Principles and Practices of Process Equipment and Plant Design

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Module - 03
Lecture - 47
Shell and Tube Heat Exchanger- Design

Hello. Now, we will discuss Shell and Tube Heat Exchangers. So, by this time we have already designed the double pipe exchanger. So, you know more or less of what are the basic steps that should be followed when you are going to design any heat exchanger. The comprehensive introduction of the shell and tube exchanger, the different parts what are the specialities as per the TEMA and the BIS code what are the classifications everything has been told to you in great details.

So, in that way your work has become much simpler now. You know more or less what is the general procedure of designing any particular heat exchanger and more or less double pipe heat exchanger and shell and tube heat exchanger - they are almost the same procedure. The only thing is just because there are several other components, there is a provision of multi passing etc. Certain things have to be taken into account.

So, certain additional design parameters will be coming into consideration. Otherwise regarding the procedure, it is completely the same. You have been already known about the shell and tube heat exchanger and the input data sheet as well as the specification sheet everything has been shown to you. So, what I plan to do is I will take up an actual problem and then we start solving it step by step and see that how each parameters can be found out. Lot of things you will find are common to the double pipe heat exchangers. The additional factors will be discussing as we proceed in this case.

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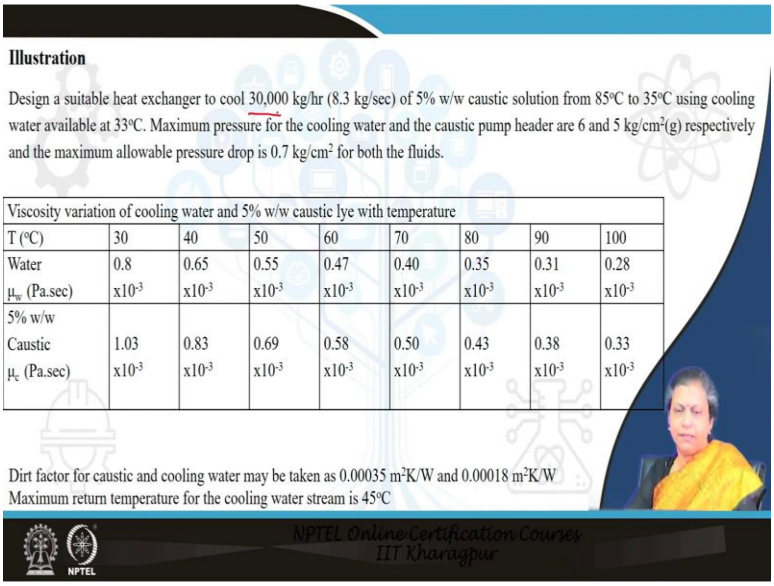

Now the problem that I have selected it is the same problem that had rather the same tool, liquids everything I have kept same the temperatures also I have kept same.

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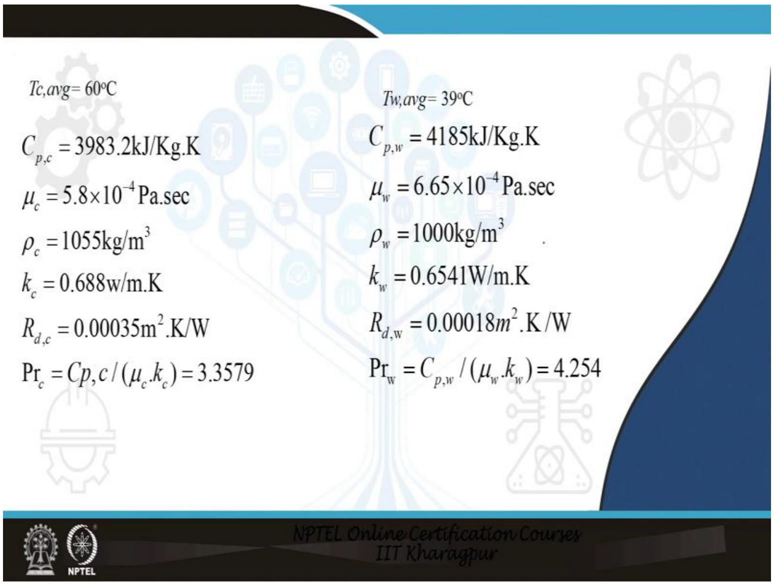

So, that you can use the properties which had been specified in the last class or rather these properties which were specified in the double pipe heat exchangers. What I have just done is that since this is we plan to run this operation in a shell and tube exchanger I have simply increased the flow rate of the caustic solution which is to be cooled from 85̊C to 35̊C.

Now in this case I would just mention one thing that when we were discussing the double pipe exchanger the problem which I had given you in the beginning I think it was about 1500 kg per hour or something. And later on at the end finish of the problem it was 2500 kg per hour; we are going to work out the problem with 1500 kg per hour itself.

In fact, you can also work with 2500 kg/hr maybe you will get the area slightly less, but I believe that it will be it will lie within 50m square area so, that a double pipe exchanger can be used. So, in this case I have just increased the amount which has to be cooled. So, accordingly you are going to do the design, everything else is as specified.

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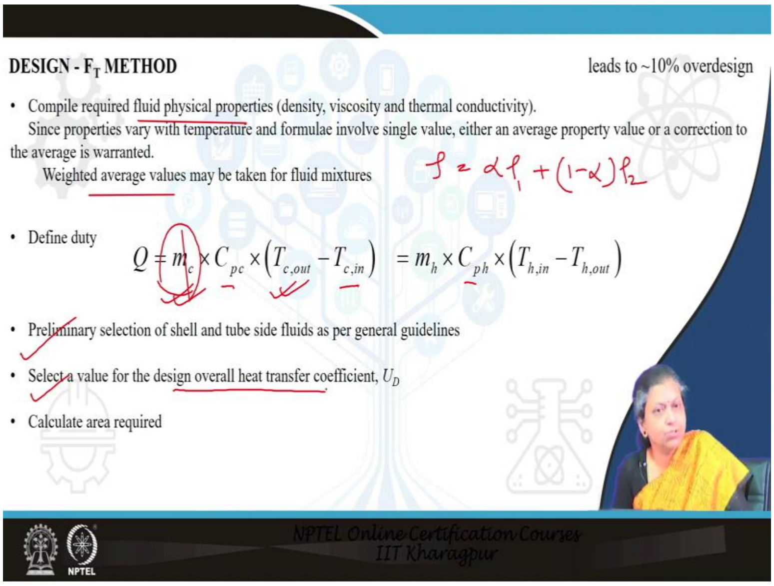

So, therefore, more or less you can use these particular things and we start designing. Now the designed part which I am going to discuss this is commonly known as an F T method which is available in Kern. Let me tell you there are different ways by which we can design as a Delaware method there is a stream analysis method, more or less we will be just discussing the F T method we do not have time to discuss all the other techniques.

