Equipment Design: Mechanical Aspects
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Lecture 05
Design of Shell

Welcome to the fifth lecture of week 1 and in this lecture we will discuss design of shell. Now if you remember the fourth lecture of week 1 there we have discussed about some of the terminologies which will be used in design of pressure vessel and those terminologies we will use in this particular lecture. So let us start the design of shell. Till now you must have the idea that whatever equipment we are going to design those are related to pressure vessels

And you must have remembered also that the pressure vessel will be designed in different components. We will design different components separately and then we will combine these components together to complete the design of pressure vessel. So first of all we will discuss what are the components and then we will start design of these components one by one.

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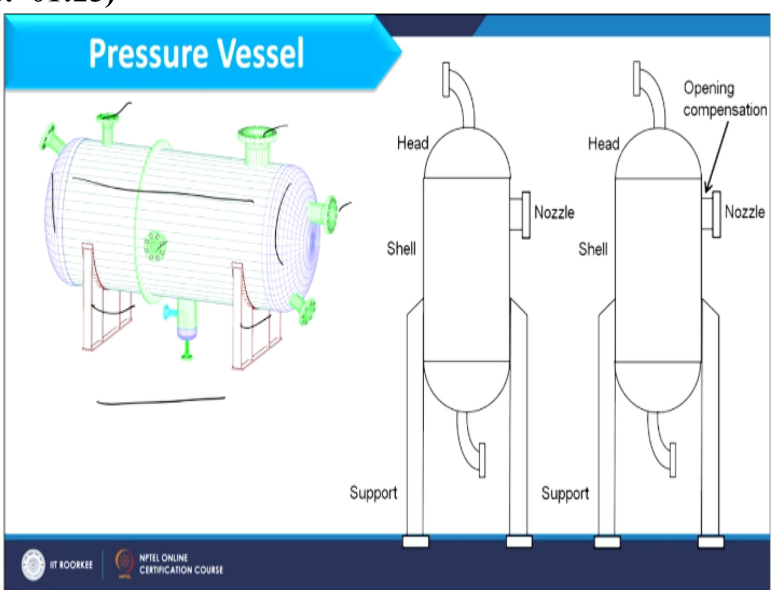

Now if you focus on this particular slide, here I am having this pressure vessel which is horizontally placed, and you see this is the shell and these are the heads and here we have the support and these are inlet, outlet nozzles. So it is horizontally placed. Another schematic I am having is the pressure vessel, which is placed horizontally okay. And it has different components which are written over here.

So here we will start design from the shell, and then we will cover heads and then nozzle, and then compensation that is related to the opening, which is placed in shell as well as head, and then finally we will discuss about the support to be placed in this pressure vessel. So here I am having different components. So first of all let us start discussion on design of shell, and as far as design of shell is concerned we will start design of cylindrical as well as spherical shells.

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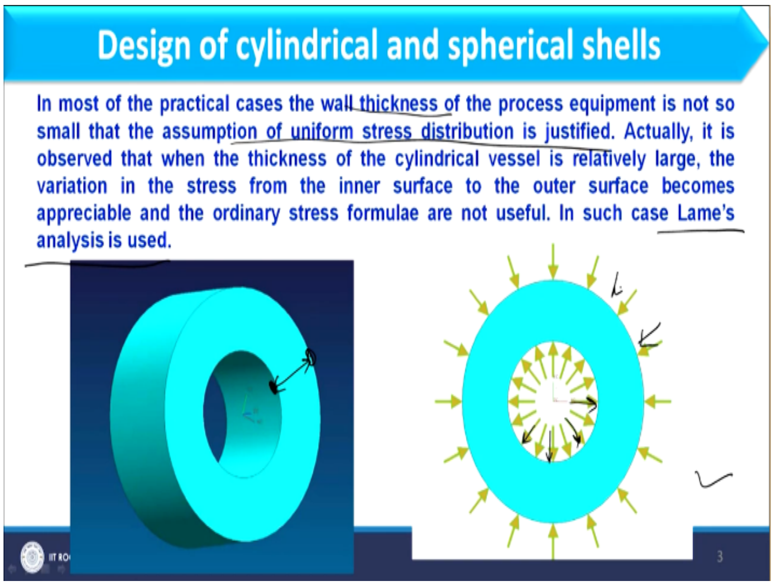

So in most of the practical cases the wall thickness of the process equipment. Now you must have the idea that what we are going to compute in designing, that is the thickness of different components. So here we will speak about design of wall thickness of shell okay. So in most of the practical cases, wall thickness of the process equipment is not so small that assumption of uniform stress distribution is justified.

Now if you remember the membrane stresses, where we have derived expression for computation of thickness for thin vessel, there we have considered very small thickness of the metal sheet or of the shell and that gives the uniform distribution of stress from inner surface to outer surface because that thickness is very small, and in that case we have considered only two stresses, hoop stress as well as longitudinal stress.

However, when I am having thick vessel, where thickness is significant in that case radial stress also matters along with hoop stress as well as longitudinal stress. So when I am having thick walled vessel, in that case whatever expression we have derived for membrane stresses those expression will not be applicable. So to design the thick walled vessel we will use different analysis and that analysis is known as Lame's analysis, as it is mentioned over here okay.

So here I am having the thick-walled vessel, this is the total thickness of the vessel where if I am considering stress at this point as well as at this point, stress will vary from inner surface to outer surface and that we need to consider in designing okay. And the cross sectional image of this cylinder is shown in this figure, where if you see inside this we have the pressure which we call as internal pressure, so in this case internal pressure as well as external pressure both are working simultaneously.

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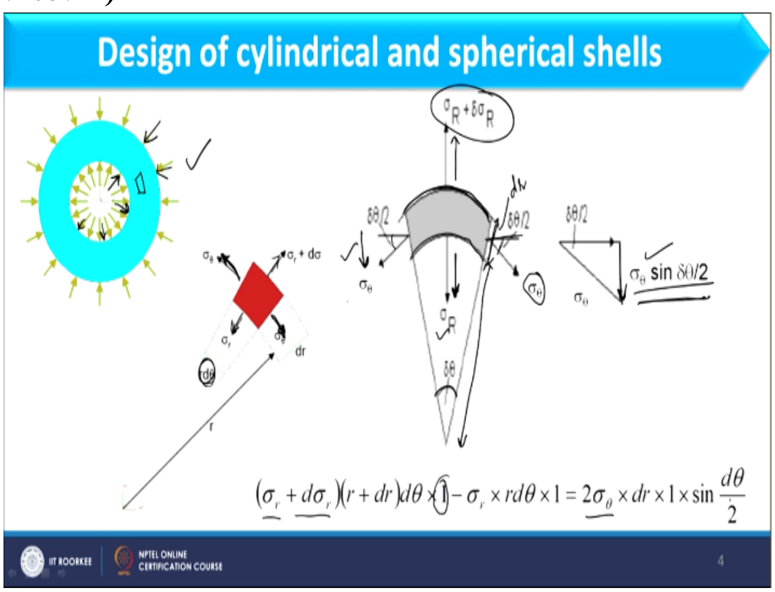

So as far as derivation is concerned, we will consider the cross sectional image, which we have just discussed and which is also shown over here where both pressures are acting where internal pressure as well as external pressure will act. Internal pressure if you consider in this image, it will try to expand this periphery, however, outer pressure or external pressure will try to squeeze this outer periphery. So both pressures are acting and this condition would be the extreme condition which can ever occur in a pressure vessel design and therefore we are starting design from this stage only.

Now to derive the expression for thickness of thick walled vessel, in this image we will consider a small section and around this small section we will make the balances. So this section I have shown over here, where if I consider inner part there stress is sigma r in radial direction and at outer side we have the stress sigma r + d sigma r. This sigma theta as well as this sigma theta will be considered as hoop stress.

And this particular section is making an angle with the centre and that will be nothing but d theta okay. And this image we have reproduced over here for better understanding where sigma r, sigma r + d sigma r, all parameters are shown and here we have the angle del theta. And please consider d instead of del for derivation purpose. So once I am having this d theta, this hoop stress is acting sigma theta in this direction and when I am considering straight line, so this angle must be d theta/2. And in the similar line the same angle is shown over here.

Now what happens when I make the force balance. Force in this direction will be balanced by force in this direction. So in this direction force will be due to sigma r as well as the vertical component of sigma theta will also act as it is shown over here. Let us see it is sigma theta. So vertical component of this will be sigma theta, sin d theta/2 okay. And the same stress will also applicable in this direction also okay.

So as far as balancing is concerned, we will consider the force balance. Now the force in upward direction that should be sigma r + d sigma r. If you consider this is the total stress and this should be the acting area. So acting area would be r because this is the r okay and here I am having this d r okay. So r + d r * d theta that would be this curve, and then L we can consider as unity, so we have considered 1 as length, that is the unit length we are considering over here. So the force over here is sigma r + d sigma r * r + d r d theta * 1 okay.

And that would be balanced by force due to this and force due to this as well as due to this okay. So that will be – sigma r and its acting area would be r, this r * d theta * 1 and that would be equal to this particular component of sigma theta and that would be 2 because both side it is acting. So 2 sigma theta * d r, d r is basically this distance into 1, that would be the length. So d r * 1 would be the acting area for this component into sin d theta/2. So in this way we make the force balance in this particular section of the thick walled vessel.

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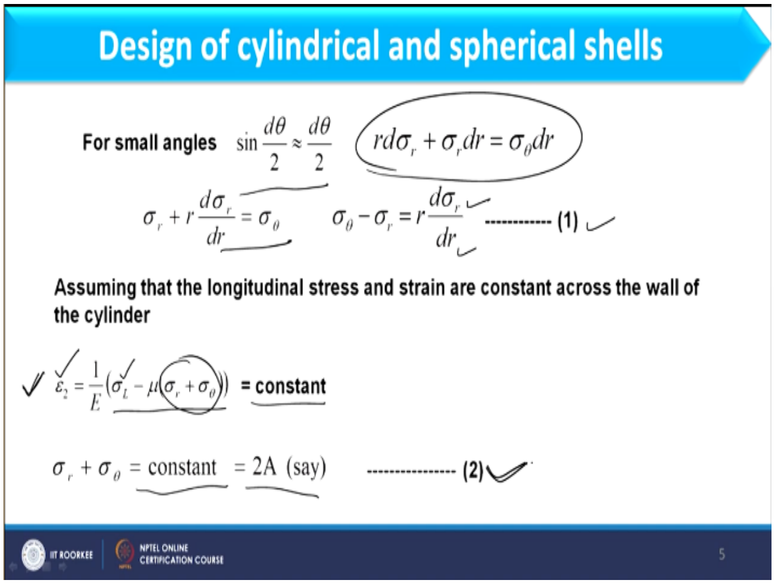

For small angles, we can replace sin d theta/2 with d theta/2. So considering this d theta/2 I am resolving the previous equation, we can get r d sigma r + sigma r d r = sigma theta d r. So this would be the equation after resolving the previous equation. Again, we can rearrange this, in this form, and then further we can rearrange this as sigma theta – sigma r = r d sigma r/d theta. So that I will consider as equation 1.

Now what happens when I am considering stress in longitudinal direction, that stress would be uniform throughout. Because whatever thickness will be there in longitudinal direction stress throughout the thickness will be uniform, and therefore because stress is uniform we can consider strain should also be uniform. Now if you remember the expression of triaxial system, which we have discussed in lecture 2 of week 1, that expression we will use over here to define stress and strain relationship in this system.

