Design of Machine Elements – I

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Lecture No - 26

Design of Joints With Variable Loading

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let us begin the lectures on machine design part one

this is lecture number twenty-six and the topic is design of joints with variable loading

in the last few lectures we have learned

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how to design ah the {st} (00:00:51) joint for strength for various static loadings

now in the last class we have learnt how to design the joint for eccentric loading and in this class we are going to study for the variable loading

now variable loading is very important for any machine components in structural ah joints ah normally the loading is static but in machine components there will be invariably some movable parts and because of this movement there will be the forces which are normally ah fluctuating in nature so these are to be taken care of while designing a joint typically for a machines

think ah the example of a machine which contains a rotating member or {reciprotect} (00:01:36) reciprocating member definitely there will be the forces which are also ah oscillatory in nature and that may cause failure because of the fatigue in the joints

so let us learn how to design the joints for variable loading

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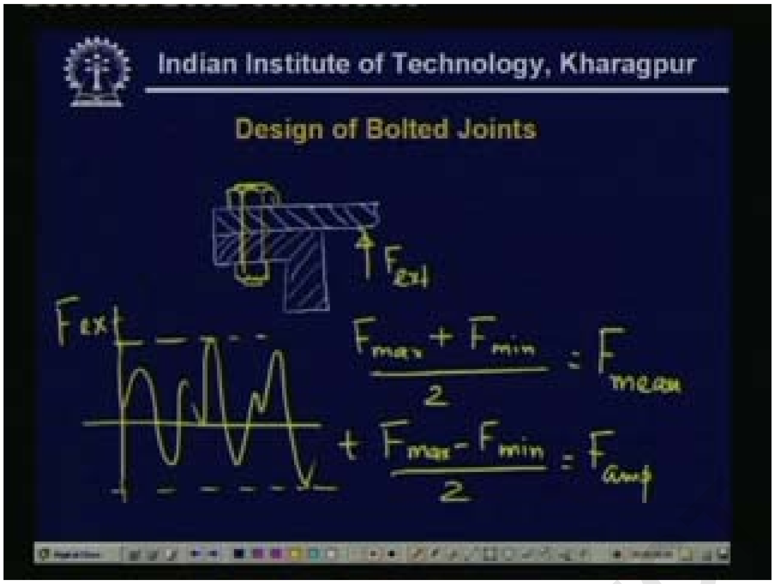

now first we study how to design the {joih} (00:02:05) bolted joints for variable loading

now in uh in regarding {connec} (00:02:11) in connection with the joints of uh um joints containing screws {fah} (00:02:17) and bolts and nuts we have studied a little bit about the variable loading so the here the discussion will be brief

we consider one example first let us take the following joint there are two members which are connected this way

now let us say this is one part and this is another component and we have a bolted connection over here so here is this bolt and here the loading is let us say here which is the F external load and that is fluctuating nature

so let us say the F external is something like this little fluctuating nature so here first of all we will have to find out what is known ah as the mean stress or mean force now {shf} (00:03:49) mean force is found out once we know the maximum force if we have F max the maximum value this is the maximum value and the minimum value F min this is the minimum value so if you take the arithmetic average then we call the force to be F min force

similarly we can define what is the force amplitude and that is equal to F max minus F min divided by two and this is F amplitude now this data must be known to the engineer who is going to design such a joint now once this is known then how to select the the ah the proper bolt that could be done by standard techniques ah by {diff} (00:04:59) different failure ah different criteria namely the Soderberg equations or Garber line or Goodman’s line let us discuss little bit a various things now here we have to know first of all

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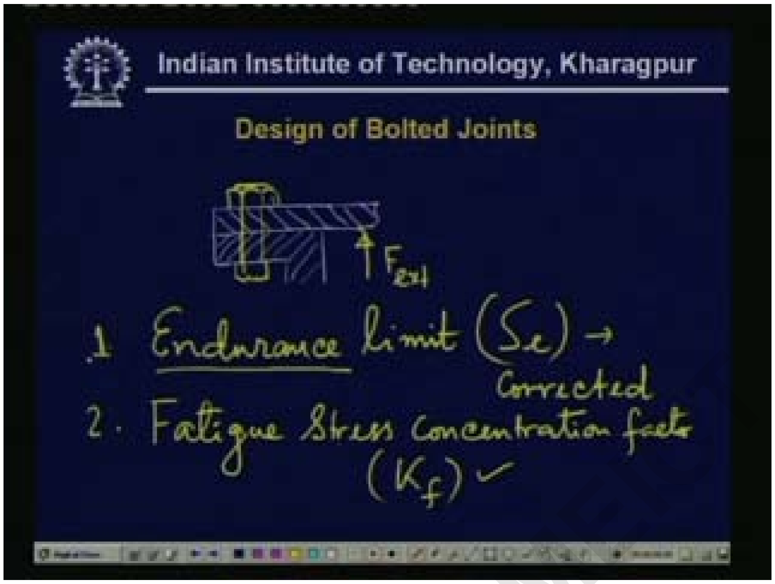

what is known as the endurance limit endurance limit normally Se now ah the corrected endurance limit depends on various factors for example it may be ah the nature of loading which is important so there {mah} (00:05:45) uh there is a factor for the uh type of loading normally the endurance loading is calculated or is experimentally obtained by making some experiments with the bending

now if the loading is something different then will have to make some provisions that is makes some corrections for the change loading and that is taken care of by introducing a load factor

then there may be factor which is ah due to the surface finish of the uh of this component

so the surface factor is there temperature factor is there so all these things must be taken into account and this endurance limit must be corrected so what we need is a corrected endurance limit for the bolts they are there are various specific um ah corrected endurance limit for different kinds of different kinds of bolts of different sizes uh they are standard so we know this endurance limit of course we know the stress concentration factor now this is again fatigue stress concentration factor fatigue stress concentration factor this is Kf so this must be known

now concentration factor ah we have theoretical concentration factor but when uh the material or the component is subjected to fatigue loading then the fatigue concentration {fac} (00:07:18) stress concentration factor is not equal to the theoretical stress concentration factor and we have to take into account what is known as the notch sensitivity

uh so when everything is taken care of then we can find out what is known as the fatigue stress concentration factor again ah there are various um uh values available for Kf and those are given in the machine design handbook

so these two things are known now we can go for designing

first of all we will have to find out we will have to see which criteria you should use