And just for your information this leads to about 10% over design which is quite acceptable. So, therefore, the initial part it same as double pipe it was also mentioned just in the last class. So, therefore, you can do these things the physical properties they are already available you can take it up you just remember that since the physical properties vary with temperature and you will find that all the formulae you will find that they involve single values.

So, therefore, either you have to take an average value or you have to make a correction to the average value. Just like for C p for the density etc., you take the average value for viscosity. there is the state correction where the correction has to be made if you find that the viscosity of the bulk fluid and the viscosity at the wall temperature are quite different. Under that condition it happens only when viscosity is very sensitive to temperature or in other words when there is a very steep temperature profiled in the shell or the tube side.

There is one more thing which you do not need for your particular problem, but just keep in mind that it can happen that we are dealing with fluid mixtures. It can be a vapour liquid mixtures, it can also be a mixture maybe of two liquids and when such a situation arises quite naturally you have to take the weighted average values.

For example, the row of the mixture is going to be the volume fraction of component 1 into say the density of component 1 plus the volume fraction of component 2 plus the density of component 2. So, in this way weighted average values can be taken. And you will remember that when we were discussing the liquid extraction that time while discussing the average properties I told you that there are several expressions to evaluate the average viscosity.

Because the finding out of suitable average viscosity is difficult you can have mass average in terms of mass fraction you can have average in terms of volume fraction all those were provided when we were discussing liquid liquid extraction you can very well refer to those and find out the average viscosity values in that way.

So, once you are ready with the input data then define the duty you know it very well in this particular problem after defining the duty what we are going to do we are going to find out the total mass of the cooling water which is required. And other things were all already known to you the T c in was to has been provided and as I have told you in Indian Standards more or less the input temperature is 33°C.

And the output temperature we allow 12°C rise in temperature; roughly it goes from 45 to 47°C you can take. So, therefore, accordingly T c out you can fix up and then you can find out m c. Once you have done that then the next thing is the you have to select the shell side and the tube side fluid. Now, these guidelines were also discussed while we were discussing the double pipe exchanger.

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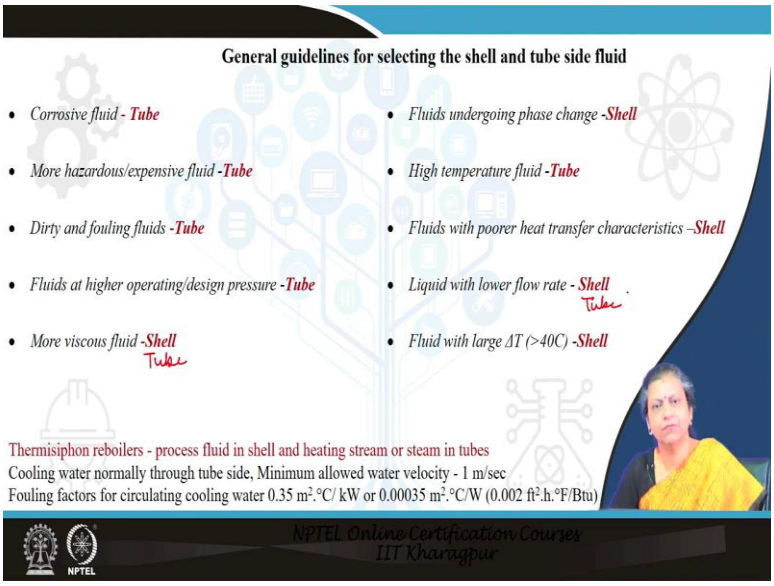

I have just put up these guidelines here once more simply for your references. In this respect there is one thing which I would like to tell you. When I was mentioning about double pipe exchanger; in fact, this particular portion it was common to both shell and tube and double pipe exchangers. I had mentioned one statement that the critical Reynolds number for the shell side fluid is 200 critical Reynolds number for the tube side fluid is 2100.

So, therefore, when we find that for a high viscosity liquid we put it in the shell side. So, that turbulent flow can happen if suppose then also we find that it viscosity such that even after introducing it in the shell side fluid also it is exhibiting laminar flow then in that case we would like to put it in the tube side fluid, but in this particular case this is specific to shell and tube exchangers.

For double pipe exchangers the critical Reynolds number it is same for both that outer fluid as well as the inner more or less the same for the inner fluid. So, specifically this is applicable for shell and tube heat exchangers due to the presence of baffles. So, therefore, there is additional mixing and the this induces additional turbulence that is why we consider that in the shell side fluid the critical Reynolds number it is around 2 200 and accordingly this particular this particular thing was said, but this is specific to the shell and tube heat exchanger itself.

So, therefore, it is definitely shell, but again just I had mentioned if even after you calculate the Reynolds number for the shell assuming it to be the shell side fluid and then also you find that it is laminar then definitely we are going to introduce it into the tube side. The same thing is applicable for the liquid with a lower flow rate as well.

And the certain other things just to mention thermisiphon reboilers mostly the process fluid is in the shell side and the heating stream or the steam they are in the tube sides; it is quite natural the you know the logic behind it. Since the most common use or rather a very common use for shell and tube heat exchangers are in thermisiphon reboilers where normally steam or any heating oil is used as the heating medium.

And for the case of coolers mostly we use cooling water and the cooling water you know why it is introduced through the tube side because of its high fouling tendency that you already know and just to reduce the fouling tendency the minimum allowed water velocity is usually specified the fouling factors are there if suppose the condition is such that we cannot use water for the cooling; frequently then we have to go for air cooled exchangers.

When we go for air cooled exchangers it was already mentioned to you that since air has a very low heat transfer coefficient we would all air cooled exchangers they are their fin type exchangers regarding fins it has already been discussed to you. So, therefore, the first thing what you will be doing? Preliminary selection of shell and tube side fluid as per guidelines. Next what is the case? Next you say you select design overall heat transfer coefficient.

