Design of Machine Elements – I

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Lecture No - 37

Design of Cylinders and Pressure Vessels - II

dear student let us begin lectures on machine design part one this is lecture number thirty-seven and the topic of today's discussion is design of cylinders and pressure vessels this is part two of the lecture

in last lecture that is in lecture thirty-six we have learnt how to design a pressure vessel when the pressure vessel is modeled as a thin cylinder now we knew about two kinds of stresses one is Hoop's stress or circumferential stress and another is longitudinal stress

now designing such pressure vessels that is designing ah the proper thickness of such pressure vessel is very very important because ah if it is not designed properly then that may lead to total catastrophe

and there are various codes available one such code as i mentioned is ah ASME code for boilers and pressure vessels

now today we shall discuss ah how to design a cylinder when the cylinder is thick

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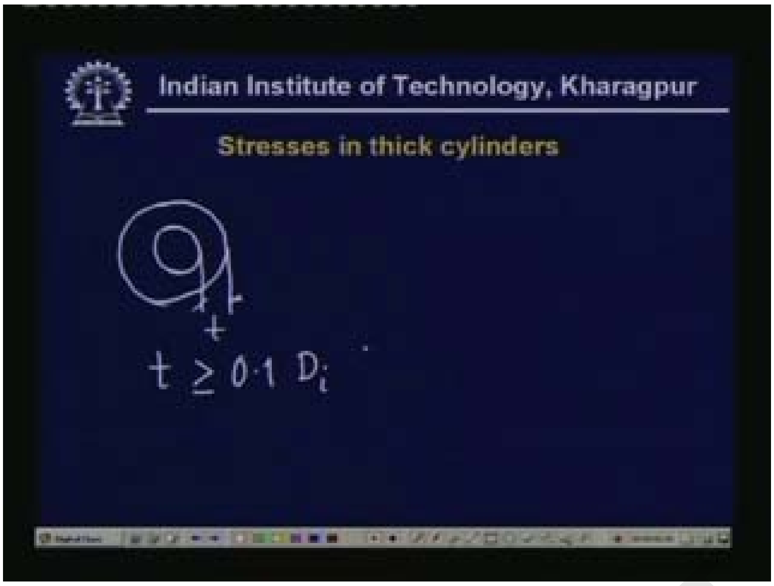

now [noise] let us see what are the stresses in a thick cylinder

now i want to remind you that [noise] a thick cylinder is a cylinder where the thickness that is this distance t is greater than equal to point one of internal diameter Di [noise] so if this is satisfied then we call this {cyl} (00:02:55) cylinder to be a thick cylinder

now let us look at what are the stresses in such a cylinder

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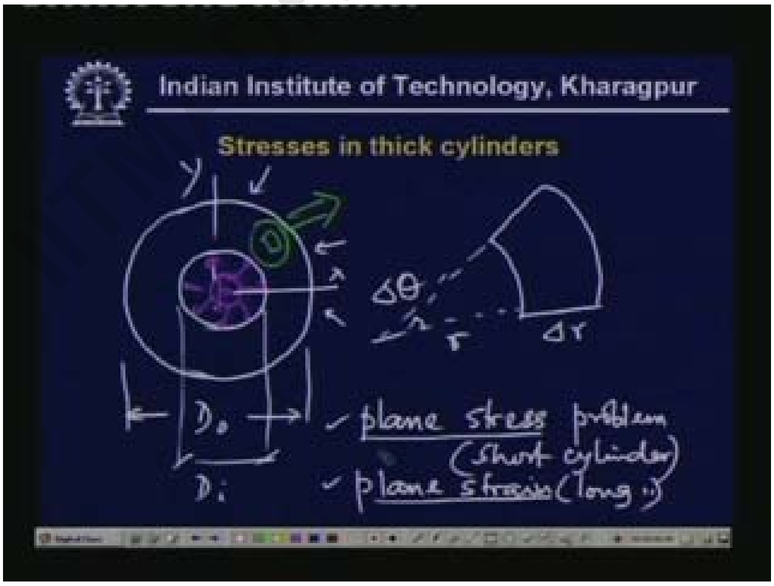

now if you consider ah the total cylinder which has outer diameter that is this is Do and this distance is Di and this is subjected to pressure from inside [noise] p is acted over here now what will be the stresses now in order to find out the stresses let us look at a {sma} (00:03:46) look at what are the forces acting on a small element

now if you increase it that is zoom this small element then [noise] we see that it looks like here where this angle is let us say delta theta which is very small and this radius is r and this is delta r

so [noise] this distance will be r plus delta r times delta theta [noise] and we want to find out the ah the different forces on that

but before doing that let me tell you that there are two possibilities ah if the cylinder is very very ah small in length that is if it is modeled to be a disc then of course we can assume and of course the pressure is acting inside or outside

but it's acting ah throughout the same ah hm the same length so therefore the stress along the z directions that is if we consider this to be xy plane so the z directions will be perpendicular to the xy plane and if the z direction ah if the stress along the z direction then zero and we call this a plane stress problem [noise] plane stress problem now this is for a very short cylinder [noise] short cylinder also for the disc

now if the cylinder is very very long then we can neglect except for the ah small deformations ah near the end we can neglect the ah shear strain throughout so the shear strain is zero and in that situation we call it plane strain problem and this is for long cylinder [noise]

now it is interesting that if we have a cylinder which is subjected to pressure of this kind or may be pressure from outside then both the ah small cylinder and long cylinder that is both the plane stress and plane strain assumptions will lead to the same result

so therefore hm we are considering here the case of plane stress ah that is there is no stress along the z directions now if this condition is satisfied now i want to ah remind that this plane stress and plane stress {pro}(00:06:46) plane problems are different if the loading is different type

we shall ah see ah in in two cases where the plane stress and plain strain problems are different but for this particular case ah its immaterial whether it is in plane stress or plane strain problem because they are going to lead to the same results

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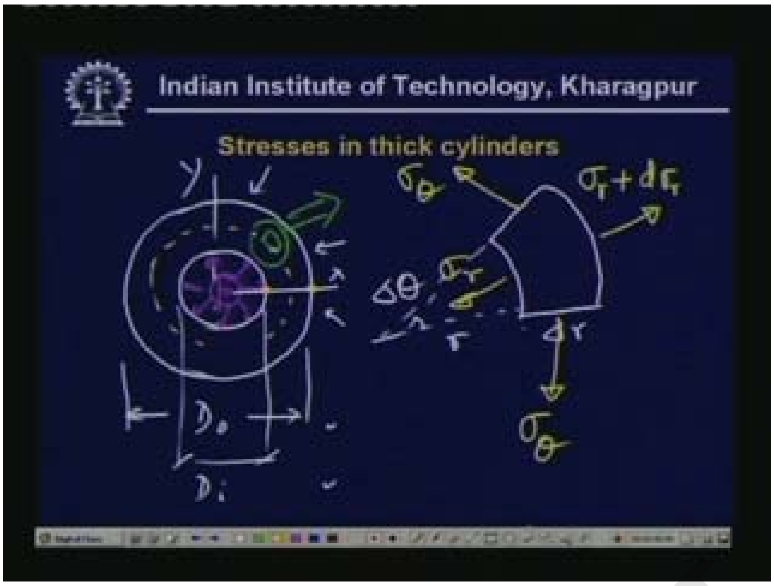

now let us see what are the forces acting on it [noise] now here there will be sigma theta and remember this is axially symmetric problem so the sigma theta will be same throughout the circumference

so here also we will have sigma theta the stress therefore the total force will be sigma theta times delta r and suppose this thickness we have taken to be unity so ah hm here this is sigma theta times delta r times unity so this is the total force

here the force is again sigma r and there of course the sigma r will vary you see here sigma r is minus p as it is obvious because there is a pressure acting over here in this phase ah there is no pressure acting so therefore sigma r is zero

so therefore a variation of sigma r is expected along the thickness so therefore sigma r if we take to this phase then it will be sigma r plus d sigma r where the d sigma r is a very small component added to sigma r