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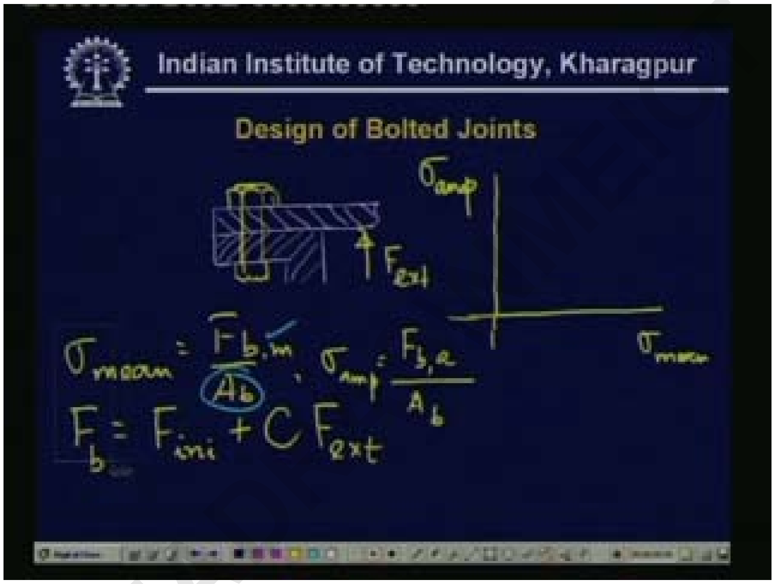

here we plot in this axis sigma m which is the mean stress and this is sigma amplitude this is the stress amplitude

now sometimes sigma for this case if the external load is given then we know that the load acting on this bolt will be equal to ah the external load fully if there is no pre-tensioning but if there is initial tension then it will be initial tension plus a factor multiplied by F external so this is a fraction which a fraction of F external which acts on Fb so first of all we will have to find out sigma m that is sigma mean with this Fb that is the bolt stress

now F external is variable F initial is normally uh constant so it is not varying pre-stress is of course a constant value so sigma mean will have to be calculated from here now that will be equal to Fd b divided by A of bolt the normal design procedure we do not know what is Ab so we calculate sigma mean in this form Fb Fb is known Ab is not known that is our aim is to how find out this Ab

similarly we can write down sigma amplitude as Fb so {th} (00:09:53) this will be the mean of that and this is Fb amplitude and divided by Ab

so we will have to find out the Fb mean and Fb amplitude and then write down in this form then we will have to select the criteria now there are several criteria's

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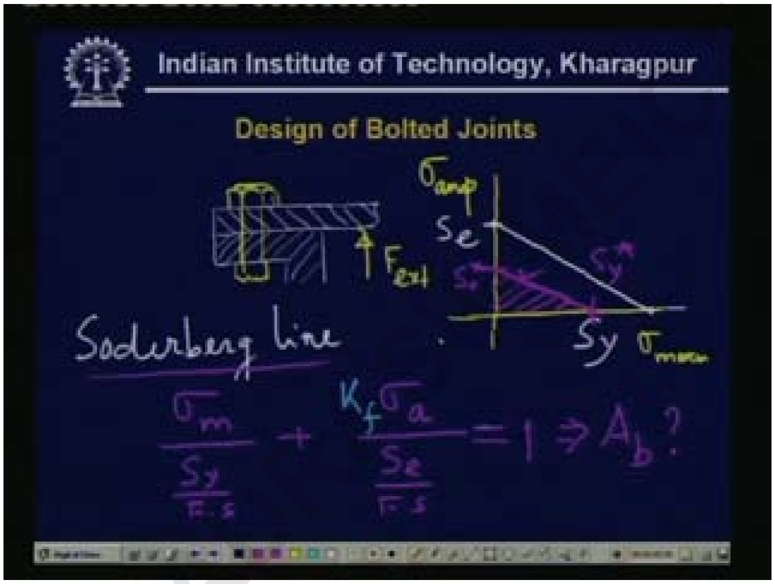

see [Noise] there is one thing which is called the Soderberg line which which is a line where this maximum limit is endurance limit or endurance strengths um and that must be fully corrected

here this goes to the yield point Sy and then this is the yield stress ah depending upon the {stre} (00:10:54) depending on the type of loading the yield stress may be different ah if this axial loading and ah that is pure tensions and pure shear the yield stress will be totally different so here the criteria is that so this is known as Soderberg line

now we select what is known as the factor of safety now the criteria is that we will have to design such a way that if this is the factor of safety we have chosen so now the design data that is sigma amplitude and sigma mean must fall within this line so what is that this is a fraction so now this point Se prime Se star and Sy star so Se star will be Se divided by some factor of safety factor of safety similarly Sy star which is this value Sy star let us say it will be Sy divided by factor of safety so all the design data must fall within this line for bolted joints normally the factor of safety is from one point five to two

so once we chose the factor of safety remember this is a reliability factor ah in order to make it ah make the joint reliable so we select factor of safety and we find out Se star and Sy star then the design criteria is that this must that is the sigma mean divided by Sy star which is equal to Sy divided by factor of safety plus this is the equations of this line particular of this line it will be equal to sigma amplitude divided by Se star which is equal to Se divided by factor of safety

here we have to multiply by this stress {con} (00:13:21) fatigue stress concentration factor Kf and that will be equal to one

so now if sigma a means again written in terms of ah F F external Fb and Ab so from this equation we can find out Ab so this is how we design according to the Soderberg line

sometimes we use what is known as Goodman's line and here the Goodman's line is so this is one way

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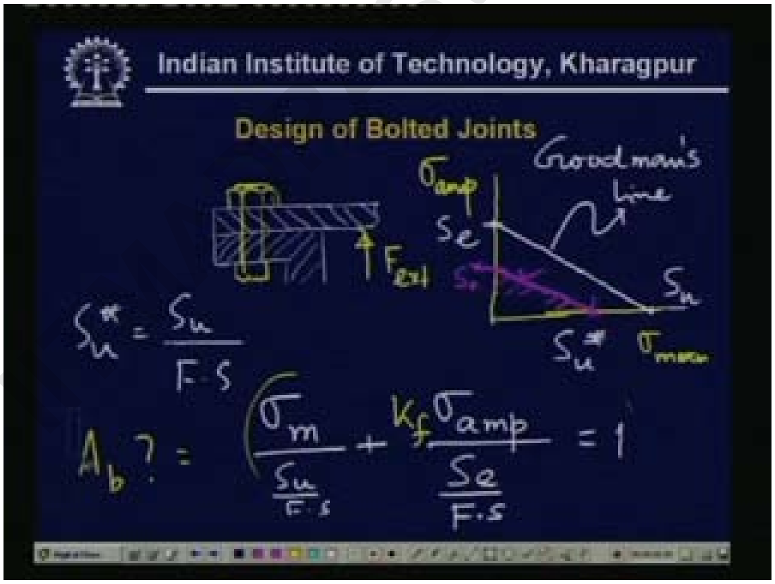

now the next approach is to use Goodman's line then then this is joint from Se to S ultimate here previously in the Soderberg's line it was uh yield stress yield strength now it is ultimate strength and again the same procedure we have Su start which is nothing but Su divided by factor of safety and this line is the Goodman's line so Goodman's line then the equation is sigma mean divided by Su star which is Su divided by FS plus sigma amplitude divided by Se star and here again just like before we will have to multiply the stress concentration factor for this ah fatigue stress

so again from this design we can find out what is Ab so that comes from these equations so this is Goodman's line again we have another criteria which is known as Gerber and line let us discuss that

