Equipment Design: Mechanical Aspects
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Lecture 14
Design of Flanges

Welcome to the fourth lecture of week 3 and here we will discuss design of flanges. This topic we have already started from lecture 2 of this week. In lecture 2 we have discussed different type of flanges and we have defined the flanges, and in lecture 3 we have discussed different type of facing of flanges, and further we have discussed the terms related to gaskets and how the gasket material should be selected, okay. So all that information will be used in designing, which we are continuing in this lecture. So let us start the design of flanges.

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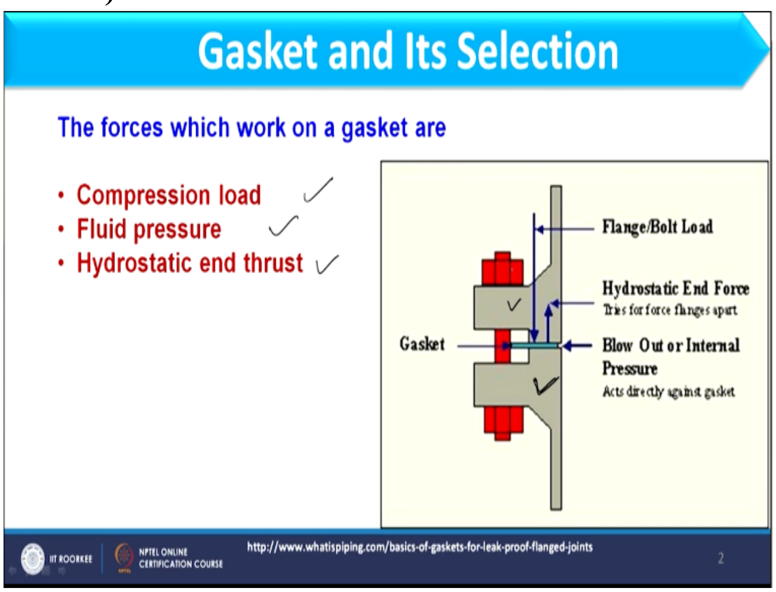

So when we are going to design the flange, what are the parameters I have to consider in designing. So if you focus on connection of flange, this basically the total connection of flange consist the flange itself, and flange you can understand it comes in a pair so that flange and in between that flange we need to place the gasket. So what we have to find out, first of all when we are starting the calculation of flange or when we are starting the design of flange.

So when we are starting design of flange we have the basic information that at which it has to place, okay. The shell or whatever we want to connect together like nozzle or some other section. So that information I am having that for which diameter you have to design the flange. So once I have that assembly where flange has to be placed then we will see based on that diameter we can find out what should be the dimension of gasket.

Once I am having the dimension of gasket I will decide what would be the diameter of bolt circle because all these flanges which we are going to design these are narrow-faced flanges where gasket lies within the bolt circle. So once dimension of gasket we can find out, which depends on dimension for which I have to design the gasket or the flanges. So after computing the dimension of gasket I will decide bolt circle diameter and once I am having bolt circle diameter I will decide the outer diameter of flange, and then we will decide about the thickness of flange plate, okay.

So in this way we will proceed. So let us start with the design of gasket. Now once I am having the design of gaskets, first of all we will discuss what are the forces which are acting on gasket because only then we can choose the correct material. So here you see I am having different forces which work on the gasket that is compression load, fluid pressure and hydrostatic end thrust that can be easily understood by this image.

Here if you consider this image, this is nothing but one part of flange, this is another part of flange, and this is the whole connection. These two flanges are connected with the bolt and in between these flanges I can place the gasket, okay. So once I am having the bolt, what is the meaning of that that it compresses this gasket, this bolt basically provide minimum seating stress, which we have discussed in last lecture. So seating stress basically compresses the gasket, okay. So it will work toward the center of gasket, okay.

And when I am having the operating condition that is operating pressure or internal pressure it will try to depart the connection, okay. It will try to depart the connection. And when it will depart the connection it will give hydrostatic end force to this connection, okay. So gasket load will act downward, however, internal pressure or hydrostatic end thrust will work upward. So when this internal pressure will try to apart internal pressure again will try to push the gasket outside.

It will try to blow the gasket out, okay. So that pressure which is putting by the fluid to gasket to blow it that we call as fluid pressure. So here I am having three things, first is compressive load due to bolt, second is hydrostatic end force due to fluid pressure or due to internal operating pressure and due to internal operating pressure of the fluid it will try to blow the gasket out. So this is the whole assembly which we have to consider in design. So let us start the design of gasket.

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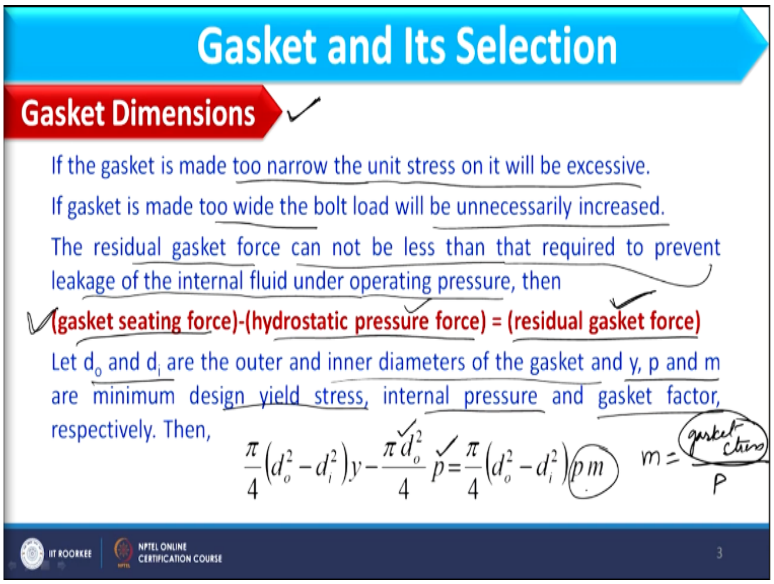

So how I can choose the dimension of the gasket, for this we have to see a few points. If the gasket is made too narrow the unit stress on it will be excessive, okay. So why it is so because it has the compressive load as well as hydrostatic end force, okay. So when I am putting the compressive load it will work on a particular area and that area will be the cross sectional area of the gasket.

So when the gasket will be very narrow or it has very small width the acting area would be very low and then in that case pressure which will act over this will be very high, so it will try to tear it. I hope you can understand that point and further if gasket is made too wide the bolt load will be unnecessarily increased, okay. So that bolt load should be decided by which load gasket requires, okay. So when the gasket width will be very high, bolt has to put excess load to assure the tight proof connection.

Therefore gasket should not be very narrow, should not be very wide. We have to make the balances to choose the exact dimension of the gasket. And that balances the residual gasket force cannot be less than that required to prevent leakage of the internal fluid under the operating condition or under operating pressure. It means residual gasket force should not be low enough so that gasket blowout occurs, okay.

Residual gasket force means when a gasket stress will act, okay. So gasket stress when it will act, when I am having seating stress minus the stress generated due to pressure. So what would be the residual gasket force, so that will be considered due to minimum seating force minus the force occurs due to the operating pressure, okay. So it will be written as gasket seating force minus hydrostatic pressure force will be equal to residual gasket force.