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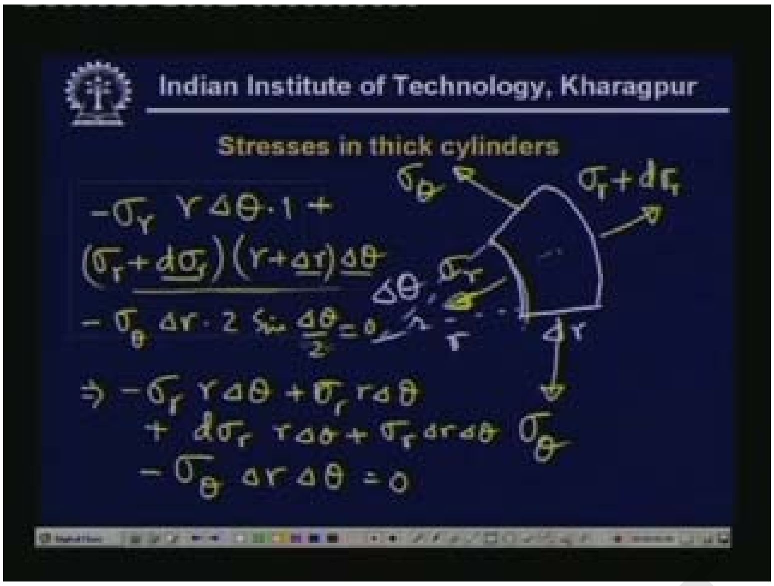

now we are ready to use our statical equilibrium equation let us look at that

if we use statical equilibrium equations then we get sigma r and here the total force is sigma r is the stress so therefore the force is sigma r times r delta theta times one

so this is the area of cross section minus sigma r that is if this is in that directions minus so it will be plus sigma r plus d sigma r times the length which is equal to r plus delta r times delta theta

and now here two forces are acting of magnitude sigma theta times delta r times one but we are now considering the force balance in the {rad} (00:09:45) along the radial plane so therefore only a component will act and that component is equal to sigma theta delta r times twice sign of delta theta divided by two and that must be equal to zero

and if you use this {caal} (00:10:06) then what you get is minus sigma r r delta theta plus sigma r r delta theta plus d sigma r r delta theta plus sigma r delta r delta theta plus there is a term {del} (00:10:36) d sigma r delta r delta theta

now which is very very small in magnitude because other terms are of the order of ah hm delta theta delta delta theta etcetera but delta r times delta theta times uh delta sigma r this is really small quantity so we can neglect that quantity and we can write down this way

that is only those three terms from this expression and from that expressions comes delta r times sign delta theta by two which is equal to delta theta by two twice of that therefore this is delta theta and that is equal to zero

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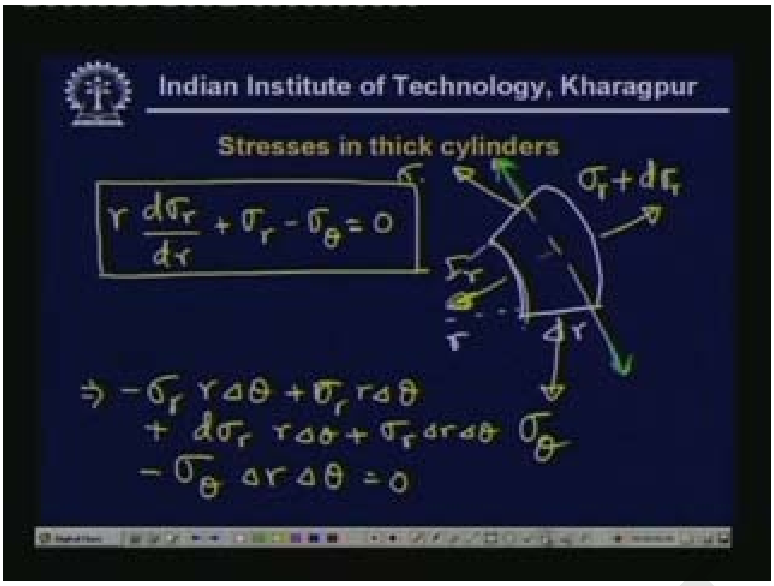

we write down this expression in somewhere compact form then what we get is this result that is r d sigma r d r plus sigma r minus sigma theta equal to zero now this is the statical equilibrium equations

the other statical equilibrium equation if you consider along the tangential direction we have considered only radial direction if you consider along the tangential direction then of course it is trivially zero as the symmetry implies so therefore this is the only equation which is available

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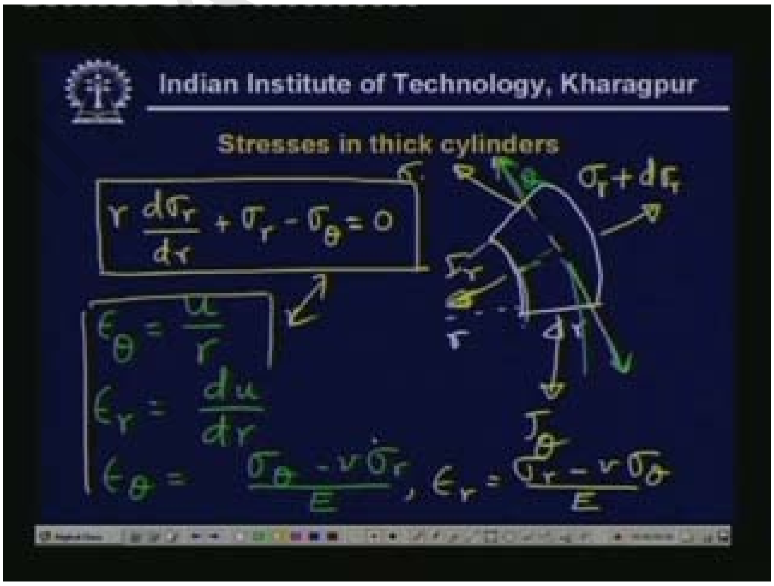

now there are two unknowns sigma r and sigma theta of course these are the two stresses which we are interested in and [noise] there is only one equation so therefore we need to take one more equations that is more things are required

it is then ah it's not possible to derive ah the expression for sigma r and sigma theta based on this equation only so we have to consider the kinematics of deformations

now this is done if we consider the deformations and for the deformations for the axis symmetric problem epsilon theta which is the (strail) (00:12:46) strain along the theta direction this is the theta direction circumferential directions

so strain along that direction is equal to u by r were u is the ah displacement along the radial directions and sigma r epsilon r is the radial strain which is equal to du dr

so these are the expressions containing sigma {del} (00:13:10) epsilon theta epsilon r and now we can combine this epsilon theta and epsilon r with sigma r etcetera by using the Hooke’s law which says that sigma theta is equal to ah epsilon theta is equal to sigma theta minus mu sigma r divided by E

and epsilon r is equal to sigma r minus mu sigma theta by E so these are the four expressions which are required and when we combine them together we get one equation in u in terms of u

that is the displacement of the ah hm of any particular point now that ah equations i am not going to derive but i give you the end result of it when we do all this calculations then we arrive at this following expression

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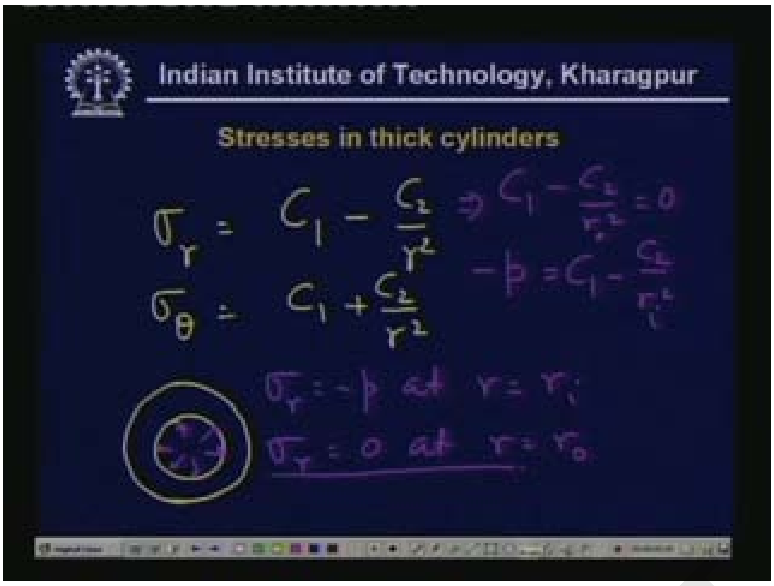

that is [noise] that is sigma r is equal to C one minus C two by r square and sigma theta is equal to C one plus C two by r square