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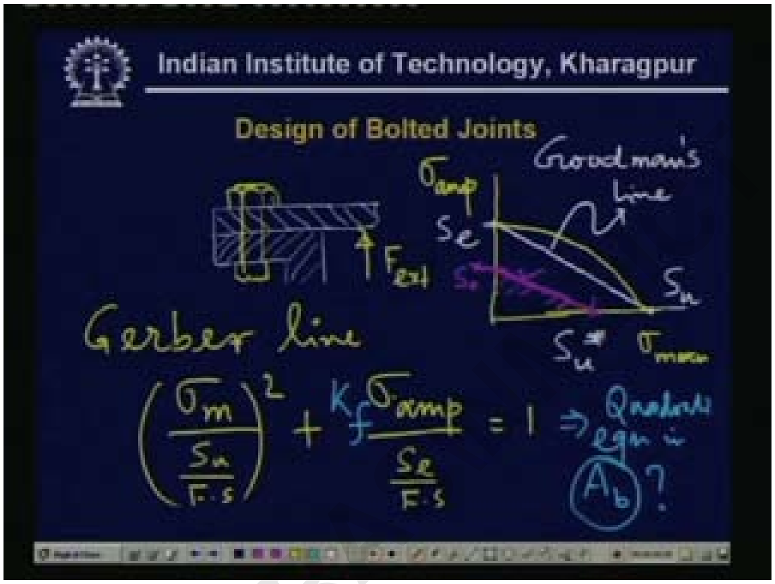

the Gerber line again is a line it's a parabola so this is the Gerber line and again the if you consider Su star and the Se star then the equation is written in this form that is sigma mean divided by Su star that is Su divided by FS square plus sigma amplitude divided by Se by FS equal to one and again just like before we will have to multiply by Kf

so here you find that this is a {quadrat} (00:17:09) this will give a quadratic relationship quadratic equation in Ab which you have to solve to find out what is this Ab

so these are different design techniques for bolted joints we discussed it ah um in some previous lectures so let us not um give more time to that but

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here you see the fatigue stress concentration factor normal fatigue stress concentration factor for different different metric grades the different metric grades are here you see so there are various materials available they are uh in ISO grade for three point six to five point eight for the rolled thread we have two point two fatigue stress concentration factor remember this is Kf for cut thread it is two point eight again because there are different ah um this is the different procedure of making the threads so they will introduce the different stress so therefore the fatigue stress concentration factor these two are different if you have fillet then we have two point one little less for metric grade of um of higher grade higher values then the stress concentration factor is again higher you see for cut thread it is much higher for fillet it is little lower side so these are roughly the fatigue stress concentration factor for the bolted design

so now now these are everything tabulated so when you want to uh design such a joint then you should make use of this um these values those are given in the machine design handbooks

for fully corrected endurance strength

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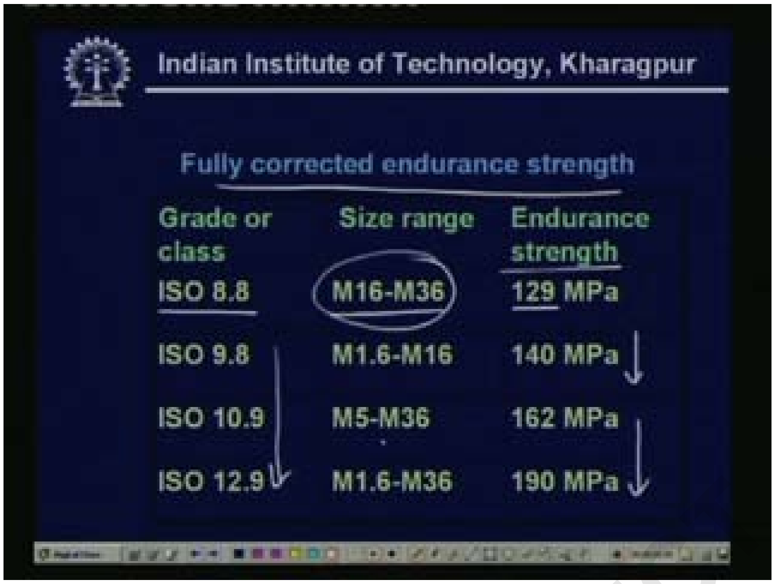

again you see for the different grades we have for different size M sixteen to M thirty-six remember this is the size of the bolt metric grade for the nominal diameter and the endurance strength again it is fully corrected endurance strength one twenty-nine Mega Pascal if you go on ah to the higher values then you see it goes on increasing okay

so these are the fully corrected endurance strength and those are given again in the handbooks so you must know that ah such data exists and you should use them while designing that

now let us come to the design of welded joint

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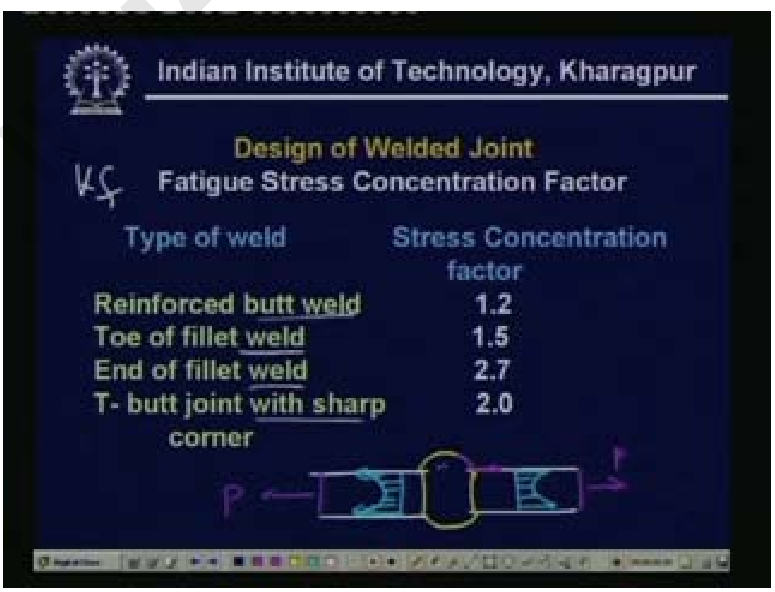

which is very important because in welding you will find the stress concentration is quite ah ah appreciable because ah because of various reasons that is we discussed already in the heat effected zone ah or during the welding process itself the residual stress may be generated or in the heat effected zone the material metallurgical transformation may takes place so this will all lead to some kind of stressing ah which will lead to the crack initiations so if crack initiates then it will propagate when the loading is alternating or variable

so the stress concentration will be important there remember in the static loading the stress concentration is not that important because always there will be some plastic deformations which will take care of the such ah enhanced loading enhanced enhanced stress but if the uh loading is fluctuating the stress is fluctuating then the crack can easily propagate so this is very dangerous

for different weldings we have different stress concentration factor so we have what is the Kf factor for here

now it also depends you see for different kinds of weld also it's not the material properties for different kind of welds here take an example of an butt weld if you have two such pieces which are welded butt welding then you see that there is a there is an abrupt change in the in the cross-sections if you apply a load here then due to this abrupt change if you find out the stress distribution there then the stress distribution is not uniform but it will be of this kind of nature so this is the stress distribution