Now I will write expression of all these, but before that we have to discuss a few parameter related to this. Let d o and d i are the outer and inner diameter of gasket and y, p and m are minimum design yield stress or seating stress, internal pressure and gasket factor respectively. So considering all these parameters we will write expression for this equation. Now if you focus on this gasket seating force, gasket seating force basically deals with minimum seating stress and that would be y and minimum seating stress where it will act, it will act to the cross sectional area of the gasket where d o and d i are outer and inner diameter of the gasket.

Further hydrostatic pressure force that would be Pi d o 2/4 x p, p is the design pressure and Pi d 2/4 where d o is the outer diameter of gasket. So if you consider hydrostatic pressure force it means what, it is considering the area up to d o, okay. It means it considers the sectional area of gasket along with sectional area of the vessel or where I am putting the gasket. So why it is considering so because hydrostatic end force whichever is at this will act at a particular layer.

So wherever it will act whole dimension I have to consider. So it will not only consider for gasket but for whole section wherever it is applicable therefore I have chosen. Therefore we have taken d o as the diameter not the d o - d i, I hope I am clear. Further residual gasket force I have to consider and that force will depend on residual gasket stress and that would be nothing but this p x m, m is the gasket factor and how we can define this, if you remember m we have defined as gasket stress divided by internal pressure, okay.

So gasket stress would be p x m and that will act on the cross sectional area of the gasket. So therefore we have considered d o 2 - d i 2 Pi/4.

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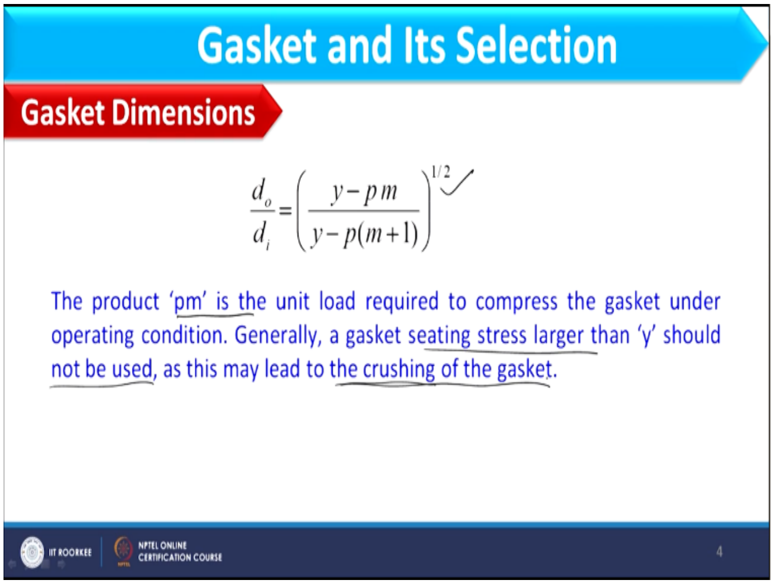

So after resolving this equation we can have d o and d i ratio as y-pm divided by y-p(m+1) power 1/2. So in this way we can decide the ratio of dimensions of gasket. If I know one dimension I can calculate second dimension using this particular equation. And where the product pm is the unit load required to compress the gasket under operating condition. Generally a gasket seating stress larger than y should not be used as these may lead to crushing of the gasket.

So y is basically the minimum seating stress and that stress is sufficient to maintain the tight connection when operation is going on or when operation is not going on and when we excess this stress it will crush the gasket material, so that we should not do. As we can choose y and m according to the gasket material as it is given in a table we have discussed in pervious lecture, okay.

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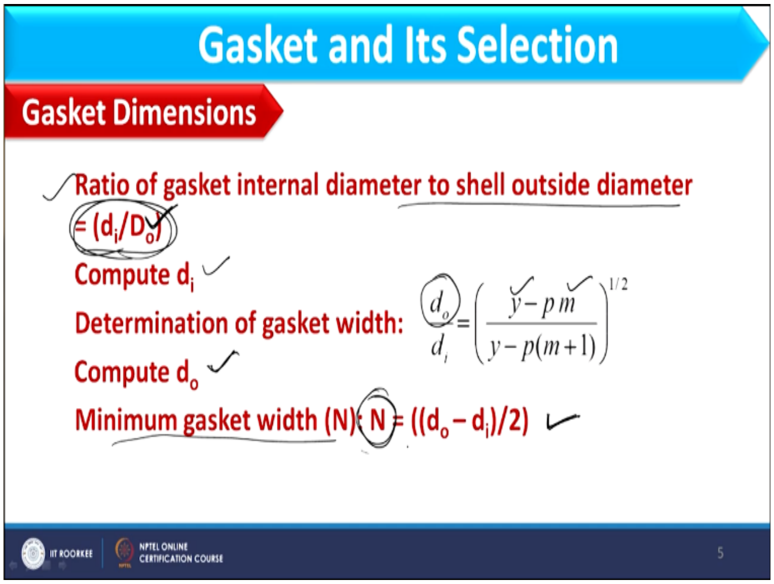

So in this way you have to find out d o and d i. Now what are the steps to calculate the gasket dimensions are. First is ratio of gasket internal diameter to shell outside diameter. This ratio I have to decide. Now what is this d i/D o. The d i is the internal diameter of gasket and D o is the outer diameter of shell or the nozzle where I have to make or where I have to connect the flange. So usually this ratio is greater than one like we can take as 1.01, 1.02, like that. It is slightly greater than 1.

So what is the meaning of that that gasket internal diameter is slightly higher than the shell outside diameter. I hope I am clear because gasket has to place slightly above or slightly away than the inner edge of the flange. So that flange inner edge will be connected to the shell so that may be equal to D o, why I am telling may be because it may be slightly higher when it is slip-on type, okay. And when it is connected over the mouth of the pipe or over the shell, so it is basically D i would be equal to the flange diameter, okay or we can call that bow diameter.

So in that case if it is placed above the pipe it means bow diameter would be equal to inner diameter of pipe, okay. So to compute the gasket dimension we have to first decide the ratio of d i/D o, okay. So that ratio we can start with let us say 1.01, 1.02, like that, okay. Once I am having this ratio I already know the diameter of shell because for that shell only I have to design the gasket or flange. Once I know D o I can find out D i, and once I know D i I can calculate D o using this expression and y and m I can choose for a given gasket material.

So then we can compute d o, once I know the d o I can calculate minimum gasket width and that should be d o – d i/2. So considering these diameters you can calculate minimum gasket width. Now once I have calculated minimum gasket width, this value should be checked with the gasket width given in the table we have discussed in last lecture. So that N should be at least greater than them or if it is less than these value we can choose the value given in the table.

So that we will exercise in the example which we will solve and then you can understand that in better way. So once I calculate N and that is the minimum gasket width we can further calculate actual gasket diameter, okay and that would be d i + 2N, okay.

