

**Heat Exchangers: Fundamentals and Design Analysis**  
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**Lecture – 39**  
**Surface Condenser (Contd.)**

Welcome back we were discussing Condensers. So, we have started from the basic and we will we are slowly developing the concepts condensation on a single tube we have seen. Now, we will see on a tube bank how condensation takes place and then, ultimately you we will move to on the design analysis of a surface condenser. That is a very big condenser, which is almost I mean which is similar to a cell and tube heat exchanger, but there are very large number of tubes on which condensation take place, typical condenser which are used in power plant. So, the topic which I start today is design of Surface Condensers.

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Effect of condensate inundation:

Suppose that refrigerant-R134a, is condensing under quiescent conditions on the shell side of a bundle of 41 tubes. The bundle can be configured in a square, inline arrangement or in a triangular, staggered arrangement, as shown in figure. Find the average shell side coefficient for each of the configurations.

Heat Exchangers: Selection, Rating and Thermal Design by Sadik Kakac, Hongtan Liu and Anchasa Pramunjaroenkij, CRC Press, 2012

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So, you will slowly go to surface condenser if you recall we have done we have done one problem were we have taken a single tube and over the tube there is condensation of refrigerant in vapor. So, here we are taking up another problem from the book of heat exchangers, selection rating and thermal design by Sadik Kakac and co authors. Suppose that refrigerant R134 a, is condensing under quiescent condition that is almost stationary vapor on the shell side of a bundle of 41 tubes.

The bundle can be configured in a square inline arrangement which is shown in the left and or in a triangular staggered arrangement which is shown to the right, as shown in the figure. Find the average shell side coefficient for each of the configuration. So, here all the tubes will take part in condensation but, the condensate from the tube above will fall on the tube below and that will thicken the condensate film of the tube below, and which will diminish the rate of heat transfer. And this diminishing of this effect of diminishing the rate of heat transfer is a progressive one because, as we go to the tube below and below there will be more detrimental effect of film storage.

So, these we have to take into consideration and average heat transfer coefficient is then, has to be estimated taking care of all the heat transfer coefficient on the different tubes so, let us proceed.

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In the inline, the arrangement would be approximately equivalent to seven columns of 6 tubes each.

$$h_{m,6} = h_1 (6)^{-1/6} = 1620.8(0.7418) = 1202 \frac{W}{m^2} \cdot K \text{ (Kern relationship)}$$

$$h_{m,6} = h_1 (6)^{-1/4} = 1620.8(0.6389) = 1036 \frac{W}{m^2} \cdot K \text{ (Nusselt relationship)}$$

$$h_{m,6} = h_1 (0.6 + 0.42(6)^{-1/4}) = 1407 \frac{W}{m^2} \cdot K \text{ (Experimental relationship)}$$

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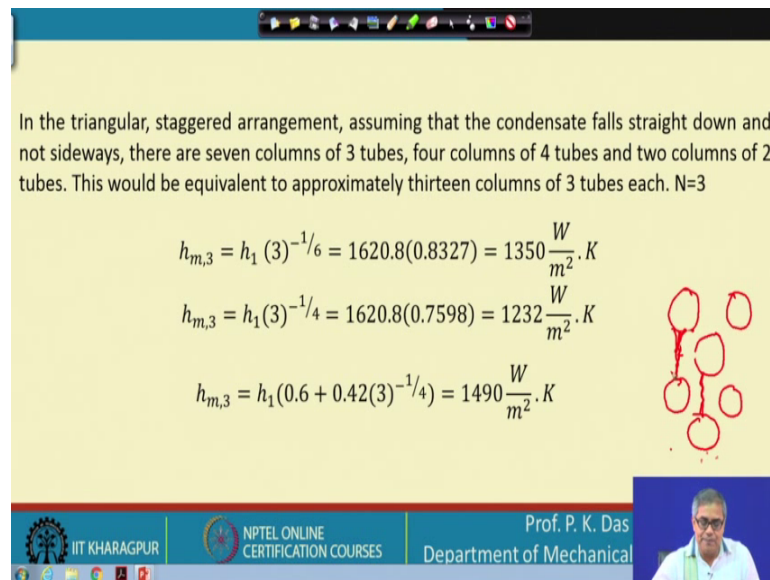
So, initially we have determined the heat transfer coefficient of a single tube and let us say, we have got the tube heat transfer coefficient which is this one, this we have already calculated. Now this is the suppose, there is a row of sorry there is a column of tubes. So, this is the 1st tube, this is the 2nd tube, this is the 3rd tube 4th tube and so on. So, on the 1st tube it is not affected by the condensate from the top so, we will have heat transfer coefficient  $h_1$ , which we have calculated earlier. So, this will give the heat transfer coefficient  $h_1$ , which we have calculated already we have calculated in the previous problem. Now, later on we will have lesser amount of heat transfer coefficient.

