Design of Machine Elements – I

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Lecture No - 13

Design for Fasteners – I

dear student let us begin the lectures on machine design part one

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this is lecture number thirteen and today's topic is design of fasteners this is the first part of the lectures on that topic [Vocalized-Noise]

now in the last few lectures you have learnt ha uh the general principles of machine design that is the how to design a machine element

now the design has various aspects and in this course we are ah studying only the strength aspects of design

while designing an element we have to make the dimensions properly such that it can sustain some load now [Vocalized-Noise] the way to design is to check against different failures

so there are various failure criteria's which you were taught in the last few classes then ah this strength of the element depends on the types of loading if the loading is static loading then of course the material may fail either because of a large tensile stress or a shear stress now if the material is a brittle material then the failure is due to the large tensile force tensile stress and there is ah the uh corresponding failure theory that the principle stress which is the maximum stress tensile stress available at a point should not uh should exceed ah a certain value if it fails

similarly if the material is a ductile material then the failure is predominantly because of shear there we have two theories of failure one is if the shear stress is ah beyond certain limit then the material may fail or what is known as the Von-mises theory of failure where it says the strain in a {ge} (00:02:42) deviatoric strain energy that is the distortion strain energy should exceed certain limit then only the material fails

if the loading is on the other hand a variable loading that is a dynamic loading which fluctuates with time then the failure criteria is completely different

if it is purely fluctuating that is if there is the mean stress is zero then the material fails uh if the stress exceeds the endurance limit that is the amplitude of the stress uh state uh stress in exceeds the endurance limit

now in all the design procedures we have to ah make the design safe against various other uncertainty you have you will notice that in designing an element there are a lot of factors which are ah uncertain that is we cannot model them and those uncertain factors may not affect the designing process

so in order to do that we will have to make a large safety margin and this safety margin is provided by taking a factor of safety this were all taught in the last few classes

now from now onwards we are going to use those theories of failure as well as ah different other ah property different other criteria in order to design few machine elements

now today's lecture will be on the design of fasteners now fasteners they are very important {ele} (00:04:17) important member in a machine or in a machine component because a machine which is a huge which may be a huge structure it cannot be fabricated or manufactured as a single piece so we will have to ah make {ma} (00:04:32) different components and then tighten them or join them together to make a complete whole

take for example a gear box which has a number of gears as well as shafts and various other thing now that cannot be designed that cannot be manufactured as a single entity

so we make instead number of gears number of shafts etcetera and then assemble them so in this assembling process we need the elements which will join one parts to the other so if the fastener that is the joining elements are weak then whatever be the or how strong the design of the members may be the material will or the machine will not serve it's purpose because the joint will fail therefore the joints are one of the very important members in a design

we study we start this uh design with the design aspects of joints because these are relatively simple and in today's class we are going to learn how to use those theories of failure effectively in order to join a very simple fastener know as the pin type joint

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so let us now look at different types of joints first we have here the classification of joints we have non-permanent joints the joints are [Vocalized-Noise] of mainly of two types

the joints where the members are connected such a way that they could be dismantled or disconnected according to our will so without damaging the components we can disconnect those two components

there are various joints available which are non-permanent type one is the bolted joint bolt it looks like all of you know so there are threaded members this will learn under the heading of threaded fasteners this is also a fasteners but this is known as threaded fasteners

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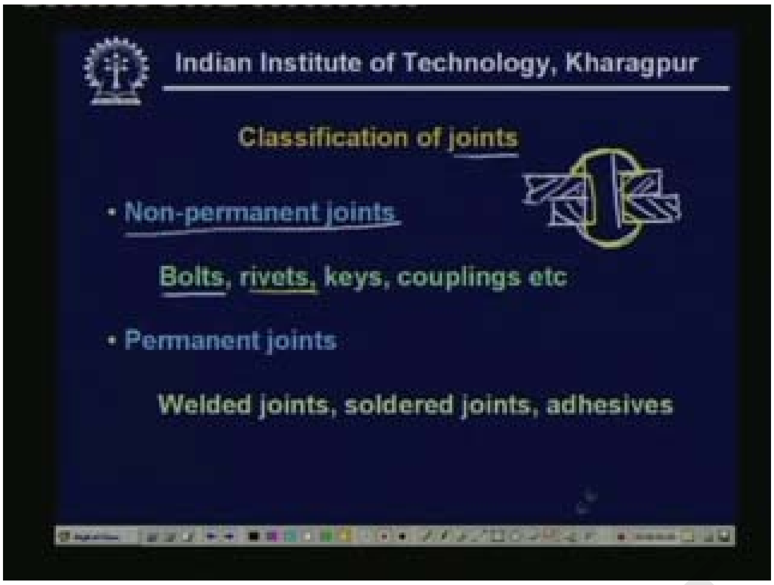

then we have rivets rivets they look like bolt but they are long rods which connect two parts suppose there are two plates which we are going to connect this is one plate and this is other plate so this is one plate and this is the other plate

then the rivet riveted joint looks like somewhat this so this is the riveted joint where this is a single piece this one is a single piece so this is a riveted joints

then we have keys we shall spend some time talking about the keys there are various kinds of keys uh known as sunk keys feather keys gip uh tapered keys etcetera

couplings couplings are normally used to join two shafts ah axially and the couplings the joints should be little flexible we will study about all about that in due course but these are few types of the non-permanent joints

permanent joints are those joints on the other hand which which is such that the two components which are joined cannot be dismantled without damaging one of them or both of them

so this is a component which is permanent as the name suggests there are various examples one is welded joints all of you know what is welding then soldered joints or braced joint there are then adhesives now industrial adhesives are {normalet} (00:08:51) are being used today's in various {ap} (00:08:56) ah various parts so lots of researches are going on and still new uh new polymers are ah fabricated which serve different purposes for adhesives so this is ongoing process going on the main point is that the joints are so important that every time we want to find out new kinds of joints which are effective which will not weaken the machine elements uh by being itself quite weak

so joints are that way important in today's lecture we are going to study some ah how to design a pin joint

so let us now go to

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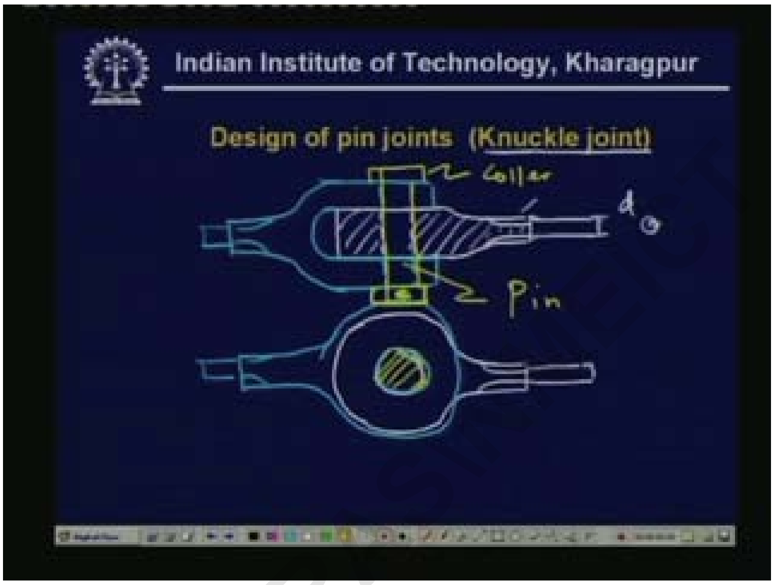

design of pin joints now there are different kinds of pin joints we take a very simple example of joint which is knuckle joint

let me {jo} (00:09:53) let me draw first and then we can explain

so knuckle joint it has two parts one is so this is one part which has hole in it now this is a rod this is this has circular cross section this part and if you look from top it looks somewhat like this

this is octagonal shape octagonal shape and there is a hole out there so this is one part and we are going to connect this this is continuous rod which has some diameter d and this is connected to the other member like this

this is another part which has again hole in it so look from above which will look something like this this is again has the same hole and through this we insert one pin

this is a pin where this part this is known as the collar of the pin so this is pin and this is the collar of this pin and this is a split pin that is we insert a pin over here there is a collar and we fixed this pin with with this arrangement with this nut kind of thing by means of a split pin we insert this split pin through a hole here and then fix it

so this is the uh this is what is known as a knuckle joint so let me erase the bottom part and explain the purpose of this knuckle joint