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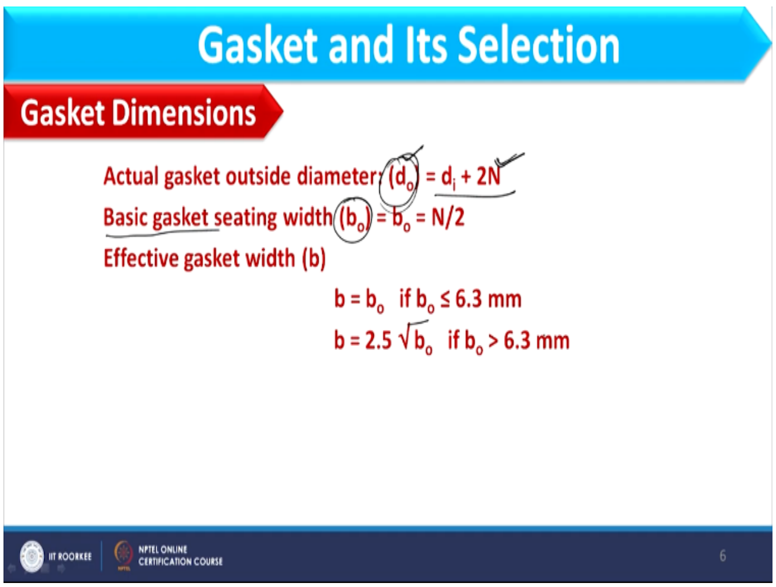

Now why this diameter is we are using when the same diameter I have used to find out value of N because N may be change now. So N value comes less than the value available in table which we have discussed in the last lecture. Then that N will be taken which is given in the table not what we have computed, okay. I hope I am clear. So this value of N will be compared the value of width given in the table which we have discussed in last lecture and whatever would be higher among these two that I have to take as N.

So N may differ whatever we have calculated in last slide, okay. So considering new value of N, we have to calculate d o. Now once I am having this width with us what I have to further find. Further I have to focus on effective gasket width. What would be the effective gasket width. For example, if I am having this two faces of flange where I am putting the gasket and if gasket placed over here and second part of flange I am putting like this, so whatever force I am applying do you think that throughout the width the force will act uniformly, okay. Answer is no.

Only certain section of that gasket would be effectively pressed. And for that purpose we need to find out effective gasket width. So to find out effective gasket width, first I have to focus on basic gasket width that is b o, b o will not have any physical existence or physical significance, it will only use to find out effective gasket width. So b o is basically N by 2 and effective gasket width can be found out as b = b o if b not is less than or equal to 6.3 mm. And if b is equal to 2.5 root over b o. If b o is greater than 6.3 mm so in that way you calculate gasket width.

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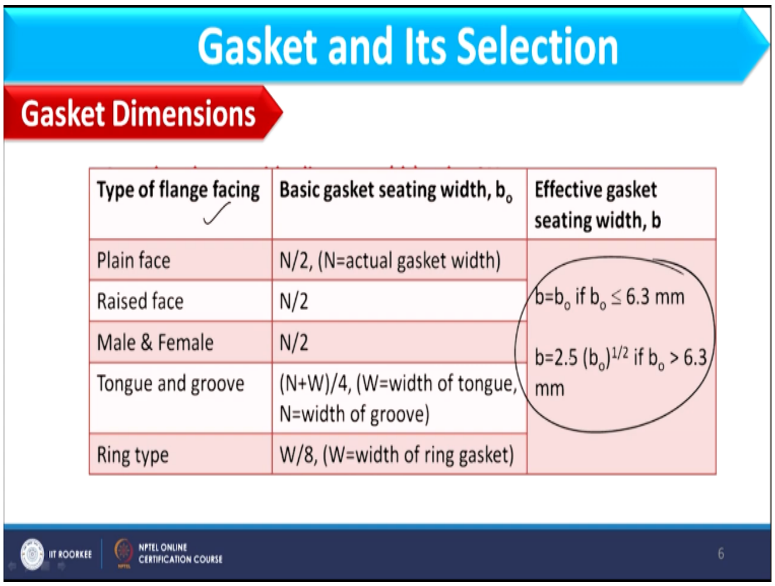

Type of flange facing	Basic gasket seating width, b_o	Effective gasket seating width, b
Plain face	N/2, (N=actual gasket width)	$b=b_o$ if $b_o \leq 6.3$ mm $b=2.5\,(b_o)^{1/2}$ if $b_o > 6.3$ mm
Raised face	N/2	
Male & Female	N/2	
Tongue and groove	(N+W)/4, (W=width of tongue, N=width of groove)	
Ring type	W/8, (W=width of ring gasket)	

Further you see this table, where I am having different type of flange faces like plain, raised and ring type and other type. And here I am having basic gasket seating width that is b o for each type of facing I am having b o expression, so b o we can calculate accordingly and then we can find out effective gasket seating width considering these expressions.

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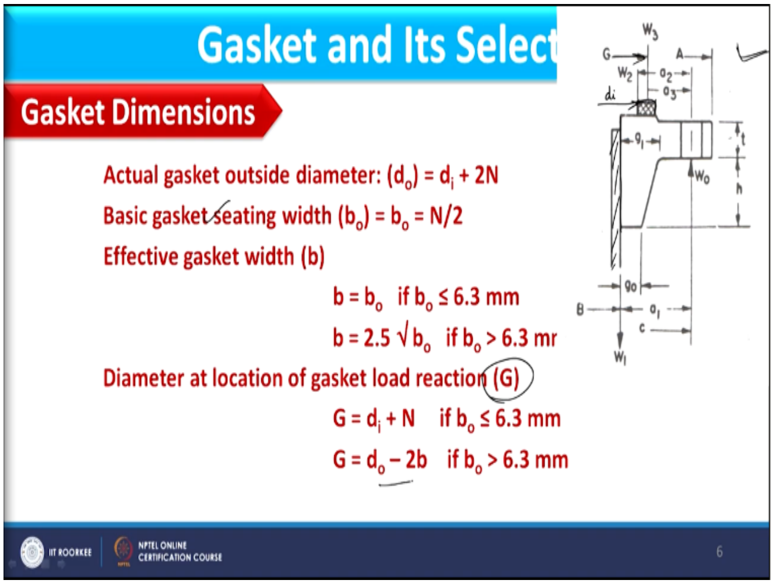

Now why I have calculated effective gasket width because I have to find the diameter where load on gasket will actually work, okay. So we can consider that diameter for load of reaction on the gasket and that would be decided by G, okay. Now here in this image we can schematically show the connection if you consider this part, this is the shell part and here this flange is connected, okay. This is basically tapered type or welded-neck and here I am having the gasket, okay.

So usually effective gasket width we consider as half of the total width or it may differ also as we can see from the expressions. Now here at the centre of this we have the load on the gasket that is total force on the gasket will act and it will be decided by diameter G, okay. So in this case G would be d i + N. So what is d i, d i basically this diameter, which is the inner diameter of gasket, okay. This is D out, so d i + N we can consider when d o is less than or equal to 6.3.

Otherwise we can see G as d o – 2b. So in this way we calculate the diameter where load on gasket will work. Till now we have discussed the design of gasket. Once design of gasket will be done, we will move towards design of flanges, okay. Once diameter of gasket will be decided based on that we will decide the bolt circle diameter and outer diameter of flange and thickness of flange, so that will come under flange design.