And that equation is given as it is mentioned over here. Where epsilon 2 = 1/E, sigma L – Mu, sigma r + sigma theta and that will be equal to constant. Now as we have discussed that longitudinal stress will be constant and accordingly strain in that direction will also be constant. So in this equation if you see epsilon 2 constant, E is constant, sigma L is constant, and Mu is constant. So only variable I am having is sigma r + sigma theta okay.

So further I can rearrange this in this form, sigma r + sigma theta equal to constant and that we can equate to 2A for derivation purpose, and this I have mentioned as equation 2. So here I am having equation 1 as well as equation 2.

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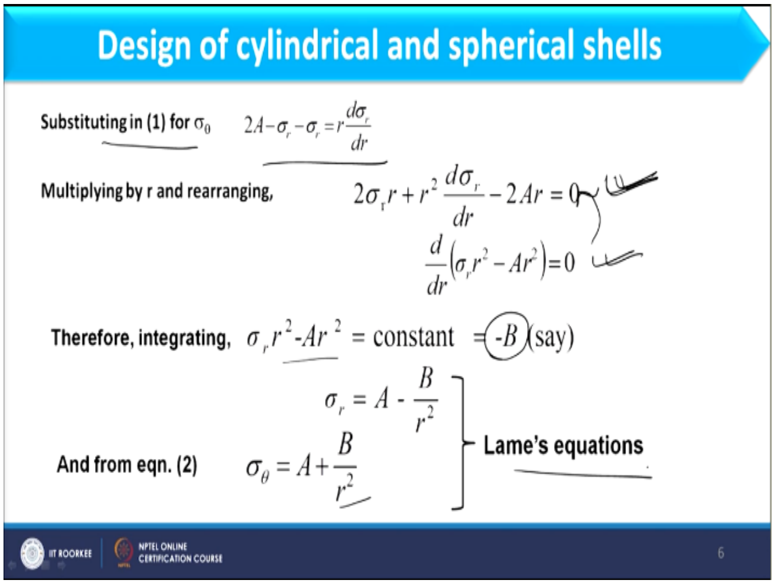

Now substituting in equation 1 for sigma theta we can remove 1 of the variable from equation 1 as well as equation 2. So we can substitute sigma theta in equation 1, and after substituting we can have expression like this.

And when we multiply both side to this equation by r and rearranging we can have this equation where this is sigma r and sigma r r added. So 2 sigma r r + r square d sigma r/d r – 2Ar = 0. Now what this equation shows. Further if I consider this equation in this form, where d/d r = sigma r r square – AR square and that should be equal to 0. If I consider this particular equation and if I derive this, I will get this equation okay.

So here I am having equation in derivative form and then we will integrate this derivative expression and we can find that sigma r, r square – AR square that should be equal to constant and that I have replaced with –B for derivation purpose. So here I am having sigma r, we can extract sigma r from this equation and we can rewrite like this. And here I am having another equation for sigma theta which is equal to A + B/R square. So if you consider these two equations are basically called as Lame's equations.

So for further analysis we will use these Lame's equations. And if you consider here I am having two constants A as well as B. Now putting conditions of actual system we can have, we can calculate, we can derive the expression of A as well as B.

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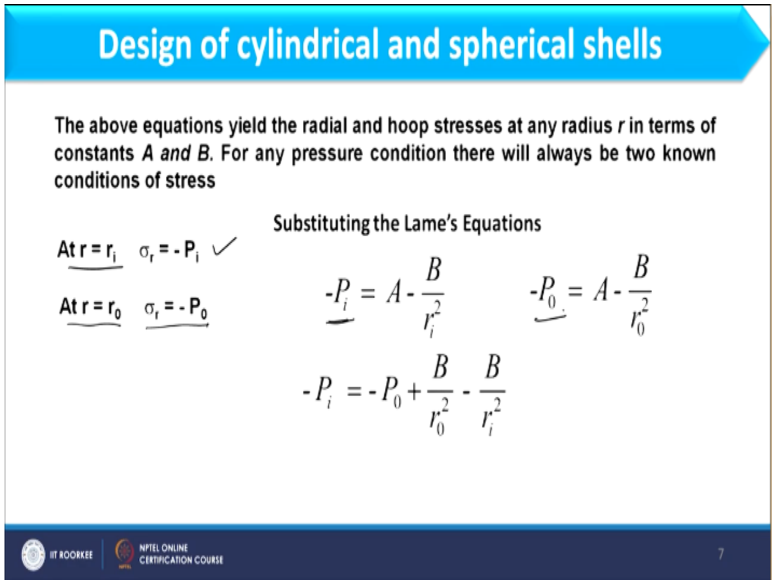

Now what would be the condition. If you consider the cross-sectional image of the thick walled vessel, where internal and external both pressures are acting. So that would be the object for which we need to derive the equation. So I will extract conditions from that image only.

Where if r = r I, sigma r would be –P i. Now why it is – P I because direction of sigma r as well as pressure are opposite to each other and further I am having when r = r not, sigma r will be equal to –P O, that is external pressure. So putting these conditions in Lame's equation we can write equation for internal pressure as well as external pressure like this.

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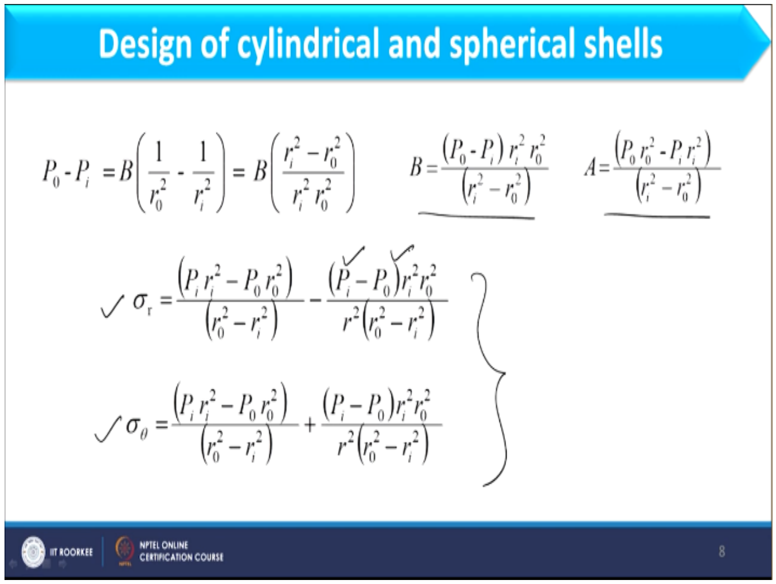

And further we can resolve this equation to find out the expressions for B as well as A, as these are mentioned over here okay. So these A and B expression I will further use in equations, which we have considered as Lame's equation and then we will go for final expressions. So let us do that. Putting A and B in Lame's equation I can find out the expression for sigma r as well as sigma theta okay.

So these two equations we can derive while putting the values of A and B. Now if you consider these two equations, where both pressures, internal as well as external pressures are acting simultaneously. However, when we operate the system that system may be internal pressure operation or external pressure operation, but here we will design the expression for internal pressure condition.

So when I am considering internal pressure condition, it means external pressure condition would be 0, so in that case P O would be 0, where that P O = 0 we will put in this equation and then we further resolve the equation.

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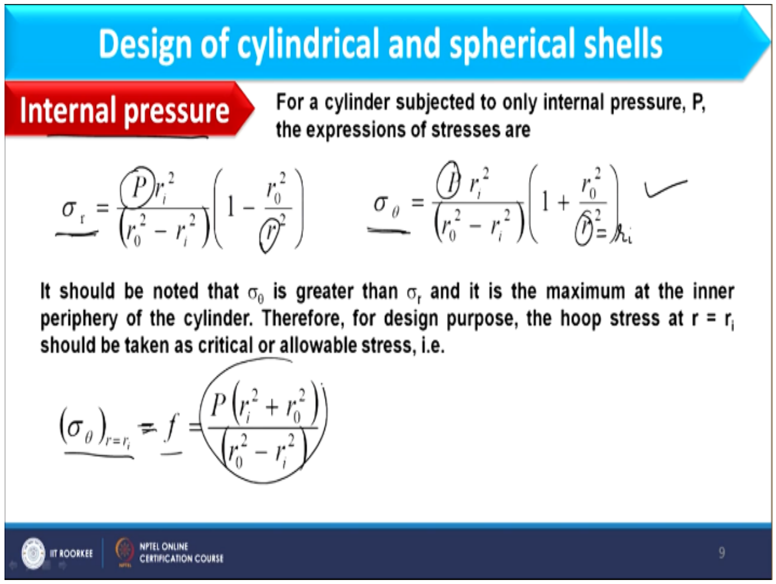

So when I am considering internal pressure only, external pressure would be equal to 0 as I have already mentioned. So putting P O = 0 and P I will be equal to P, we can have expression for sigma r as well as sigma theta okay.

Now if I focus on these two here, I have only one pressure that is internal pressure. And what is the variable, variable is this r in both cases, okay. Now if you consider expression for sigma r as well as sigma theta, which will give the maximum possible value. Obviously when we compare these two expressions, sigma theta will give the maximum possible value and that value would be maximum when this r will be equal to r i okay. I hope you are getting it.

So here I have to maximize the stress, which can be generated in the system okay. So sigma theta would be maximum when r equal to r i and that would be the maximum possible stress in the system, and that stress we can replace with the allowable stress value because at that value we will design the system okay. So that sigma theta if you sigma theta r = r i that we can replace with f and that would be equal to P (r i square + r o square)/(r o square – r i square).

This expression we have obtained by placing r = r i and that we can equate to allowable stress okay. What is allowable stress, that we have discussed in terminology in lecture 4 of week 1. And further if you consider the terminology lecture, there we have discussed another term, which is weld joint efficiency factor, and what was that. When we make any welding joint in a metal sheet, strength of that joint will be weaker in comparison to the regular sheet.

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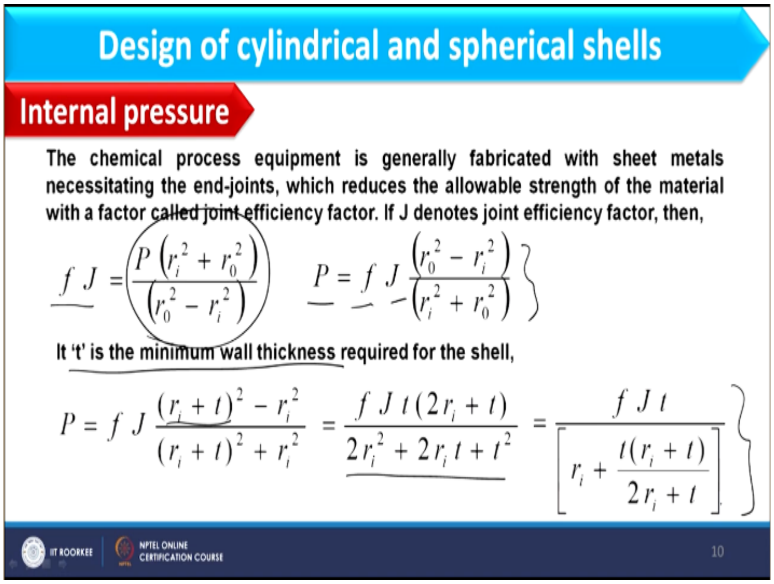

Therefore, as far as allowable stress is concerned, that we have considered for regular material. So that must be multiplied by J factor to get the actual stress okay. Therefore, f * J we have considered over here and right hand expression will be seen as we have discussed in last slide. And further resolving this we can extract pressure and that can be shown as a function of allowable stress J and radius.