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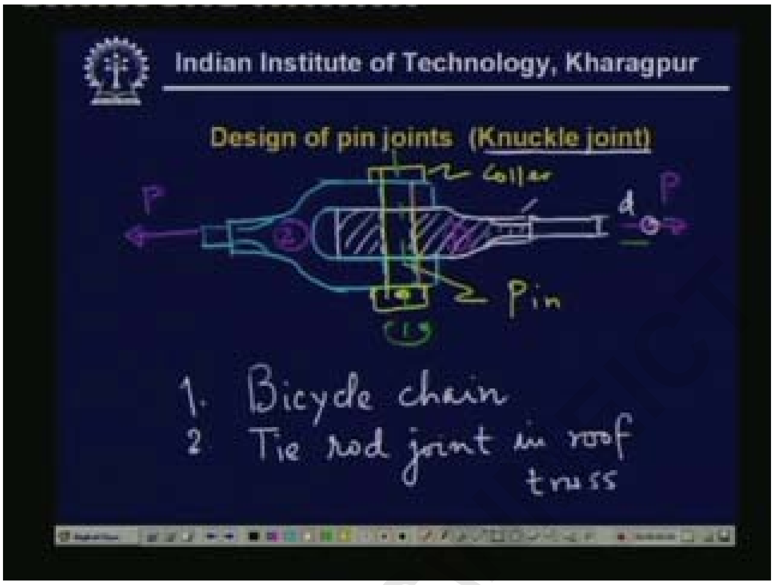

so here what we see here this is a joint now there are two rods which are connected together

now normally we apply two forces so this joint it takes a large tensile stress what happens when we apply a {tenh} (00:14:04) tension here so this pin arrest the motion of this part part one this let us call this as part one and let us call this part two so it arrest this part one and this part arrest the motion of part two

so this is the purpose of this uh of this pin now you see that this two rods can rotate {ev} (00:14:26) if you remember the the uh top view then it can rotate freely about this axis

so this joint cannot take normally a compressive stress it can take large tensile stress it cannot take much compressive stress because this this is flexible so it can rotate so when once we apply compressive stress here then it will just buckle ah if you if you ah let me call it buckling so it will just buckle

but if this pin is guided properly then it can really take some compressive load now this kind of joints are ah used very often in the bicycle chain you have seen this kind of pin joint sometimes {ith} (00:15:22) we used in the tie rod in the roof truss walb rod and various other purposes so this is very commonly used joint now here we want to design such joint so the design criteria uh may various we are going to discuss them suppose this load P is given so we want to design this joints such that it can take some load

now let us go to the design and see how this could be done

now first of all we have learned various criteria ah of various failure theories and we are going to use them

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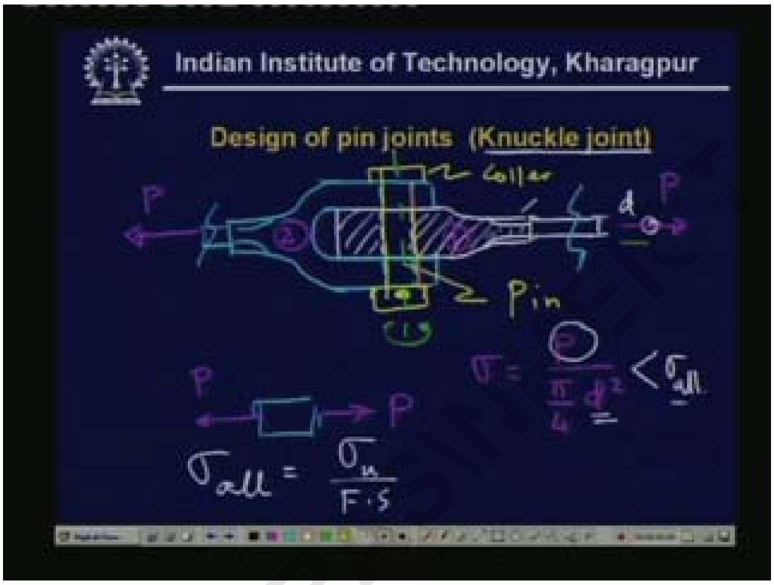

now when the P is larger than some value then there may be various kinds of failure one is that {sup} (00:16:24) if P is really large then this tie rod or this rod could be broken so or this one may be broken

now then if you if you want to apply the theory of failure then you will have to take a cut across this sections so if you take a cut there then the pressure P is at force P is acting so definitely normal force P is there so the stress will be now normal stress is P divided by the area pie by four d square if d is the diameter of this rod and this sigma should not exceed certain limit so this sigma must be less than sigma allowable

what is sigma allowable sigma allowable will be equal to sigma ultimate divided by some factor of safety so if the factor of safety is more then of course sigma allowable is less and for that for a given P will have to increase the value of d

so if you take a large factor of safety then the dimension uh becomes larger so ah with the increasing dimensions there will be lots of other factor coming into play that is the ah um the weight of the joint may be ah high or the material cost incurred may be more so we will have to compromise we will have to find out what is the correct factor of safety

this ah this are done ah well there it factor of safety usually ah this is taken from the previous experiences and they are uh different for different manufacturers they could be obtained from the design hand book as well so there are various guidelines for them {bu}(00:18:19) but the but the fact is we have to find out what is this factor of we have to first select what is the factor of safety ultimate stress this is tabulated will have to select the material material selection is one of the very important thing in design first of all we will have to select proper material then material selection depends on {varial} (00:18:37) various other factors the {emh} (00:18:39) the environmental conditions the temperature etcetera etcetera

so select when the material is properly selected then sigma u it could be looked from the table and factor of safety is selected we we choose the factor of safety

now for in this formula what we know we know this P and {witd} (00:19:02) we know this sigma allowable so from this we find out what is the smallest d ah required

now [Vocalized-Noise] sometimes in the design procedure here of course it is not so sometimes we ah we have already designed the uh value of d we have founded found the value of d and we know this P so we want to check for safety now if this is satisfied then we can say that this is quite safe

so this is how the diameter has to be selected from the given information about P and sigma allowable