so were C one and C two are the constants which are to be determined ah knowing the boundary conditions now what are the boundary conditions

in this case where the cylinder is subjected to internal pressure cylinder is subjected to internal pressure then of course the boundary condition is that sigma r is equal to minus p at r equal to ri the inner radius and sigma r is zero at r equal to r zero

so therefore if we use this expression this expression first let us say then we get C one C one minus C two by r zero square is zero and if we use this expression then we get minus p equal to C one minus C two by ri square

now using these two equations what we get is the following

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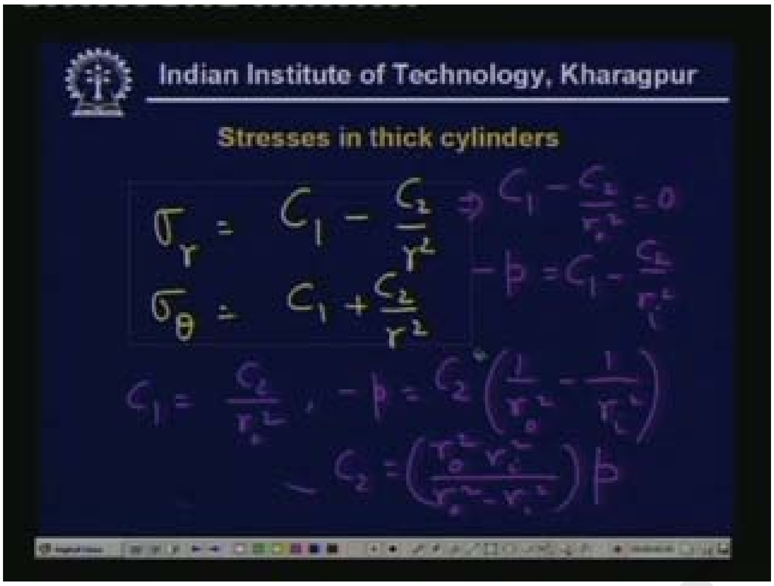

that is C one will be equal to C two by r zero square and minus p equal to C two one upon r zero square minus one upon r i square

so when we calculate C one and C two so from this expressions we can get C two in terms of r zero square ri square divided by r zero square minus ri square times p and C one is this divided by r zero square

so when we use those expressions then ultimately for that case we get this following relationship that is

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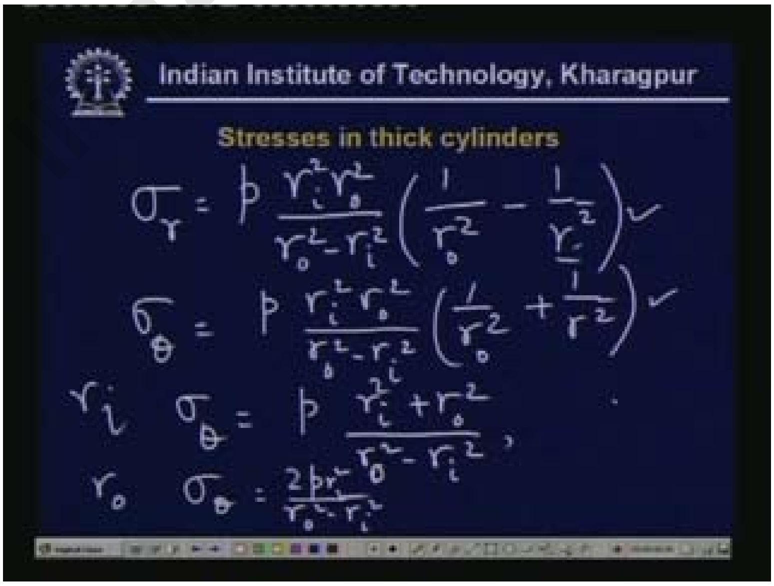

that is sigma r sigma r is p times ri square r o square divided by r o square minus ri square divided by one upon r not square minus one upon r square and sigma theta will be equal to let me write down sigma theta

sigma theta will be equal to p ri square r zero square divided by r zero square minus ri square one upon r zero square plus one upon r square so what will be the distribution

you see at r equal to ri this becomes minus p which is very very clear from here if you substitute here in this expressions ri you will get sigma r is minus p if you substitute here r equal to r zero then you get sigma r is zero

and here if you get if you substitute here r equal to ri then what you get is at i at ri sigma theta is equal to p ri square plus r o square divided by ri square ro square minus ri square and at r equal to r o sigma theta will be equal to twice p ri square divided by r o square minus ri square so that's that is clear

now if you represent it graphically then it looks like thes two formula are very important and ah we can design the entire ah cylinder based on this expressions

now we shall uh come to that little later but for this time being let us see the ah the distribution of the stresses across the thickness [noise] let us see what will be the distribution of sigma r

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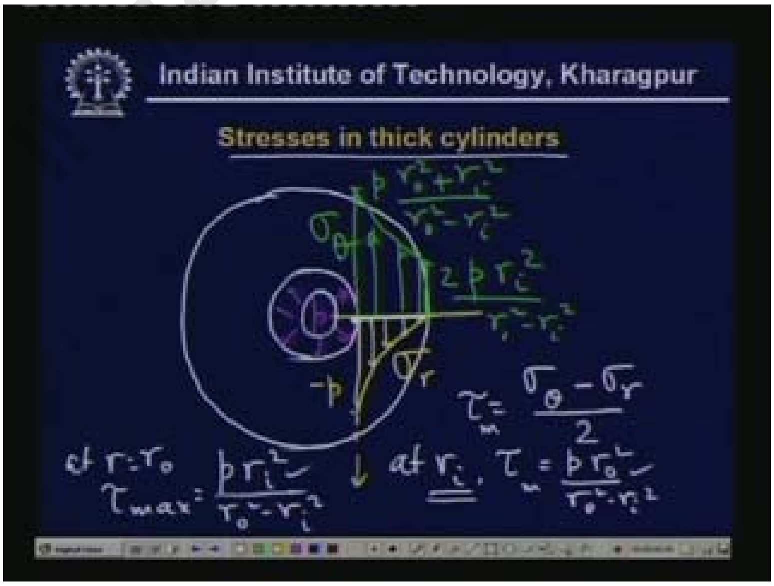

here we have this cylinder and this is subjected to internal pressure p from outsides [noise] here the distribution of of sigma r will be now at ri here sigma r is minus p] minus p and it goes to zero in this position

so so this is the distribution and this is always [noise] so this is the distribution of sigma r and what is the distribution for sigma theta sigma theta is distributed little different way sigma theta is quite large here

here this is p times so it goes to decreases this way here this value is p times r o square {mi} (00:20:42) plus ri square divided by r o square minus ri square and here it goes to twice p ri square divided by r o square minus ri square

so these are the stresses this is sigma theta so now these are the two kinds of stresses which are there now knowing this we can calculate what will be the shear stress

now the shear stress will be as we see ah hm it's a plane stress problem then the shear stress is defined tau will be equal to sigma theta minus sigma r divided by two this is the ah maximum shear stress

now what would be maximum shear stress at ah um this location will be equal to definitely this minus this one divided by two that is ah sigma r ah if you if you do that then it becomes the shear stress here at ri tau max so this is tau max will be equal to if you do that

it becomes p r zero square divided by r zero square minus ri square and at r equal to r zero so this is at r equal to ri at r equal to r zero tau max will be equal to sigma theta which is equal to p ri square divided by r zero square minus r i square and sigma r which is equal to zero so therefore this sigma theta minus sigma r by two that gives the result p ri square divided by r zero square minus ri square

so definitely as r zero is larger than ri we can say that the shear stress will be ah the maximum shear stress will be larger at r equal to ri so when we want to design that then we have to take ah this into considerations

now ah how to design the thickness the problem is that we are given suppose this value p and ri the internal radius now what will be the um the thickness of the cylinder so that will be the next problem but in order to do that we must remember what are the different stresses in a thick cylinder now as i said that the ah hm the stress distribution for ah sigma r and sigma theta values for ah the hm a plane stress problem and plane strain problems they are same but the design will be different if you go from plane stress to the plane strain problem as we can see very easily now this is about the stress distribution in a thick cylinder