Now what we need to find from this equation is the thickness of the shell. So if I consider t is the minimum wall thickness required for shell, so r o we can replace with r I + t and the whole expression became rewrite like this and we can further rearrange this expression, and then further resolving it we can have this particular equation, where P can be defined in terms of f J t and radius. So this expression we can further rearrange in this form, where P will be equal to f J t by this whole expression.

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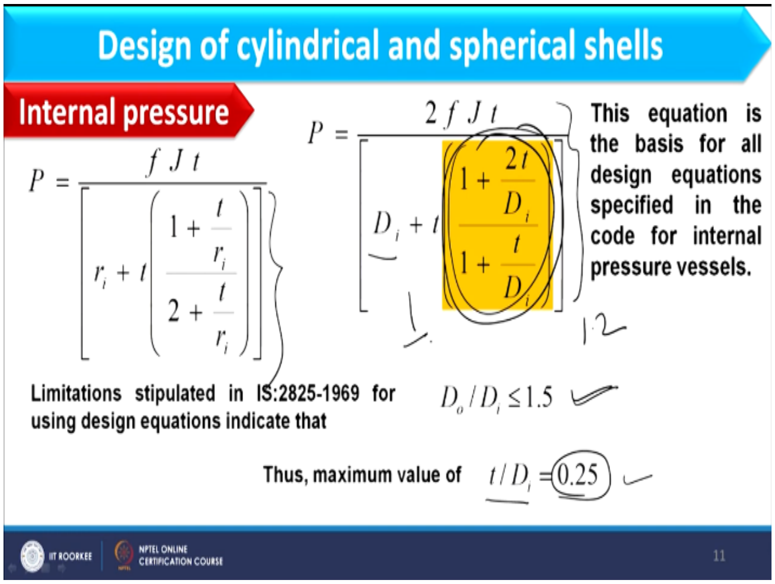

So this would be the basic equation for design of shell when internal pressure is acting. And this equation is also used as a basic equation in code, which is IS: 2825 okay. So further representing this equation in terms of diameter instead of radius, so we can have equation like this where P in terms of diameter is shown. Now as we have discussed that this equation will be used as a base for design of shell when internal pressure is acting.

So here we will apply the limitation of IS: 2825, and if you remember there is the limitation where D i/D o should be less than or equal to 1.5. So this is the limitation of IS: 2825, where D o/D i should be less than equal to 1.5. So this is the expression for this D o/D i less than 1.5. After resolving this, we can have expression of t/D i would be equal to 0.25. How this 0.25 has come because this Do we have replaced with D i + 2 t, and after resolving it, we can have t/D i as 0.25.

Now what happens when this t/D i 0.25 we will put, so this whole expression will become 1.2 okay. And that would be the maximum possible value of t/D i. If I am considering minimum value of t/D i that value would be 0. And if I put 0 over here, the whole expression will give the value as 1. So this whole expression vary from 1 to 1.2. So for derivation purpose we have considered this particular expression as 1 because when we are considering 0.25, it means that would be the maximum possible value, which should not exceed. So therefore instead of 1.2 we have taken value of this expression as 1.

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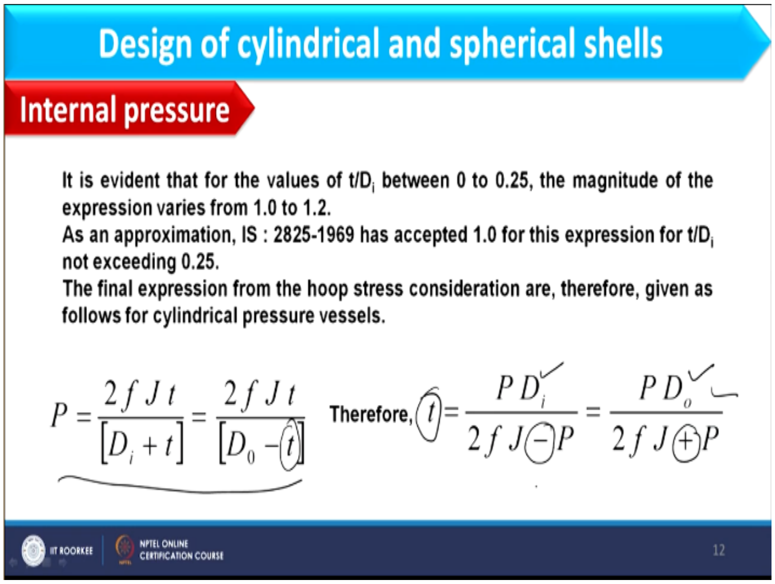

So this would be the final equation for pressure, where P is the internal pressure and that 2 f J t/D o – t because here the whole expression I have replaced with 1. So this would be the final equation and from this we can extract the term t, which is the minimum thickness that should be equal to P D i/2 f J – P and further it can be written as P D o/2 f J + E. So when I am considering outer diameter I have to take plus over here, when I am considering inner diameter I need to take minus over here.

So this expression is basically the minimum thickness of the vessel. Why I am calling this minimum thickness because at least this thickness should be provided to withstand the pressure P, therefore, it is called as minimum thickness.

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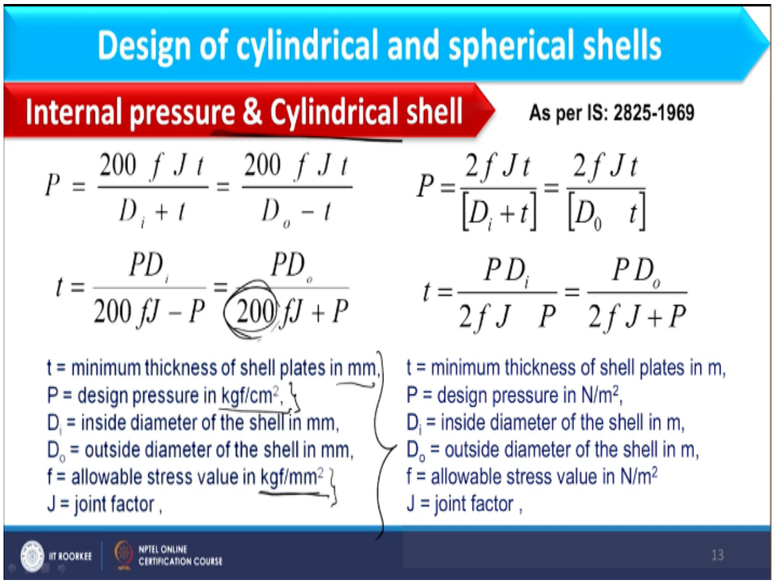

So here I am having the expression of internal pressure and cylindrical shell, and as per IS: 2825-1969, the expression given is this, like P = 200 f J t/D o – t and if I need to extract t from here, the expression should be P D o/200 f J + P.

Now how this 200 has come. Because when you see the nomenclature for this, here t is in mm and p is the design pressure in kg force per cm square and allowable stress that is f value is given as kg force per mm square. So due to these difference in units, pressure is shown in kg force per cm square and f is shown in kg force per mm square. Due to this we are having this 200 factor and J is the joint efficiency factor, which we have discussed in lecture 4.

So these expressions are basically given in IS: 2825-1969. However, in books you can find the expression 2 f J instead of 200 f J. So in books you can have this expression, where we have only factor 2 because there pressure as well as allowable stress both are shown in same unit that is in newton per meter square, and thickness we can compute in meter and other parameters are also shown over here.

So due to this difference in units, we can have different expressions in code as well as in book. And difference is only for the factor, where in code 200 is available and in books 2 is available, that is the only difference.

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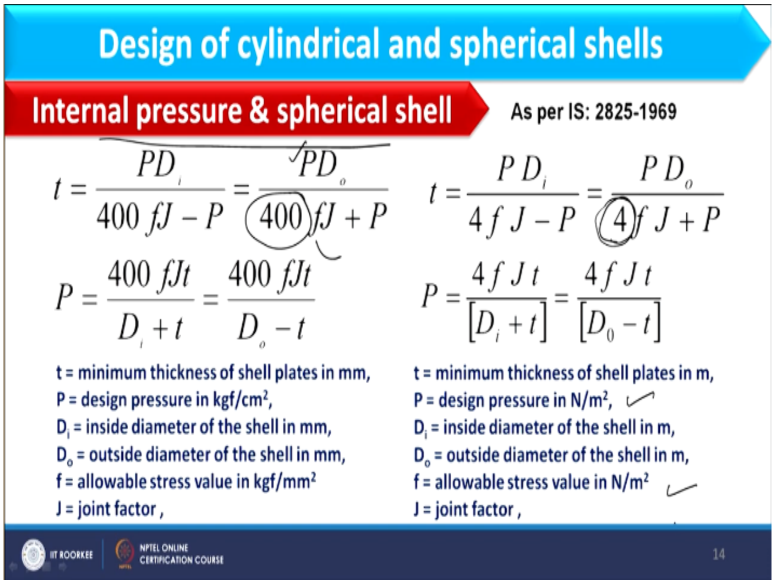

Further if I am considering internal pressure and spherical shell as per IS: 2825, here instead of 200 I have 400 because acting area for pressure will be different, so due to this this 400 appears. Further P and f will have different units and therefore this 400 is there.

Otherwise in books you can find 4 instead of 400 because pressure and allowable stress both are shown in same units. So in this way you can derive the expression for thickness of cylindrical as well as spherical vessel when internal pressure is acting okay. And we have already discussed that this t is the minimum possible thickness which should be provided to withstand pressure P. So that thickness is the minimum possible thickness that will be maintained throughout the operation of the plant, of the vessel okay.

Now what happens, if you remember the terminology lecture which is lecture 4 of week 1, there we have discussed another parameter which is corrosion allowance. Corrosion allowance is used because there are so many ways through which there are deposition or the material will extract from the metal sheet, all that wastage occur. So to account that we will have corrosion allowance.

So that corrosion allowance will be added to minimum possible thickness because as time passes only that thickness which we have provided as corrosion allowance that would be affected. Rest of the thickness will remain as it is to carry out the process in regular manner. So minimum thickness plus corrosion allowance we will add to get the thickness of vessel. Now what happens, after that, there are sheets available in the market, which have standard thickness. So whatever thickness are available above to that.

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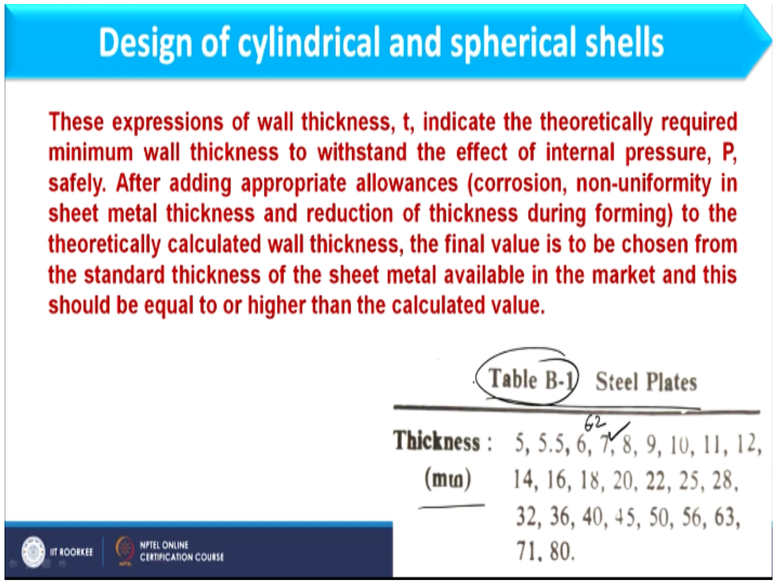

Above to that means t plus corrosion allowance, above to that whatever value is available that we will consider as standard thickness, and that value you can see from table B1 and here we have standard thickness of steel plates. So let us say I am having the value 6.2 for minimum thickness as well as corrosion allowance, so I will consider 7 as standard thickness. This table B1 is available in book B. C. Bhattacharya and you can refer this appendix B in that book. So in that way we will calculate.