now about the other parts we will have to break them into different components

first of all let us see the component number one

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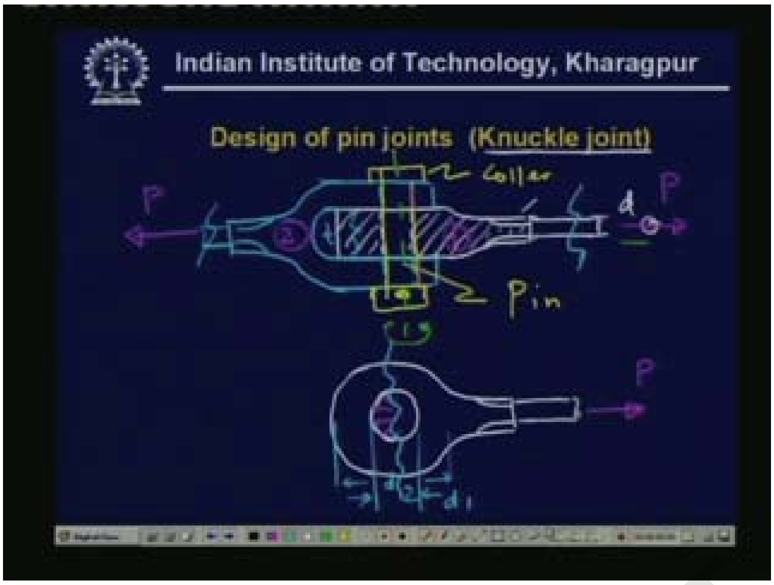

so if you draw the component number one then what you see is so this is component number one and we have this force applied force here

definitely this force will be balanced by this ah contact force between the pin and member one and therefore there will be a contact force here

what is that distribution of the counter {forc} (00:20:53) contact force we shall come later but [Vocalized-Noise] what will but if there is a clearance then of course there will be no contact pressure on here but there will be a large contact force on this side on the left hand side

so now we will have to see see what are the critical sections that is what are the first modes of failure which way the material can fail and those are the critical sections and will have to design ah the dimensions so that the critical sections doesn't fail

now first of all one once we see this then obviously the material may fail also in tear so it may so happen that it fails by breaking from this

so we have to now design this diameters etcetera now if this length is this thickness is t let us say this much is t so this diameter d one and d two let us take this is d one the inner diameter is d one and let us take this as d two okay so [Vocalized-Noise] back to the design this d one d two

now once we know that this is a critical sections then we will have to draw the free body diagram or draw draw the internal forces in internal forces developed in that critical sections

so the free body diagram will look like

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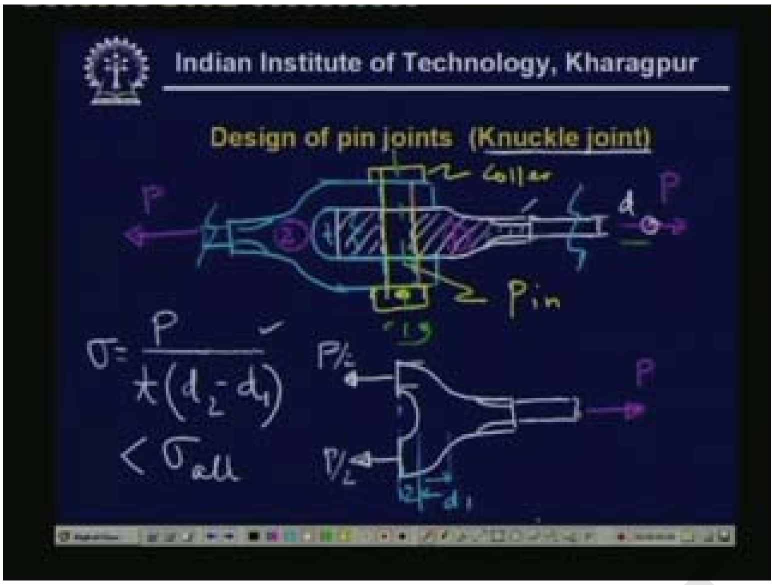

so this is the part and here of course we get this force so the total force is P so this is roughly you can take P by two P by two

so total force is P so therefore the average stress here will be along this sections average stress will be P divided by the total area area is t t is the thickness and times this distance which is equal to which is equal to d one d two minus d one

so this this will be the stress developed and for designing the ah thickness or d two minus d one etcetera we need to satisfy this expression that is P allowable divided by t times d two minus d one this is the normal force it should be less than sigma allowable so this condition has to be satisfied

now once we know that then we can design what is the value of d one or d two if t is properly chosen so this is one way of failure and we have one expressions here

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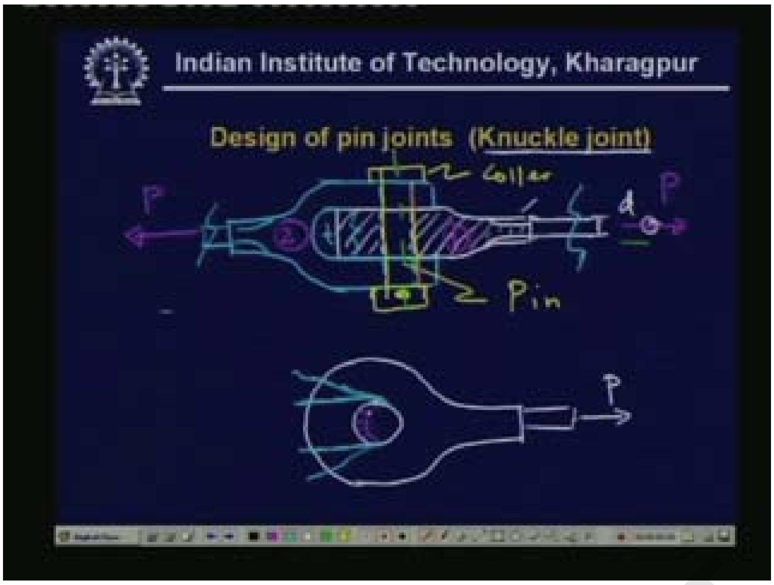

now what is the other way of failure let us look at the again the same since there is pressure over here on this region so the material may also tear away that is this part if the load is too high then this can come over here

so it can it can just tear away so when it tears away then of course this is one such critical section it just goes away

normally it should not go like {strait} (00:25:17) low go straight it may go in a radial way also so but we can design roughly based on on this ah this mode of failure so let us consider the free body diagram of this part

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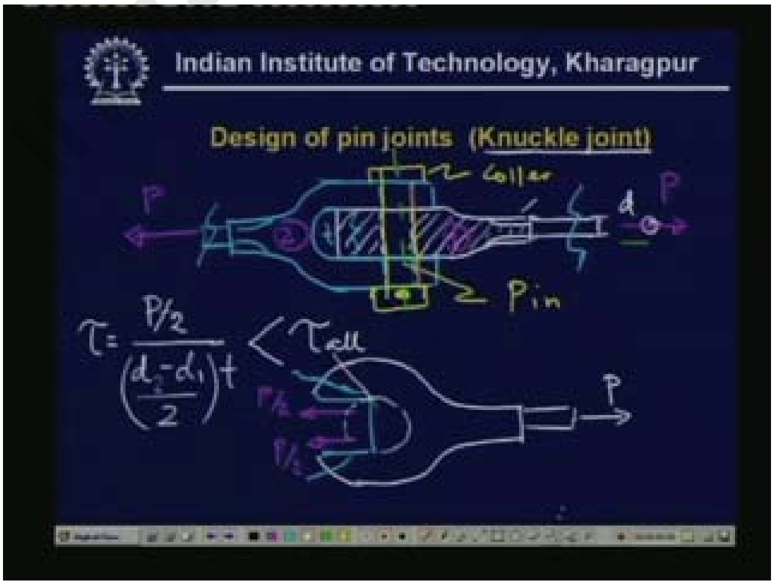