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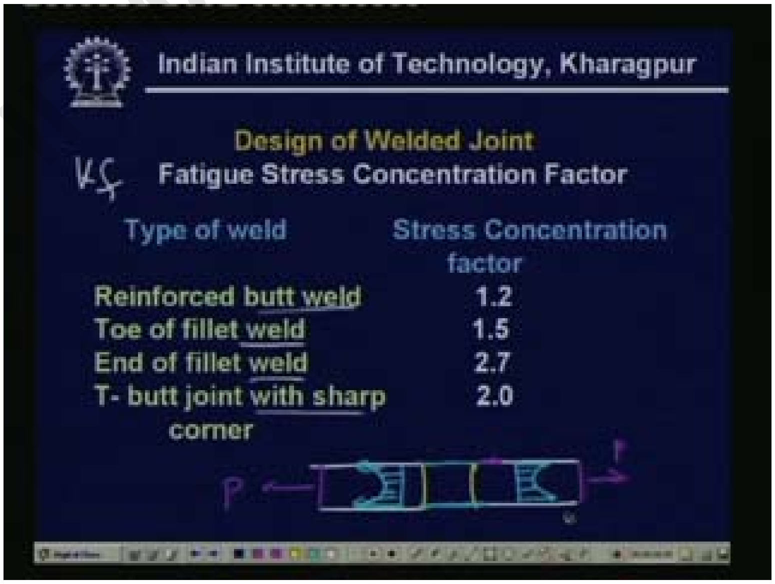

now how to avoid that we do sometimes we cut it off that is here and we remove this part similarly we remove this part so then the stress may be stress concentration may be reduced

if we have a fillet joint then the stress concentration is again quite important

if you consider a fillet joint

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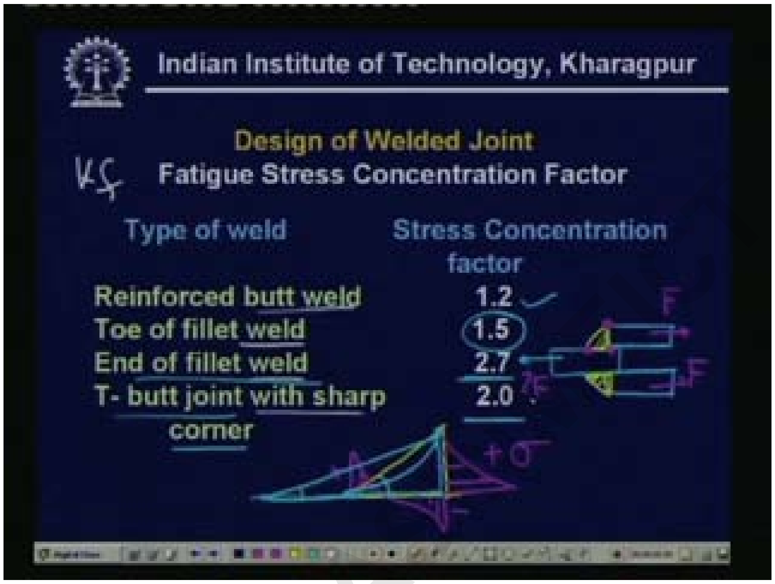

let us say we have a fillet joint given this form so this is one bar connected to so let us apply a load twice F it will be taken F and F and remember the fillet is here

now after careful ah experiments it was found out that if you consider this fillet then the normal stress distribution is something of this form so that is a positive normal stress and negative normal stress you see the shear the stress concentration appearing here similarly on this side the stress distribution is something like this again this is positive so this is sigma so therefore in this region this region and this region the stress concentration exist there are various ways to decrease the stress concentrations one is by ah making this shape concave profile or if we increase the leg distance that is normally this is forty-five degree angle but if we decrease this angle that is make this larger than that then the stress {conc} (00:24:14) concentrations may be reduced

nevertheless there will be stress concentration always and the typical stress concentration values are plotted here for reinforced butt weld this is one point two for toe of fillet weld this is one point five for end of fillet weld this is two point seven for T-butt joint with sharp corners remember {theh} (00:24:34) butt joints are possible T-butt joints are possible and so here we have uh two point zero stress concentration factor

so these are roughly the stress concentration factors which are again tabulated in the design handbooks

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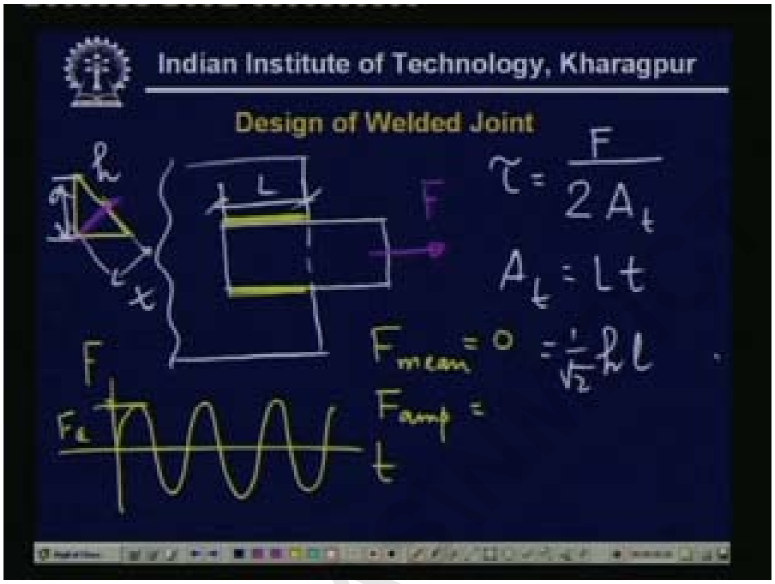

now with this data let us try to design some of the welded joints for variable loading let us first consider this case

again we have a plate which is connected to another plate by welding welding is done this is again this is the fillet welding now this is parallel fillet a longitudinal fillet the force is applied over here

now let us see let us take that this distribution of this force with time let us see this totally sinusoidal in this case you can find out this F mean equal to zero and F amplitude is nothing but this amplitude that is Fa

now in this case uh Soderberg line or Garber line or ah um what is called a Goodman's line will be of no use because right now we are having purely uh fluctuating stress purely oscillatory stress

so therefore we will have to consider ah the design first we find out the stress what will be the stress now definitely here the stress is shear in nature and if you consider this fillet there is the fillet cross-sections but the critical cross-section is not the entire cross section it is this section therefore the sheer force stress here we have definitely shear stress the shear stress will be equal to F divided by the total throat area that is two for each fillet and At where At is nothing but the length times the throat length which is t L times t and this is nothing but one upon root two times h times l where h is this height the size of the weld

so if we find out this tau then definitely this tau this maximum value of this tau will fall below certain limit

now what is that ah let us see for the successful design we must have

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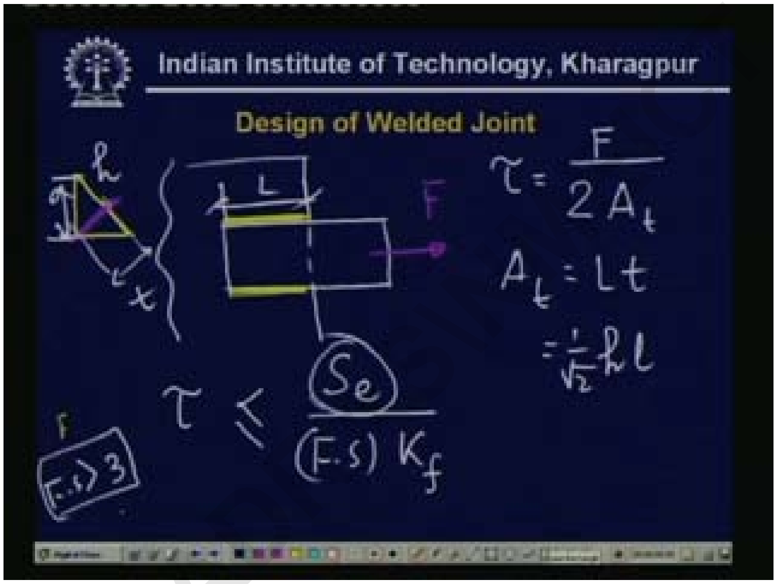

tau must be less than equal to the endurance limit now here endurance limit is to be calculated for ah um endurance limit for the shear stress because now the loading is shear type so this is again corrected and we will have to take into account load factor here so this divided by factor of safety and what is that factor of safety factor of safety is normally taken to be three to four so this is greater than three that is roughly the practice factor of safety is greater than three and then we have Kf that is the fatigue stress concentration factor

so we must have tau that is tau is less than Se divided by factor of safety times Kf

now remember Se endurance limit is calculated ah its tabulated when the ah longevity is infinite that is it will never fail but if we want the loading for sometime then we may not be so restrictive in this design we can find out the corresponding corresponding endurance strength for different loadings so normally if we have this Sn curve then