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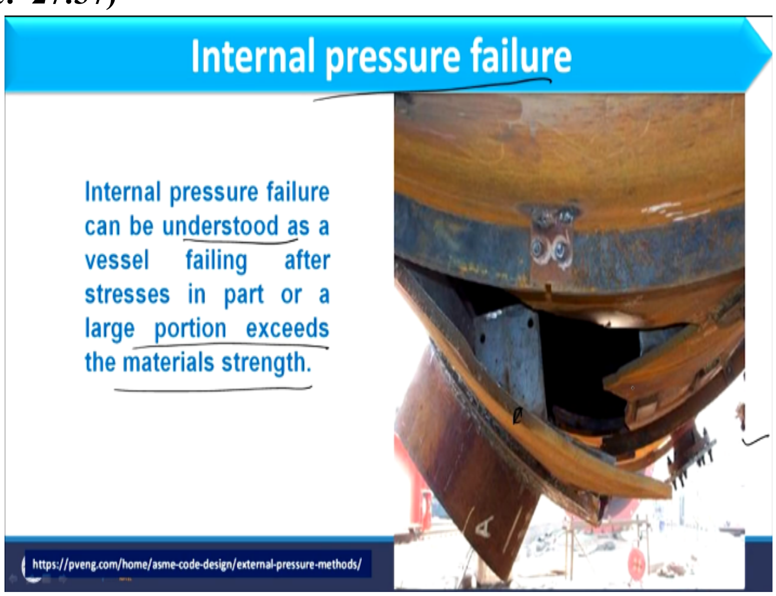

Now in this slide we will discuss the internal pressure failure. How that internal pressure failure will occur, that can be understood as vessel failing after stresses in part or a large portion exceeds the material strength. Whatever stresses are generated, if that exceeds the strength of the material then that failure will occur. So as far as internal pressure failure is considered, in that case bursting of the vessel will occur okay. As we have shown in this image, that you can imagine that thickness is so much but still failure will occur because stress will exceed the strength of the material

And that can be defined as damaging stress of the material. Till now we have derived the expression for thickness of cylindrical as well as spherical pressure vessel. And considering that we will solve a few examples, but before that I want to demonstrate that how code IS: 2825-1969 will be used. And I assume that this code will not be available to you, so that code I will provide to you so that you can use it, but now we will discuss that how it should be used.

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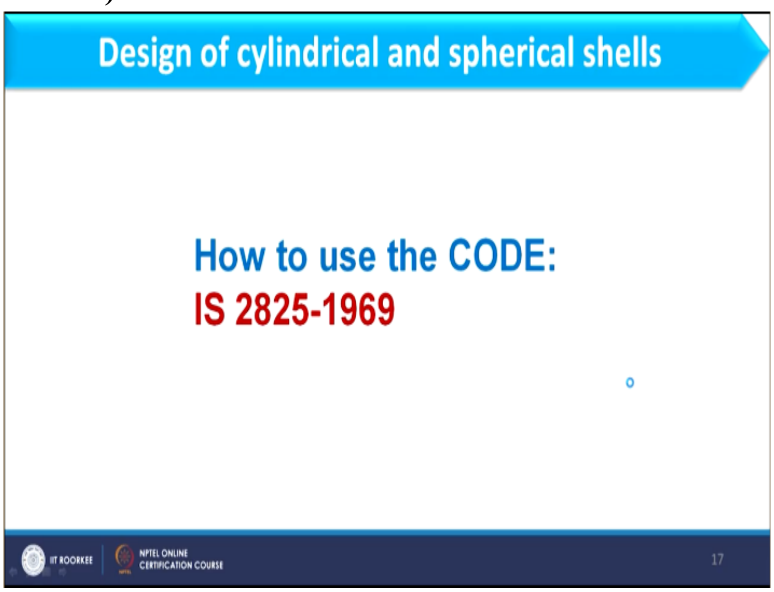

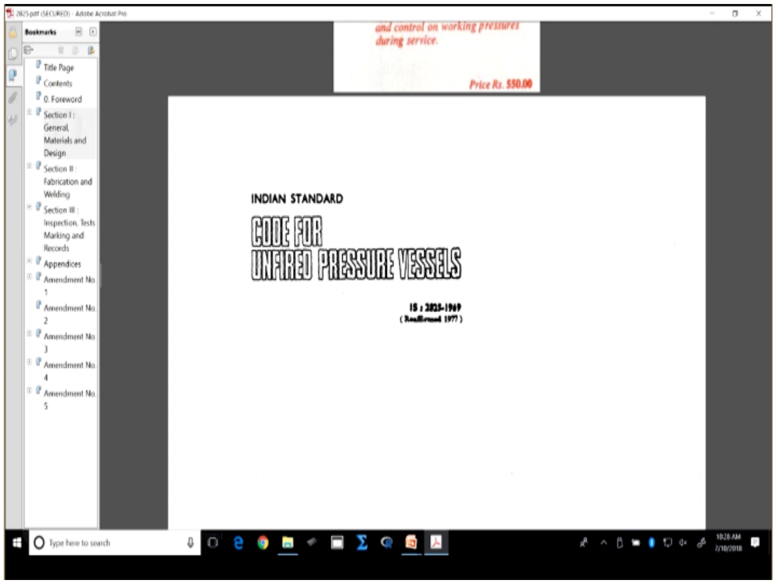

So if you see here, this is basically the code IS: 2825, which is made for code for unfired pressure vessel and which is given by Bureau of Indian Standards as we have discussed that these standards are given by governmental bodies, so in India that is Bureau of Indian Standards okay. So this is basically the code, where we are having IS: 2825-1969. And if you see the content of this code, here I am having section 1, section 2, and section 3, and as far as this particular course is concerned, we will only focus on section 1, which is on generals, materials, and design.

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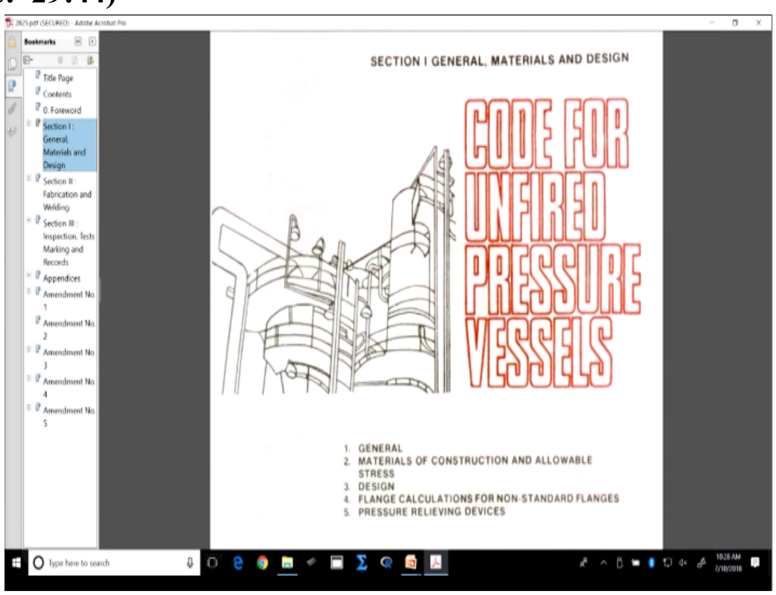

For design, section 1 should be considered. Therefore we are focusing in this only, where design are given like cylindrical as well as spherical shells, and then we have different section that we will discuss when times comes.

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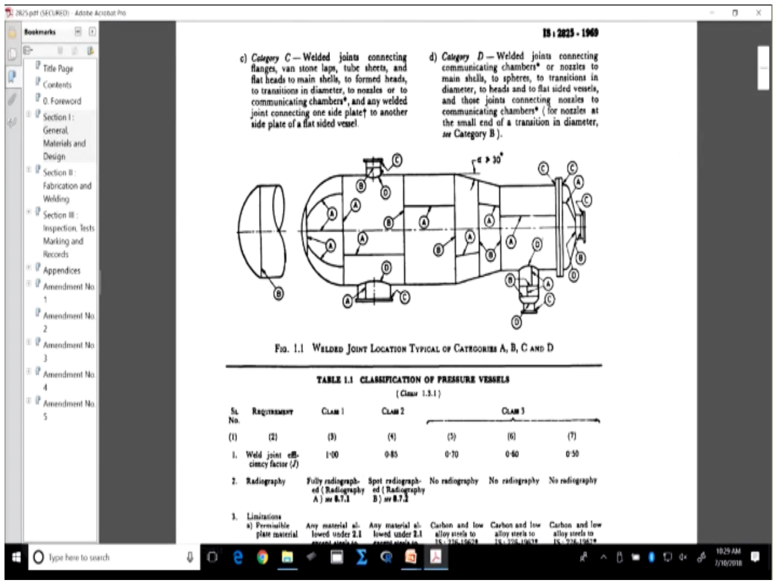

So if you consider this, here value of J are defined and if you remember this particular diagram, we have already discussed in lecture 4, and here we have table 1.1, which also we have discussed in lecture 4.

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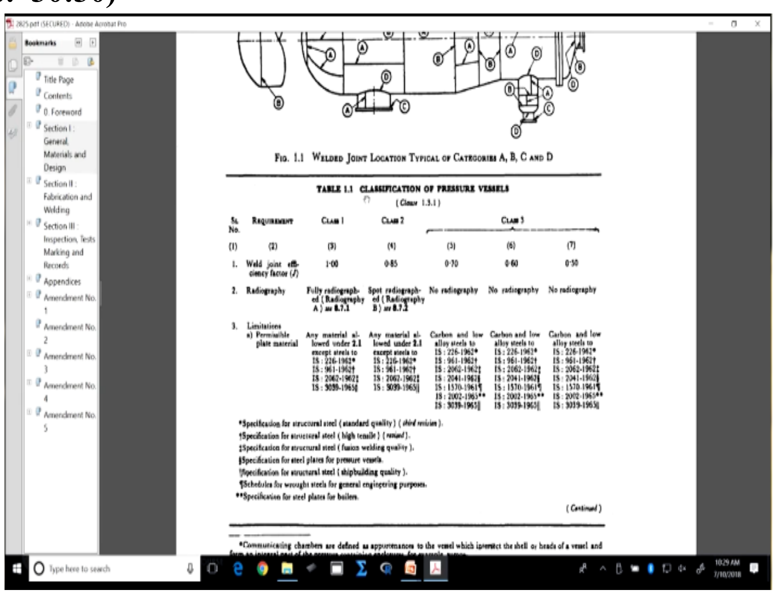

So in this way you can use this. There are so many parameters which are given over here okay.