free body diagram of the part will be now so here we have this one now the the shear stress now here it will be you will have P by two P by two so then the material fails under shear so it because the force is ah is a shear force so we will have to calculate for the shear stress then the next step is to calculate the shear stress and this shear stress will be equal to P by two divided by this is the average shear stress this is divided by this length and this length although it ah it it may look complicated but roughly this length that is making some approximations this length could be taken as the same that is this one can be taken to be the radial length which is equal to d two minus d one by by two and times t

so this is the shear stress so this shear stress must be less than allowable shear so this has to be satisfied now once this is satisfied then it will not fail in in this configurations

so this is one such failure theory we have used again the failure theory that is this material breaks for maximum shear

what is the other way of failure let us look at them in somewhat care form

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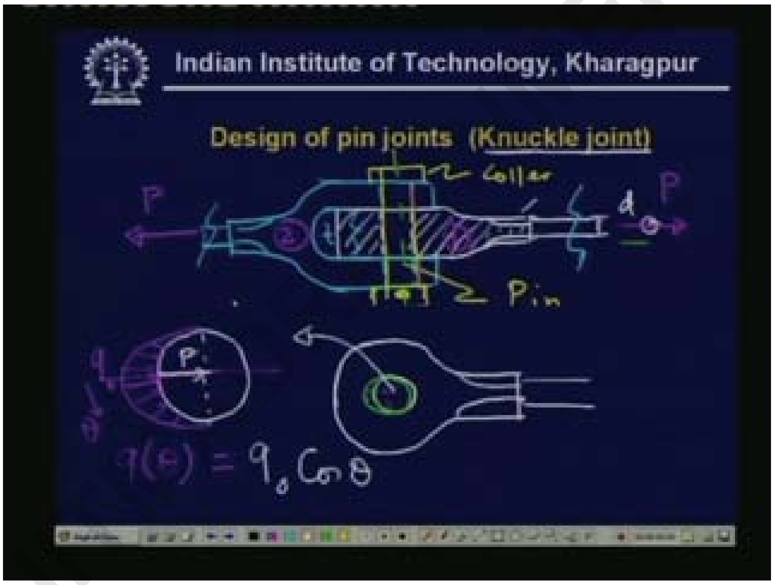

again let us draw so now if this pressure here is the pressure acting if the pressure is too large then this surface gets damaged

so although it doesn’t fail completely but there may be elongation here so the pin shifts here by damaging this surface so that has to be checked because if this is this is damaged then of course the material gets weakened and ah there there is a tendency for further propagation of different cracks and ah failure ultimately leading to the failure of the entire joint

so what is that bearing stress this is known as the bearing stress that is stress developed due to the contact what is that bearing stress that clearly depends on the pressure profile developed here so here we do not know exactly what is the pressure profile developed we have to make some judicial assumptions and there are few assumptions there are two assumptions which could be made for example if you take the large okay the the hole over here that is this part then the pressure may be distributed such that this is there is maximum is q knot

so the distribution may be a cosine functions if you measure theta from here theta from this direction so q theta may be equal to q knot cosine theta

so this is one such distribution so there may be another distribution thus q theta is independent of q ah of theta so q theta is um is a constant so this is uniform pressure distributions

now if you take this kind of pressure distribution then what will be if you ah then what will be the value of q knot because we have to satisfy again the force balance equations so if the force balance equation is taken so the entire um the summation of this force that is the total force this is distributed hence the total force will be equal to P

so if you take that into account and then you can write down the force balance equation

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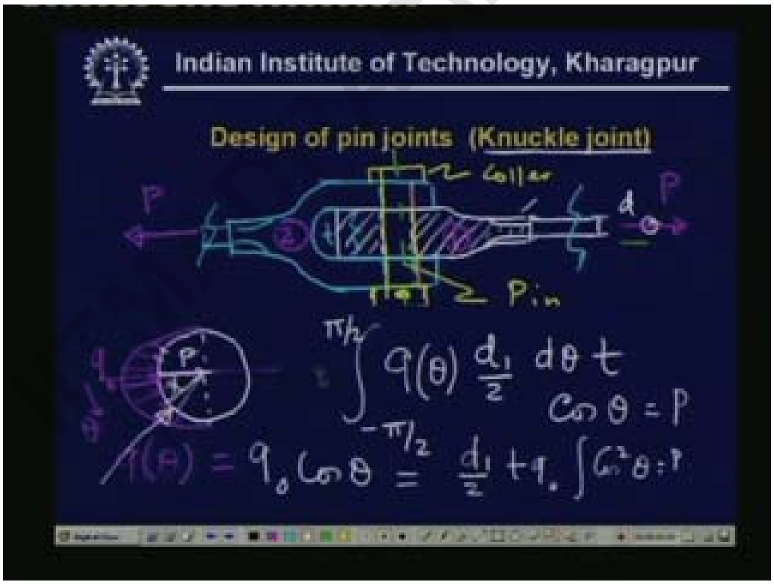

that is if you take a small area here at angular distance theta so we get the total force acting is q theta times r or d one by two d theta

so q knot d one by two d theta times t i am sorry this is q theta so this is the force developed on on this surface and this acts at an angle theta

so therefore the horizontal component will be this times cosine theta and will have to integrate from minus pie by two to pie by two and that must be equal to your given pressure P so from this relation we can find out now you substitute ah in place of q theta this expressions q knot cosine theta so therefore it becomes ah d one by two times t so this becomes d one by two times t q knot and cosine square theta from minus pie by two to pie by two and this becomes pie by two so so therefore ah what you get after that if you just do that then it becomes ultimately

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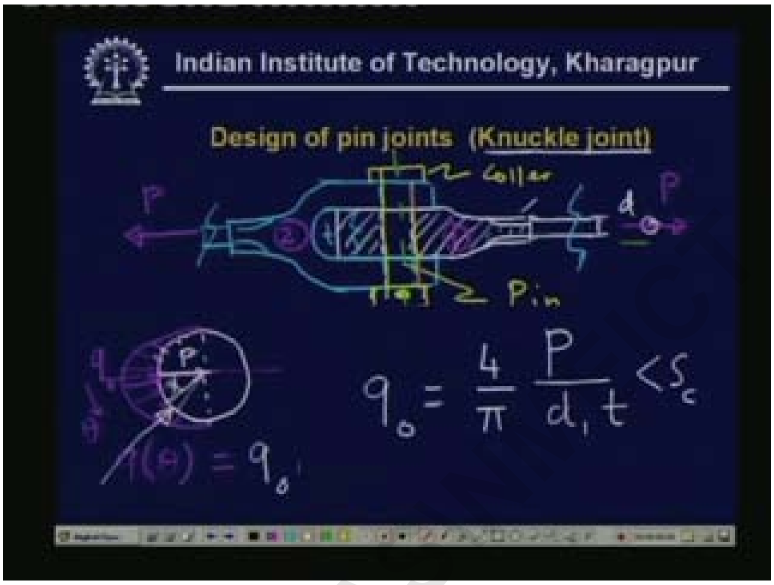

q knot is equal to four by pie P divided by d one t so this is the maximum pressure developed here

now this this is the again the maximum pressure and this must be less than Sc that is the crushing strength of this material

so the material may be crust so q knot may be less than Sc but this is very complicated assumptions but one simple assumptions could be that q knot is a constant ah q knot is constant and q theta is constant again equal to q knot so therefore we have in the second assumptions we have the we have this following case