Let us see what we have written. In the inline arrangement in the inline the arrangement would be approximately equivalent to 7 columns of 6 tube each. This is approximate because, you see at the cell is of circular cross section, so on all the column the same number tubes will not be there, so we make an approximate. And this approximation for the small problematic loops, that it is little bit unjustified, but when we consider a very large condenser so then, these approximation of considering average number of tubes on in a particular column is not unjustified.

So, this kind of surface heat sorry surface condensers are used in power plant and already I have shown some diagram where very large number of tubes are there in a column. And then, we are using three relationship, these relationship has been described earlier. So, from Kern relationship we get what is the average heat transfer coefficient, where as the heat transfer coefficient of the 1st tube is 1620 and the average heat transfer coefficient is 1202. Then we use Nusselt relationship, nusselt relationship gives around 1000 and then there is some sort of correlation predicted from experiment, so that also we use and we get something around 1407.

So, what we can see the Kern co relationship sorry Kerns correlation or Kerns relationship is in between the other two. So, it is I mean it is many a times it is recommended that, we should use Kern correlation and we will use in the later problem also the Kern correlation or Kern relationship. And kern relationship what we can see that, the average heat transfer coefficient is given by the heat transfer coefficient of the top most tube. Then the number of tubes comes in two places the 6 is the number of tube in a particular column, so with this we get the average heat transfer coefficient.

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In the triangular, staggered arrangement, assuming that the condensate falls straight down and not sideways, there are seven columns of 3 tubes, four columns of 4 tubes and two columns of 2 tubes. This would be equivalent to approximately thirteen columns of 3 tubes each.  $N=3$

$$h_{m,3} = h_1 (3)^{-1/6} = 1620.8(0.8327) = 1350 \frac{W}{m^2} \cdot K$$
$$h_{m,3} = h_1 (3)^{-1/4} = 1620.8(0.7598) = 1232 \frac{W}{m^2} \cdot K$$
$$h_{m,3} = h_1 (0.6 + 0.42(3)^{-1/4}) = 1490 \frac{W}{m^2} \cdot K$$

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So, this is for inline arrangement, let us go to the staggered arrangement. Assuming the condensate falls straight down and not sideways, because you see this staggered arrangement is something like this staggered arrangement we can get, so this is the staggered arrangement. So, what we assume that let us say this is the staggered arrangement so; condensate from the top tube falls on the tube below. So, it does not fall from here to here so, this is not, I mean this does not fall from the tube on the side. So, rather it falls directly from the top tube to the bottom tube, so this is the way it falls so if we assume that.

Then there are 7 column of 3 tubes, 4 columns of 4 tubes and 2 columns of 2 tubes. So, this would be equivalent to approximately 13 columns of 3 tubes each with  $N$  is equal to 3. So, here we have made an assumption that, some sort of approximation we could have calculated separately, also the changes will be not be much, but in design calculation, there are many assumptions and this kind of assumption we have made that there are 7 columns and approximately sorry there are 13 columns and approximately 3 tubes in each column.

So, let me go back to the previous figure then, again let me go back to the previous figure it will be so, here I would like you to do this calculation of your own. So, we can see this is the staggered, this is the staggered arrangement. So, just see what we have told that there are 13 columns and approximately these many tubes are there so, whether you

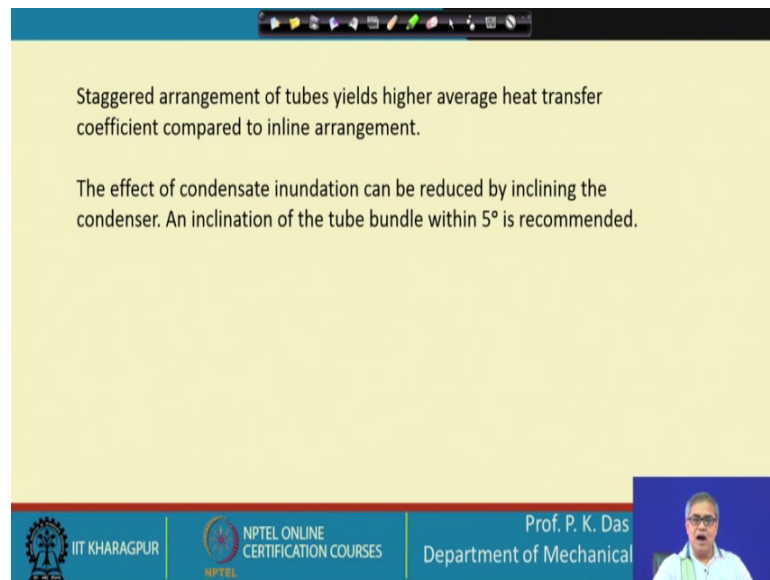
can get from the figure out from this particular diagram or not. So, this is what I like you to do this exercise.