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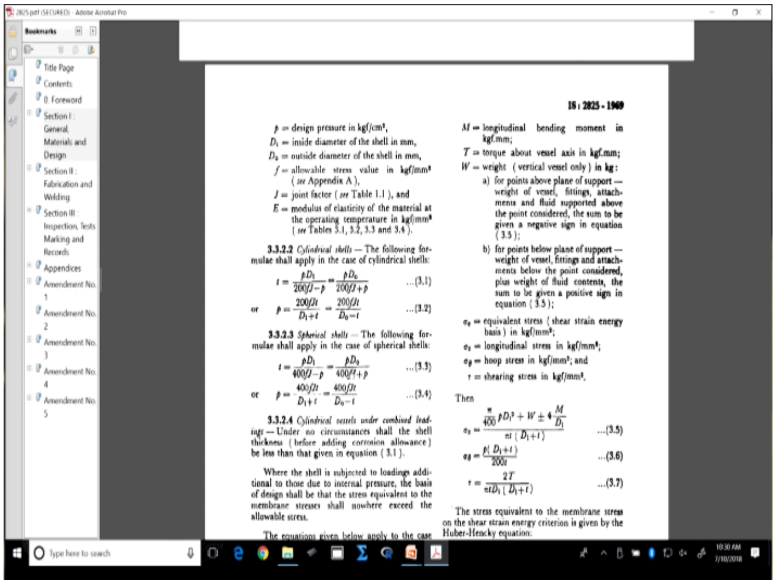

IS : 2825 - 1969

p = design pressure in kgf/cm²,
D_i = inside diameter of the shell in mm,
D_o = outside diameter of the shell in mm,
f = allowable stress value in kgf/mm² (*see* Appendix A),
J = joint factor (*see* Table 1.1), and
E = modulus of elasticity of the material at the operating temperature in kgf/mm² (*see* Tables 3.1, 3.2, 3.3 and 3.4).

3.3.2.2 *Cylindrical shells* — The following formulae shall apply in the case of cylindrical shells:

$$t = \frac{pD_i}{200fJ - p} = \frac{pD_o}{200fJ + p} \quad \ldots(3.1)$$

or

$$p = \frac{200fJt}{D_i + t} = \frac{200fJt}{D_o - t} \quad \ldots(3.2)$$

3.3.2.3 *Spherical shells* — The following formulae shall apply in the case of spherical shells:

$$t = \frac{pD_i}{400fJ - p} = \frac{pD_o}{400fJ + p} \quad \ldots(3.3)$$

or

$$p = \frac{400fJt}{D_i + t} = \frac{400fJt}{D_o - t} \quad \ldots(3.4)$$

3.3.2.4 *Cylindrical vessels under combined loadings* — Under no circumstances shall the shell thickness (before adding corrosion allowance) be less than that given in equation (3.1).

Where the shell is subjected to loadings additional to those due to internal pressure, the basis of design shall be that the stress equivalent to the membrane stresses shall nowhere exceed the allowable stress.

The equations given below apply to the case

M = longitudinal bending moment in kgf.mm;
T = torque about vessel axis in kgf.mm;
W = weight (vertical vessel only) in kg:

a) for points above plane of support — weight of vessel, fittings, attachments and fluid supported above the point considered, the sum to be given a negative sign in equation (3.5);
b) for points below plane of support — weight of vessel, fittings and attachments below the point considered, plus weight of fluid contents, the sum to be given a positive sign in equation (3.5);

σ_e = equivalent stress (shear strain energy basis) in kgf/mm²;
σ_z = longitudinal stress in kgf/mm²;
σ_ϕ = hoop stress in kgf/mm²; and
τ = shearing stress in kgf/mm².

Then

$$\sigma_z = \frac{\frac{\pi}{400} pD_i^2 + W \pm 4\frac{M}{D_i}}{\pi t (D_i + t)} \quad \ldots(3.5)$$

$$\sigma_\phi = \frac{p(D_i + t)}{200t} \quad \ldots(3.6)$$

$$\tau = \frac{2T}{\pi t D_i (D_i + t)} \quad \ldots(3.7)$$

The stress equivalent to the membrane stress on the shear strain energy criterion is given by the Huber-Hencky equation:

And if you see here I am having the expression for cylindrical shell and for spherical shell instead of 2 here 200 is available and instead of 4 we have 400 over here. And why this 200 and 400, that we have already discussed in this lecture only, okay. So in this way you can use this expressions.

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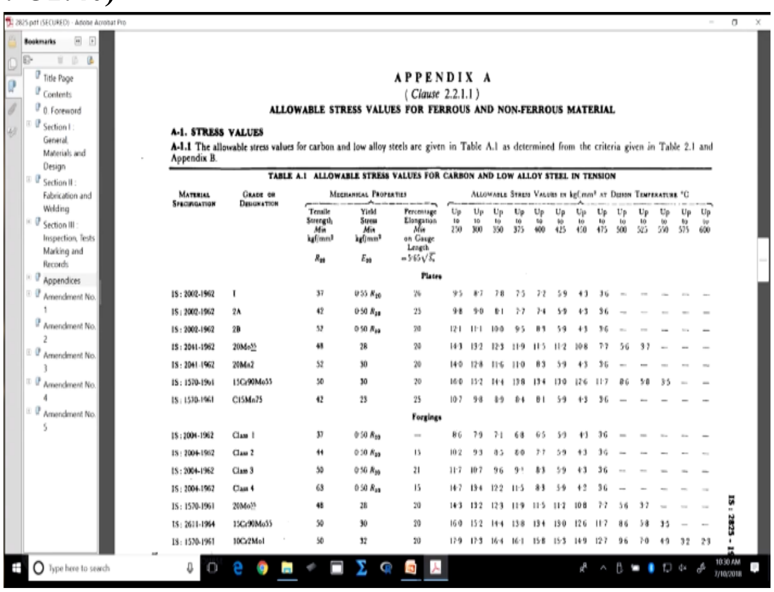

APPENDIX A

(*Clause* 2.2.1.1)

ALLOWABLE STRESS VALUES FOR FERROUS AND NON-FERROUS MATERIAL

A-1. STRESS VALUES

A-1.1 The allowable stress values for carbon and low alloy steels are given in Table A.1 as determined from the criteria given in Table 2.1 and Appendix B.

TABLE A.1 ALLOWABLE STRESS VALUES FOR CARBON AND LOW ALLOY STEEL IN TENSION

Material Specification	Grade or Designation	Mechanical Properties: Tensile Strength Min kgf/mm² R_{20}	Yield Stress Min kgf/mm² E_{20}	Percentage Elongation Min on Gauge Length $= 5.65\sqrt{S_o}$	Allowable Stress Values in kgf/mm² at Design Temperature °C: Up to 250	Up to 300	Up to 350	Up to 375	Up to 400	Up to 425	Up to 450	Up to 475	Up to 500	Up to 525	Up to 550	Up to 575	Up to 600
				Plates													
IS : 2002-1962	1	37	0·55 R_{20}	26	9·5	8·7	7·8	7·5	7·2	5·9	4·3	3·6	—	—	—	—	—
IS : 2002-1962	2A	42	0·50 R_{20}	25	9·8	9·0	8·1	7·7	7·4	5·9	4·3	3·6	—	—	—	—	—
IS : 2002-1962	2B	52	0·50 R_{20}	20	12·1	11·1	10·0	9·5	8·9	5·9	4·3	3·6	—	—	—	—	—
IS : 2041-1962	20Mo55	48	28	20	14·3	13·2	12·3	11·9	11·5	11·2	10·8	7·7	5·6	3·7	—	—	—
IS : 2041-1962	20Mn2	52	30	20	14·0	12·8	11·6	11·0	8·3	5·9	4·3	3·6	—	—	—	—	—
IS : 1570-1961	15Cr90Mo55	50	30	20	16·0	15·2	14·4	13·8	13·4	13·0	12·6	11·7	8·6	5·8	3·5	—	—
IS : 1570-1961	C15Mn75	42	23	25	10·7	9·8	8·9	0·4	8·1	5·9	4·3	3·6	—	—	—	—	—
				Forgings													
IS : 2004-1962	Class 1	37	0·50 R_{20}	—	8·6	7·9	7·1	6·8	6·5	5·9	4·3	3·6	—	—	—	—	—
IS : 2004-1962	Class 2	44	0·50 R_{20}	15	10·2	9·3	8·5	8·0	7·7	5·9	4·3	3·6	—	—	—	—	—
IS : 2004-1962	Class 3	50	0·50 R_{20}	21	11·7	10·7	9·6	9·1	8·3	5·9	4·3	3·6	—	—	—	—	—
IS : 2004-1962	Class 4	63	0·50 R_{20}	15	14·7	13·4	12·2	11·5	8·3	5·9	4·2	3·6	—	—	—	—	—
IS : 1570-1961	20Mo55	48	28	20	14·3	13·2	12·3	11·9	11·5	11·2	10·8	7·7	5·6	3·7	—	—	—
IS : 2611-1964	15Cr90Mo55	50	30	20	16·0	15·2	14·4	13·8	13·4	13·0	12·6	11·7	8·6	5·8	3·5	—	—
IS : 1570-1961	10Cr2Mo1	50	32	20	17·9	17·3	16·4	16·1	15·8	15·3	14·9	12·7	9·6	7·0	4·9	3·2	2·3

Now how you need to see the value of J, that you can refer table 1.1, which we have just seen. And then f value you can see from here. If you consider, here I am having appendices okay. And when we see these appendices, appendix 1 will be on allowable stress value for ferrous and nonferrous material. And screenshot of that I have also shown in lecture 4, while discussing allowable stress.

So this is I think you can resemble this table from there. There we have shown different materials and allowable stress. And along with this we have mechanical properties of that particular material also in this table. So in this way you can use different values in expression of thickness. Now we will solve a few examples to demonstrate how the thickness of cylindrical as well as spherical vessel will be computed.

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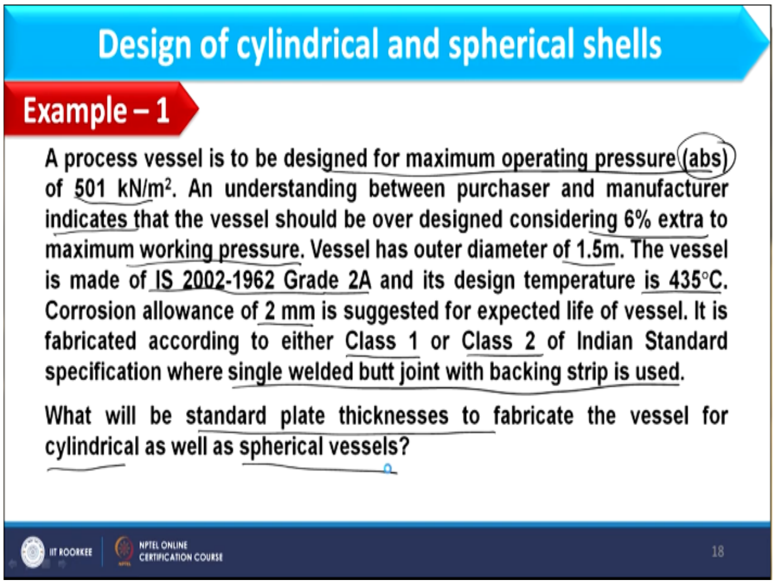

So let us focus on example 1. Where a process vessel is to be designed for maximum operating pressure absolute, you see here absolute is given, which is 501 kilonewton per meter square. And understanding between purchaser and manufacturer indicates that vessel should be over designed considering 6% extra to maximum working pressure. So usually we consider 5% extra, but here as it is required by the customer that 6% extra is required, so we will design accordingly.

So vessel has outer diameter of 1.5 meter. It is made up of the material IS: 2000-1962 grade 2A and its design temperature is 435. Corrosion allowance of 2 mm is given. Now here we need to design the vessel for class 1 or class 2 of Indian Standard specification, where single welded butt joint with backing strip is used. So this information is given to extract the value of j for class 1 as well as for class 2. What we need to find is, we need to compute the standard plate thicknesses to fabricate the vessel for cylindrical as well as spherical vessels.