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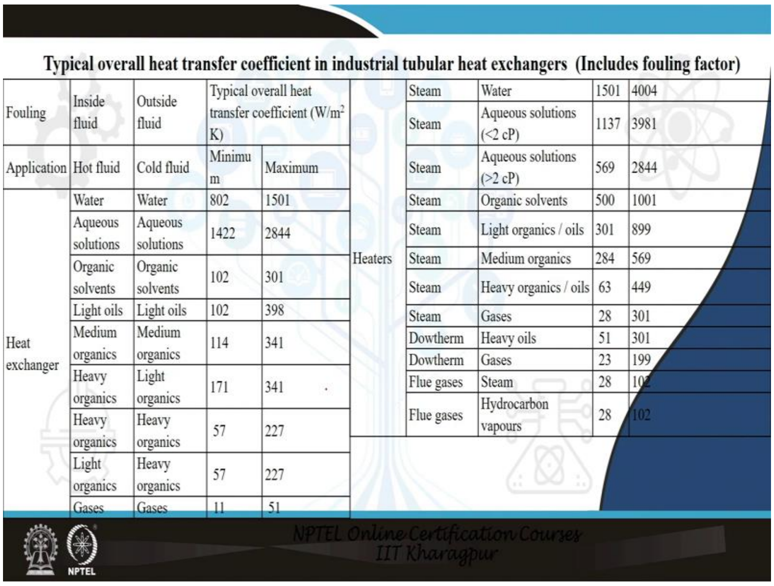

Typical overall heat transfer coefficient in industrial tubular heat exchangers (Includes fouling factor)

Fouling	Inside fluid	Outside fluid	Typical overall heat transfer coefficient (W/m^2 K)	
Application	Hot fluid	Cold fluid	Minimum	Maximum
Heat exchanger	Water	Water	802	1501
	Aqueous solutions	Aqueous solutions	1422	2844
	Organic solvents	Organic solvents	102	301
	Light oils	Light oils	102	398
	Medium organics	Medium organics	114	341
	Heavy organics	Light organics	171	341
	Heavy organics	Heavy organics	57	227
	Light organics	Heavy organics	57	227
	Gases	Gases	11	51
Heaters	Steam	Water	1501	4004
	Steam	Aqueous solutions (<2 cP)	1137	3981
	Steam	Aqueous solutions (>2 cP)	569	2844
	Steam	Organic solvents	500	1001
	Steam	Light organics / oils	301	899
	Steam	Medium organics	284	569
	Steam	Heavy organics / oils	63	449
	Steam	Gases	28	301
	Dowtherm	Heavy oils	51	301
	Dowtherm	Gases	23	199
	Flue gases	Steam	28	102
	Flue gases	Hydrocarbon vapours	28	102

How do you select this? For this particular case also the table had already shown you rather we had gone through the stable when we had discussed the shell and tube sorry the double pipe heat exchanger. And let me tell you that when you design a double pipe heat exchanger and that time you need not refer to this table. At least for the problem that you had have been given and the methodology that I have discussed in that methodology you do not need to assume a heat transfer coefficient earlier.

On the contrary what we do? We assume or rather we select some dimension of the inner tube as well as the outer tube we go through the design and we find out the heat transfer coefficient for the shell for the outer fluid and the inner fluid. Find out the overall heat transfer coefficient after that you can just check whether it complies with this particular table or not.

But for the shell and tube heat exchanger usually we start with assuming and then we come back later and find out that after we have checked whether the heat transfer coefficient that we have considered that is less than what you have assumed or that is greater than what you have assumed. Suppose, the heat transfer coefficient that you get at the end that is less; that means, you had already designed with a higher cross sectional area.

Little amount of over design it is permissible, but over design its preferable that it is within 10 to 15 % and not more.

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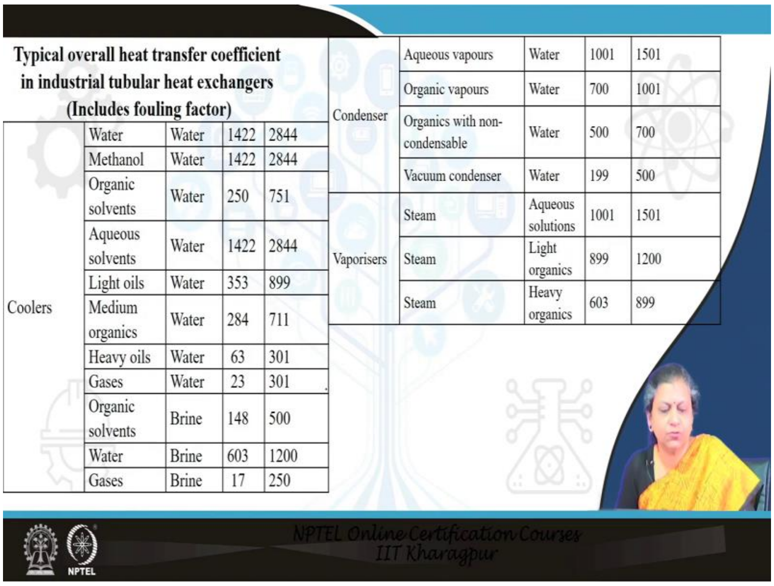

Typical overall heat transfer coefficient in industrial tubular heat exchangers (Includes fouling factor)

Coolers	Water	Water	1422	2844
	Methanol	Water	1422	2844
	Organic solvents	Water	250	751
	Aqueous solvents	Water	1422	2844
	Light oils	Water	353	899
	Medium organics	Water	284	711
	Heavy oils	Water	63	301
	Gases	Water	23	301
	Organic solvents	Brine	148	500
	Water	Brine	603	1200
	Gases	Brine	17	250
Condenser	Aqueous vapours	Water	1001	1501
	Organic vapours	Water	700	1001
	Organics with non-condensable	Water	500	700
	Vacuum condenser	Water	199	500
Vaporisers	Steam	Aqueous solutions	1001	1501
	Steam	Light organics	899	1200
	Steam	Heavy organics	603	899

So, therefore, these are the two tables depending upon the application you can take up the value of your this one and the overall design heat transfer coefficient and these values they have typically they also include the fouling factors.