Then there is mathematical calculation, very simple mathematical calculation. And just like before what we will get the Nusselt correlation will give the most conservative value of average heat transfer coefficient that is the lowest value. Kern correlation or Kern relationship will be in between and the experimentally derived relationship, which has been shown lastly that will give the maximum heat transfer coefficient. So, now it is up to the designer that, what he will select. So, with this let us go the next slide, this is the end of the problem, but at the end of the problem, there is certain thing to be noted.

What we find that when we have arranged the tubes in the staggered arrangement what we get? We get a higher value of average heat transfer coefficient for the tube side condensation so, this is one thing to be noted. So, many places in practice we will see that, staggered layout of tubes are preferred compare to the inline layout of the tubes; one of the reason we can see here particularly for condensation.

And staggered layout generally, in case of single phase flow, it gives a better coefficient of heat transfer incase of single phase flow also. So, many cases if there are no other reason people will prefer to have staggered arrangement. Staggered arrangement gives a compact design on a small design you can on a small volume you can pack up more number of tubes that is first thing. It gives higher rate of heat transfer, that is another thing and in case of condensation, what we can find that it gives lesser inundation of the condensate film. So, with this message let us move to next slide.

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Staggered arrangement of tubes yields higher average heat transfer coefficient compared to inline arrangement.

The effect of condensate inundation can be reduced by inclining the condenser. An inclination of the tube bundle within  $5^\circ$  is recommended.

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Staggered arrangement of tubes yields higher average heat transfer coefficient compared to inline arrangement, so this is the first thing. The effect of condensate inundation can be reduced this is another way. So, what we can see the average heat transfer coefficient on surface condensation is getting reduced because of inundation of the condensate film. One way of course, is to have staggered arrangement to reduce this effect to some extent, but in if we go for higher reduction then what we can do.

We have thought of horizontal layout of the tube bundle, we can go for laying out the tube bundle in a slightly inclined way. So, in that way it will help into free film drainage because, gravity force is there. So, it will thin down the liquid fill and at the same time the effect of or the detrimental effect of condensate inundation will be reduced.

So, generally some inclination we cannot go for very large inclination that will have other effect on the design of the condenser. So, generally, within 5 degree we try to limit the inclination of the tube to get some sort of a relief from the effect of inundation. So, this problem then gradually what we have seen how to calculate film condensation on a single tube and how to calculate the film condensation if there is a tube bank and there are tubes in a column large number of tubes in a column.

Another thing I like to say that, in all this cases we have assume the nusselt type film condensation where, there is no velocity of the vapor. This is true in many cases in surface condenser particularly for power plant etcetera we will find that the vapor

velocity is really very small. But where the vapor velocity is there, it will have some effect on the film and obviously, it will reduce the sorry it will modify it will change heat transfer coefficient. Now sometimes we will have a small discussion on this.

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An example of a shell and tube surface condenser:

Problem: A shell-and-tube type condenser is to be designed for a coal fired power station of 200 MWe. Steam enters the turbine at 5 MPa and 400°C ( $i_1 = 3195.7$  kJ/kg). The condenser Pressure is 10 kPa (0.1 bar). The thermodynamic efficiency of the turbine  $\eta_t = 0.85$ . The actual enthalpy of steam entering the condenser at 0.1 bar is calculated to be  $i_2 = 2268.4$  kJ/kg with 80% quality. The condenser is to be designed without subcooling. A single tube pass is used and the cooling water velocity is assumed to be 2 m/s. Cooling water is available at 20°C and can exit the condenser at 30°C. Allowable total pressure drop on tube side is 35 kPa. The tube wall thermal conductivity  $k = 111$  W/m·K.