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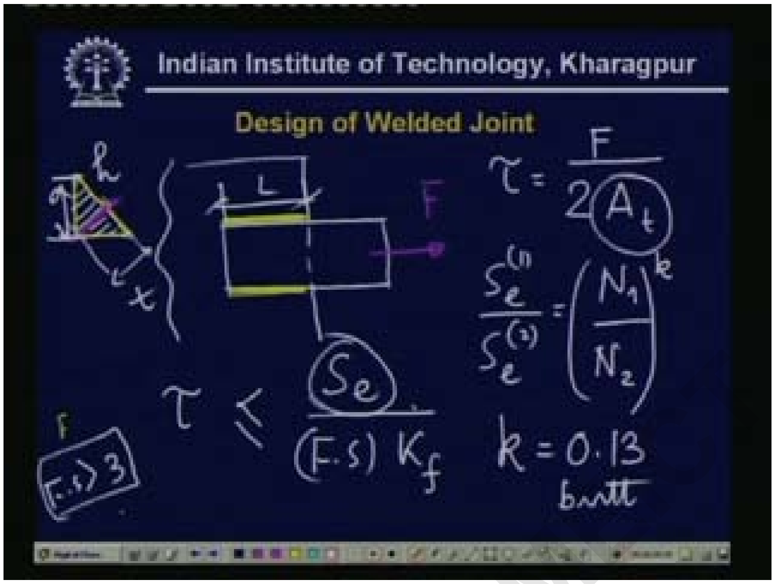

the endurance strength will be say Se one divided by Se two will be equal to N one that is N one is the number of cycles in which it will operate divided by N two times a factor k

so now this factor k is about point one three for butt joint and from point one five to point one eight for the fillet joint so this is how we calculate this Se for or that is the allowable fatigue stress we should not call exactly a theoretically speaking we should not call Se but this is allowable fatigue stress for different ah {cyc} (00:30:34) number of cycles N one and N two so this is how this is ah this design is done

now when we use this then we can find out what is this At and what will be the maximum value of this minimum value of this At i am sorry and with this minimum value we can design what will be the minimum length h or this minimum minimum length of the weld h or minimum length uh um of of this fillet minimum length of this fillet h and minimum length of the weld l to be precise

now this is done for the complete reversal of loading suppose we do not have ah complete reversal then what to do then of course we have to recourse to our original formulation that

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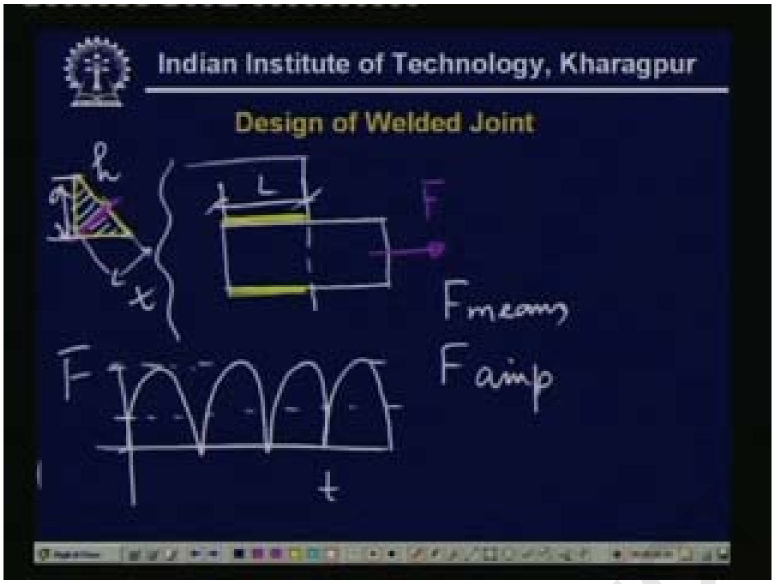

in is this case let us assume that F is not totally fluctuating but something like this so then we can find out what is F mean and what is F amplitude that comes from this F maximum value and F minimum value is zero over here so you can find out F mean and F amplitude and then we ah take the help of that ah um Goodman's diagram which we have already used for the bolted joints

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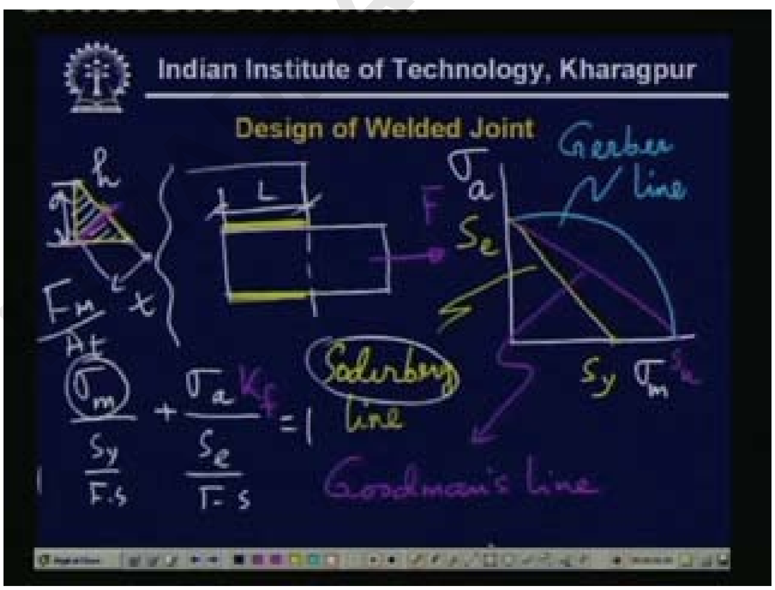

Goodman's diagram again for sigma m sigma a we can use different strategies we may use Soderberg line where this is from sigma e Se to Sy then this is Soderberg line we may take another approach this is with Su and this is known as Goodman's line or sometimes with Gerber line this is this is Gerber line there is of course this is the standard technique we can write down the formula that is for the Soderberg line of course it will be sigma m divided by Sy divided by factor of safety plus sigma a divided by Se divided by factor of safety and here we will have to multiply by Kf and that must be equal to one so this is for the Soderberg line

if we use Goodman’s line as we have just so here here ah what is not known is this At so that comes directly from this expressions remember sigma m is fm fm is known and that is divided by the total uh throat cross-sectional area so ah we do not know what is At we want to calculate At so here we replace sigma m to be Fm divided by At

now you see once you substitute Fm by At in place of sigma m and a by At in place of sigma a then this equation gives the unique value of At so from this we can design what is t or l or h if ah that depends upon what you want

so this is for the Soderberg line if we use the Goodman’s line then we have instead of Sy the only change is that