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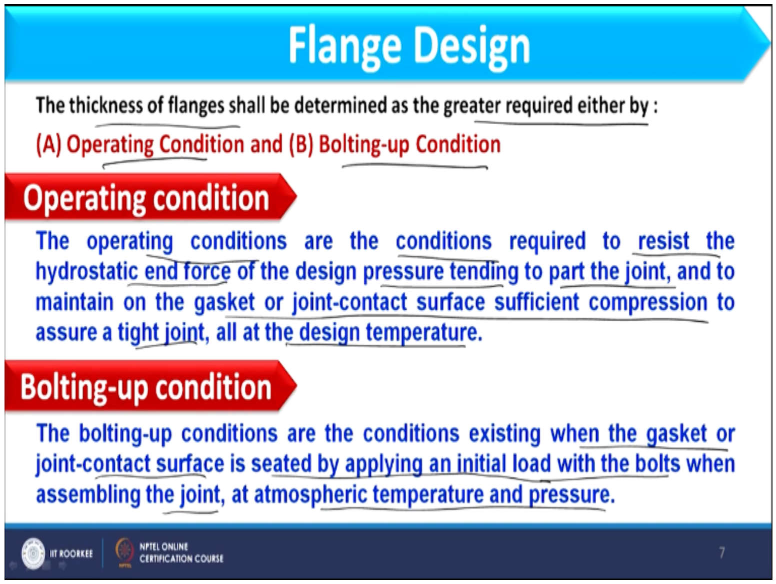

So as far as flange design is concerned the thickness of flanges shall be determined as greater required either by operating condition and bolting-up condition. So when the flange is being operated it has two conditions, first is operating condition when operation is going on, second is bolting-up condition where operation is not going on, still the connection should be tight enough, okay. So we consider we calculate parameter for these two conditions separately and then we will choose the higher one.

So first of all we will discuss, we will define operating condition and bolting-up condition in detail and here we will start with operating condition. So you see operating conditions are basically those conditions which required to resist the hydrostatic end force of design pressure, which tends to apart the joint, okay and to maintain the gasket or joint-connect surface sufficient compression to ensure the tight joint at all design temperature. So while operation is carrying out the joint should be tight enough or compressed enough even though hydrostatic end force will act.

So when we are going for operating condition whatever parameter I am choosing for designing, let us say allowable stress value. So all that value I have to take as design temperature. Second condition I am having is bolting-up condition and in this bolting-up condition, so these conditions existing when the gasket or joint contact surface is seated by applying an initial load with the bolts when assembling the joint at atmospheric temperature and pressure. So that bolting-up condition will be the initial condition, where operation is not going on.

So in that case whatever parameter I am considering for designing, for allowable stress that I have to see at atmospheric temperature not at design temperature. So this is the basic difference between operating and bolting-up condition that in operating condition we choose values at design temperature and design pressure, and for bolting-up condition we choose value at atmospheric temperature and pressure.

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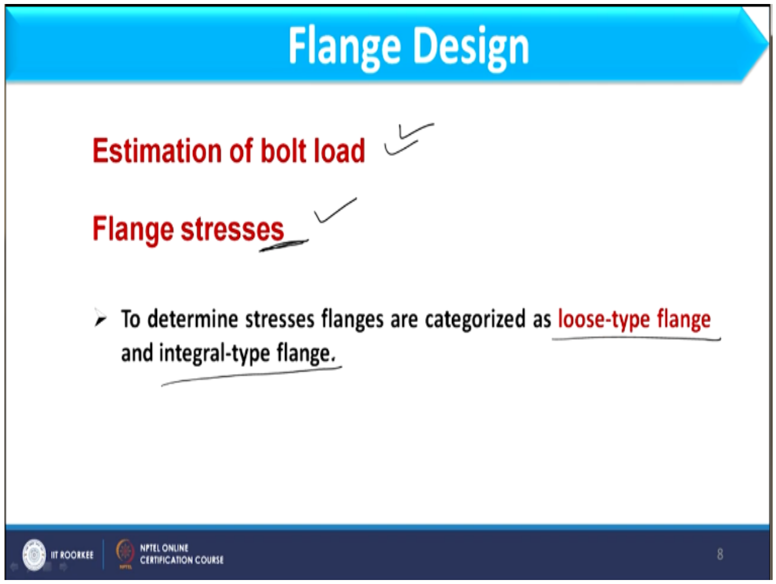

Now as far as flange design is concerned what I have to estimate or find out, I have to estimate the bolt load first and then flange stresses. Because bolt load will speak about how many bolts are required, flange stresses speak about how much thickness of flange to be provided. So to determine the stresses flanges are categorized as loose-type flange and integral-type flange. Now what is the loose-type flange and integral-type flange.

When I am having this pipe okay and then I connected the flange just above this, okay. This is the pipe and this is the flange.   When it is connected just above this, it is basically called integral-type, okay. When I am having this pipe and flange is connected like this, I hope you are getting this, so this is basically called loose-type, which is available at outer diameter of the pipe. Integral-type whatever opening in flange or whatever inner diameter of the flange that will be equal to inner diameter of pipe.

However, in loose type inner diameter of flange will be equal to outer diameter of pipe that is the basic difference. So in this particular lecture we will discuss loose-type flange designing. But first of all we have to estimate the bolt load. So for estimation of bolt load we have to decide parameter for operating condition as well as bolting up condition. So let us start with operating condition calculation. Now as far as operating condition is considered what parameter I have to focus on.

First of all I have to focus on internal pressure or the design pressure and second I have to focus on gasket stress because that gasket stress will work when operation is going on, okay.

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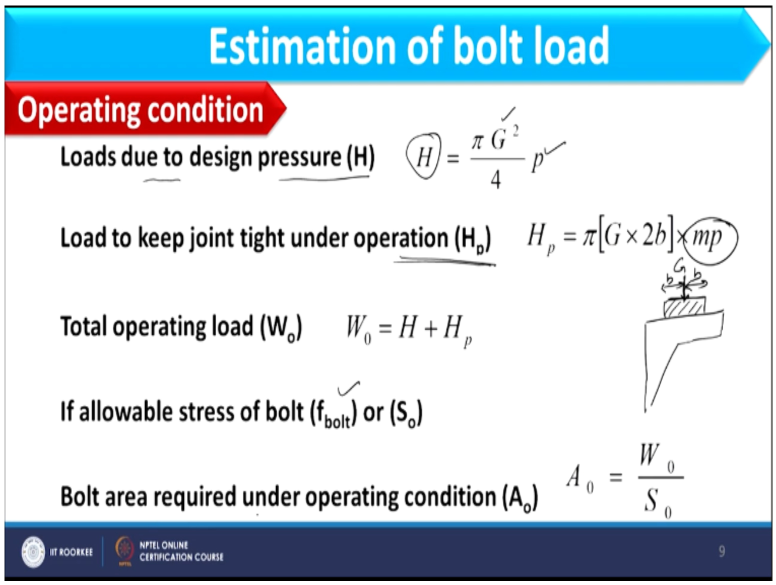

So first of all we will find out load due to design pressure, okay and that is denoted by H and in this case Pi G 2/4 x p. The p would be the design pressure and G would be the diameter where load on gasket will act. So till here that pressure will work. And second we have to find out load to keep joint tight under operation condition. So here we will discuss gasket stress. So this H p can be found as Pi G x 2b x mp.