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Solution:

$$m_s = \frac{W_t}{\Delta i} = \frac{200 \times 10^3}{3195.7 - 2268.4} = 215.68 \text{ kg/s}$$

Flow rate of steam

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Now, let us move to the next topic. With our knowledge of single tube condensation over a single tube, condensation over a tube bank now, we move to a condenser where we are, where the problem is more rigorous and where the problem is more involved. So, this is again taken from the book of heat exchanger book called heat exchangers selection rating and thermal design by Sadik Kakac and his co authors. And the problem bits like this, a shell tube type condenser is to be designed for a coal fired power station of 200 mega watt electricity. .

Steam enters the turbine at 5 mega pascal and 400 degree Celsius,  $i_1$  is denoted as  $i_1$  is denoted for or  $i_1$  is used as a symbol for enthalpy. So,  $i_1$  the enthalpy value has been given, which can readily be picked up from a steam table. And the condenser pressure is 10 kilo pascal, so 0.1 bar so, steam enters the turbine and comes out at 0.1 bar, the thermodynamic efficiency of the turbine is 0.85, this is isentropic efficiency.

The actual enthalpy of steam entering the condenser at 0.1 bar is calculated like this. So, isenthalpic sorry isentropic efficiency is given so that, we know the actual enthalpy of this steam when it is entering the turbine. So, that we have got and then the condensation is to be designed without subcooling.

So, in the condenser only condensation take place and we assume that, the liquid is coming out as saturated, the condensate is coming out as saturated liquid. A single tube pass is used and the cooling water velocity is assume to be 2 meter per second. Though in the surface condenser used for steam power plant, what we do is that steam is condensed on outside the tubes and through the tube, we circulate cooling water, coolant is water only.

So, this I have told and again I am repeating because, people should have good idea regarding at least the basic heat exchangers very important heat exchangers for our industry. Cooling water is available at 20 degree Celsius and can exit the condenser at 30 degree Celsius.

So, cooling water there will be a change in temperature, but for steam there will not be a any change in temperature because, sub cooling is not consider over here. So, only thing is that this steam quality will change from inlet to the heat exchanger to the from the inlet to the heat exchanger to the outlet and at the outlet the x value or dryness fraction will be equal to 0. Allowable total pressure drop on tube side is 35 kilo pascal the tube wall thermal conductivity is given, what is the thermal conductivity. So, first what we have to do, we have to calculate the mass flow rate of the streams.

So, first let us calculate the mass flow, mass flow rate of the vapor side, shell side or steam, the condensing steam. So, how do we find out, we know the total amount of watt, that has to be extracted from the that can be extracted from the turbine and we know it is a 200 mega watt power plant. So, 200 mega watt will be the work done and from there enthalphy we know, enthalphy of the steam entering the turbine, enthalphy of steam leaving the turbine we know and from there we can get the steam flow rate that is 215.8 kg per second.

So, this is the steam flow rate we are getting, so basically, this one the flow rate of steam. So, this is what we have got and this is a very important, very basic information regarding the heat exchanger. So, now let us move to the next slide.



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Cooling water properties at a mean temperature of 25°C	Saturated liquid properties of condensed water at 10 kPa, $T_s=45.8^\circ\text{C}$
$c_{ps} = 4,180 \text{ J/kg}\cdot\text{K}$	$i_g = 2392 \text{ kJ/kg}$
$\mu_c = 0.00098 \text{ kg/m}\cdot\text{s}$	$\mu_l = 0.000588 \text{ N}\cdot\text{s/m}^2$
$k_c = 0.602 \text{ W/m}\cdot\text{K}$	$k_l = 0.635 \text{ W/m}\cdot\text{K}$
$\rho_c = 997 \text{ kg/m}^3$	$\rho_l = 990 \text{ kg/m}^3$
$Pr_c = 6.96$	$i_l = 191.8 \text{ kJ/kg}$
	Outside diameter of the tubes, $d_o = 0.0254 \text{ m}$
	Inside diameter of the tubes, $d_i = 0.02291 \text{ m}$
	Tube wall thermal conductivity, $k = 111 \text{ W/m}\cdot\text{K}$

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Next slide let us just have a look into the properties. Cooling water properties at mean temperature of 25 degree Celsius; you see the cooling water temperature changes at inlet it is 20 degree Celsius, at outlet it is 30 degree Celsius. So, the mean temperature will be 25 degree Celsius at this point probably we have mention it and we will mention number of times that, in heat exchanger as there is a change in temperature there could be change in property.