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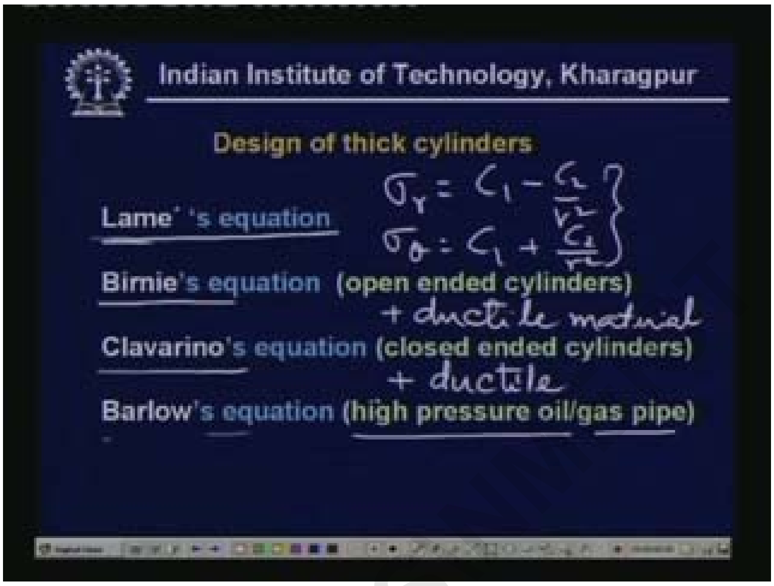

now how to design that ah in order to design we have various equations one is Lame's equation now probably i mentioned that the derivations which i have done just now is ah known as Lame's equations

that is sigma r is equal to C one minus C two by r square and sigma theta is equal to C one plus C two by r square so this is the Lame's equation and we design the thickness based on this Lame's equations

now if the cylinder is open cylinder open ended cylinder then and of course open ended cylinder and with ductile material [noise] ductile material then what we have ductile material then ah the design criteria is given by Birnie's equation which is a little different ah we shall see ah very soon

if the cylinder is closed ended and the material is ductile [noise] then it is governed by Clavarino's equation and for a high pressure oil or gas pipe the ah hm the design is guided by what is known as Barlow's equations

now let us see how to design how to get the ah proper thickness of a thick cylinder which is subjected to internal pressure

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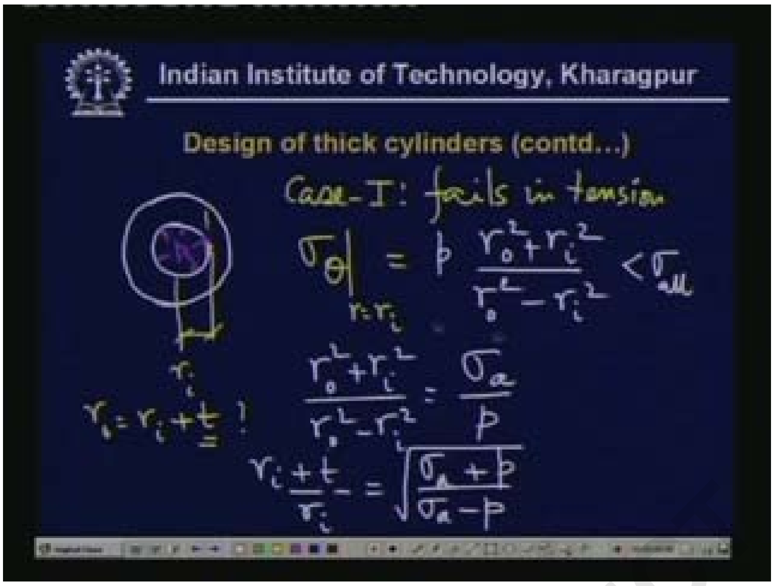

now first one is based on Lame's equation that is here this is subjected to internal pressure and ah inner radius let us say r this one is ri now how to get ro that is ro will be equal to ri plus t so that is the design criteria design ah problem is to select this t

now if we say that the material fails under tension then definitely the maximum shear is maximum tensile stress will occur in the inner surface and that will be sigma theta so therefore if it fails in tension

so the case one is fails in tension then of course we have ah sigma theta the maximum tensile stress will develop sigma theta at r equal to ri which is equal to p times ro square minus ri square divided by ro square

ah i'm sorry plus here and minus in the denominator so this is the maximum tensile [noise] ah stress developed and this must be less than equal to that is in the design limit this must be equal to sigma allowable ah sigma allowable

then [noise] how to do how to make this calculations that is very very simple so r naught square plus ri square divided by r naught square minus ri square is sigma a divided by p

now if you make this ah hm component dividend uh then what you get ro square divided by ri square is equal to sigma a plus p divided by sigma a minus p now if you use this relationship that is ro is equal to ri plus t then what you get is r

now here the next step is just to this one will be equal to that and if you substitute here ro to be ri plus t then it is very easy to see that ah t will be equal to [noise] sorry

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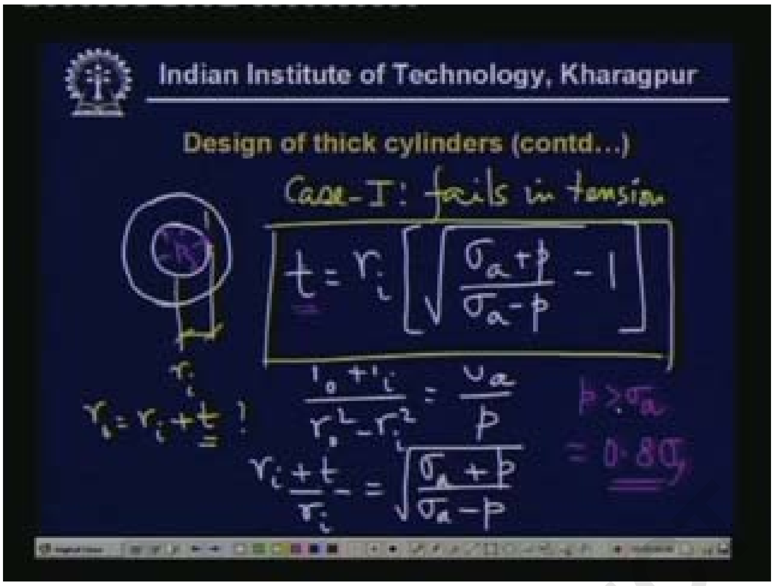

so again t will be equal to ri times under root sigma a plus p divided by sigma a minus p minus one so this is the [noise] ah sorry this is how one has to design when the material fails in tension

now for such cylinder the material may fail {als} (00:29:18) also in compressions but before going to that stage let us see one thing that if sigma a if p is greater than sigma a then t is ah not real

so therefore it is not possible to design any cylinder which will ah sustain a pressure more than the allowable stress [noise]

so ah this is very important that we cannot design any cylinder using one single cylinder that will {ta} (00:29:49) will sustain an internal pressure beyond the ah allowable stress limit an allowable stress limit is roughly equal to point eight of ah hm sigma ilt ilt strength

ah and if it is brittle material is about one ah hm one eighth of ultimate stress so for ductile material this is so and for brittle material it is {poi} (00:30:17) um point one to five times sigma u which is the ultimate stress

now if ah um that condition is satisfied that is p is greater than sigma a then no real value of t is possible and we cannot design any cylinder so how to avoid that

in the next class we shall learn if we use another cylinder then on top of it and place fit then it can sustain a very large pressure a pressure higher than what is sustainable with single cylinder so this is a case of compound cylinder and we shall we shall study it ah um in some more detail in the next class but for the time being let us go to the case two which is ah the case when the material that is it fails in shear so it fails in shear