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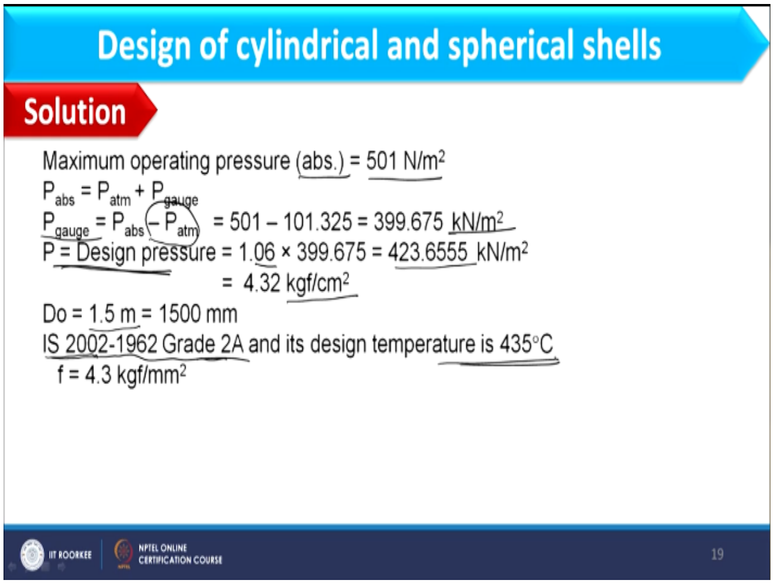

So let us focus on the solution of this. In this example, we are given maximum working pressure in absolute. However, if you see in expression of t we consider pressure in gauge. Now why we are considering in gauge. Because when we have equated external pressure to 0 okay.

What is the meaning of that. It means that when external pressure is 0, it means external pressure is 0 in gauge. In that case, outer pressure should always be atmospheric pressure and internal pressure would be higher than that okay. So when I am considering P o = 0, it means that I have accounted in gauge. So whatever P I = P I have considered that should also be taken in gauge.

Therefore in expression of t, whatever pressures are given that should be taken in gauge okay. So here maximum operating pressure in absolute is given 501, so first of all we will convert this into gauge, while deducting atmospheric pressure from the absolute so here we have 501 – 101.325, so 399.675 kilonewton per meter square is the maximum working gauge pressure internal.

Design pressure should be computed as 6% extra, so 1.06 * 399.675 and that should be equal to 423.6555 kilonewton per meter square and we need to convert that into kg force per cm square because in this unit it will be used in expression provided in code. So that value comes as 4.32 kg force per cm square. Outer diameter is given as 1.5 and material I have been given is IS:

2000-1962 grade 2A, and its design temperature should be 435. So this information is provided to extract the value of allowable stress.

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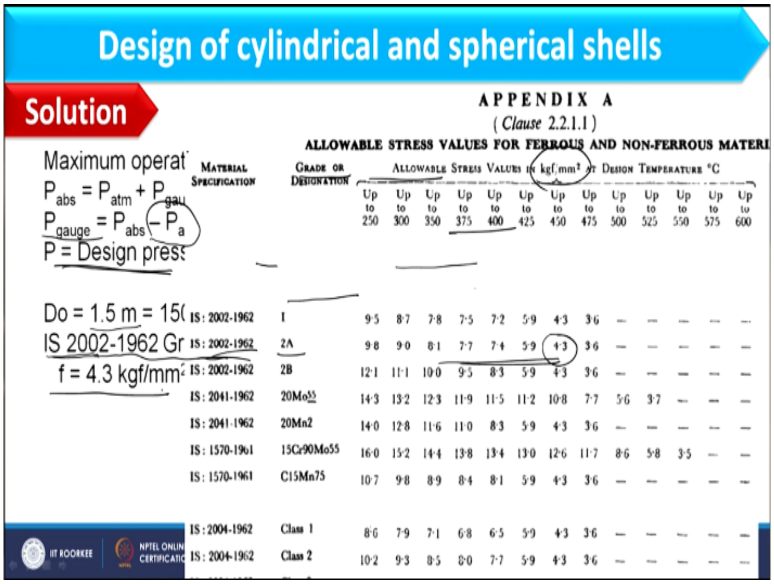

Material Specification	Grade or Designation	Up to 250	Up to 300	Up to 350	Up to 375	Up to 400	Up to 425	Up to 450	Up to 475	Up to 500	Up to 525	Up to 550	Up to 575	Up to 600
IS: 2002-1962	1	9·5	8·7	7·8	7·5	7·2	5·9	4·3	3·6	—	—	—	—	—
IS: 2002-1962	2A	9·8	9·0	8·1	7·7	7·4	5·9	4·3	3·6	—	—	—	—	—
IS: 2002-1962	2B	12·1	11·1	10·0	9·5	8·3	5·9	4·3	3·6	—	—	—	—	—
IS: 2041-1962	20Mo55	14·3	13·2	12·3	11·9	11·5	11·2	10·8	7·7	5·6	3·7	—	—	—
IS: 2041-1962	20Mn2	14·0	12·8	11·6	11·0	8·3	5·9	4·3	3·6	—	—	—	—	—
IS: 1570-1961	15Cr90Mo55	16·0	15·2	14·4	13·8	13·4	13·0	12·6	11·7	8·6	5·8	3·5	—	—
IS: 1570-1961	C15Mn75	10·7	9·8	8·9	8·4	8·1	5·9	4·3	3·6	—	—	—	—	—
IS: 2004-1962	Class 1	8·6	7·9	7·1	6·8	6·5	5·9	4·3	3·6	—	—	—	—	—
IS: 2004-1962	Class 2	10·2	9·3	8·5	8·0	7·7	5·9	4·3	3·6	—	—	—	—	—

So let us see the table of allowable stress. Here I am having the table of allowable stress. And IS: 2000-1962 grade 2A, this is the material. And 435 is the design temperature. So you see here, up to 450 we can use value 4.3 kg force per mm square okay. Therefore, f I have taken as 4.3 kg force per mm square.

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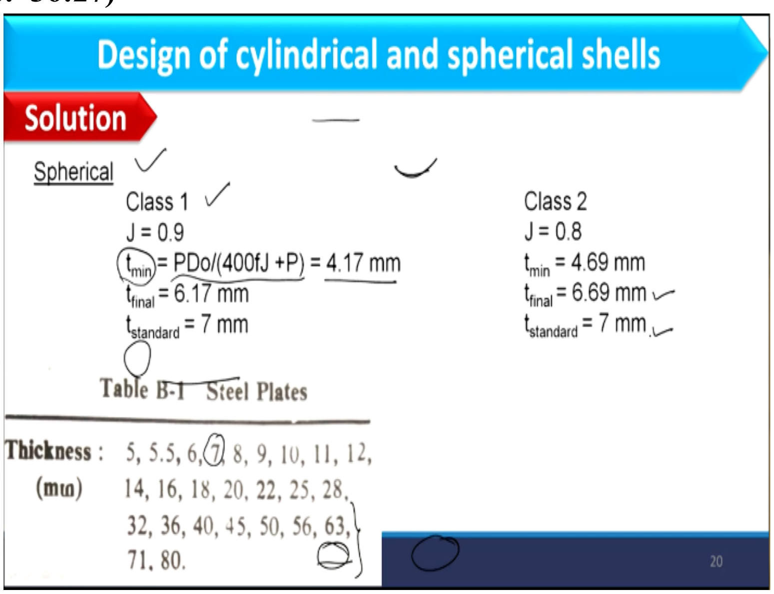

Now here we are designing spherical vessel where for class 1, we have to extract the value of J from table 1.1 from code. So let us focus on this.

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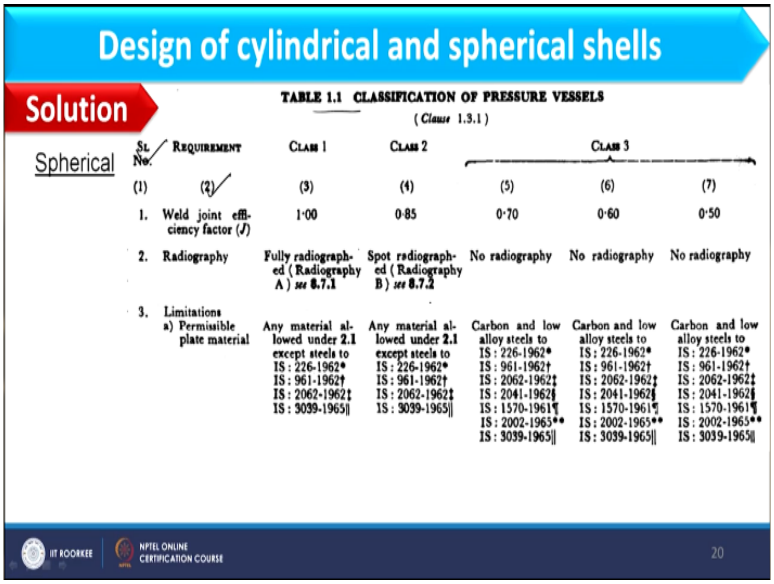

Here I am having table 1.1, where joint efficiency factors are already mentioned, but we need to find that corresponding to single welded butt joint with backing strip.

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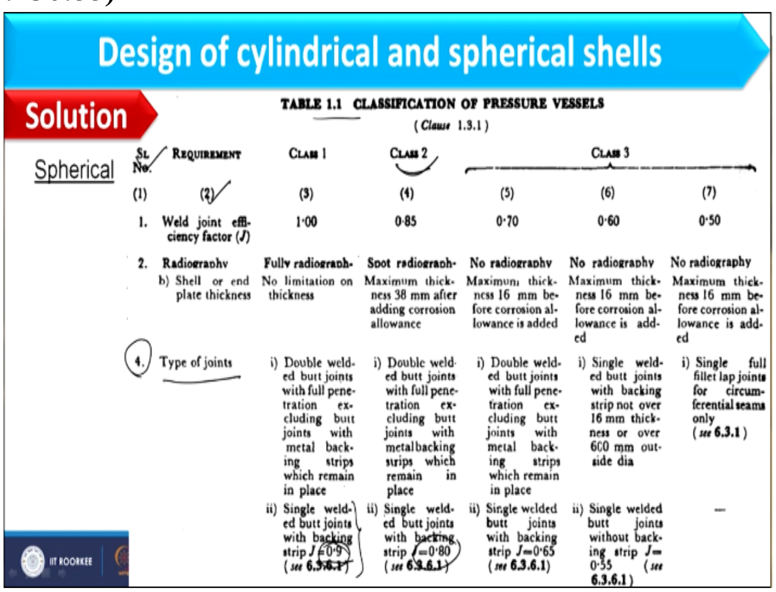

So when we further see this table, in second page of this table we have 0.4, where type of joints are mentioned, and here you see for class 1 single welded butt joint with backing strip is there and J I can take as 0.9 okay. And similarly for class 2 we can have value of J as 0.8 okay. So for class I am having 0.9, for class 2 I can take value as 0.8.

And considering the values of J, allowable stress, pressure and outer diameter, we can find out minimum thickness using this expression and it comes as 4.17 that is the minimum possible thickness. We will add corrosion allowance into this and that should be equal to 6.17. Now once I am having the value 6.17, the next value available in table B1 is 7, so 7 I can take as standard thickness.

In the similar line, for class 2 minimum thickness is observed as 4.69, adding corrosion allowance we can have 6.69, and further 7 we can take as standard thickness of spherical vessel for class 2.