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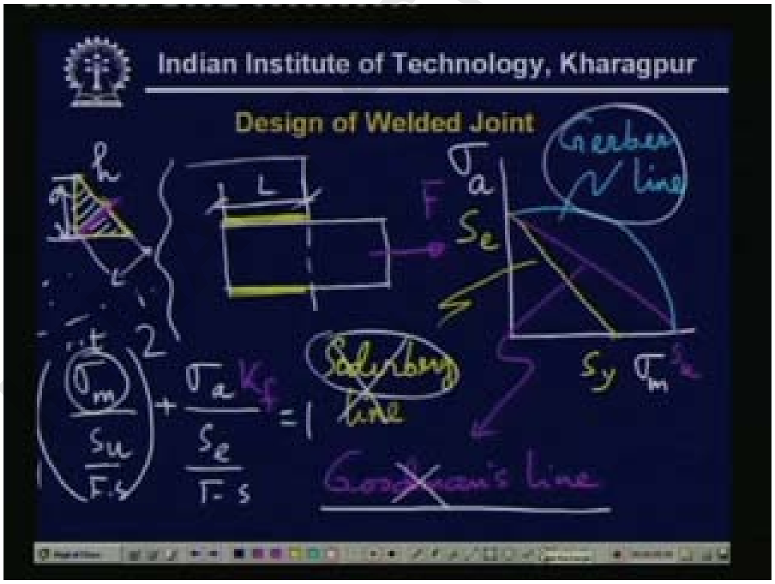

instead of Sy we have Su so this is now Goodman’s line

when we have so now again we can get the value of At when we have the uh Gerber’s line then what is done here the Gerber’s line equation is nothing but square of that let me erase all these unnecessary parts so this gives not he Goodman’s line for the Gerber line so from this here if you substitute again sigma m and Fm what you get the quadratic relationship in At which is to be solved you take the maximum value of At and with this you calculate ah either l or h or t h and t are ah related by this formula that t will be equal to h divided by under root two

so this is how the design is done in the normal way there is or another alternative way which I would like to discuss here

again we take this same example but now we want to discuss another way this is what is known as

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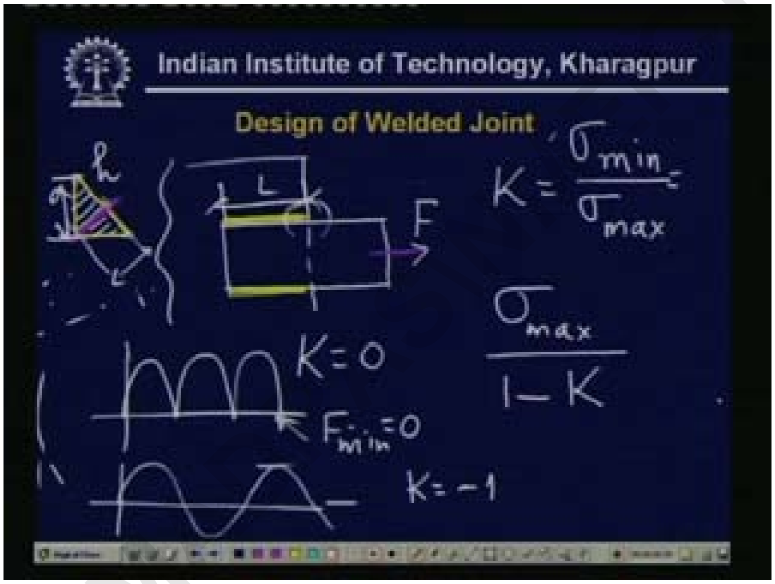

using what is called this maximum allowable fatigue stress for the weld

here the approach is that we find out what is the maximum stress remember we have the ah fluctuating stress component corresponding to F and then we have then the maximum value of F and with this data we calculate what is the ah maximum fluctuating stress and this maximum fluctuating stress that is the maximum value of this fluctuating stress should be below the allowable fatigue stress

now how to calculate the allowable fatigue stress those are again tabulated but there are various techniques in various countries ah for example in United States the the maximum stresses are tabulated in this form first we calculate for the {maximumable} (00:38:22) allowable stress for uh repeated loading that is the loading of this type where the mean value is zero that is the minimum value is F minimum will be here and that is equal to zero

with this we calculate what is the maximum allowable stress and uh that is again done by experiments and then these values when is calculate it that is sigma max let us say this is divided by one factor one minus K now this K factor takes care of this asymmetry in loading if K is defined as sigma min that is the minimum divided by sigma max

if we consider one cross sectional area the same cross sectional area then this is nothing but F min divided by F max

you remember here k will be equal to zero in this case because F min is zero for the repeated loading loading of this type so here K is zero and the allowable stress is sigma max itself for this kind of loading sinusoidal then F min is minimum value of F is nothing but minus of minimum maximum value of F so therefore K will be equal to minus one

K is plus one then this is not uh um well this formula should not be used but remember for equal to one minimum is equal to maximum that is only the static loading where of course we do not uh need to consider this ah this design procedure

so for different values of K's we have ah these values so this is the allowable loading for the {variab} (00:40:25) for the variable loading case this is the allowable stress and the maximum stress should be lesser than this allowable stress

in Russian literature of course the procedure is little complicated what we have there is again

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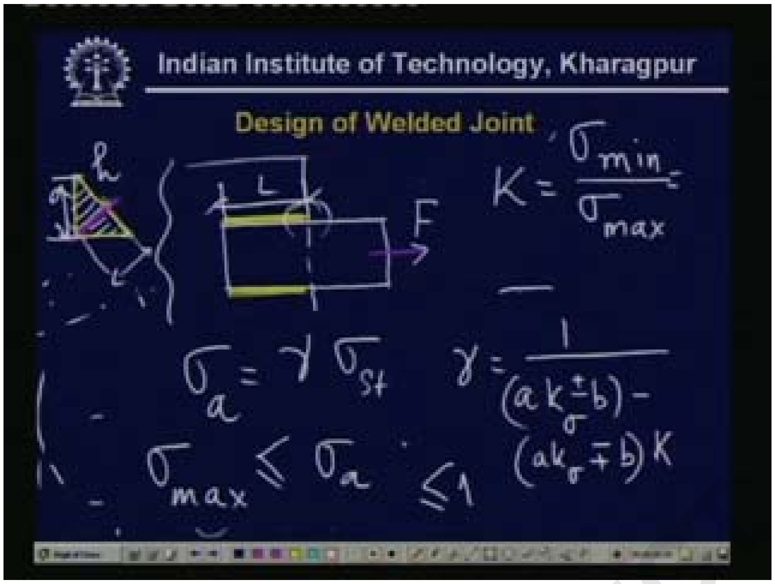

we calculate what is the static stress and that is that is maximum value for the static loading and multiply by a factor gamma then this gamma is it looks little complicated this is something there are few factors ka sigma minus at b there may be plus or minus depending upon compression or ah tension then this is ak sigma then reverse n times this K K is again defined this way

so gamma is definitely less than one so the value allowable stress is gamma times st

again if the number of cycle is not infinite infinite means more than two million cycles if it is less than two million cycles then of course this K sigma that is the stress concentration factor remember this K sigma is nothing but stress concentration factor corrected stress {conc} (00:41:44) concentration factor and those are again ah multiplied by another factor and then ah this is how the ah effect of number of cycles is taken care of then again ultimate design objective is that is sigma maximum that is the maximum value of the sigma must be sigma less than sigma a

so this is roughly the procedure

in Indian design handbooks you will find again this kinds of well the values are given for