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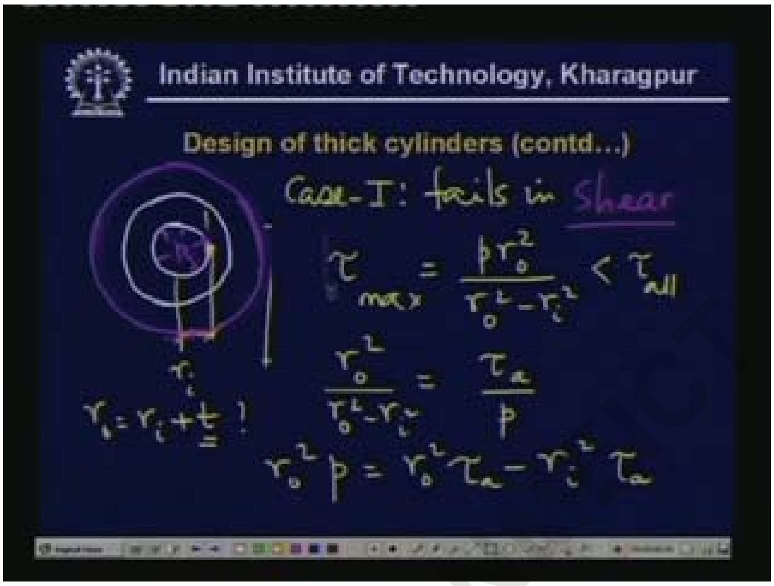

then let us see what is the maximum shear's force what is the maximum shear stress the tau max we have seen that maximum shear stress occurs in the inner boundary and that is equal to p ro square divided by ro square minus ri square

and in order to have a safe design this must be tau less than tau allowable so now how to do that this is r zero square divided by r zero square minus ri square is equal to tau allowable divided by p

and if we just [noise] ah hm subtract it so then it becomes r zero square divided by or let us not do that so this is r zero square times p is equal to r zero square tau a minus ri square tau a therefore r zero square by r i square from this expressions we can see that

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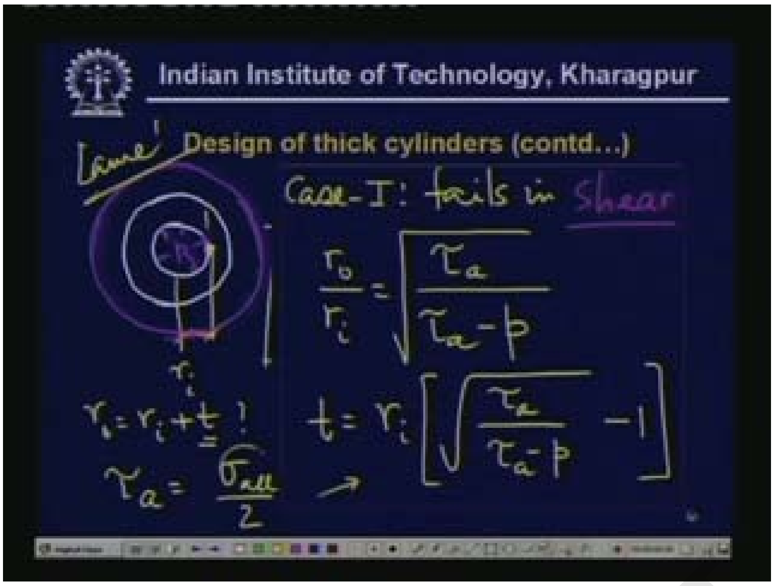

this becomes r zero square by ri square it becomes tau a divided by ah um tau a minus p so [noise] then again is the routine method we just make r zero by ri which is equal to under root of that

and we use this expression that is r zero is ri plus t then ultimately what we get is the same this result that is t will be equal to ri times under root tau allowable divided by tau allowable minus p minus one

again you see that in this case if pressure is greater than ah tau allowable then this material will will fail in shear and we cannot design any ah any cylinder any single cylinder is incapable of doing if of sustaining that high pressure

and tau a is normally taken to be sigma allowable divided by two this is the rough estimate and ah if we use that you can get this relationship in terms of sigma allowable which are {cott} (00:34:31) which are coded

so there are codes available uh which says what will be those values of sigma allowable now uh this is the case of design of thick cylinder when the when it fails in shear now so these are based on the Lame's equation Lame's equation

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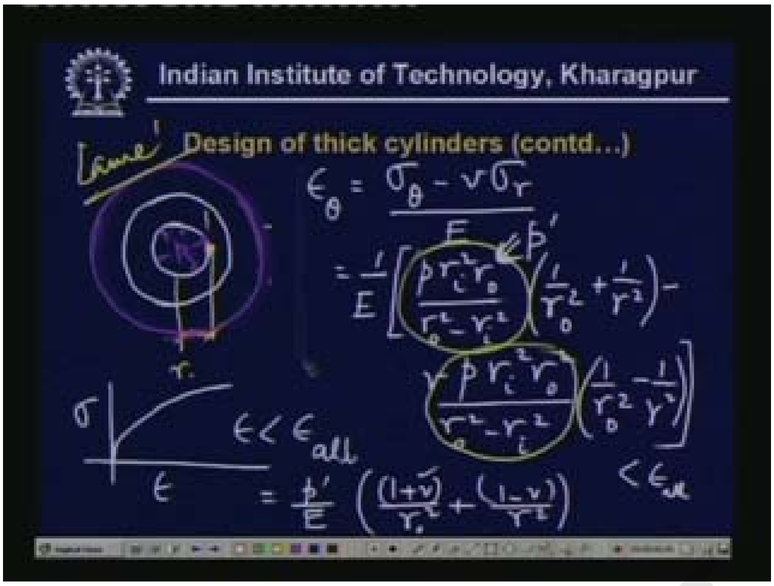

now let us look at the other conditions that is when the cylinder is now open ended then of course the failure criteria is that and this material is ductile remember when the material is ductile then the stress turn relationship is somewhat like this sigma and epsilon

so the failure criteria is that epsilon is less than some epsilon allowable [noise] so this is the failure criteria which is used in Birnie's equations now what is epsilon that is the [noise] ah um that is the maximum strain

and this is equal to definitely epsilon theta and ah you can you can verify that and this is equal to this divided by r divided by E so one upon E times sigma theta is equal to p ri square ro square divided by ro square minus r i square times one upon ro square plus one upon r square minus mu times p ri square ro square divided by ro square minus ri square one upon r naught square minus r square so this must be lesser than epsilon allowable

now if one uses that that is after somewhat some simplification we get the following form that is this part is constant this one and that one so what we get here is this part let us say p prime let us denote this as p prime

so p prime divided by E and it will be one plus mu times one upon r square plus one minus mu times one upon r square

so this is the expression and we see that now ah there there is a term Poisson's ratio appearing here because we are now ah considering the {minem} (00:37:33) the maximum strength theory now when we do all these calculations ultimately i shall give you the end result which of course you may derive and it comes from this line itself

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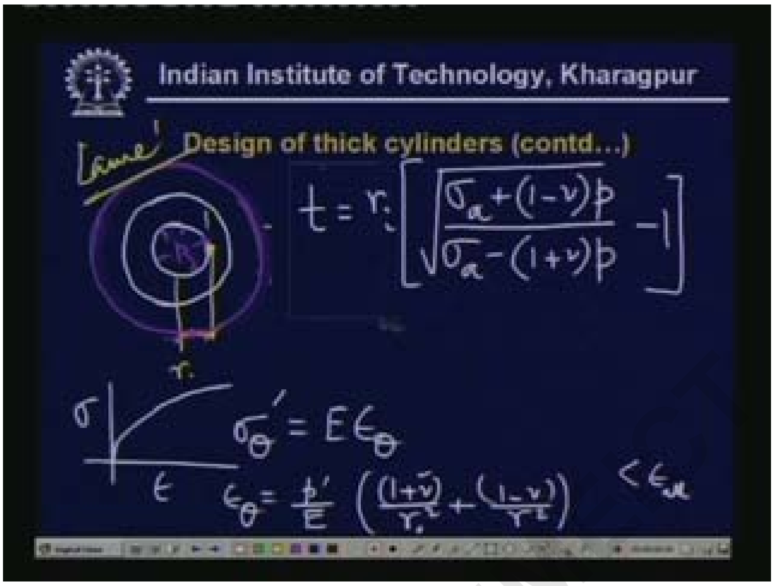

ah that the t thickness is given as ri times under root again again ah here we have that ah um epsilon theta is equal to p prime by E times this one

now if we take that epsilon theta times E that is we consider epsilon sigma theta to be epsilon E times epsilon theta then ah really a limiting this {val} (00:38:21) ah value um of epsilon theta is limiting this value of sigma theta prime and this is normally taken to be the allowable stress limit

so now if we use that what you get is this form that is sigma allowable plus one minus mu times p divided by sigma allowable minus one plus mu times p minus one so this is the results from Birnie's equations and this is based on the maximum strength energy that is the maximum strength within the cylinder

now this is very important to design a cylinder a long cylinder ah which is subjected to internal pressure and which is made of ductile material but remember that this cylinder is now open ended