So, due to this change in property, the heat exchanger calculation ideally becomes non-linear and which is difficult to handle. So, in many cases the non-linearity is so high that we have to handle it accurately and probably, we have to go for numerical solution. In there are very large number of cases industrial example, where the non-linearity is not that high and we cannot afford going for fully numerical solution, so what we do we go for some sort of averaging.

So, here also we have average the liquid temperature, coolant side temperature 25 degree Celsius, so we have got the specific heat, we have got the viscosity, we have got the thermal conductivity and then, we have got the density of liquid water and ultimately, we have got the prandtl number of the liquid water.

Similarly, saturated liquid properties of the condensed water on the shell side, we have got the latent heat of vaporization, the latent heat of vaporization we have got sorry, I have to go back to the previous slide. So, we have got latent heat of vaporization, we

have got liquid viscosity, liquid thermal conductivity, liquid density and we have got when the liquid is in the saturated condition what could be the enthalphy of the liquid. So, all these things we have got.

Outside tube diameter and inside tube diameter we have given. Now these two things of course, one can think of calculating but, particularly the inside tube diameter one can think of calculating but sometimes it is selected. In a practical heat transfer equipment design or heat exchanger design, many of the things are to be selected a priory when you start the design.

Sometimes this selected quantity do not change, sometimes this selected quantity we change, depending upon our preliminary result. So, here we are we had considered the inside and outside diameter of the tube sometimes it happens like this that, these are the tubes which are generally manufactured for this kind of heat exchanger so, we have got limited selection.

And sometimes people use or the designer use is intuition and experience for selecting certain parameter. Keeping everything floating, designing of a heat exchanger it becomes almost impossible if not a herculean task, so certain thing we have to assume or take and that this things are given. So, now, let us go back to the next slide.

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Cooling water velocity  $u_c = 2\text{m/s}$   
Cooling water velocity is selected from the consideration of fouling and erosion.

Low velocity gives higher fouling.  
High velocity gives higher erosion.

Fouling resistances have been picked up from TEMA.

$$R_{fi} = 0.00018 \text{ m}^2 \cdot \text{K} / \text{W}$$

$$R_{fo} = 0.00009 \text{ m}^2 \cdot \text{K} / \text{W}$$

$$Q = m_s (i_{in} - i_l)$$

$$= 447.9 \text{ MW}$$

$$m_c = \frac{Q}{(T_c - T_{c1}) c_{pc}}$$

$$= 10,717.4 \text{ kg/s}$$

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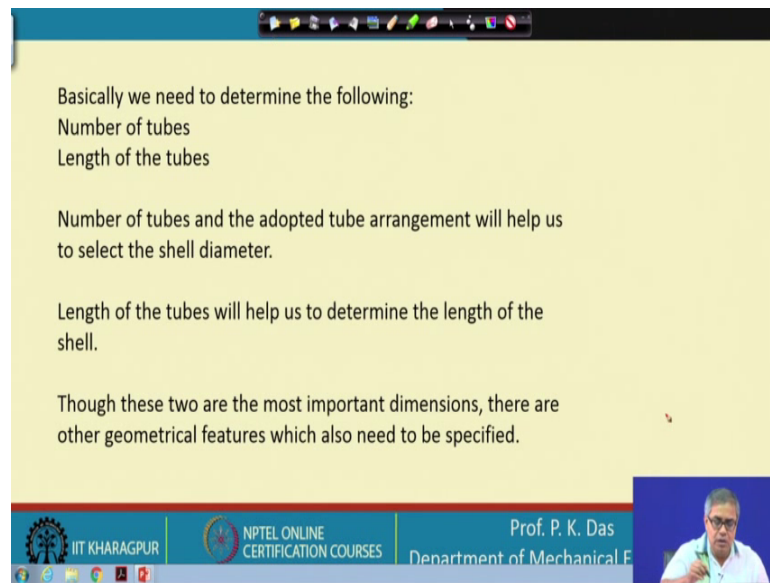
So, next slide the cooling water velocity has been selected as 2 meter per second, this is a problem statement. Now cooling water velocity is selected depending on two consideration, if we have got very low velocity then we will have tendency for fouling, if we have got high velocity then we will have erosion lesser tube life and problem of tube leakage etcetera that may be there. And again higher velocity will give higher pressure drop generally and higher pumping power.