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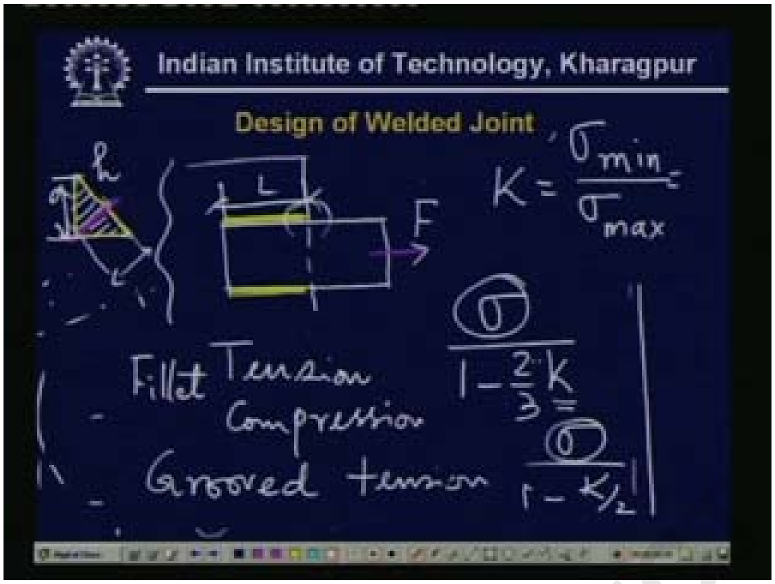

various size like the tension fillet weld for for tension for compression so they are given in the form of tables normally they look like some {cons} (00:42:46) some value sigma divided by one minus two third K where this factor is two third now and K is the same so once you know the sigma for different {valu} (00:42:57) type of loading you may calculate what is the corresponding K and then with this we calculate the allowable fatigue stress

for grooved ah butt joints for tension again we have something normally this is one minus K by two something where these values are again tabulated everything is tabulated so we look for ah that particular data for to calculate what is the maximum ah uh or allowable fatigue stress

so remember the design the maximum design stress must be less than maximum ah fatigue stress allowable fatigue stress so this is the design criteria

now for this type of joint it is it was relatively simple if you consider another little complicated joint

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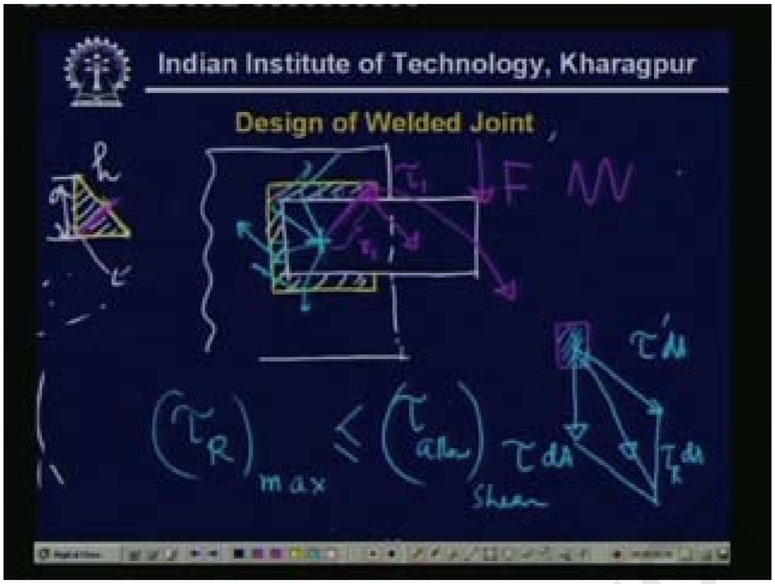

so if you consider somewhat complicated joint like this fillets are all around and and this is subjected to a load something like this and this is again fluctuating in nature

now how to use that conditions remember we studied in the last class that here the effect uh this kind of joint will have two kind of shear stress the shear stress which is the primary shear that is that will be equal to tau will be equal to F divided by the total throat area that is the throat area for this entire length the entire throat area so this is the primary ah um shear stress there will be secondary shear stress because now because of this kind of loading that is eccentric loading it will try to rotate about its center so there is a tendency of rotations

so what will be the stress developed again just ah like what we have done in the last class this part will have a stress tau one let us say tau one will be proportional to the {lec} (00:46:03) the magnitude will be proportional to this distance r one so similarly we can find out what is what will be the stresses individually at various locations

the proportionality constant can be find {frah} (00:46:18) out from the techniques which we had discussed before but the end result is that we know what is the secondary stress for various points

now if you consider a small section let us say if you consider this section enlarge little bit then what we have you have the direct shear tau and the shear tau prime then we calculate what is resultant

now remember we cannot find out the resultant vectorial resultant stress vectorially but what is done is that this area is constant so the force will be equal to this times dA the small area and this times dA will be the force over this sections so therefore this in resultant force is tau times tau resultant dA so therefore the resultant stress is tau resultant dA divided by dA will be equal tau r

so now we calculate what is the r maximum as a resultant stress we make these calculations for various points for different points we make these calculations so we know the secondary shear force and the primary is always there so therefore we calculate the resultant shear force for all the points and we calculate the maximum shear {forh} (00:47:46) shear stress acting over this sections and then this maximum value will be less than equal to this allowable stress that is what we now have is tau R the maximum value must be less than this tau allowable again this is for shear only for shear only

so this is how this kind of joint ah um this ah this kind of joint is to be designed

now if we have another kind of joint let us say a joint where different kinds of stressing is possible remember up till now we have considered only those joints where the loading is of ah uniform nature that is either shear or in tensions but suppose you consider this kind of a joint

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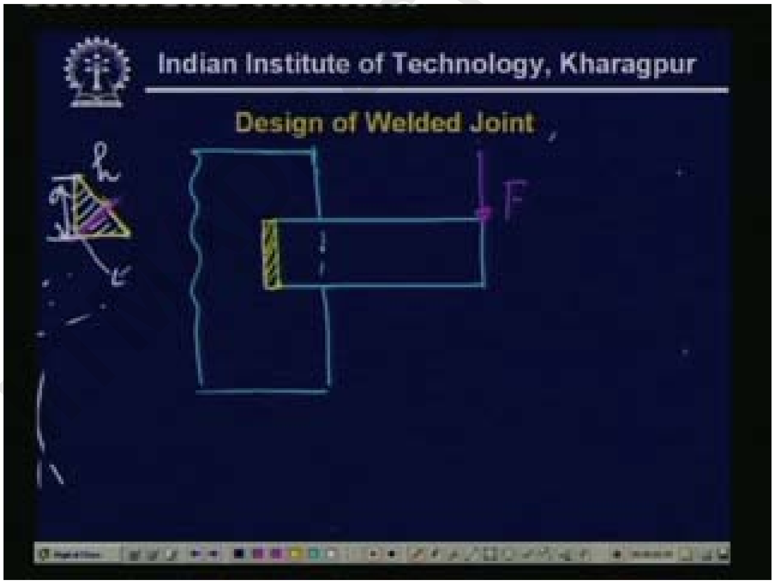

this is another plate now here we apply load this way F but now the welding is here in this sections or ah in order to make it much more precise we may take little variation of this example