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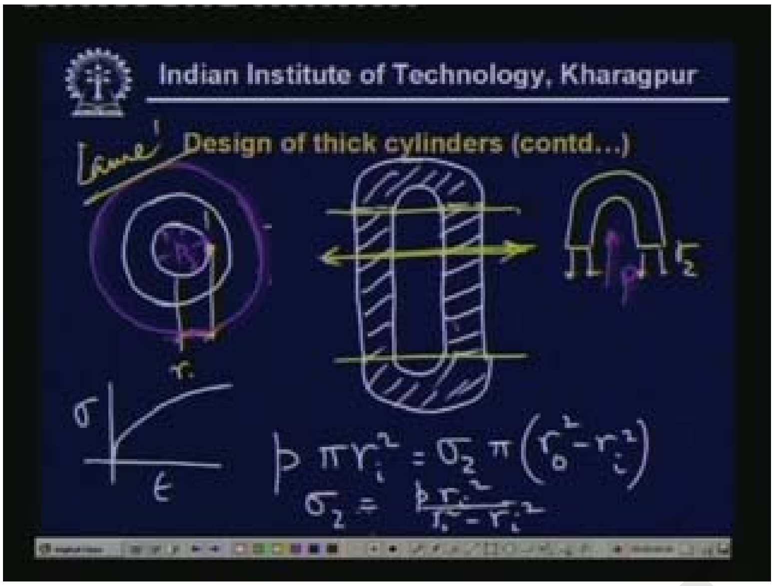

Now if we have a close ended cylinder then the Clavarino's equation is used and that looks little different let me explain somewhat here

if we have if we have a close cylinder then what we have so this is the cylinder which is um this is a thick cylinder and we have a gap out there so there's a cap and a cylinder so now you see this is no more a plane stress problem

here we have sigma z what is the value of sigma z sigma z average ah longitudinal stress ah is taken to be the {fo} (00:40:22) that is if you take across a section over here then what you get the total force will be equal to sigma z is governed this way that is sigma z

let me clear it out {sig} (00:40:38) we are interested in finding out the sigma z then how to do that we first we take the cut take the cut over here and then write down this equation there will be sigma z and pressure is acting throughout

so therefore if you balance the equation along the vertical directions which you get is p times pi ri square is equal to sigma z times pi ro square minus ri square so this is equal to ah this gives sigma z equal to p times ri square divided by ro square minus ri square so this is the expression for sigma z

now ah if the material is ductile we can use the same principle that is the principle of maximum strain and in that case the strain looks different from the previous one

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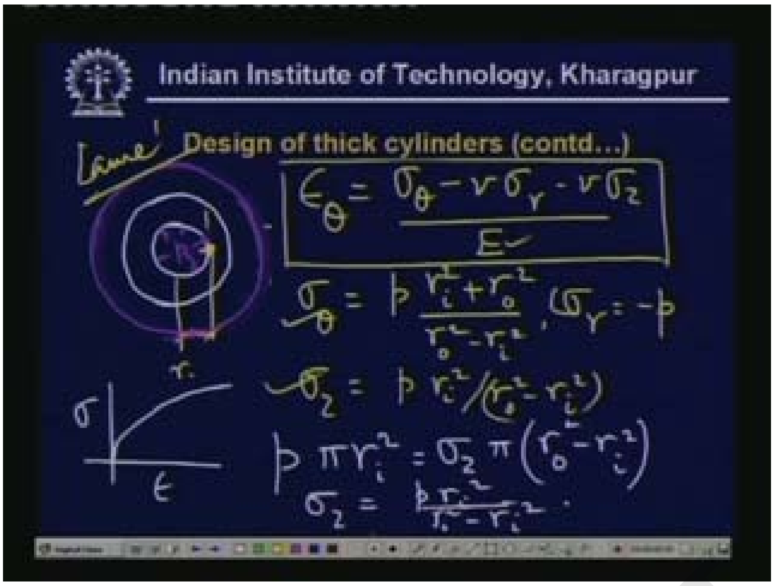

because we have to consider again sigma z but in this let us see what how does the strain expression look like epsilon theta which is the maximum strain along the circumferential directions and this will be the maximum strain which you can verify is equal to sigma theta minus mu sigma r minus mu sigma z divided by E

now remember and ah the maximum strain will be in the inner surface and here sigma theta is equal to p times ri ri square plus ro square divided by ro square minus ri square sigma r is equal to minus p and if sigma z is given this way that is sigma z is equal to p times ri square divided by ro square minus ri square

now if you use all these expressions then the next step is to insert these three expressions here in this expression for strain and noting that E times that is Young's modulus times epsilon theta gives somewhat an estimate of the stress ah along the circumferential directions and limiting this sigma epsilon theta is just limiting that value of the stress and that stress is taken to be the fracture stress or the tensile {ulti} (00:43:37) allowable stress of the material

so if you use {eve} (00:43:40) all these things then ah um after making ah all such calculations you get the following expression that is

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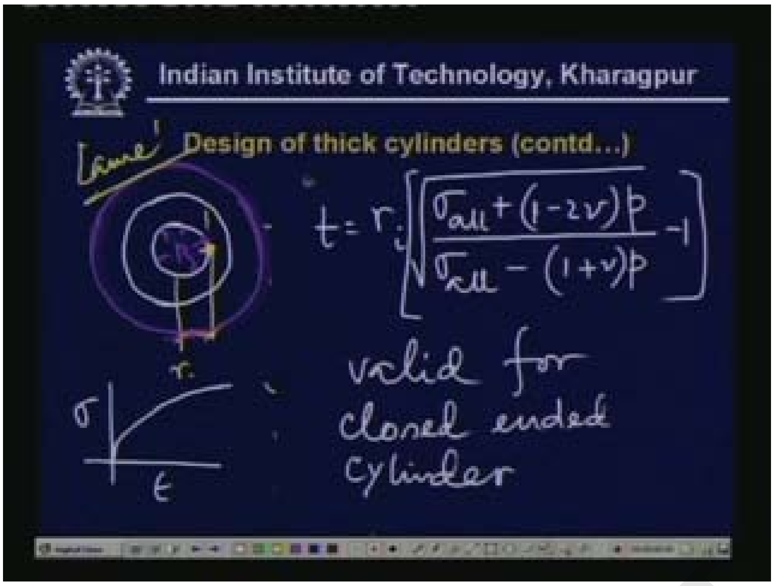

t will be equal to ri times under root sigma allowable plus one minus two mu earlier it was one minus mu now this is one minus two mu plus p divided by sigma allowable minus one plus mu times p minus one so this is the results obtained for Clavarino's equation and this is valid for valid for closed ended cylinder

ah again um these are the three expressions which are normally used while designing it but if we want to design the pipe ah for a very high pressure then sometimes {uu} (00:45:15) we use the Hoop's stress formula that is the same expression for the thin cylinder

but instead of taking the ah hm the average diameter we take the outside diameter that is now we see ah ah what will be the thickness for a high pressure

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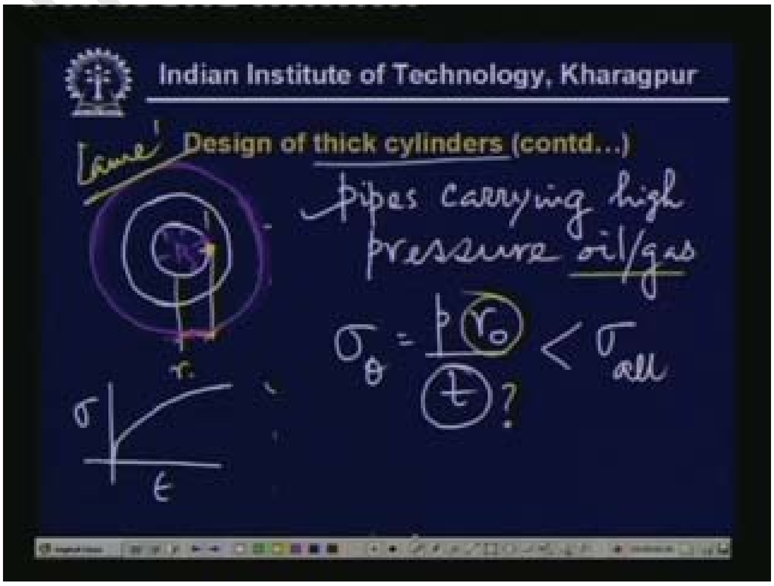

so pipes carrying high pressure oil or gas in that case ah remember the ah hm design criteria for thin pipes thin cylinders was that sigma theta was pr by t where earlier case r was the mean diameter for a thin cylinder which is equal to the average of the ah outside diameter and the inside diameter

but when we want to calculate for the thick cylinder in this expressions we put instead of r we put r zero definitely the value is the maximum over here sigma theta and this must be lesser than sigma allowable now this is the design condition from which the value of t is calculated

again ah depending upon the nature ah the nature of the liquid it carries we have to give some allowances as ah um like the thin cylinders so these are the ways ah few ways of designing a thick cylinder now remember we have used Lame's equation