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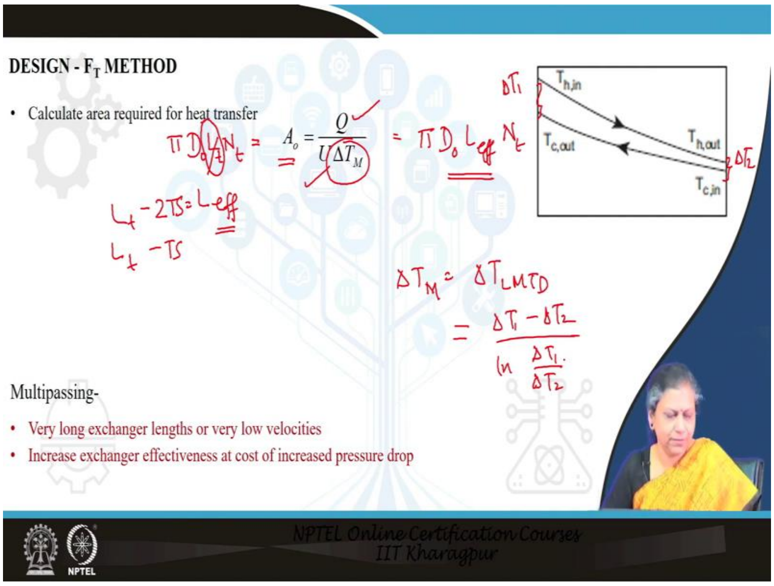

So, that you can assume them and then go ahead to calculate the heat the overall area of required for heat transfer. Where does heat transfer take place? It takes place on the across the tubes. So, therefore, it takes place across the tube wall. So, therefore, just like double pipe heat exchanger in this case also the outer surface of the tubes; they provide the heat transfer area.

So, you know Q you have assumed U once you can calculate the delta T effective delta T, you are in a position to find out the heat transfer area which gives you the total area of the outer surface of all the tubes that are there inside the heat exchanger. So, therefore, this will be equal to pi the outer diameter of all the tubes into the total length of all the; of all the tubes that have been there or for a single tube it is going to be L t into N t the number of tubes.

Now in this particular case also we have to remember that the entire tube length is not involved in heat transfer. So, therefore, instead of L t we replace it with L effective. The effective length of heat the effective length of the tube which is involved in heat transfer by this stand you know that the tubes they are fitted across the two tube sheets. So, therefore, whatever is there inside the tube sheets will not be available for heat transfer.

So, therefore, if it is a fixed or a floating head type naturally from this total length what we do? We will be subtracting the thickness of the two tube sheets and we are going to get the L effective which will be which is the length that we are going to get from this particular expression definitely it is if it is a U tube exchanger there is one heat there is one tube sheet. So, therefore, it is going to be L t minus T S in that particular case.

But again you have to remember something which also has to be considered when we decide on the effective length. Because it is not just the length which is covered inside the tube sheet or which is there inside the tube sheet, but there are also certain other areas which are not available for heat transfer. I leave it to you to think about it in a while I we will be discussing a little about the L effective.

So, therefore, once you have known everything you are in a position to find out A 0 once you can find out A 0, then you have to decide I will just write it down once more this is pi D 0 L effective into number of tubes. So, therefore, then we make some adjustment among these parameters and try to come up with the tube layout. Now in this particular equation if you find that in order to find out A 0 quite naturally just like double pipe heat exchanger.

You have to find out the delta T now in double pipe heat exchanger normally we have counter current flow I have told you when we have co current flow that has also been discussed. So, when we have counter current flow what is the what is delta T M in that particular case this is from derivation you know that this is the log mean temperature difference. What is the log mean temperature difference? It is delta T 1 minus delta T 2 by L n delta T 1 by delta T 2 this has been derived for assuming U and A to be constant that has also been told to you where delta T 1 is this, delta T 2 is this.

So, therefore, accordingly you can do it is not a big problem. In shell and tube heat exchangers when the introduction or the description of the shell and tube heat exchangers were being provided to you then you have come across something known as the multi passing especially for the tubes we very often go for multiple passes. Just to avoid very long exchanger lengths and also if the velocity is very low.

So, for those cases we go for multi passing now. Remember one thing while multi passing definitely it increases the exchanger effectiveness, but quite naturally if you are going for longer tubes it is going to entailer higher pressure drop. But multi passing is something which we have to consider when we are going for a shell and tube exchanger. In fact, this is one thing which distinguishes shell and tube exchanger from double pipe exchanger.

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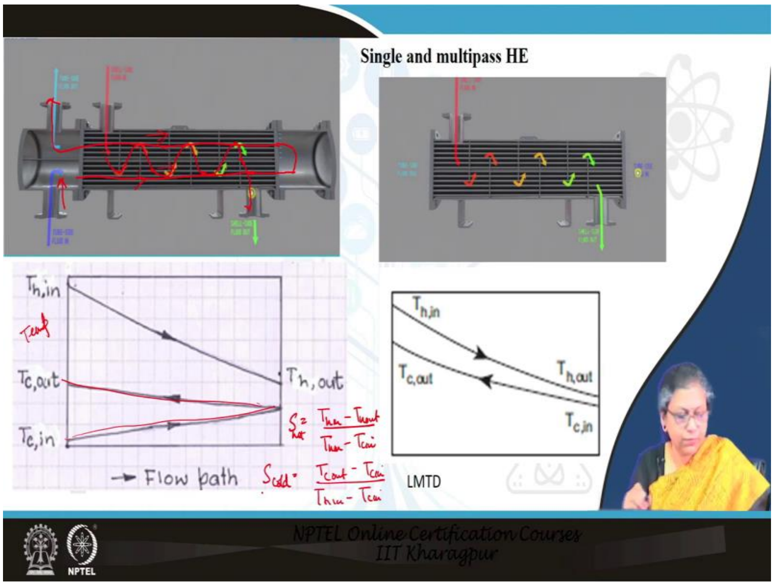

Now, when you have multi passing under that particular case what happens say for instance in this case this has been repeatedly told to you that you can have the tube side entry and this shell side entry both at one end or you can have at different ends as well. In this particular case the tube side fluid is entering here, it is going like this there, it is going there, it is coming out it is coming in this way and then it is going out.

What about the shell side fluid this has been said it comes here naturally it flows across the baffles, but more or less if you find that grossly if you find if you just consider the inlet and outlet you find that more or less it is a counter current arrangement. But if you see it insight, then in that case what do you see? You find that in some portion say for instance in this particular portion you will find that both the tubes sorry I have I will just corrected you will find that in this particular case the shell side fluid is flowing in this way the tube side fluid is flowing in this way.

So, therefore, the flow is counter current. In this part in the lower part of below the back the pass partition plate if you observe what do find the tube side fluid is flowing in this way the shell side fluid also grossly it is flowing in this direction only. So, therefore, what do you find? We find that in the portion lower than the pass partition plate the temperature the temperature profile if we plot that is the temperature versus the flow path if we plot we find that the for the tube side fluid it is flowing in this way and for the shell side fluid it is flowing in this way.