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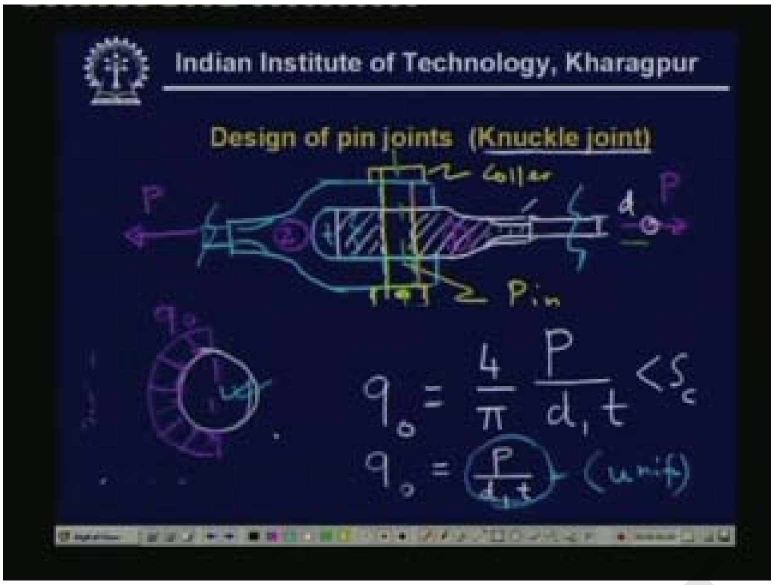

that is pressure is uniformly distributed

in that case if you do this same analysis then you can find out this is nothing but we can have this following expressions that is q knot is equal to ah q knot times the total area which is equal to d one times t is equal to P so therefore q knot will be equal to P divided by d one t equal to P divided by d one t

so {thereof} (00:35:13) therefore you see that if you use this ah let us say that this is based on uniform pressure theory uniform pressure so in the uniform pressure theory we have the q knot which is little lesser than that here this is multiplied by a factor four by pie which is greater than one so therefore we have more pressure over here

so therefore if you design with this then we have a conservative design that is ah now we ah we have q knot is more more than this value so if you want to have a design if you want to make the ah or select the value of d if you use this then you get a larger value

but nevertheless normally we use because the we want a very simple design not very complicated because we should not ah we we want uh some answers in the simplest way therefore we design based on this second assumptions and here in future also when we want to find out the crushing strength then normally we use this uniform pressure assumptions

so in this uniform pressure assumptions we have this following expressions now this strength criteria based on that is again

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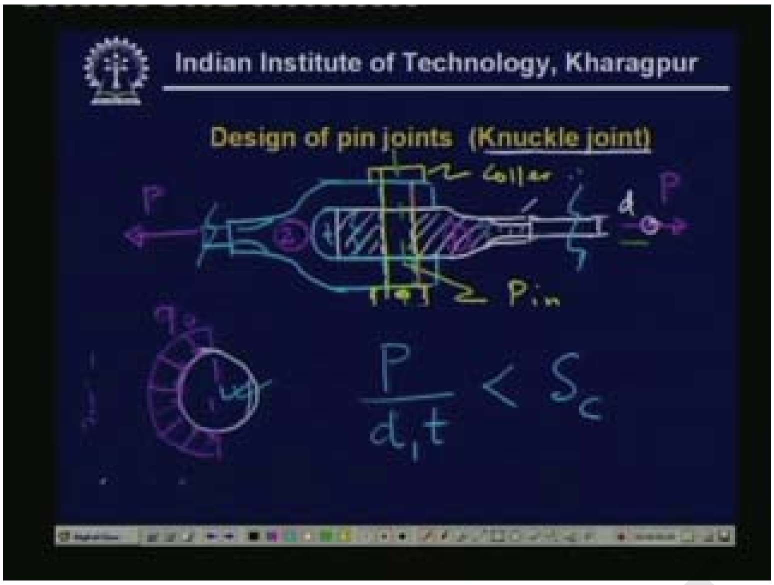

P divided by the total area that is d times d one times t this is the projected area and this must be less than Sc this is the crushing strength

now the crushing strength is more than definitely the tensile strength because the material ah is very tough against crushing

so this is in another again one another uh one formula which is to be used to check the uh safety of the design of this I bar

now this this this are the three ah things which one has to consider while designing this I bar

now the next important thing is the pin now will have to see what are the different modes of failure of a pin

so here let us see the pin

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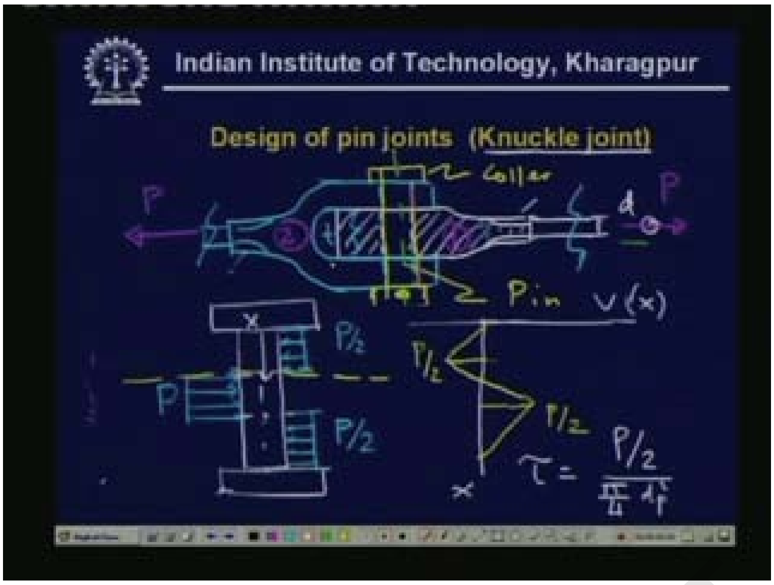

so this is the pin part we have a collar here and this is again the bottom part which is a rink now this part so therefore this part is connected to this fork end and this part the middle part is connected to the ah to this I bar end

so we have a different kinds of force pressure distributions on this pin again we have to make various assumptions the first simple assumption is that the force is uniformly distributed so here the pressure is the total force acting is P by two here the total force acting is P and again here the total force acting is P by two

so now this is one very simple assumption and we have to check against failure now you see this is just like a beam which is subjected to transverse uh force

so we can draw various things like this shear force diagram if you draw the shear force diagram let me draw the shear force diagram v x and x is the length along this beam so let us measure this length x here

so the shear force diagram will look like so there will be this one and then its starts increasing and it reaches maximum over here and then it decreases so this value is P by two and this value is P by two

again we can have bending moment diagram as well but let us now ignore this bending moment for the first time and see ah how does it fail

so since the shear force is P by two maximum shear force is P by two and it appears in this sections so therefore the shear stress the average shear stress can be taken to be tow is equal to P by two divided by the area of cross section which is equal to pie by four dp square so this is the shear stress developed now again we know from the basic theory of ah oiler beam that the the exact distribution of shear stress is not the average but the maximum shear stress occurs somewhere in the neutral section in the neutral plane and this is roughly for a ah for a ah for a rectangular bar it is {bou} (00:41:30) roughly about one point five times the average value the maximum shear stress but for a {recta} (00:41:35) for a circular cross sections we have something different we have to use the formula that tow is equal to v we have to use this formula where if you want to use a precise will have to use precisely the form for the shear stress then