What is this mp, this is nothing but the gasket stress, okay under operating condition. Now Pi G, Pi G is what, Pi G is nothing but the periphery and if you consider 2b is basically the total width of the gasket, okay. So G is the point and this is b and this is b in the gasket if you consider this in flange like this. So here I am having the G value, so Pi G is nothing but this periphery and 2b is basically total width of the gasket. So when I am considering mp, it is basically acting on total gasket, okay. I hope I am clear.

And then total operating load in that case would be H + H p, we will join this two. And if allowable stress of bolt that is F bolt or S o we can consider. Then bolt area required under operating condition can be defined as W o/S o. Now what is this S o. S o is basically allowable stress of bolt material at design temperature, okay because here I am considering operating condition and all parameters should be considered at design temperature.

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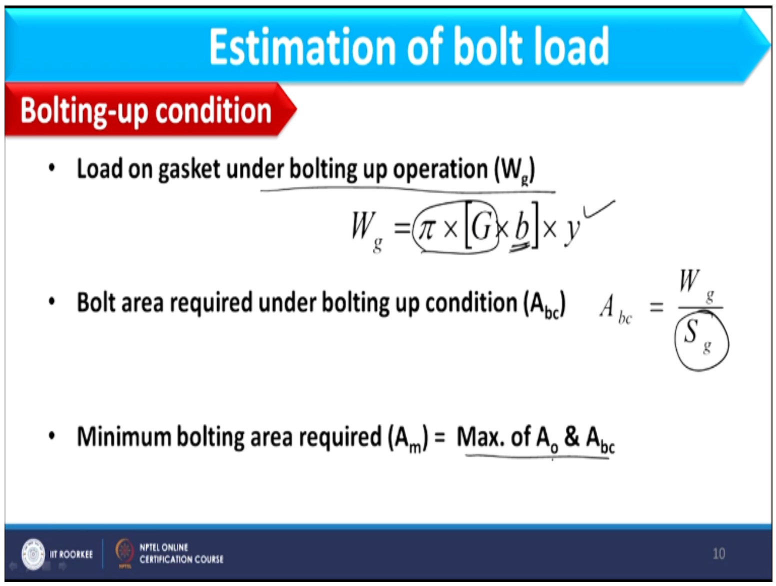

Further we will focus on bolting-up condition. And in bolting-up condition we have only one force that would be the seating force which act on the gasket because at that time pressure will not work. So load on gasket under bolting-up condition we can define that as W g, which is equal to Pi x G x b x y, okay. So that y is the seating force. Pi G is basically the periphery and b is the width of gasket. So b you can consider as effective gasket width.

So here you should see that in bolting up condition we have considered only b, which is the effective gasket width and in operating condition we have considered 2b because when operation is going on whatever stress will be applicable on gasket that will act on total width because until unless it will not act on total width it will not push it, okay. Because when I am having internal pressure it will try to push the gasket out, okay.

So in that case we have considered total width. However, in bolting-up condition we have considered only effective gasket width because when y will be applied it will only compress the b dimension effectively, okay. Therefore we have considered b over here. Now bolting area we can find out as A bc, which is equal to W g/S g. So Wg is that load at bolting-up condition and S g is the allowable stress of bolt material at atmospheric temperature. So minimum bolting up area would be maximum of these two, maximum of A o and A bc.

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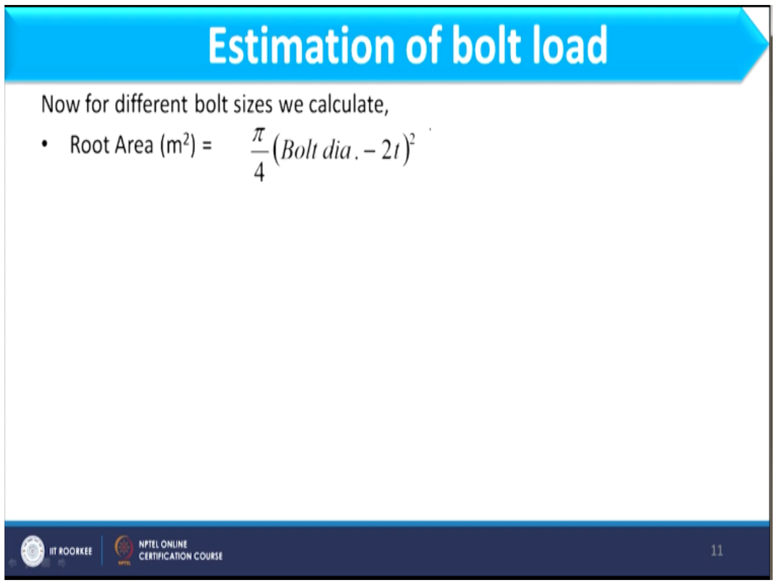

So till now we have found the minimum bolting area and now we will decide that how many bolts will be required, okay. So for that purpose we have to find out root area, okay. And root area is related to a particular bolt and root area can be defined as Pi/4 bolt diameter - 2t whole square, okay. Now what is bolt diameter and what is t.

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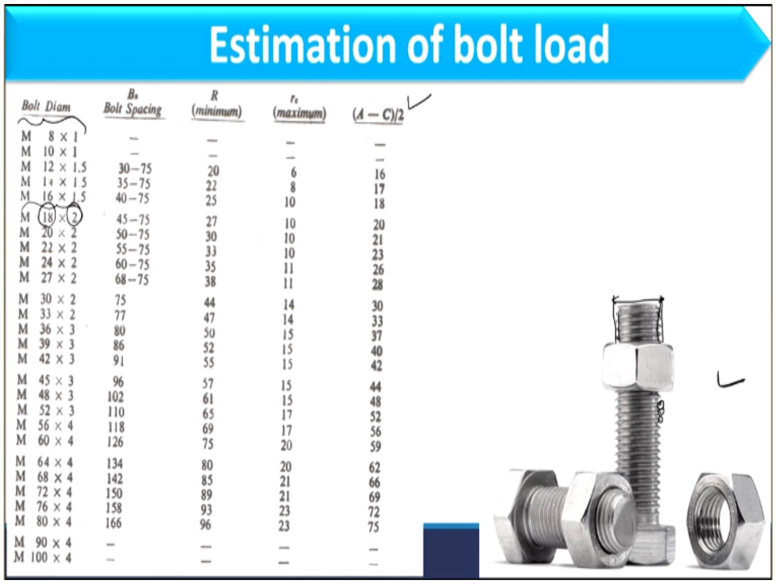