And above the pass partition plate we find that the tube side fluid is flowing in this way and the shell side fluid it is continuing here. So, for one half of the path you find that the two flow are co current for another half of the entire flow path you find that the flow is counter current. How to account for this thing? For how would you going to account for this thing?

Because moment this happens you cannot take just the LMTD and you cannot proceed in that way. So, one thing definitely you can do you can start with the basic mass balance sorry the basic heat balance and the rate equation from that particular rate equation you strike a differential balance you integrate it over the entire flow path and then you can get an expression an implicit or an explicit expression, but that is going to be extremely complicated.

So, therefore, just to make matter simple what we think is that let us assume that with the same temperature boundaries and also the same heat capacity parameters suppose we design a counter flow heat exchanger. Then in that case if the exchanger is counter flow we can find out the LMTD, but the LMTD cannot be put for this particular one two shell and tube heat exchanger.

So, therefore, with that LMTD expression let us add a correction factor the correction factor is termed as the F T correction factor.

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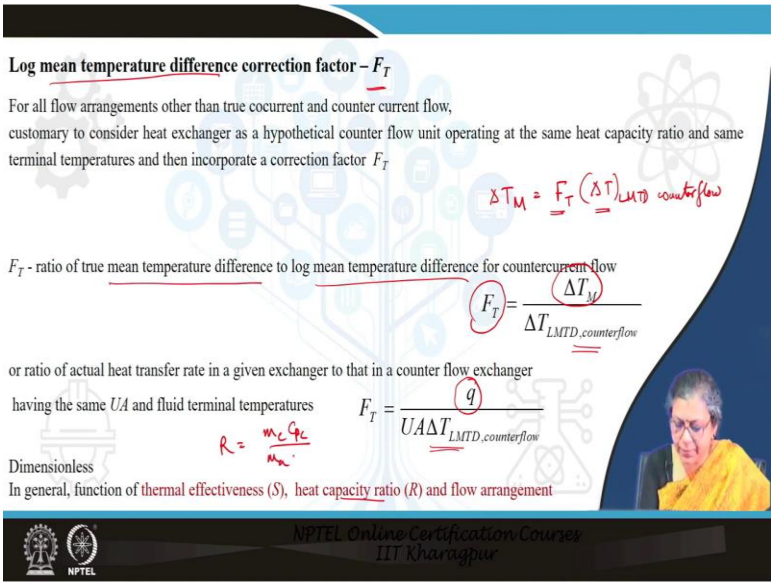

Which is a correction factor, which is made to the log mean temperature difference assuming that the log mean temperature difference has been calculated for counter flow. I hope I am pretty clear to you what we do is that is and remember one thing this we do for all cases where the flow is neither truly counter current not truly co current; that means, even for a double pipe exchanger with multiple tubes.

We adopt the same procedure and we calculate the LMTD for a counter flow heat exchanger and then we incorporate or we correct that particular LMTD with a correction factor F T to get the effective temperature difference which should be used in the rate equation to find out the heat transfer area. Because if we just use the counter flow expression definitely this is going to be higher.

And so, therefore, the area requirement you get will be less. If you design the heat exchanger with that area requirement definitely the temperature, the heat duty that you want, we are not going to get that. If you assume a co current flow you will get a higher the LMTD will be less and therefore, you will go for a higher area than what is required.

So, just to strike an optimum balance and to ensure that you do not go for an either a extremely over design or an under design what we do? We propose this particular F T correction factor which is the ratio of the true mean temperature we assume that this is the ratio of the true mean temperature difference to the log mean temperature difference for counter current flow.

Or in other words it is the ratio of the actual heat transfers which is occurring in the multi pass shell and tube heat exchanger and the heat exchanger which would a heat exchange which would have occurred had this heat exchanger exhibited truly counter current flow. So, accordingly now what happens? So, in that case we find that our delta p effect delta T effective that is nothing but your delta T LMTD mind it this LMTD is for counter flow multiplied by the correction factor.

So, quite naturally this is just a dimensionless parameter and its quite evident that since it is being multiplied to a LMTD counter flow. So, F T always has to be less than 1. And so, if there are ways by which F T can be found out then multiplying that with delta T LMTD counter, you can find out delta T M. Normally people have found is that F T, it is a function of two things; one is the thermal effectiveness.

What is the thermal effectiveness? Thermal effectiveness is the difference in temperature across any particular fluid divided by the inlet approach temperatures divided by this particular inlet temperature. So, the inlet temperature difference. So, it is T h in minus T c in and when we are deriving it say for the hot fluid, then in that case S becomes what? S becomes T h in minus T h out by T h in minus T c in. When we are defining say this is S hot. When we are defining it for the cold fluid then in that case what it becomes T c out minus T c in divided by T h in minus T c in. So, therefore, the s is different for the hot fluid and the cold fluid and the other parameter on which F T depends this is the heat capacity ratio heat capacity ratio is quite naturally this R is equals to m c C p c by m h C p h naturally.

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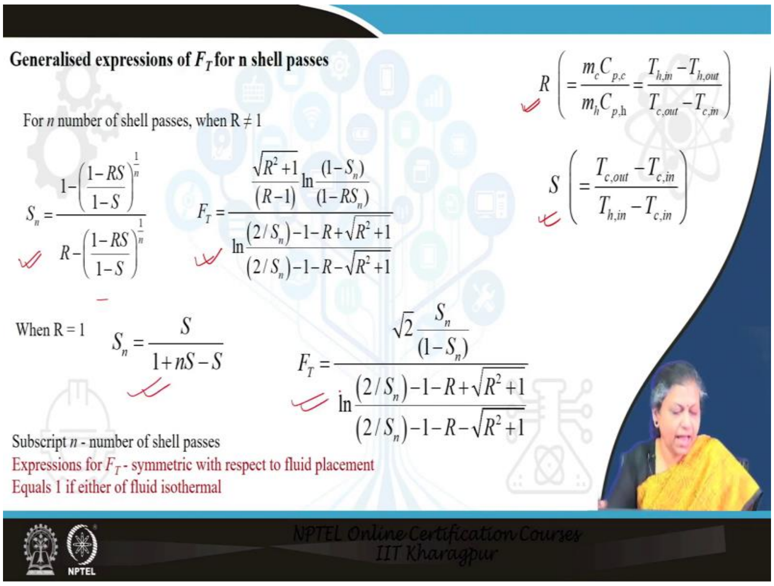

So, therefore, by convention the S is defined in terms of the cold fluid temperature range divided by T h in minus T c in and R is defined the way I have told you. So, you find that R and S both of them they are functions of the terminal temperatures all the four terminal temperatures are known to you once the terminal temperatures are known you can find out R and you can find out S.