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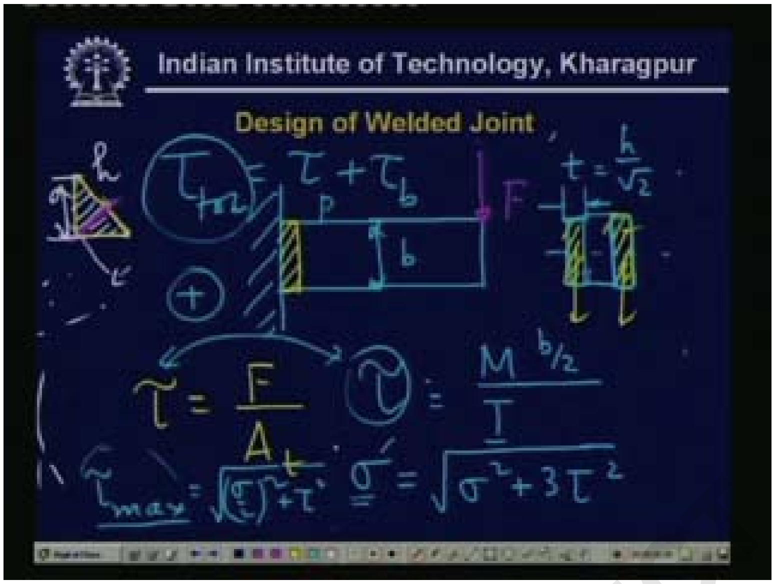

just consider a cantilever beam where the welding is done over this part

now what will happen here here you you will have the direct shear stress first so this tau will be equal to F divided by At so remember if we look from this side then we have here and this weld melts are on these two sides remember this entire load will be carried by this weld melt ah this part so we have the shear stress here acting direct shear and there may be bending stress so because of bending there will be the tensile stress

now because of bending then sigma will be M the maximum bending stress will be here if you consider this to be length b then M b by two divided by I what will be the I I will be the I of this welded region only only the {ih} (00:51:21) weld and remember here we must take the critical cross-sections that is instead of taking the true size we must take this size to be t that is the throat length which is equal to h divide by root two

then with this configurations geometry of this cross-sections we calculate what is I and this will give the value sigma

now in the {shea} (00:51:51) in the endurance limit they are uh there is load factor so load factor factor will depend on whether tensile load or ah shear stress but here we have both kinds so how to do that the procedure is simple we calculate the equivalent normal stress using one Mises criteria again sigma square plus {thrih} (00:52:14) thrice tau square and then we calculate the entire thing again only considering the tensile stress just now we can ignore this shear stress because we are now ah doing dealing with sigma prime which has already taken care of this tensile uh of the shear stress we can also use the maximum shear stress theory that is tau max we calculate and that will be equal to of course it is very known this is under root sigma by two square plus tau square then we can use this theory as well tau max

so we can reduce this {multip} (00:53:04) multiple loading to unique loading if if you consider the maximum shear stress or Von Mises stress theory

there is another way which is ah sometimes used that is instead of taking to be sigma here that is there is one way that is we calculate the same but taking this to be again another tau so instead of taking this to be sigma we take this to be tau and make this calculation that is the total stress will be this plus this together

so the total stress will be t total is tau plus tau direct that is uh primary plus tau into bending then we can use this formula now here we see that the factor of safety is little larger if you consider this therefore this is much used and this is mostly used in Soviet Russia and many other countries but ah this gives a conservative value

so now we have learned various ways of calculating ah the {joih} (00:54:12) designing the joints for the variable loading so there are few techniques two techniques as at least for welding welded joints namely the usual Soderberg Goodman or uh Gerber line criteria Geber lines uh um but there is another technique using the allowable fatigue stress

so the maximum design stress will be less than this allowable fatigue stress so this is how normally this welded joints are designed

so now in this class we have learned how to design bolted joints and welded joints when the loading is variable and that is very important you should read many books on the same topic and try to solve some problems so uh that is all for we today thank you very much

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good day today we will start our lecture which is lecture number twenty-seven and uh this is the design of springs

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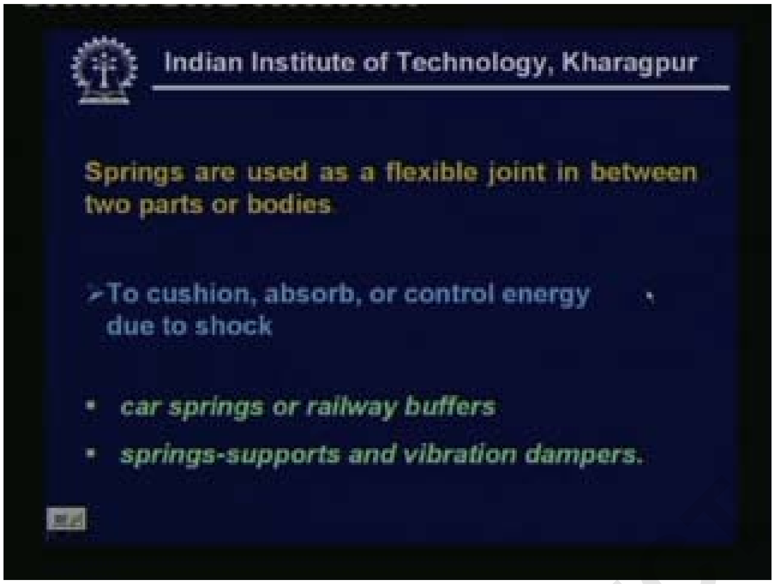

now for the design of springs ah we start with the how we define basically a spring all of us know what is spring we have used springs in our everyday life so basically we can say that a springs are used as a flexible joint between two parts of bodies

now here you are having some sort of objectives for the spring what is that one first of all to cushion absorb or control energy due to shock this is a very common thing what we find out the application of the springs in car springs and railway buffers and to control energy due to shock is a well known situations for the springs-supports and vibration dampers

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next to control motion what is this idea to control motion means maintaining contact between two elements as cam and it’s follower

you understand that when ah if you just think of the cam and follower arrangements uh widely used in so many applications then we will finding out a spring is there which always keeps a contact between the cam and the followers thus it controls the motion

now whenever ah you see someone driving a car he uses a break or a clutch for the car control of the ah car control as far as the motion is concerned and there what you do we apply a break which again eventually is having a spring system and the clutch is also having a spring system which engages and disengages ah at it which the mechanism of a system of springs or a single springs like that

and another one is the restoration of a machine part to its normal position when the applied force is withdrawn a typical example you know ah that uh in ah turbines ah in in the turbines suppose if you take an example then what we see that uh whenever you are having to control the speed okay which is going on increasing or decreasing a governor system is used then which controls the motion or the rotation of the turbine wheels and does the say the flow through the ah turbine veins by the governor mechanism which controls a series of ah say veins for inlet water impingement like that and that is also controlled by springs

and uh here we use again as an some sort of a control motion and uh that is also using a spring governor and another example the some valves actuator valves are also having the spring which can have ah some arrangements for the control motion