Bolt Diam	B_s Bolt Spacing	R (minimum)	r_1 (maximum)	$(A - C)/2$
M 8 × 1	–	–	–	–
M 10 × 1	–	–	–	–
M 12 × 1.5	30–75	20	6	16
M 14 × 1.5	35–75	22	8	17
M 16 × 1.5	40–75	25	10	18
M 18 × 2	45–75	27	10	20
M 20 × 2	50–75	30	10	21
M 22 × 2	55–75	33	10	23
M 24 × 2	60–75	35	11	26
M 27 × 2	68–75	38	11	28
M 30 × 2	75	44	14	30
M 33 × 2	77	47	14	33
M 36 × 3	80	50	15	37
M 39 × 3	86	52	15	40
M 42 × 3	91	55	15	42
M 45 × 3	96	57	15	44
M 48 × 3	102	61	15	48
M 52 × 3	110	65	17	52
M 56 × 4	118	69	17	56
M 60 × 4	126	75	20	59
M 64 × 4	134	80	20	62
M 68 × 4	142	85	21	66
M 72 × 4	150	89	21	69
M 76 × 4	158	93	23	72
M 80 × 4	166	96	23	75
M 90 × 4	–	–	–	–
M 100 × 4	–	–	–	–

Now here if you see this table is available in the book given by B. C. Bhattacharya in Chapter 7. So there we have this bolt diameter, okay. Now this if you see here I am having M and here I am having some dimension, so this basically M8 x 1. This is the way dimension of bolt is given, okay. For example, if I am having M18 x 2, so it means 18 is the diameter of bolt and 2 is the T value, okay. So considering these two values you can find out root area.

Further if we see what is root area, for that if you focus on this particular bolt here, you see the basic area we are considering that would be the root area. So what is the bolt diameter, bolt diameter is this diameter, okay. When I am considering this and this, so this is basically bolt diameter. And if you consider this particular thickness of thread that will be nothing but t.

So root area we can define as the base area of the bolt. We are not considering this thread area because when this bolt is entered into the nut, you see for this thread it has already assembly, it has already construction like that so that this thread will be fitted in this and only the area required to be entered is the root area, okay. So in the similar line whatever hole I am preparing in flange that will depend on root area not the total diameter of bolt, okay. I hope it is clear to you.

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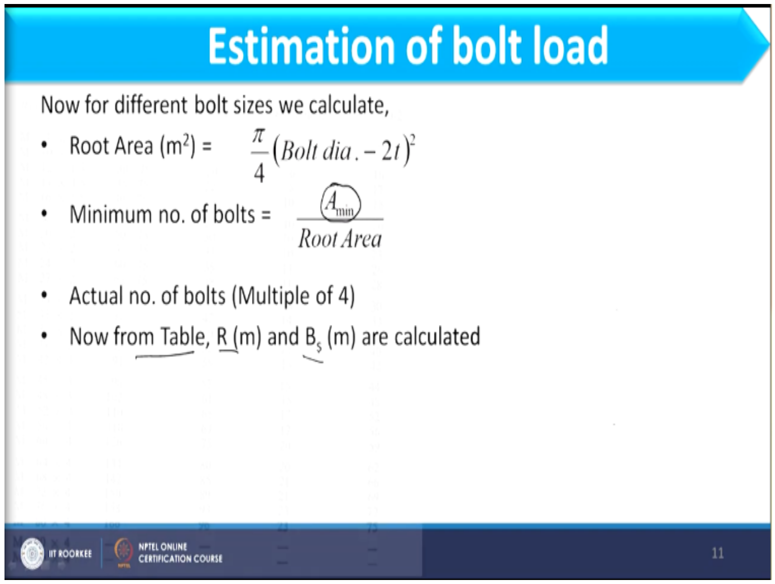

So once I am having root area I will calculate minimum number of bolts that is AM, which we have seen in the last slide divided by root area it will give minimum number of bolts, okay. Now if you remember the second lecture of week 3 where we have defined the flanges, there we have seen that all flanges will have bolt hole in multiple of 4, okay, either 4 or 8 or 12, in this way the number of bolts vary. So whatever minimum number of bolts I have found out I have to choose actual number of bolts as multiple of 4, which is available next to that value.

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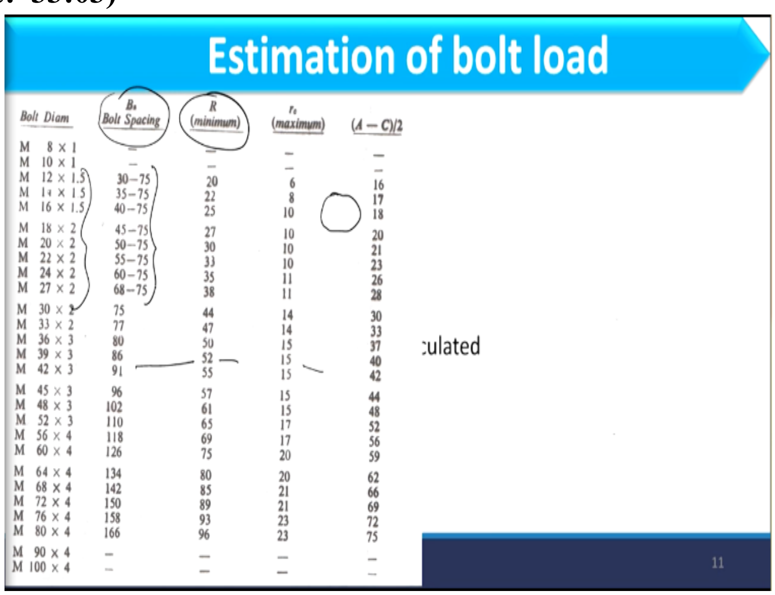

Bolt Diam	B_s Bolt Spacing	R (minimum)	r_s (maximum)	$(A - C)/2$
M 8 × 1	–	–	–	–
M 10 × 1	–	–	–	–
M 12 × 1.5	30–75	20	6	16
M 14 × 1.5	35–75	22	8	17
M 16 × 1.5	40–75	25	10	18
M 18 × 2	45–75	27	10	20
M 20 × 2	50–75	30	10	21
M 22 × 2	55–75	33	10	23
M 24 × 2	60–75	35	11	26
M 27 × 2	68–75	38	11	28
M 30 × 2	75	44	14	30
M 33 × 2	77	47	14	33
M 36 × 3	80	50	15	37
M 39 × 3	86	52	15	40
M 42 × 3	91	55	15	42
M 45 × 3	96	57	15	44
M 48 × 3	102	61	15	48
M 52 × 3	110	65	17	52
M 56 × 4	118	69	17	56
M 60 × 4	126	75	20	59
M 64 × 4	134	80	20	62
M 68 × 4	142	85	21	66
M 72 × 4	150	89	21	69
M 76 × 4	158	93	23	72
M 80 × 4	166	96	23	75
M 90 × 4	–	–	–	–
M 100 × 4	–	–	–	–

Once I have actual number of bolts further I will focus on table and then I will see value of R as well as B s. So if you see this table B s is basically bolt spacing that is the space between two bolts and R is the distance. What is that distance that we will see. So for further calculation we need B s value as well as R value. And if you see for some of the bolts B s value is given in the range. So if you are choosing any bolt among these you have to choose highest value of that spacing not the lower than. Why I have to choose highest value that we will discuss.