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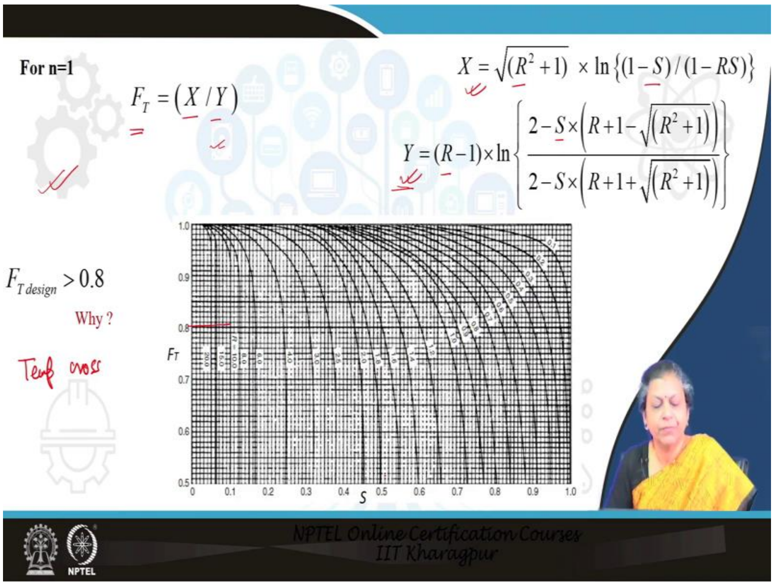

And we find out that if we go for a single shell pass which we normally prefer to do multiple two passes and singles shell passes. Then we can find out that F T can be expressed as a ratio of X and Y; where X and Y are functions of R and S the expressions are given here. So, therefore, what you need to do to find out F T?.

For a single shell pass mind it you can find out X provided you know R and you know s you can find out Y from R and S and then you can divide one by the other and you can find out the F T and you can use it you can correct the LMTD using this particular F T. Now, there are certain things which I would like to mention regarding this F T.

Quite naturally you know F T is less than 1. Is there a lower if can be adopt any particular F T value for designing or heat exchanger, remember one thing we would like to approach the counter current flow as closely as possible just to ensure that we have we get an exchanger with the lowest possible area.

When you are designing whatever you are designing optimum design is something which is very important, it was told to you in the very beginning of this particular course itself. So, therefore, we would always like to optimize cost by optimizing the capital cost as well as the operating cost. So, therefore, we would always like to go or rather we would always like to design an exchanger which can have the lowest possible area while respecting the pressure drop constraints and while performing the heat duty that we want.

So, therefore, naturally if you go for a lower and lower F T definitely we are deviating more and more away from counter flow we would not want that. But normally we do not go below F T equals to 0.8. It is not only that when we go to lower values we end up with an exchanger with a larger area; this is not the reason for this.

The particular reason is that see in this particular graph where I have plotted F T as a function of S and R if you observe that below F T equals to 0.8 below this you find that all the lines they are extremely steep; which tells you that as we go to lower and lower values of F T even with a small variation in temperature also there will be a large variation in the or rather the heat exchanger will become very sensitive to changes in any particular operating temperatures.

So, therefore, the design then becomes extremely sensitive to operating conditions and we would not like it. And then apart from that there is also one other thing if we go for F T low values there is always a chance of temperature cross. Now what is this particular term the temperature cross? This is a term which means that at any point of the heat exchanger the condition is such that the heat is actually transferred from the cold to the hot fluid.

And other words at some point of the exchanger the temperature of the coke of the cold fluid becomes more than the temperature of the hot fluid, we will definitely not want that. So, therefore, considering the temperature cross factor, considering that the steepness of the curves below F T equals to 0.8 more or less we would like to will not like to go below F T equals to 0.8.

But remember one thing this 0.8 is not again a sacrosanct value we would like we would always opt for a much higher F T values particularly when we are going for very large R values or sorry very small R values or for very large R values. under these particular conditions we find that more or less we need to go for a F T much higher and would always prefer then F T near about 0.9296 for doing our design. So, therefore, what you do? You calculate x y you calculate F T.

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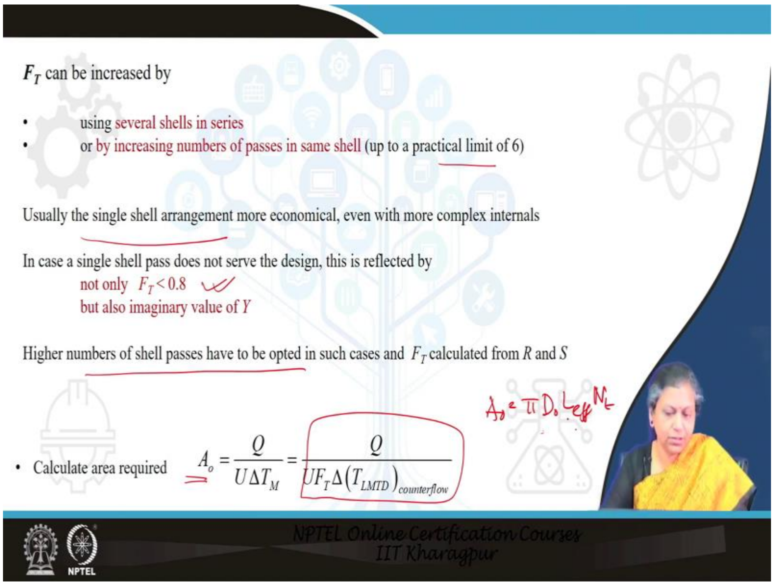

But again I will repeat this is just for a single shell. Suppose you have calculated you find that F T is very less now you would like to increase the F T. What are the two options that you have? One is either you can use several shells in series or you can use the single shell you can increase the number of tube passes usually from economic considerations.

We all we usually we try to opt for the single shell arrangement this is definitely much more economical even if it involves more complex internals. Because if the multiple passes naturally the internal arrangement is going to be more complex, but even with that we would we would like to prefer a single shell arrangement.