Lame's equation is based on the ah principle that ah the ah hm the stress ah the material fails ah the material fails when the stress ah um normal stress exceeds a certain limit sigma allowable ah also we may use that the material fails when the shear stress gets a maximum limit now these are based on Lame's equation

in Birnie's equation this is valid for an open cylinder and here the design criteria is that the strain that is the circumferential strain which is of course larger than the radial strain the circumferential strain reaches a um ah {haa} (00:47:49) a um an upper limit

and with this expression we calculate the Birnie's {equas} (00:47:54) um the thickness using Birnie's equations Clavarino's equation uses the same principle now it is applied for a cylinder which is closed ended

and lastly we calculate the thickness of a pipe which carries high pressurized oil or gas using (ve) (00:48:15) a very similar familiar looking formula used for {ca} (00:48:18) used for calculating the thickness in the thin cylinders

the difference being here instead of taking the mean radius we take take the outside radius so this is how these things are calculated

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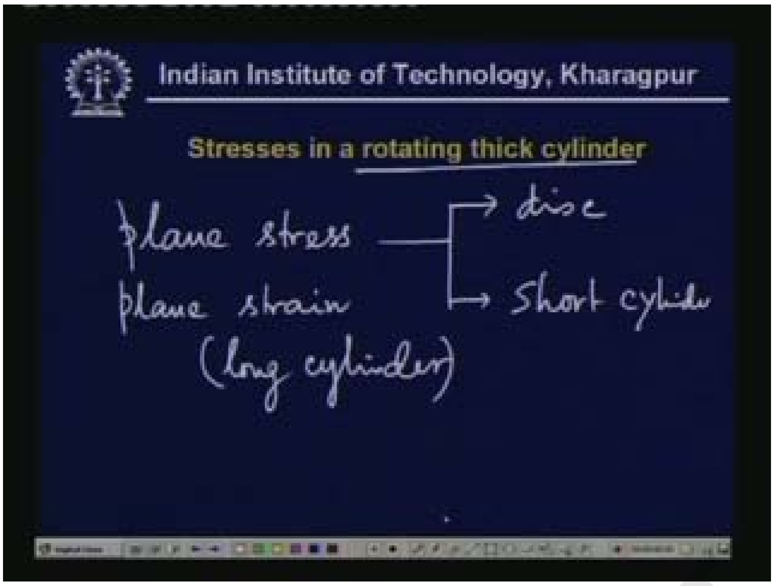

now let us come to stresses in a rotating in a rotating disc rotating thick cylinder now how this is important

you see we have also talked about how to design a flange which ah of course the um the purpose of the flange is that it it connects to shafts and the shafts rotate so therefore along with the shaft the connections the flanges also rotate and this stress is developed in that flange

then in this case when we want to find out the stress we have to differentiate between the plane stress problem and [noise] plane stress problem and plane strain problem now plane stress is for short cylinder disc or for short cylinder and plane strain is for long cylinder

now how to calculate the stresses then we revert back to ah the same procedure just by first finding out the ah equilibrium equation now if you consider again a very small element as we have done for a thick cylinder then ah it looks like

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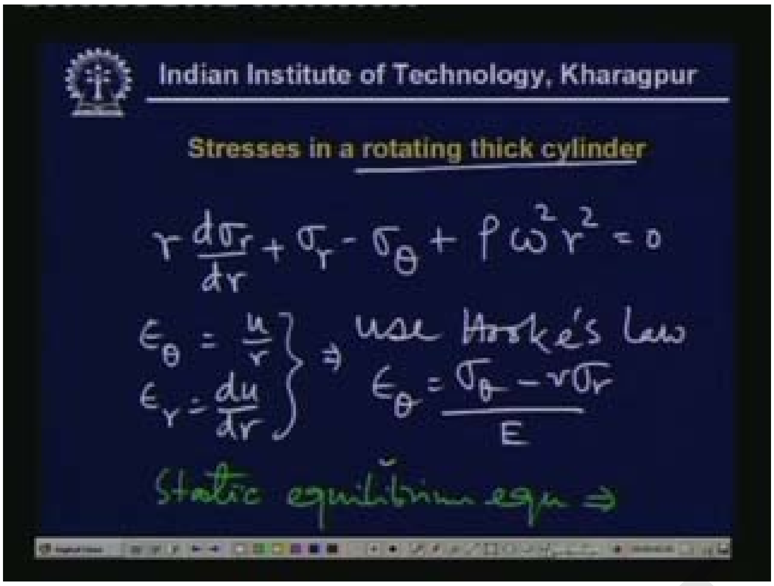

again the same stress sigma theta sigma r sigma r plus d sigma r

in addition here there will be one {ssh} (00:50:44) one force which acts along the radial that is the centrifugal force because now the entire thing is rotating with omega so therefore again there will be another force which has a magnitude this is the entire mass of it

the force is now ah the mass delta dm and omega square r and d m is mass as before is the total volume times the density row is the density of this material let us say and the total volume is delta r times r delta theta times unity

as before we take the thickness to be unity we consider here in the plane stress problem so it take unity thickness because ah in the um in the um axial direction there is no stress

so this is the total mass and if you write down this equation ah again the static equilibrium equation just by balancing the force different along the radial directions then what you get is that very similar expressions that is r d sigma r dr plus sigma r minus sigma theta and plus here we have omega row omega square r square this is equal to zero

now for plane stress problem again we'll have to find out the expression of epsilon theta in terms of u which is equal to u by r epsilon r is du dr and then we calculate we use the the ah Hooke's law after that we use Hooke's law

that is epsilon theta is sigma theta minus mu sigma r by E and so on and so forth

ultimately when we solve this {ex} (00:53:07) expressions we get the following results

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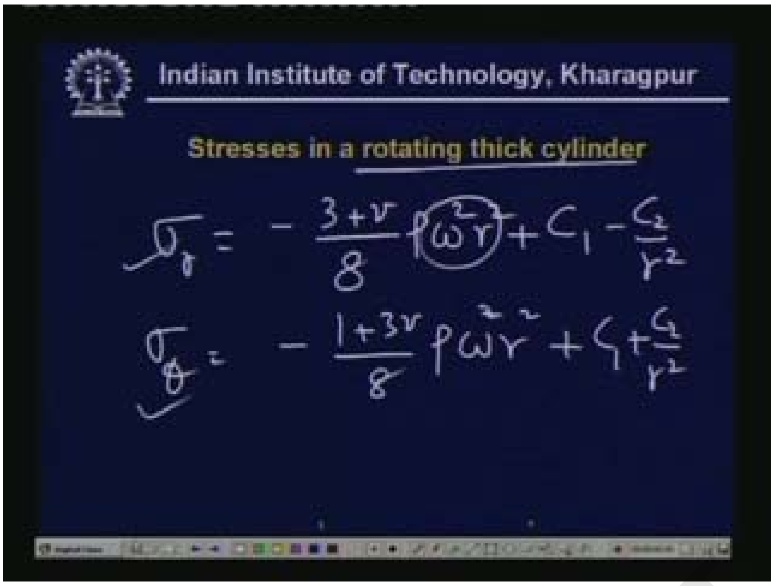

that is ah now we solve everything and i shall write down the final form of this expressions sigma theta will be equal to sigma r is first three plus mu divided by eight row omega square r square plus C one minus C two by r square and sigma theta is minus one plus three mu by eight row omega square r square plus C one plus C two by r square

again this is for the plane stress problem for the plane strain problem again it looks different but i am not going to that details

so now you see the stress develops because of the rotation and that may limit the speed of the ah flange because the maximum ah rotating speed of the flange may get limited by this sigma theta or sigma r whatever it may be so now these are the stresses for a rotating thick cylinder