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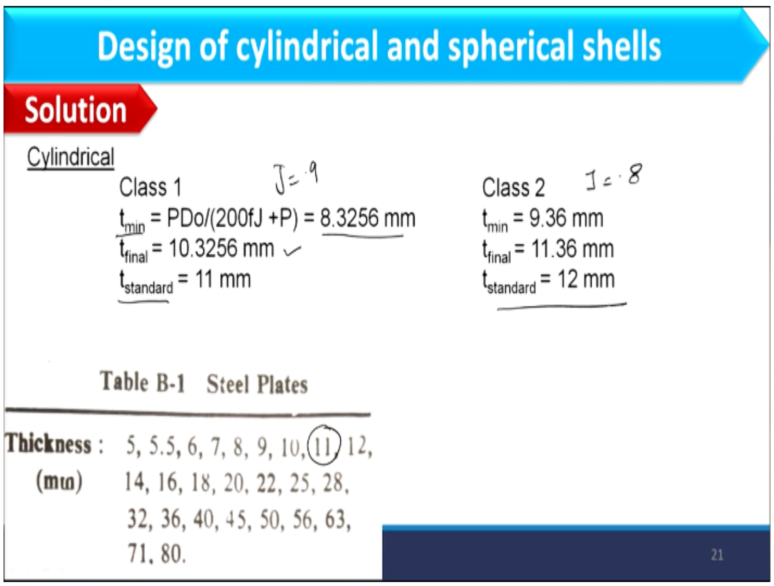

And similarly for cylindrical vessel for class 1 J value will remain 0.9 in this case and 0.8 in this case for class 2. So for class 1 we have minimum thickness as 8.3256, adding corrosion allowance we can have 10.3456. And then t standard I can take as 11, as it is next value, then 10.3256. And similarly for class 2, I can find standard thickness as 12 mm. So in this way we can calculate the thickness of cylindrical as well as spherical vessel okay.

So minimum thickness, thickness with corrosion allowance and standard thickness. These three thickness we need to find for any system. And further we will discuss example 2 for better understanding.

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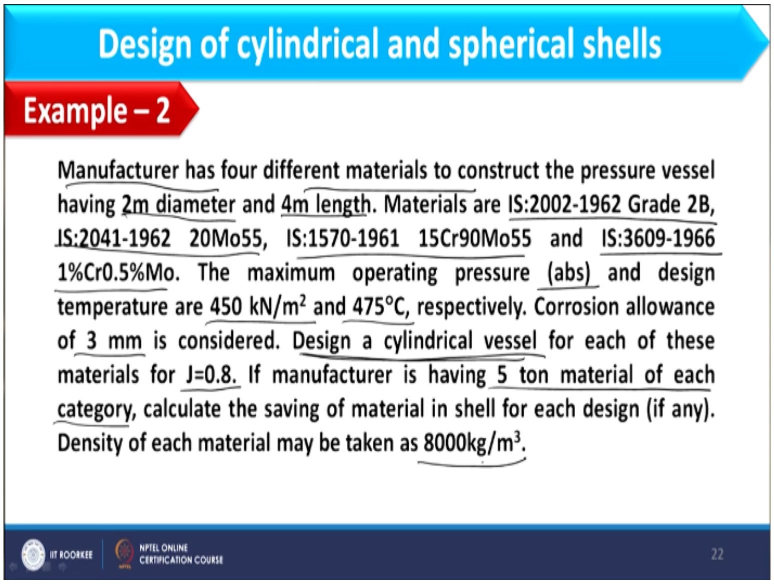

And in this example, manufacturer has four different materials to construct the pressure vessel having 2 meter diameter and 4 meter length, as diameter and length are defined, we can consider this as a cylindrical shell. Materials are IS: 200-1962 grade 2B. This is another material and fourth material is this. So all these four materials are used to construct the pressure vessel.

The maximum operating pressure absolute and design temperature are given as 450 kilonewton per meter square and 475 degrees Celsius. Corrosion allowance of 3 mm are considered. In this problem, we need to design a cylindrical vessel where j factor is given as 0.8 and if manufacturer is having 5 ton material of each category, calculate the saving of material in shell for each design. Density for each material we have considered as constant and that is 8000 kg per meter cube.

So in that case what we need to find is the saving in each material and how I can find that saving, once I will calculate the minimum thickness, thickness with corrosion allowance, and standard thickness. Considering that standard thickness, we will calculate how much material is used to prepare the shell. And then that material will be compared with 5 ton, which is given in the problem for each category of the material. So let us start the solution of this problem.

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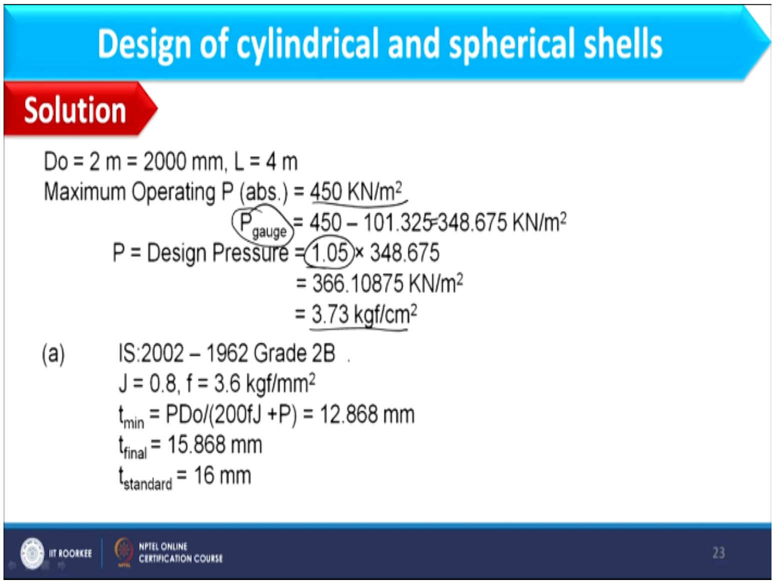

So here maximum operating pressure 450 is given. We will first convert that into gauge and then we can consider 5% extra. Why 5% extra because here I do not have any guideline from the customer, so I will take at least 5% extra pressure as design pressure okay. So here design pressure comes as this. Now IS: 2000-1962 grade 2B is the material okay.

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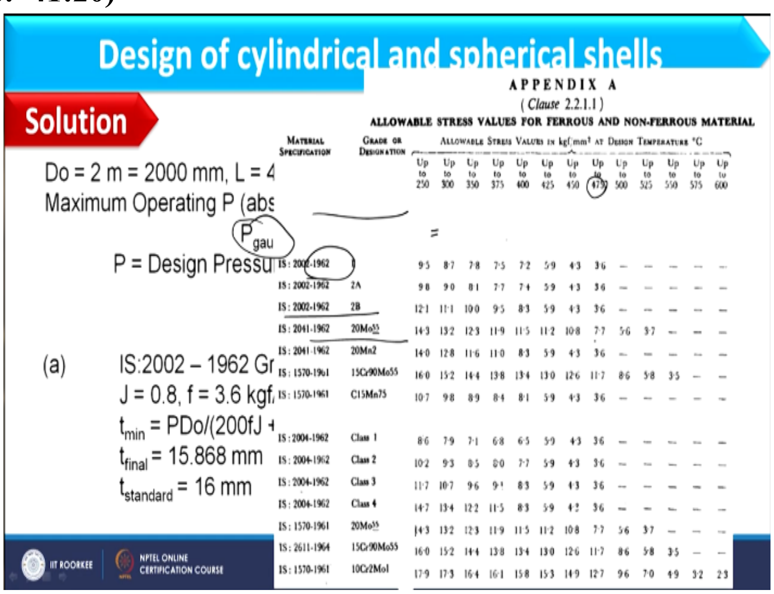

And here in allowable stress table, if you see this is the material, where 475 is the design temperature, so corresponding to this, 3.6 kg force per mm square can be taken as allowable stress. So that is given over here. Considering this expression, we can find minimum thickness, J factor I already know as 0.8, other parameter we have already extracted from table or formulae and then adding 3 mm to this as corrosion allowance, we have thickness 15.868.

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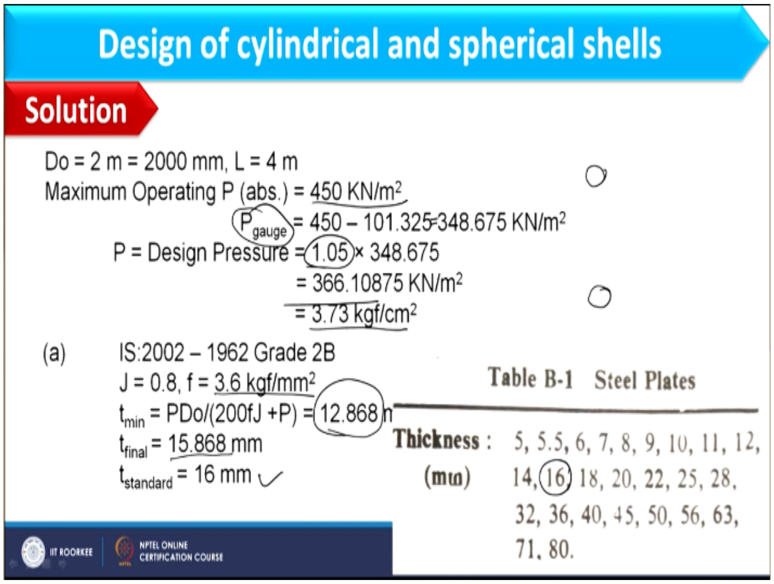

And then you can see table B1 where next value, then this is available as 16. So 16 mm we have considered as standard thickness.

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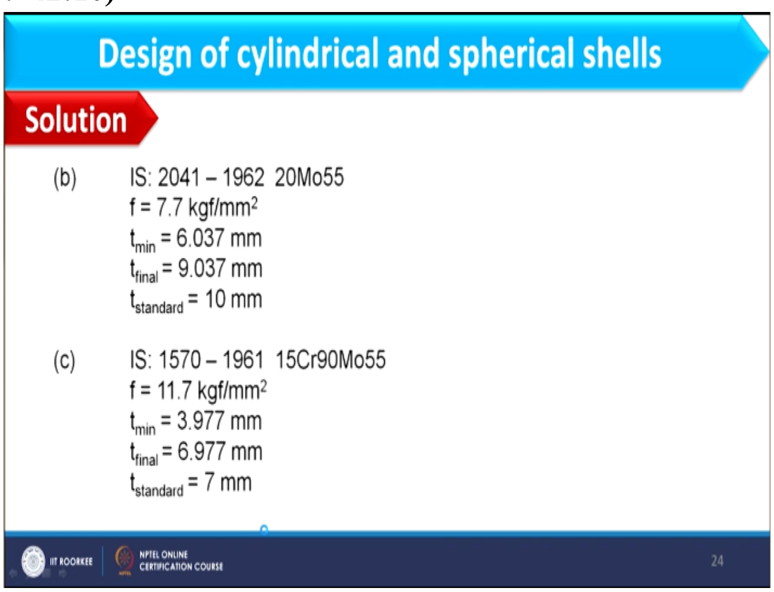

In the similar line we can compute the thickness for other material okay.