If suppose by some means we find that in a single shell arrangement the F T it is not possible to perform this design and this will be reflected not only by F T less than 0.8, but also band imaginary value of this y. So, both these would suggest that a single shell design is not possible if that happens then in that case we have to go for higher number of shell passes when we go for higher number of shell passes.

Then under that condition we find that we cannot use this particular formula anymore, this formula then in that case we have to go for much more involved formulae. In fact, this expression it has been derived from these expressions only the considering n to be the number of passes, then in that case you have to find out S n and once you have found out S n you have found out r then you are supposed to find out F T. And then you can use this F T and you can correct your delta T LMTD and then you can go ahead with your design.

Now remember one thing that it does not matter if you look at these expressions and even if you look at these expressions you will find that these expressions are symmetric and does not matter where the fluids are placed if the hotter fluid is on the shell side or if on the cooler fluid is on the shell side it does not matter the F T is the same.

So, therefore, if you interchange the fluids the F T is not going to change. It remains the same and F T is equal to 1 for truly co current truly counter current and when either of the fluids are symmetrical. So, therefore, once you have found out the F T you have decided on the number of tube shell passes and number of tube passes generally.

It will be again I repeat generally we would prefer one shell pass and multiple tube passes now in this case I would like to mention that even if you have multiple tube passes even in a single shell a shell pass. More or less this particular figure is its applicable because we have found starting from 1 2 to 1 8 passes more or less the F T value does not change.

If the terminal temperatures and the t capacity parameters are the same there will be at most a 2 percent 4 percent not more than that change. So, therefore, for a single shell pass whatever has been has been discussed here that suffices. Only when we have to go for multiple passes only we have to go for these in these particular expressions which have been mentioned here.

And so, therefore, for a single shell pass you go for 2 tube passes, 4 tube passes, 6 tube passes more or less it is a till a practical limit of 6 passes you can continue in this way. So, therefore, now you have you know what to do with how to find out F T. Once F T can be found out then you know that in this particular expression you know everything.

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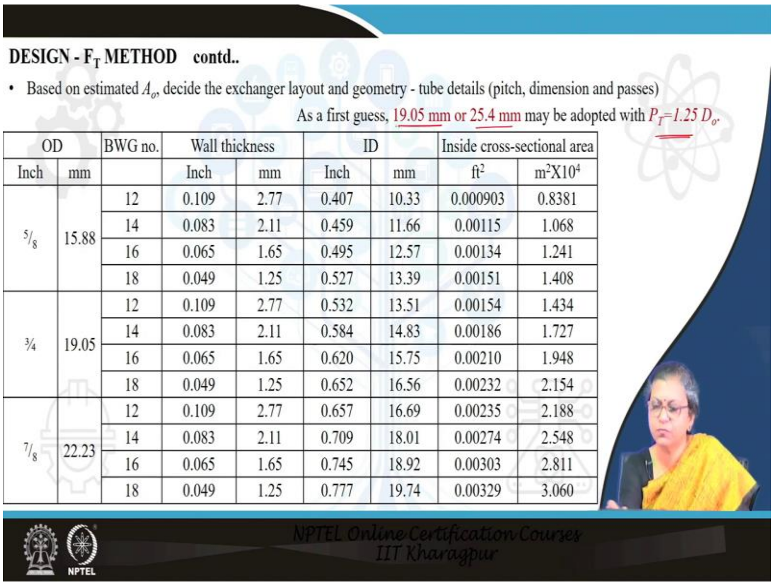

OD		BWG no.	Wall thickness		ID		Inside cross-sectional area	
Inch	mm		Inch	mm	Inch	mm	ft^2	$m^2 X 10^4$
5/8	15.88	12	0.109	2.77	0.407	10.33	0.000903	0.8381
		14	0.083	2.11	0.459	11.66	0.00115	1.068
		16	0.065	1.65	0.495	12.57	0.00134	1.241
		18	0.049	1.25	0.527	13.39	0.00151	1.408
¾	19.05	12	0.109	2.77	0.532	13.51	0.00154	1.434
		14	0.083	2.11	0.584	14.83	0.00186	1.727
		16	0.065	1.65	0.620	15.75	0.00210	1.948
		18	0.049	1.25	0.652	16.56	0.00232	2.154
7/8	22.23	12	0.109	2.77	0.657	16.69	0.00235	2.188
		14	0.083	2.11	0.709	18.01	0.00274	2.548
		16	0.065	1.65	0.745	18.92	0.00303	2.811
		18	0.049	1.25	0.777	19.74	0.00329	3.060

So, therefore, now you are in a position to calculate A 0. Once you are in a position to calculate A 0 I have already mentioned to you that more or less once A 0 is mentioned what is A 0 equals to once more I repeat pi D 0 L effective N t. So, therefore, we have to guess certain things and then we can find out the other things. So, the first thing that we try to find out is we try to fix up the tube OD.

Now as a first guess that I have mentioned more or less we either start with you have 1 inch or three fourth inch tube and this is the standard pitch that we adopt. It can happen that this does not suffice then in that case the standard your dimensions are put here from this particular table this table is for five eighth inch three forth inch seven eighth inch so on and so forth.

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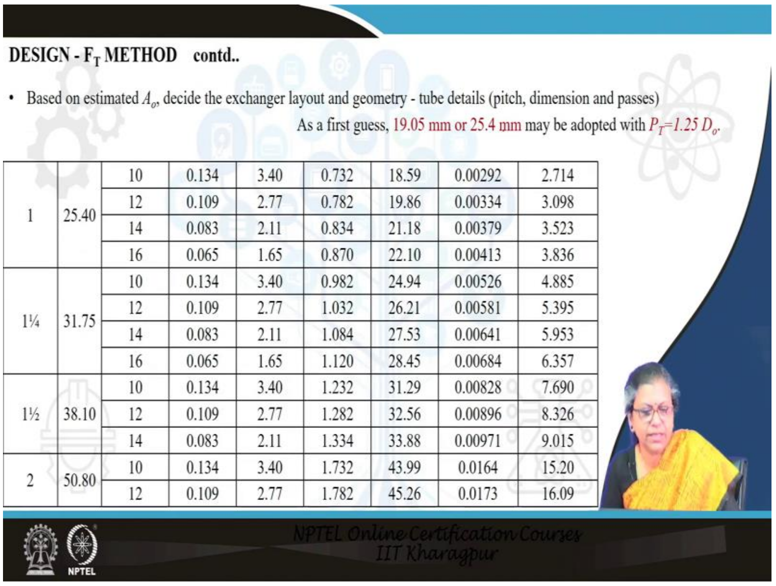