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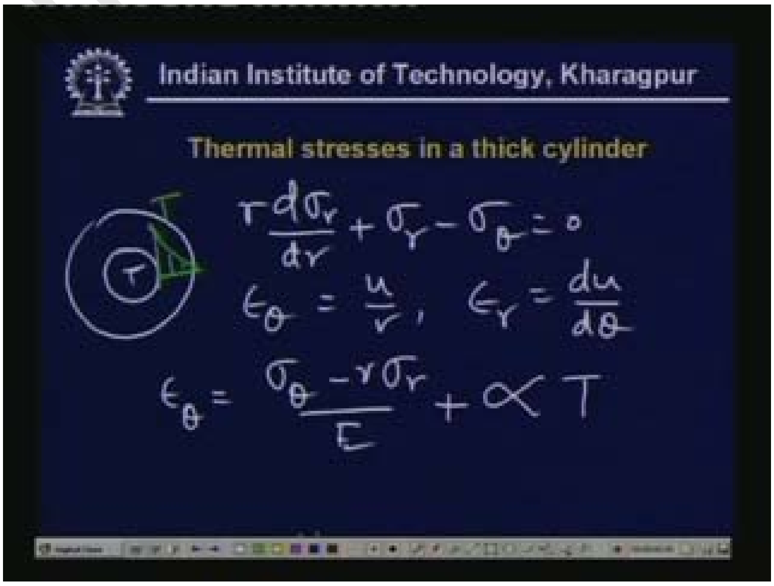

then we come to the last case that is the thermal stresses in a thick cylinder which i shall tell you briefly the procedure everything is same

first we write down these expressions sigma r dr plus this comes from the balance equations then we {wrout tha} (00:54:36) write down epsilon theta is mu by r and epsilon r is du d theta but now you see the Hooke's law changes

epsilon theta is sigma theta minus mu sigma r by E but here we get alpha T which is the temperature distribution now what is so important because it carries say the the cylinder carries a liquid which is at very high temperature

now definitely the temperature will vary from a inner surface to outer surface so this variation can be ah found out by ah um by heat transfer methods by equations or considering the heat transfer so this kinds of equations are now used

and ultimately what result we get is that

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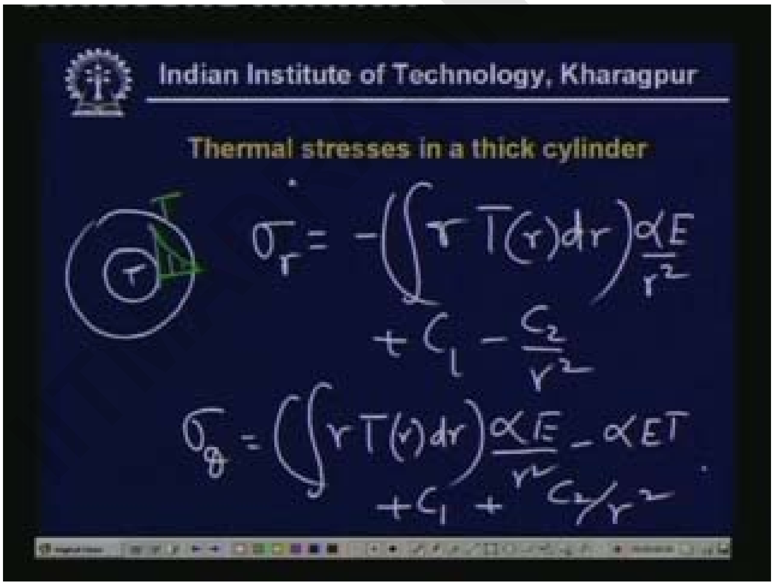

epsilon theta {epsil} (00:55:27) say first of all epsilon r sigma r is equal to minus here r T T is a function of r dr times alpha E divided by r square plus C one minus C two by r square and {eps} (00:55:48) sigma theta is equal to again r T is a function of r dr times alpha E by r square minus alpha E T plus C one plus C two by r square

again you see because of the temperature changes a thermal stress is developed so all these are very important aspects for designing

now let us sum up what we have learnt today we have learnt how to calculate the thickness of the thick cylinder {whe} (00:56:19) if it is subjected to internal pressure

there are different procedures procedures using Lame's equation Birnie's equations Clavarino's equations or {fo} (00:56:27) just like the Hoop's stress formula

and also we have learnt how to get the stresses {fo} (00:56:32 min) when the ah the cylinder is rotating

ah we {aal} (00:56:36) we also calculated ah the thermal stresses that is if it is subjected to internal ah hm internal thermal that is the temperature distribution is there then stress is develop

so all these things are to be consider properly when we want to design a thick pressure vessel which is containing fluid ah at a very high temperature sometimes it may rotate sometimes it may not but that depends upon the ah system to system

so we come to the end of this lecture

thank you very much for today

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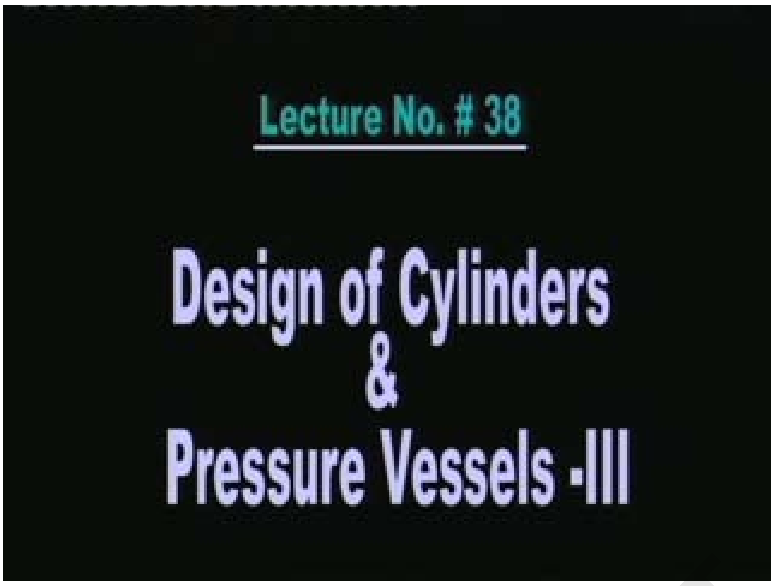

let us begin lectures on machine design part one this is lecture number thirty-eight and the topic is design of cylinders and pressure vessels this is the part three and the concluding part of the lectures on the same topic

now in last two lectures we have discussed how to design a cylinder when the cylinder is modeled to be a thick one or a thin one and also we have discussed how to take care of the effect {whe} (00:57:44) ah when the cylinder rotates also the temperature distributions

just before going further let us recapitulate once again what we have learnt so far in a very brief way

so ah let us for this time being let us again go back to the earlier discussions and ah just ah refresh our memories about the stresses in cylinders

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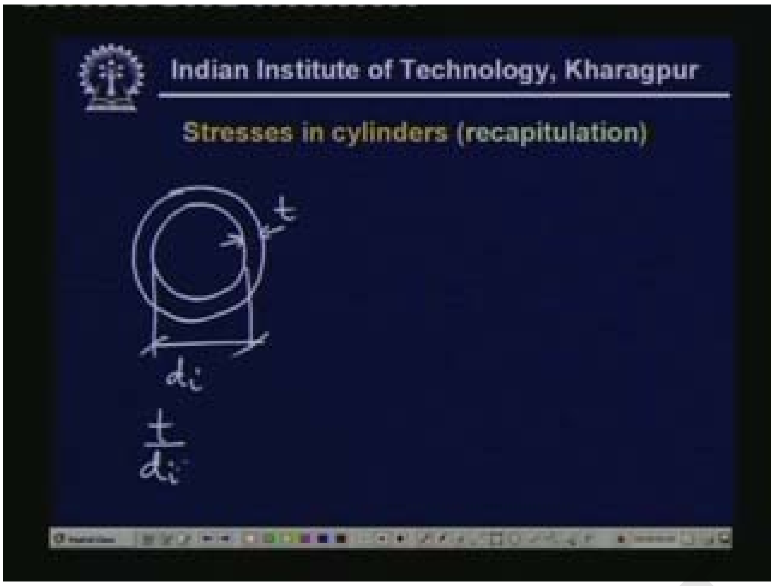

now the discussion started with the different kinds of cylinders you see

the cylinders are classified according to the thickness and the so this is the thickness and the inner diameter di so it is divided if this is t by