So, based on all these considerations, we have to select the velocity. And again I have told that the designer is an experienced one or he has got good knowledge or he has got accessed to hand books etcetera codes and standards so from air velocity can be picked up. Selection of the velocity is important because, the inside heat transfer coefficient that is also dependent on velocity so, let us work with a velocity of 2 meter per second. Then fouling resistance for a practical heat exchanger design one has to consider fouling. And particularly, an application like this where you have to use water, water is a very good reactant with many kind of metallic surfaces.

So, that is one thing, another thing is that there is sufficient amount of temperature difference so, heating of the tube is there on one side, cooling of the tube is there on the other side. So, there will be the tubes will be prone to fouling on both sides; inner side and outer side. So, similarly we have to calculate or estimate pickup the fouling resistance. So, inside and outside fouling resistances we have picked up and here we used TEMA, TEMA I have mentioned earlier that is it is TEMA is a code gives many codes and standard for heat exchanger. It is tubular exchanger manufacturers association.

So, from there, we have picked up and then from the data etcetera which have been supplied to you, so we get what is the total amount of heat transfer and we get what is the coolant flow rate. So, these two are again very important information, which we get from this calculation, preliminary calculation, simple calculation. With these let us go back to I mean let us proceed to the next slide.

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Basically we need to determine the following:

- Number of tubes
- Length of the tubes

Number of tubes and the adopted tube arrangement will help us to select the shell diameter.

Length of the tubes will help us to determine the length of the shell.

Though these two are the most important dimensions, there are other geometrical features which also need to be specified.

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Basically, we need to determine the following now. The number of tubes, length of the tubes number of tubes and the adopted tube arrangement will help us to select the shell diameter, we will discuss about this later on. So, initially we want to know that how many tubes we need to have to have this heat transfer rate of this heat transfer has been given to us. Tube through the tube liquid velocity we have got, liquid flow rate we have got, so number of tube we have to find out.

And then, length of the tube that is very important, so length of the tube we have to find out. So, when we get number of tubes and length of the tube then gradually, we are moving towards the entire size of the heat exchanger, entire size of the shell and tube heat exchanger. Though these two are the most important dimension, there are other geometrical features which also need to be specified.

So, this is also one thing we will gradually see what are the other features we need to specify for heat exchanger, for a complete design of heat exchanger. At this point I like to mention that, obviously mechanical design we are not doing. In mechanical design of heat exchanger, there are many details like what should be the shell thickness etcetera, why we have calculated this tube thickness, whether really this tube thickness are good from mechanical design point of view or not. But the mechanical design is not within the your view of present course.

So, we will concentrate basically to thermo hydraulic design, more precisely the methods of analysis to arrive at a thermo hydraulic design. So, this is what we are going to do and we are proceeding slowly towards that.

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Calculation of number of tubes:

$$m_c = u_c \rho_c \left( \frac{\pi d_i^2}{4} \right) N_T$$

$$N_T = \frac{4 m_c}{u_c \rho_c \pi d_i^2}$$

$$= 13,038 \text{ tubes}$$

For calculating the condensing side heat transfer coefficient an estimate of the average number of tube rows in a vertical column is needed. Considering typical designs the number is taken as 70.

The coolant heat transfer coefficient  $h$  can be estimated from in-tube heat transfer correlations.

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The number of tubes once we have got the coolant flow rate, so the this equation will give us the number of tubes ok. So, basically the number of tubes we can calculate like this. For calculating the condensing side heat transfer coefficient and estimate of average number of tube rows on a particle column need is needed. And there again comes some sort of a calculation procedure that, now we have got total number of tubes. The total number of tubes is quiet high 13000 tubes around 13000 tubes.

Now, how the tubes will be arranged. So, there will be rows of tubes, there will be columns of tubes. Columns of tubes in a column how many tubes will be there, that is very important. So, because the heat transfer coefficient due to inundation of condensate depends on how many tubes are there on a column this we have explained in a very in a in a great detail we have explained. So, here based on some consideration again we are considering that there will be 70 tubes like this.

So, like this there will 70 tubes so, this is 70, this is first tube so, on a particular column there will be 70 tubes, so these we are considering. And the coolant heat transfer coefficient  $h$  can be estimated from in tube correlation. So, outside heat transfer coefficient we have to now determine, inside heat transfer coefficient we have to

determine. And how we have to determine we will see and you can see that, we are fixing certain numbers, we are calculating certain numbers, so ultimately, this heat transfer coefficient can calculate. Let us go back lets go to the next slide, so, I think we have come to a an end of this lecture and next lecture we will carry from this point.

Thank you.