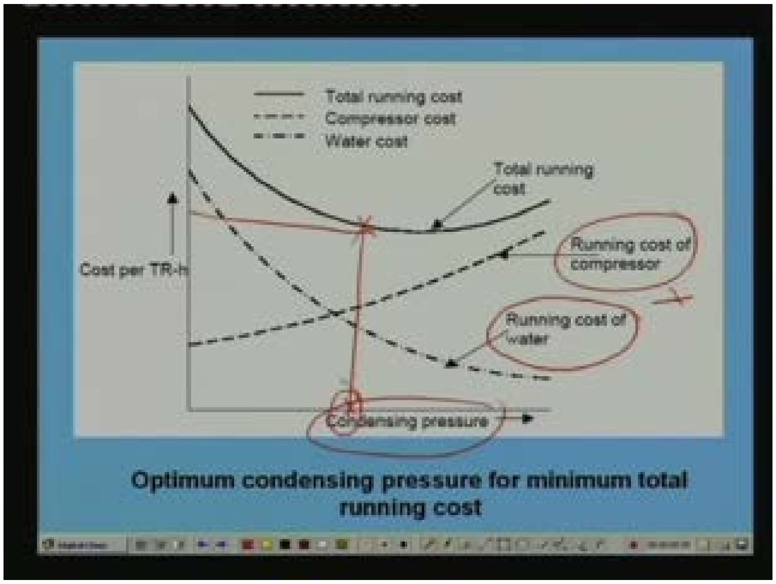

Let me show a typical picture this is for example I have plotted the cost versus condensing pressure okay. So you find that as the condensing pressure increases running cost of compressor increases at the same time running cost of the water reduces in the total cost which is nothing but running cost of compressor plus running cost of the water may show a minimum at a particular condenser pressure okay. So this condenser pressure becomes the optimized optimum condenser pressure for this particular plant okay. So if you know the plant characteristics you can plot this curves and you can find out what is the optimum condenser pressure at which the total cost is minimum okay.

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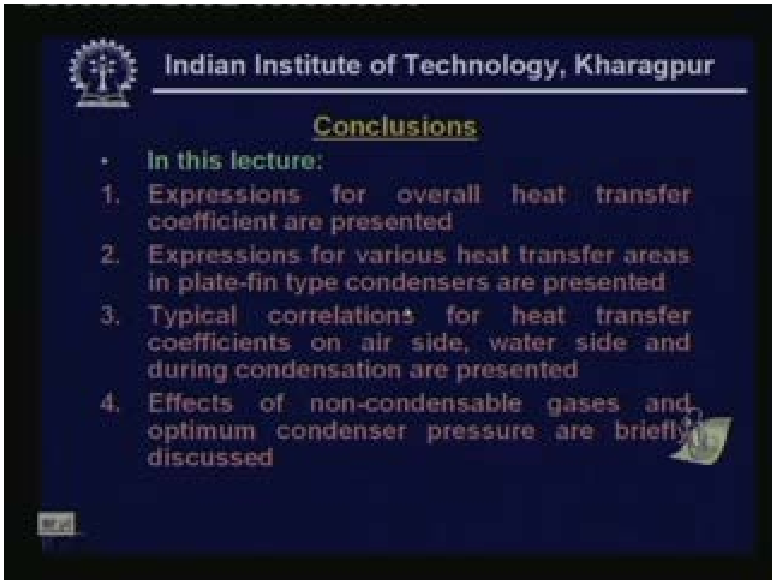

So with this I conclude this lecture in this lecture expressions for overall heat transfer coefficient are presented expressions for various heat transfer areas in plate fin condenser are presented typical correlations for heat transfer coefficients on air side water side. And during condensation are presented and effects of non condensable gases and optimum condenser pressure are briefly discussed. So in the next lecture I will work out problems on the design of air cooled and water cooled condensers thank you.