1	25.40	10	0.134	3.40	0.732	18.59	0.00292	2.714
		12	0.109	2.77	0.782	19.86	0.00334	3.098
		14	0.083	2.11	0.834	21.18	0.00379	3.523
		16	0.065	1.65	0.870	22.10	0.00413	3.836
1¼	31.75	10	0.134	3.40	0.982	24.94	0.00526	4.885
		12	0.109	2.77	1.032	26.21	0.00581	5.395
		14	0.083	2.11	1.084	27.53	0.00641	5.953
		16	0.065	1.65	1.120	28.45	0.00684	6.357
1½	38.10	10	0.134	3.40	1.232	31.29	0.00828	7.690
		12	0.109	2.77	1.282	32.56	0.00896	8.326
		14	0.083	2.11	1.334	33.88	0.00971	9.015
2	50.80	10	0.134	3.40	1.732	43.99	0.0164	15.20
		12	0.109	2.77	1.782	45.26	0.0173	16.09

So, therefore, accordingly you can select the OD corresponding to the OD the wall thickness ID in inside cross sectional area everything is provided. So, you can go for it. But first we start with these particular guesses.

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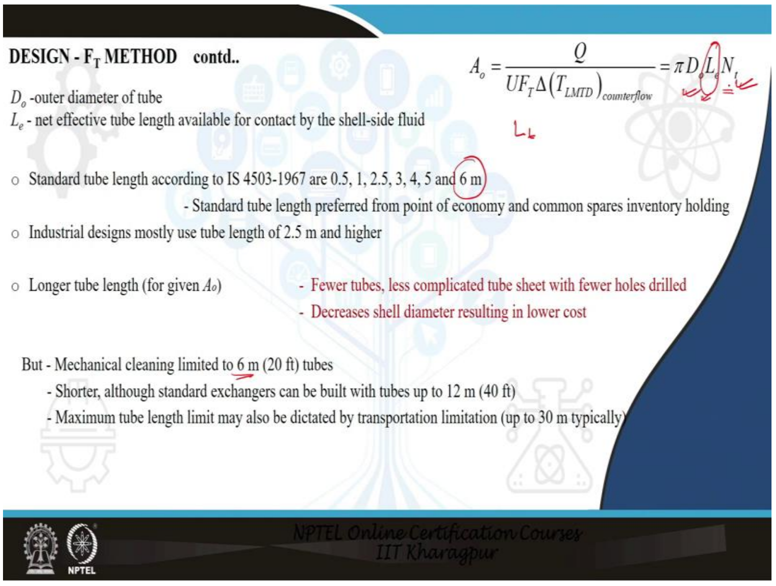

Once this has been done; so, therefore, your D 0 is known now the thing is if you can consider some L e you can find out N t. What are the different again when we go for finding out L e we first need to find out L t, then we have to find out what are the areas which are not available then we are we can find out L e. Now considering finding out the tube length again we referred to the course.

Now, we would like to go for longer tubes under what conditions? When we go for longer tubes then in that case we would need fewer number of tubes, when we have fewer number of tubes fewer holes can be drilled in the tube sheet. So, therefore, the tube sheet fabrication becomes less complicated and the other thing is when we go for longer tubes it decreases shell diameter results in a lower cost at the same time.

When we go for very long tubes mechanical cleaning becomes very difficult at least mechanical cleaning is limited to 6 meters of the tube the 6 meters of tube length. Although we can go up to 12 meters, but for mechanical cleaning 6 meters is maximum. The other thing is when we have very long tubes then that can also be a problem for transportation.

So, considering those things normally what we will be doing is we will go for 6 m long tubes and we would start with the design constrain the 6 m long tubes and subtracting the thickness of the tube sheets these will discuss later we will find out L e and then we would try to opt for the number of tubes that we can accommodate here.

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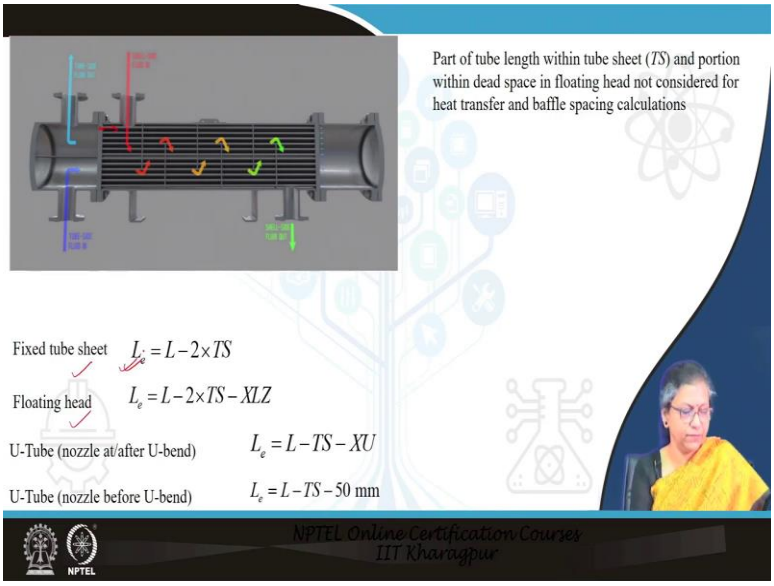

Well regarding L e we are going to discuss in the next class that how we are going to find out L e for different different conditions. One thing you have already understood that the part of the tube which lies within this particular thickness of the tube sheet is not available for heat exchange. So, therefore, this part has to be subtracted. So, in you need some way to find out the tube sheet thickness. There is one other part also where the tube length is not available.

This is this particular portion beyond the nozzle of the shell side this particular portion this is also known as the dead space this is also not considered for your tube length and also for baffle spacing. So, these things have to be considered accordingly depending upon the tube sheet that you have decided we are going to find out the effective length. From the effective length; we are going to find out the number of tubes that are required.

Remember one thing after you whole thing is done we again need to come back and make a number of checks to see that the design is acceptable. So, with this we have defined F T we have defined LMTD the effective temperature difference for multi pass heat exchanger based on that we know how to calculate the heat transfer area.

We assume the diameter, to start with we assume the length and we find out the number of tubes. Once the tube layout is fixed, then we go to find out the shell diameter and after that we proceed with the heat transfer coefficient and the pressure drop. So, this part we will be doing in the next class. This much for today.

Thank you so much.