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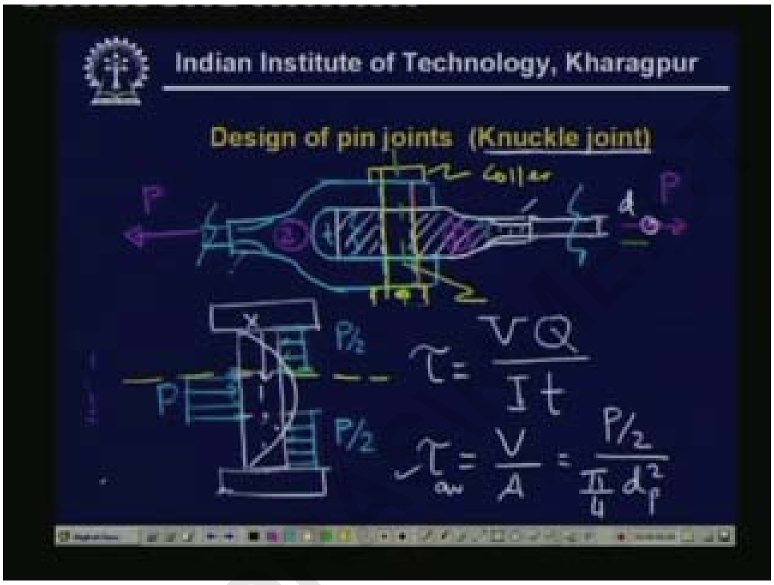

tow is equal to V times Q divided by I t where I is the second moment of area of the cross section and Q is that section ah this is the first moment about some area which you know of course then V is the total shear force and t is the thickness of that particular sections

so this gives more exact result but for ah preliminary calculations we can of course use the average value tow average which is equal to V by A which is for this case is P by two pie by four d pin square

now t average if you use this formula t average must be lesser than some value and we have to calculate dp from it

so this is one way of calculating but there are {furth} (00:42:54) further complications here we have ignored the bending moment now there is always some clearance between between ah the pin and ah this I bar or the fork end so therefore there is a tendency that it will bend and it bends something if if this part is so it bends like this so we can draw the bending moment diagram and see what is the effect of this bending stress

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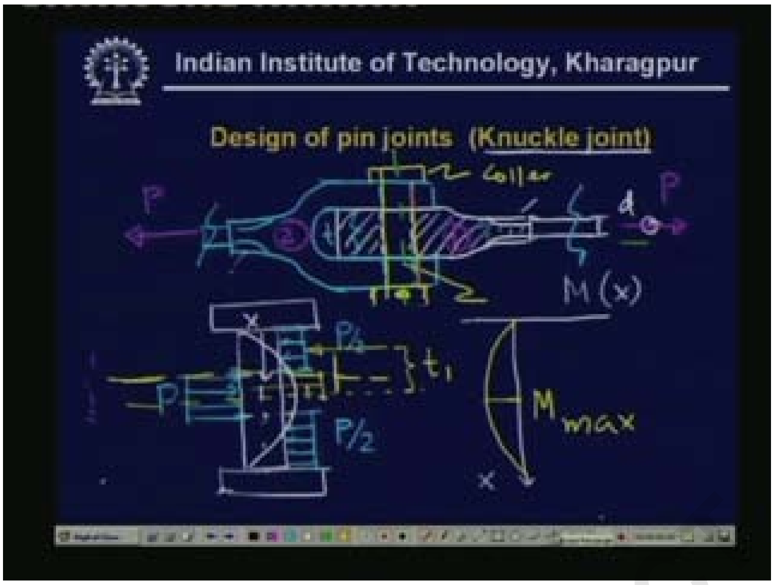

now if you draw the bending moment diagram again the same axis then if you remember the earlier uh drawing for the shear force then the bending moment diagram looks somewhat like this

so this is the bending moment diagram and the maximum bending moment occurs over here and what is the value of maximum bending moment that maximum bending moment of course will be equal to P by two yes this force is P by two times this distance minus P again P by two here times this small distance

so you can you can calculate this knowing this few distances one is this distance let us say t one another is distance from this point to this midpoint or or this sections so we can find out what is this M max over here

now once this M max is found out from this chosen distribution then we can find out the bending stress the bending stress formula is very well known that

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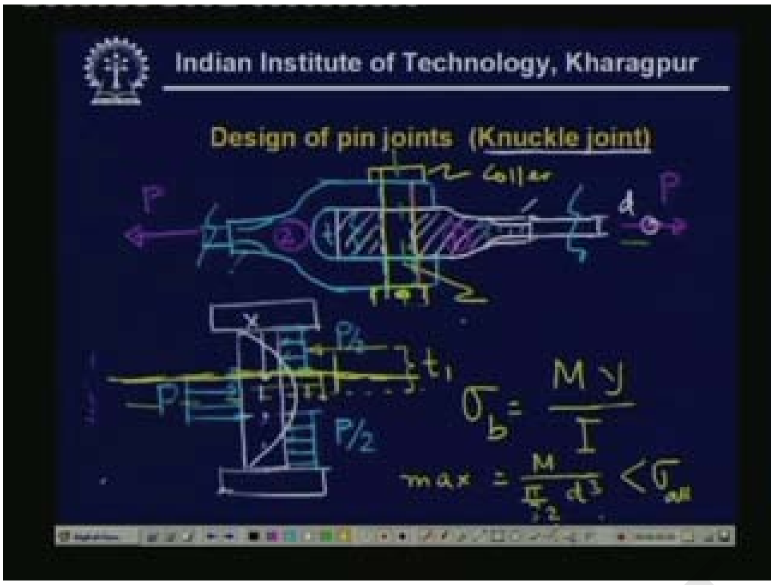

sigma bending is equal to M y by I where y is the distance from uh distance of any point from the neutral axis and I is the second moment of area now if you take the circular ah cross section then of course the y by I by the maximum value of this bending stress

now this has a maximum value the maximum value will be equal to M by Z where Z is equal to pie by thirty-two d cube so if you use this formula then we can find out maximum bending stress and this maximum bending stress must be less than sigma allowable

now this is not the end because here the where the maximum bending stress occurs the shear stress is zero but there may be other sections where the shear stress and bending stress are simultaneously present

for example take this sections this sections along this we had the maximum shear stress if you look the shear force diagram

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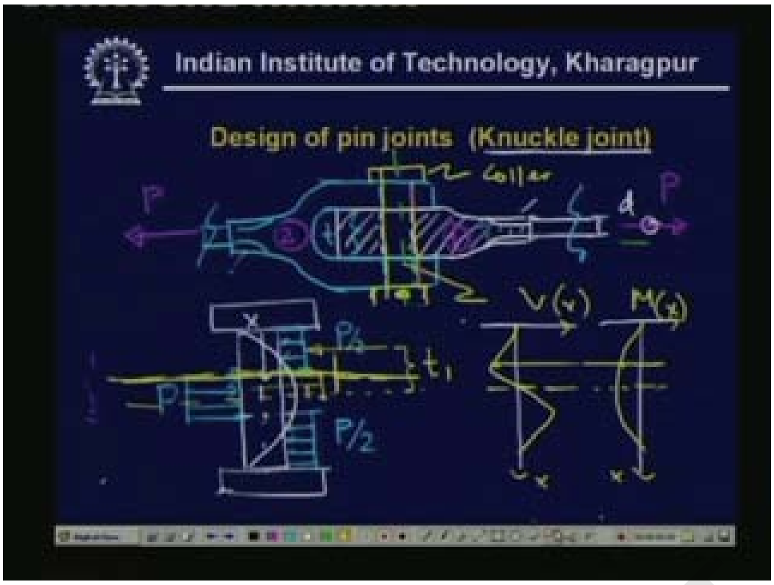

if you looks the shear force look at the shear force diagram this was roughly the shear force diagram and the bending moment diagram so this is V x this is M x positive here so x x

so you see that here bending moment is bending moment is maximum but shear force is zero in the mid pin but here we have definite some value of shear force as well as bending moment

once we have that then we have both the normal stress and the shear stress then we will have to use {dif} (00:46:53) other fail theory of failures

we if the material is a brittle material then of course we will have to find out the maximum principle stress and the maximum principle stress if there are uh then the maximum principle stress is given by this form