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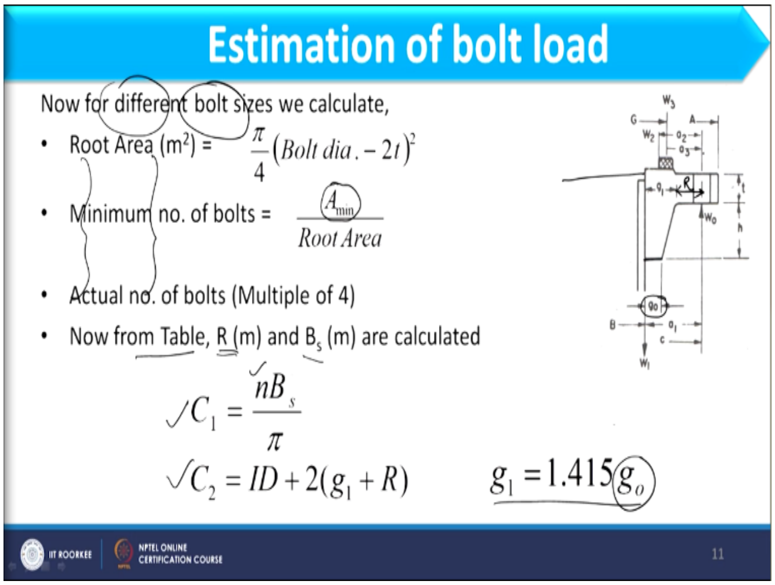

So once I am having this R as well as B s I will find value of C 1, so C 1 is basically the bolt circular diameter which is given as n B s/Pi, where B s I have seen from the table and n is the actual number of bolts, okay so this is C 1. And C 2 I have to find is equal to ID + 2(g 1 + R). So this C 1 and C 2 both are bolt circle diameter, where g1 is 1.415 g o, okay.

Now if you focus on this particular connection of flange g 1 is this, okay and this is the g o which is available over here, this is basically the lower width of the welded-neck flange and g 1 is the tapered width, and R if you see R is this value. So when I am considering C 2 as internal diameter it means this is the internal diameter of flange plus 2 x g 1 + R. So considering this I can find value of bolt circle diameter.

Now here for bolt circle diameter I have computed C 1 as well as C 2, why I am doing so because till now you have no idea that which bolt should be suitable for the connection, okay. So here we have to find out, we have to carry out the whole calculation for number of bolts and then we will compare value of C 1 and C 2 for selection of proper bolt, okay.

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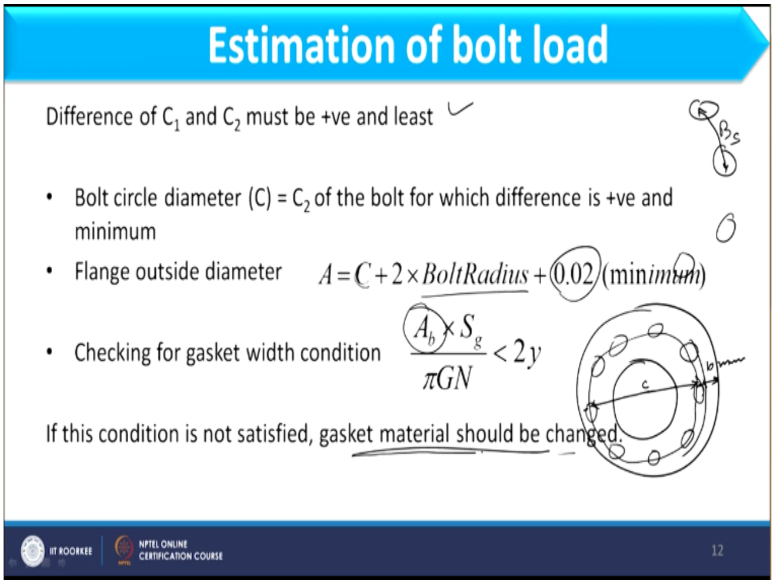

And that can be chosen as difference of C 1 and C2 must be positive and least, okay. We have C 1 and C 2 value, let us say I am having four different bolts or five different bolts, for each bolt I am having C 1 and C 2 value and I will see the difference of C 1– C 2, not C 2 – C 1. It should be C 1 – C 2 and for positive and least value wherever I am finding corresponding bolt I have to select as optimum bolt for design, okay.

So once I am having this C 1 – C 2 value least and positive, accordingly I will choose the bolt, okay and C 2 for that bolt will be now actual bolt circle diameter, not the C 1, okay. Now why I am taking that C 2 as bolt circle diameter because if you see for some of the bolt B s value vary, okay. So in that case because there is the variation I cannot take decision on B s value, however, R value is one value for all and therefore I have taken decision based on that value.

So C 2 of optimum bolt will be chosen as bolt circle diameter. And if you remember we have discussed that if range is available B s value should be chosen as highest one. Now if I am having C 2 value as bolt circle diameter, okay and if C 1 – C 2 is positive and least, what is the meaning of that, that C 1 value would be higher and now I am choosing C 2 value as final bolt circle diameter. And corresponding that if I am considering C 1 as equal to C 2 I have to recalculate value of B s, okay.

And then that B s will be found less than the highest value because B s value can never exceed the highest value. So therefore we have taken C 1 – C 2 should be positive and least and accordingly select bolt circle diameter and then accordingly you have to find out value of B s. I hope you understand what is B s. For example if I am having this bolt circle, B s is basically this value. So we have decided number of bolts, we have decided bolt circle diameter and then we can decide B s value, okay.

So once I am having bolt circle diameter, we can find out flange outside diameter, what is the extreme diameter of flange and that can be found as A = C that is bolt circle diameter + 2 x bolt radius, okay. Now why it is so because for example if this is the flange, okay, here I am having this bolt circle, I will connect all these like this, okay. So that C is basically this value, C is this, okay.

And here I am having the bolt, which is half this way and half this way, therefore 2 x bolt radius I have added otherwise you can simply add the bolt diameter + 0.02, it means this section which is you can say edge of this bolt circle and this outer diameter or outer edge of the flange this value is at least 10 mm, therefore we have added 20 mm minimum. So considering this you can find out flange outside diameter, okay.

And further we will check for the gasket condition, where A b x S g/Pi G n should be less than 2 y, where A b is the actual bolt diameter where we will consider actual number of bolts into root area of that bolt. If this condition is not satisfied, gasket material should be changed.

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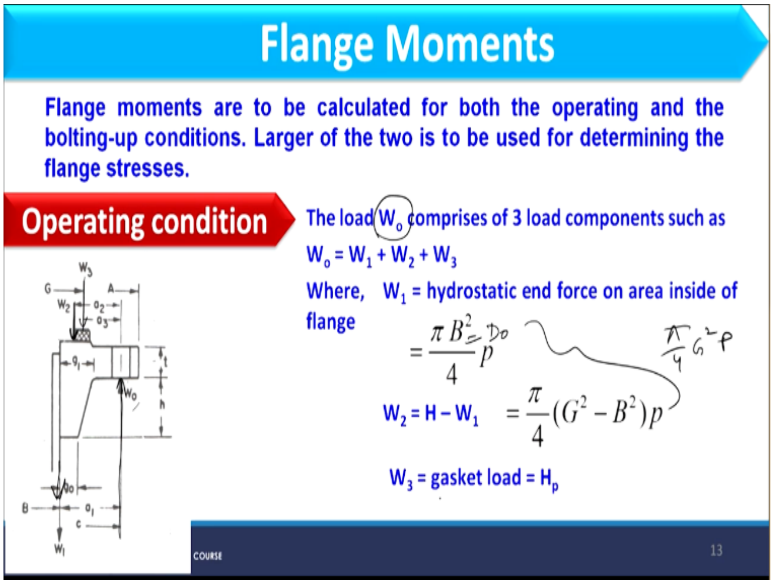

So in this way we have decided bolt circle diameter as well as flange outside diameter. The final parameter I have to compute is the thickness, okay, thickness of flange. So to calculate thickness of flange I have to focus on flange movements and again these flange movement will be calculated for operating condition and bolting-up condition separately. So for operating condition total load W o which comprises three components.