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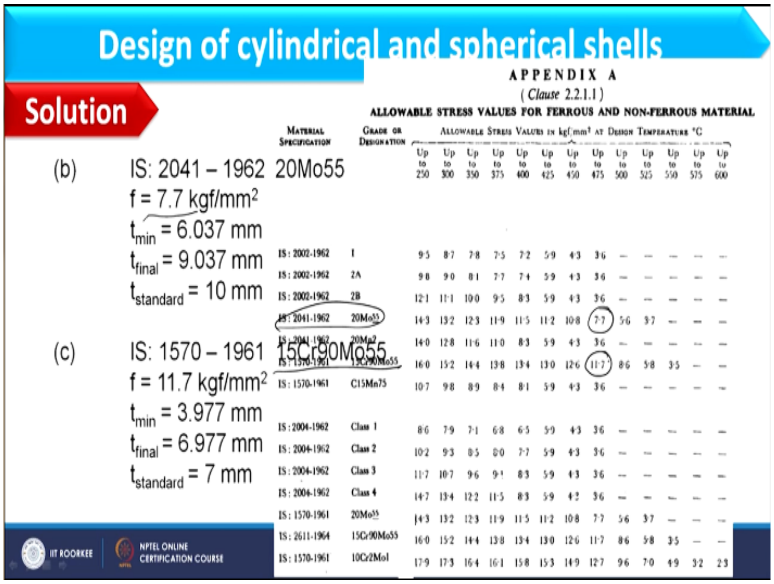

And here you see material is IS: 2041-1962, 20 molybdenum 55, so this is the material and corresponding to 475, 7.7 is the value, which we have considered over here. And similarly IS: 1570-1961, 15 chromium, 19 molybdenum 55, so this is the material. And corresponding allowable stress value is 11.7 at 475 degrees Celsius. So that we have taken here like this, and then we can compute the standard thickness.

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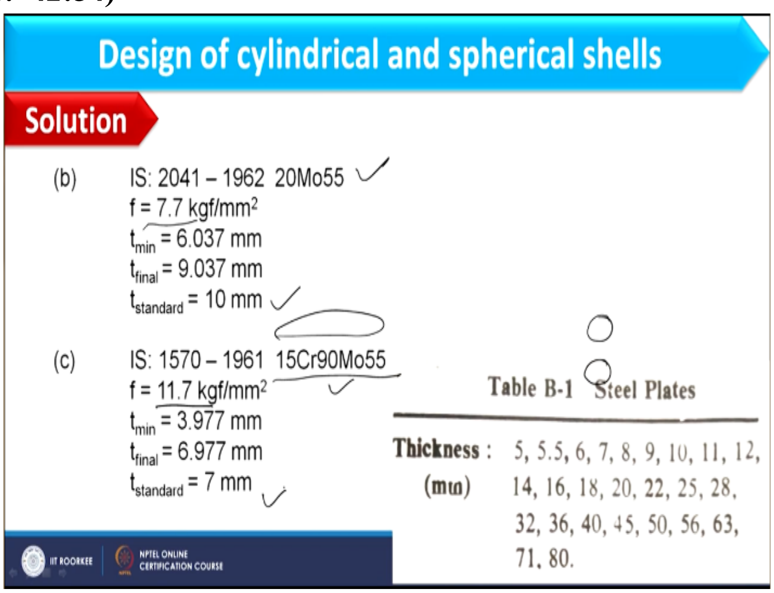

For this material we have 10 mm as a standard thickness, and for this material we have 7 mm as standard thickness.

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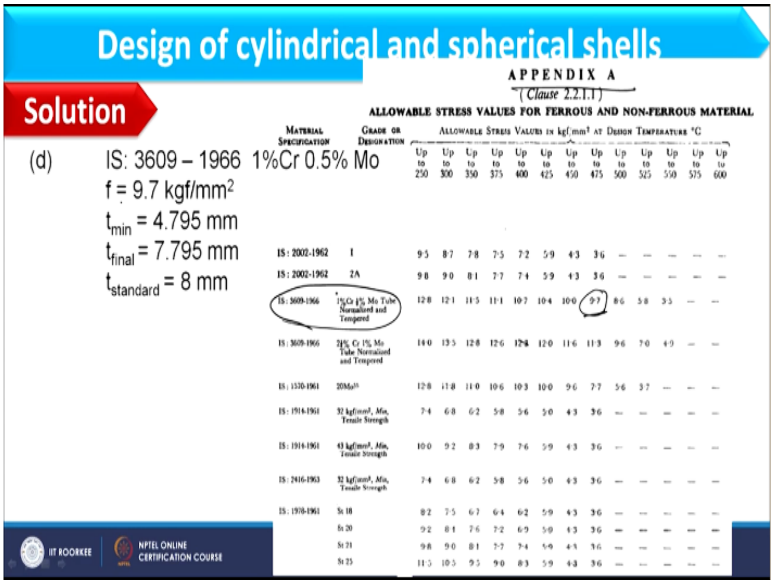

And similarly we can compute for material D and where this material that is IS: 3609-1966 1% chromium, 0.5% molybdenum. It is available in next page of this table. So here you can find this material and corresponding to 475 we have f value as 9.7 okay. So considering this we can compute the thickness for the material and then it is coming as 8 because 7.795 is the thickness after adding corrosion allowance and next value to this is 8 mm okay.

So in this way we have computed all thickness of four material. And now we will speak about the savings. How I can find the savings, I need to find out the volume of material of each category, which is used to design the shell and then we will multiply that with the density and then we will compare that with 5 ton material available.

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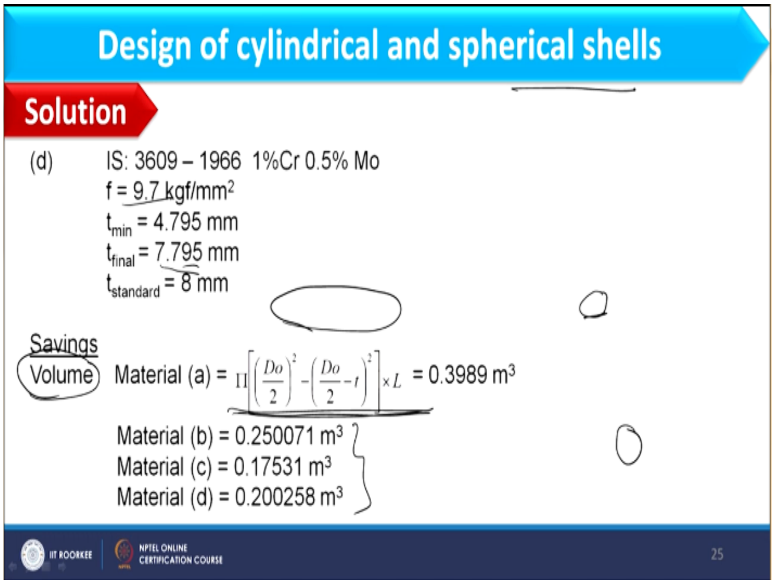

So for saving, we need to calculate the volume. Volume you can calculate by this expressions, that is a common expression. And for material A we have 0.3989 mete cube as a volume, and similarly we can consider volume for other materials also.

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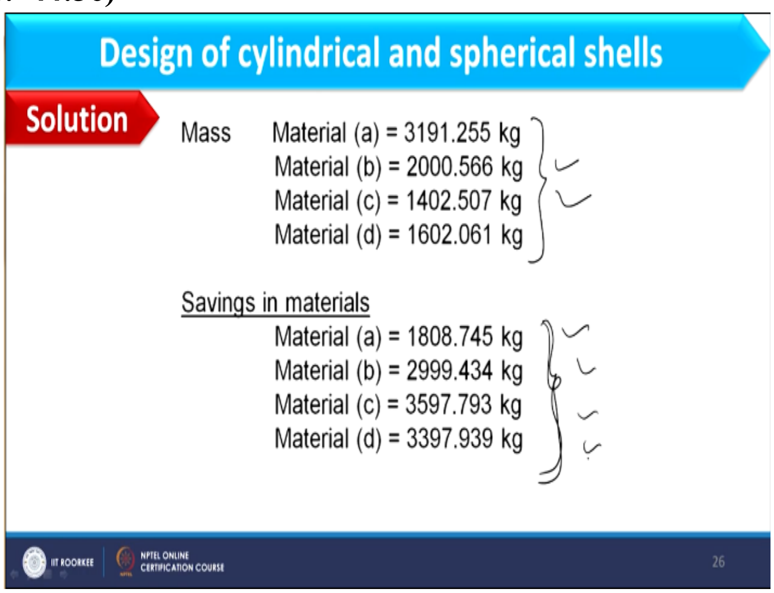

Multiplying that volume with the density, we can find mass of each material as these are shown over here. And when we compare these with 5 ton, we can find out that all material are below 5 ton, so we can have savings in each material in these magnitude as it is shown over here. So in this way we can find out the savings in the material, which is used to design the shell okay. So here we have discussed two examples.

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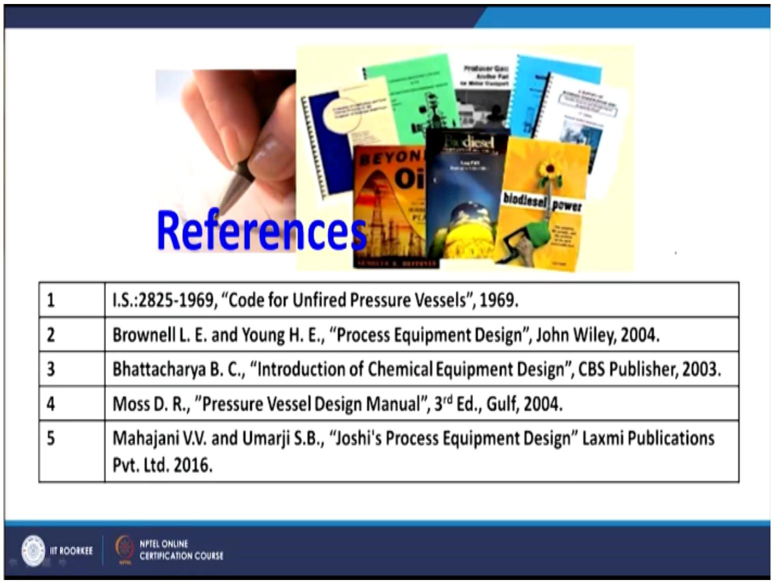

And here we have the reference books. This you can use for your study purpose and IS: 2825, how it will be used that we have already demonstrated. And you can follow other books also okay.

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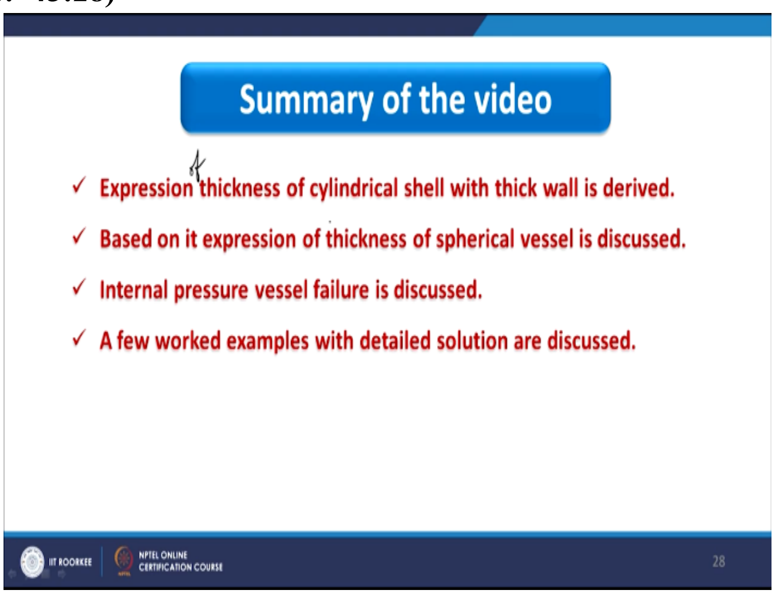

Now we have summary of this video as expression of thickness of cylindrical shell with thick wall is derived, based on it expression of thickness of spherical vessel is discussed, and internal pressure vessel failure is discussed, a few examples with detailed solution are discussed. And that is all for now, thank you.