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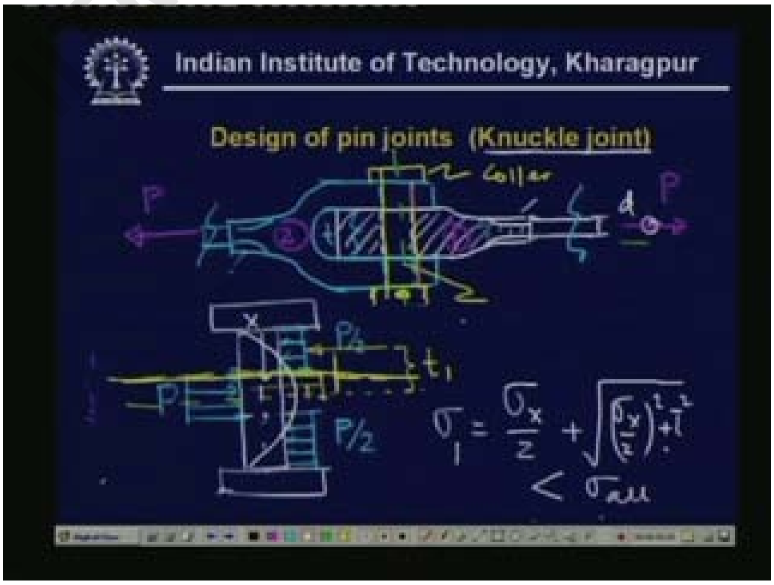

it is sigma max sigma one is equal to sigma x by two plus sigma x by two square plus tow square so this is the maximum principle stress and this must be less than sigma allowable

so this gives one expression from which the value of d could be found out

if the material is again a ah {bri} (00:47:48) a ah a ductile material then we have to use the other theory of failure criteria namely the maximum shear stress for that purpose the maximum shear stress

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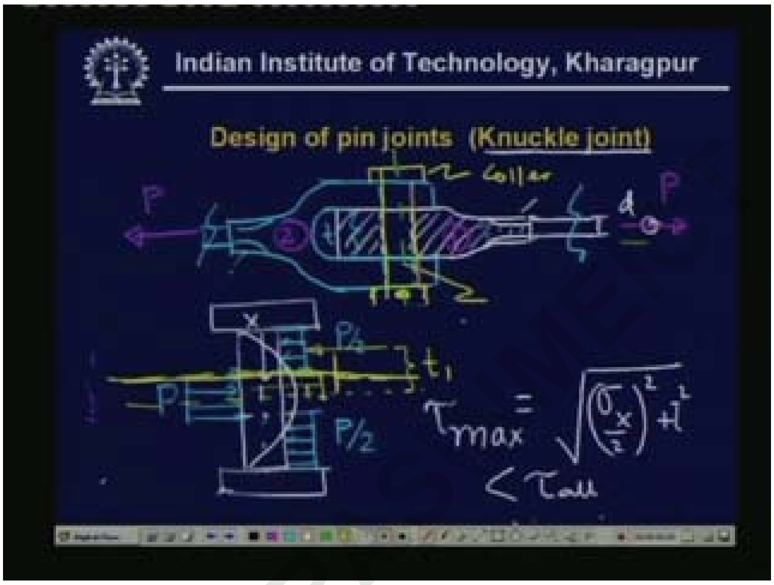

tow max is equal to sigma x by two square plus tow square

if we use the Von-mises theory of failure then it {liks} (00:48:34) looks somewhat different they are the equivalent normal stress so this is ah for the {shesr} (00:48:41) maximum shear stress and this must be less than tow allowable

if you have the Von-mises theory of failure criteria then we can use different thing

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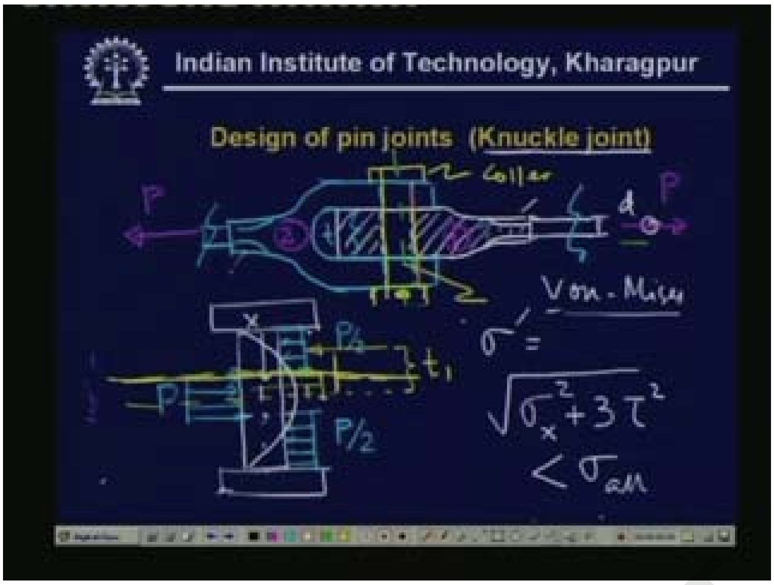

for Von-mises theory we calculate what is the equivalent stress and equivalent normal stress and that is equal to sigma square plus thrice tow square this is sigma x square plus tow square so this this is the equivalent normal stress and this must be less than sigma allowable this is Von-mises theory of failure Mises criteria okay

there are various strategies which could be used and ah we have to select accordingly

now this is for the this is for that case where the distribution is such if sometimes we have to use different other kinds of distribution that is let us take a different thing