If you this schematic here whatever load is applicable on this flange that would be counter balanced by bolt. Therefore this W o basically having three different components, W 1 plus W 2 plus W 3, where W 1 is hydrostatic end force on the area inside the flange because when operation will be carried out it will try to depart the flange connection.

So depart flange connection how it will do because that one part of force total force will act here, another part of total load will act over here and third part will act at the centre of gasket. So W 1 would be the hydrostatic end force on the area inside that is Pi B 2/4, where B is basically **bow diameter** and then I am considering loose-type connection B is equal to D O, outer diameter of pipe or inner diameter of flange.

W 2 = H – W 1 that is Pi/4 (G 2 - B 2) p. Now if you consider this two expressions when we join this would be equal to Pi/4 G 2 x p. This is nothing but the force which we have considered during operating condition to calculate the area if you remember, okay. So this total load we have divided according to the point these are acting. And similarly I am having W 3 which is gasket load and that can be decided by H p which we have already found in area calculation.

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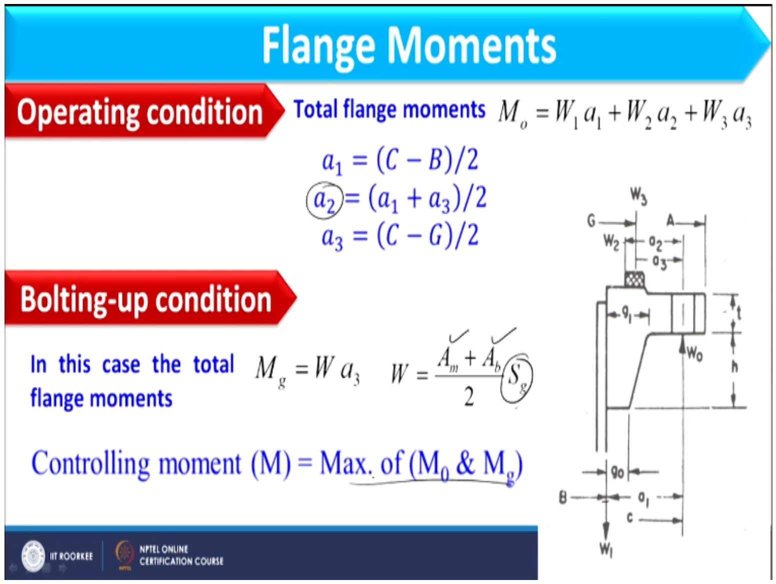

So total flange movement that would be calculated as M o, which is equal to W 1 a 1 + W 2 a 2 plus W 3 a3, so a 1, a 2 and a 3 you can see in this image, all these distances will be found on the basis of W o, so W o will act over here. This is a, this is a 1 you see and here I am having a 2 and here I am having a 3. So at what distance W 1, W 2 and W 3 will act from W o that would be given by a 1, a 2 and a 3.

And according to image, according to this figure you can find out a 1 as (C-B)/2, a3 as (C-G)/2 and a 2 as (a 1 + a 3)/2. In the similar line for bolting-up condition we have movement M g which is equal to W x a 3 because that gasket bolting-up condition will only consider the seating force and which will act a 3 distance apart from W o.

So here W can be defined as A m, which is minimum bolt area, A b is the actual bolt area that is actual number of bolts into root area of the selected bolt by 2 x S g, S g is basically the allowable stress of gasket material, okay. And maximum of M o and M g would be the controlling movement.

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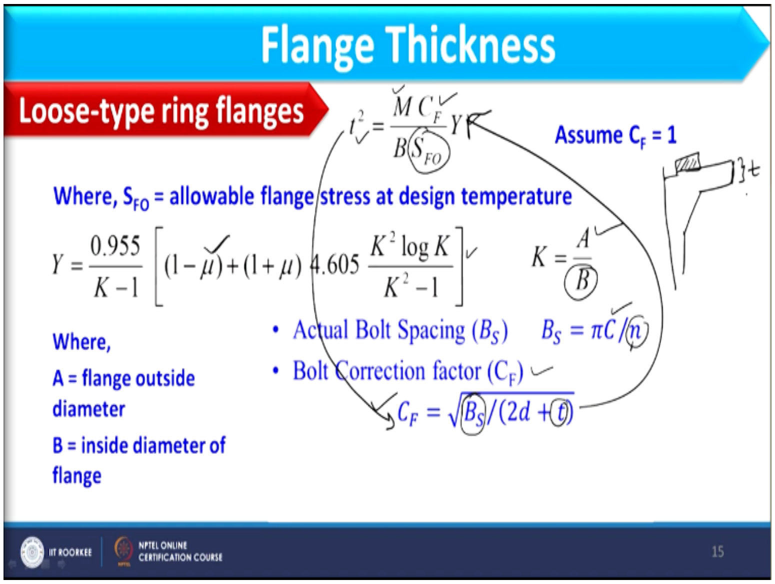

So once I have calculated the controlling movement, I will find out the flange thickness as t 2 which is equal to M C F/B S FO x Y. S FO is the allowable stress of flange material, understand B is basically the bow diameter or inner diameter of flange, which in this case is equal to D o and M which we have already seen in this last slide and C F is the correction factor and Y is the factor which is computed by this expression, where Mue is the Poisson's ratio and K is defined as A/B, A is the outer diameter of flange, B is the inner diameter of flange.

Now how you have to calculate. First of all we will consider C F as 1, okay, Y I have already calculated from this expression, other parameters I already know. So considering all these parameters I will find value of t. Now once value of t we can find we can calculate bolt correction factor that is C F and which is given by B s divided by 2 d + t. So B s is basically bolt spacing which is given by Pi C/n, where C is bolt circle diameter, which we have chosen corresponding to optimum bolt and n is the actual number of bolts.

So B s divided by 2d, d is the diameter of bolt and t is the thickness of this flange, okay. So considering this flange, we will find out C F value. Once I am having the C F value I will put the C F value in this expression and then I will calculate t and then that t value will further used in this expression to find out revised value of C F and then t and then C F till consecutive values of t and C F would be equal. So in this way we can complete design of flange where we have computed dimension of gasket, bolt circle diameter, number of bolts and outer diameter of flange and then thickness of flange, okay.

I hope you understand what is the meaning of thickness of flange. If I am having flange like this, this is the gasket, so this is the t which we have just discussed. So in this way whole design of flange you can carry out. And in the next lecture we will discuss a few examples to illustrate the procedure in better way. So that is all for now, thank you.