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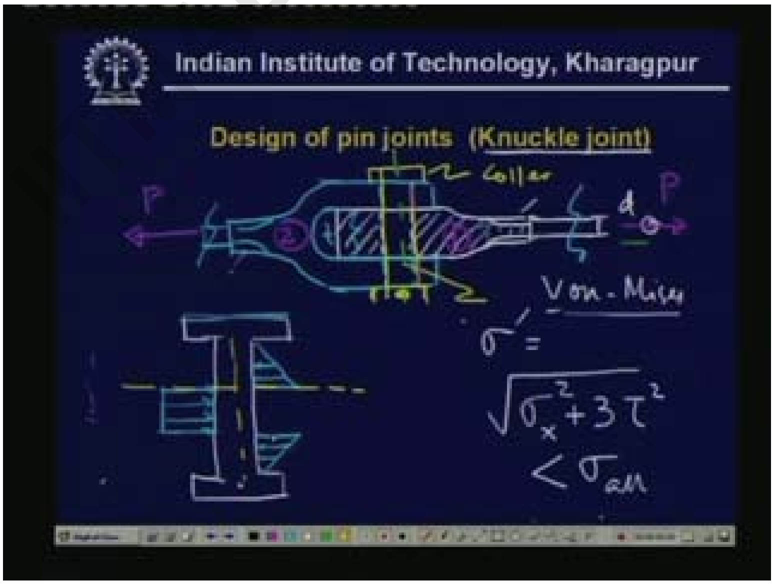

we see this is the beam and sometimes the following distribution is used here we have uniform distribution the middle part but there we take some time a triangular distribution this is one kind of distribution now here {als} (00:50:52) of course the analysis remains along the same line we have to calculate the maximum shear force and the maximum shear force definitely occurs over here again the value is equal to P by two so the maximum shear force is P by two

so the average shear stress is again the same P by two divided by pie by four d pin square but the moment distribution is totally different as well as the shear stress distribution is ah shear force distribution is different moment distribution is entirely different and we have to calculate then if you want to calculate ah accurately using those criteria that is Von-mises criteria or shear stress criteria or {ma} (00:51:38) principle stress criteria

then we will have to calculate again ah what is the bending moment {firh} (00:51:45) first at different sections what is the maximum bending moment then we will have to calculate the bending stress over there and also what is the ah maximum stress developed throughout this along the length of this pin and with with the help of those data we select the proper dimension of this uh pin

so this is the way how pin is designed then what remains is the part two and part two design goes along the same line because if you open up that part two then what you see the again

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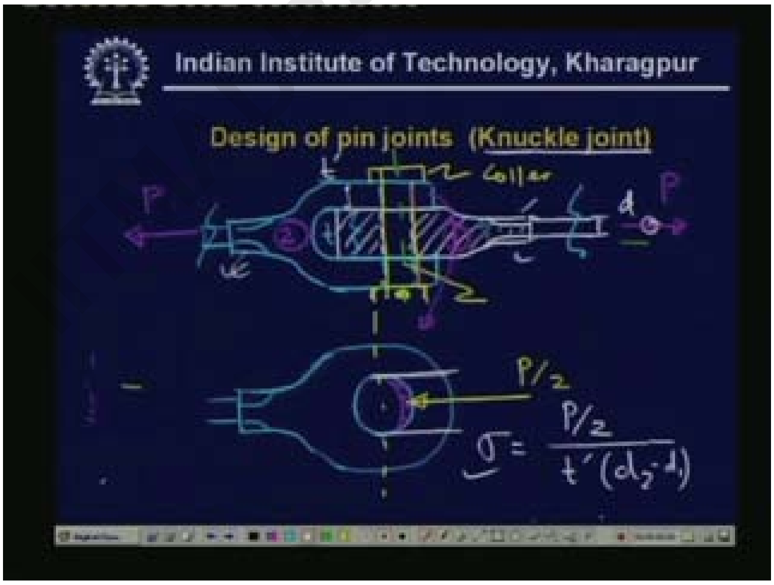

it looks like the same but now instead we have here for each fork we have P by two load

so here the distribution pressure distribution is such that this this one takes each one that is let us consider the top one this takes a load of P by two

so this is nothing but P by two

so each fork here if it has to break then the dimension this sigma will be equal to P by two divided by the area area is now if this is t prime let us say t prime times the difference that is d two minus d one if we use the same notations

so this is the sigma here again it can fail along this line so so this is one way of failure it can fail by shear along this line so we will have to use the um free body diagram and use the the same formula of failure as that of the I bar uh which we had already considered so instead of they there here instead of P we have P by two all the time because the net force taken by each of the fork is P by two again it can just crush over here so the crushing strength has to be checked so all the three criteria of failure uh ah all the three modes of failure for uh the I bar will be same as that of the uh this fork end

so this is how a machine element has to be checked for failure and this using those failure theory we will have to design ah ah the proper dimensions here the the measured dimensions are this rod ah diameter then the pin diameter as well as the thickness of the of this I bar thickness of this fork end etcetera

there are other things which are not of importance that is say this ah um the roundness ness that ’s it’s the ah the curvature of it it is important because you see if there is no curvature here then lot of stress concentration will be developed but that analysis is normally not done and this is left to ah um left as i am left to the practise there are hand books available will have to always look to those hand books and there are standards there

so will have to comply to some standards so in standards if you see if you look at ah any of the standards in a hand book good design hand book then see that those dimensions that is the curvature of this I bar here ah of this fork end this is related to the measured dimensions d so this is how a normally ah those those things are to be selected

once this is selected then of course we can go for further very ah um very detailed analysis using various numerical methods like ah fem or many other things and then we can check the safety of the design but for preliminary safety design we will have to ah rely on the theories of failure using proper sections and ah using the methods described in the theory of failure uh which was ah which were taught in the last classes

so this is roughly a ah um the guidelines for a pin joint design we will learn many things about the fasteners in the coming classes

next class we shall devote we shall study some other kinds of fasteners which is known as cotter joint and of course the key joint and ah lastly some spine elements so this is all for today we shall continue uh in the next class so thank you very much

Lecture No. #14

Design for Fasteners – II

lectures on machine design part one

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this is lecture number fourteen and the topic is design of fasteners this is the continuing part of the lecture on the same topic

now in the last class that is in lecture number thirteen we studied how to design a fastener that is a mechanical joint now in that class we have designed how designed what is known as a knuckle joint that is the fastener is pin type joint now ah we are going to study different kinds of joints today ah mostly we are going to concentrate on a type of joint which is which uses a uh mechanical element which is known as cotter

so let us go to the topic proper and let us see what are the different types of cotter joints

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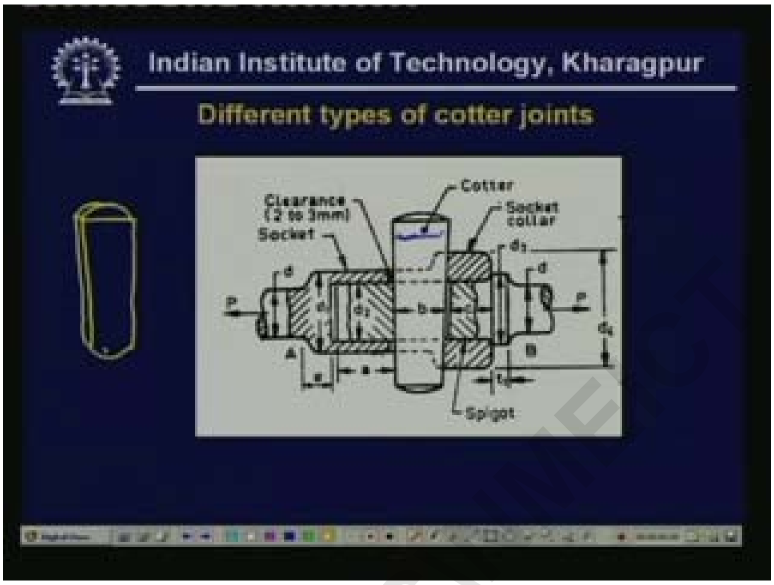

now [Vocalized-Noise] what you see here is one kind of cotter joint and here you see this part there are two parts which are to be joined by means of a cotter

now this is known as a cotter a cotter is like a pin but it has square cross sections if you if you want to draw the cotter then it looks something like this so this is a cotter in three dimensional