Design of Machine Elements – I

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Lecture No - 34

Design of Shafts

good day we continue our lecture on design of shafts and ah today's lecture is lecture number thirty-four today we will take up design of the shaft for the fatigue loading

and in the last class we have seen that we have already learnt about the design of the shaft based on ASME code

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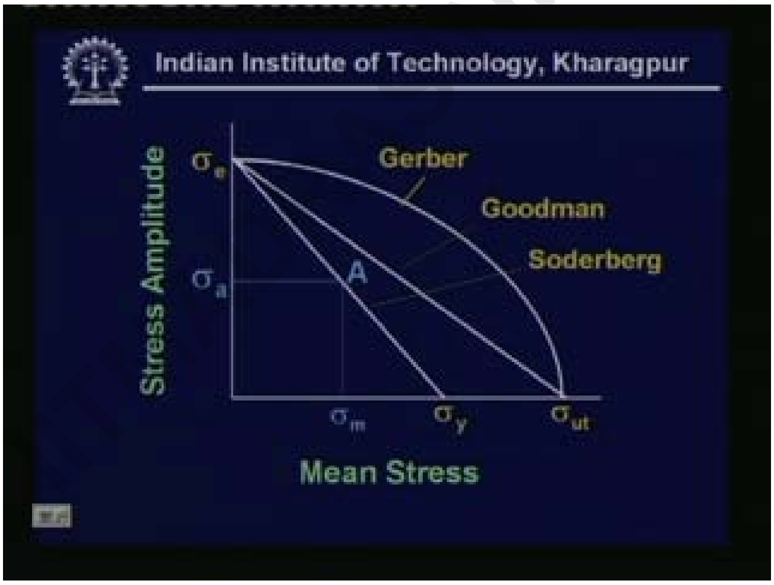

so to start the design of the shaft for fatigue loading you see that we again come down to the familiar diagram of the stress amplitude and mean stress diagram what we have seen in our earlier class

here you can see the three sets of curves one representing the Soderberg line another representing the Goodman line another representing the Gerber line and this A is a point where whose stress amplitude is represented as sigma A and mean stress is represented as sigma m

you know that the Soderberg in the Soderberg criteria the material property is the yield point whereas in the Goodman and Gerber criteria we utilize the material property as a ultimate stress whereas for the fatigue loading property we have the endurance limit

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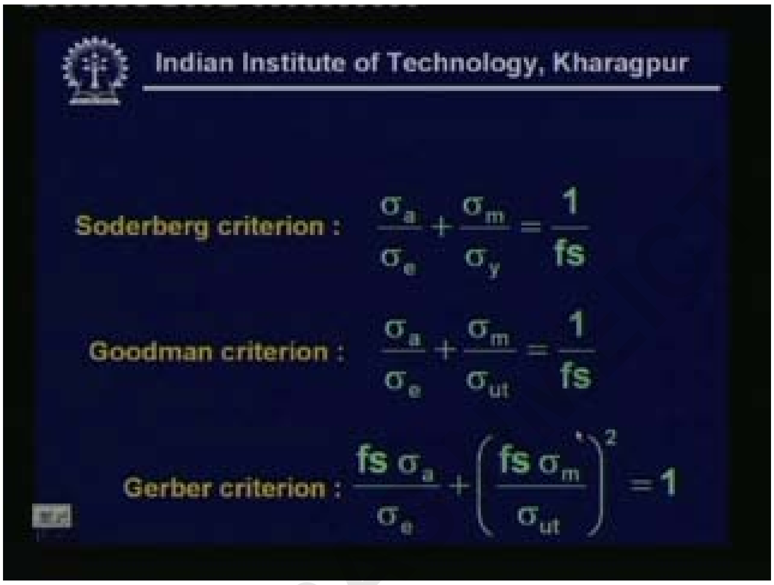

now if we look for the familiar equation that we uh we can write down these equations which we which is again a repetition of the equations what we have already seen in the earlier lecture

so Soderberg criterion is given by sigma amplitude by this is the endurance limit sigma mean stress yield point equals to one by factor of safety similarly the Goodman criterion is given by this expression and Gerber criterion is given by this expression the third expression [noise]

now these expressions are quite common to you and the next one if we look then you can see that this is the expression when we consider the situation of combined normal and shear stress

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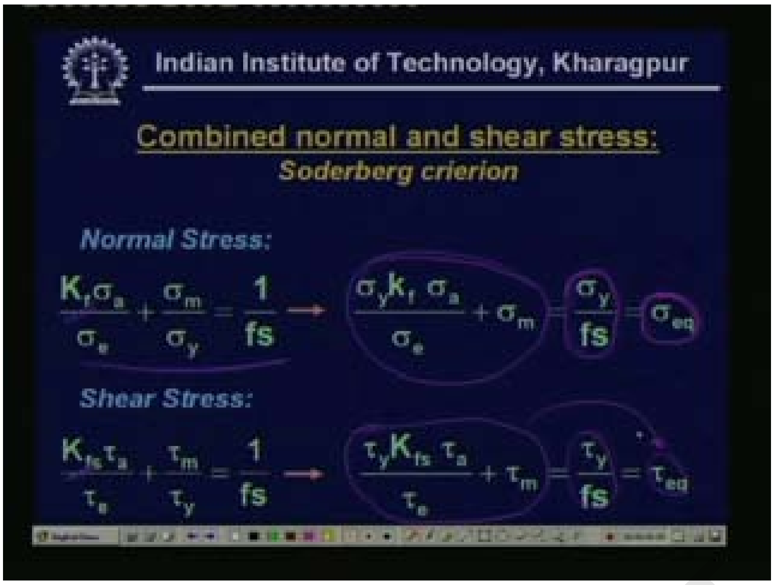

and here we are considering the Soderberg criterion only as you know that in case of the ductile material we multiply the stress amplitude by the fatigue stress concentration factor Kf this is rhe fatigue stress concentration factor Kf and similarly in case of shear we use a fatigue stress concentration factor for the shear

now here ah this particular equation this {expre} (00:04:26) this expression is just retained in another form by multiplying both the sides of this equation by sigma yield point and we get a equation of this nature where this particular component is important

he important lies in the fact that this sigma yield point by factor of safely can be called as what it can be called as simply an allowable stress or the design stress so one can conclude that this entire expression which is equivalent to the deign stress will be nothing but somewhat an sigma equivalent

so what is the idea that means if we can find out this value of the sigma equivalent so this sigma equivalent will be containing the terms of the design parameters and that if we equate with the material property then we get the desired design situation

similar is the case for the shear stress equation where the shear yield point is multiplied on the both sides of this equation to give a expression of the equivalent shear stress of this nature

so this is primarily we are writing down as in [noise] i mean ah this shear stress equivalent and this shear stress yield point by factor of safety is nothing but the design stress

so this idea will be taken up for getting an equivalent normal stress or a equivalent shear stress

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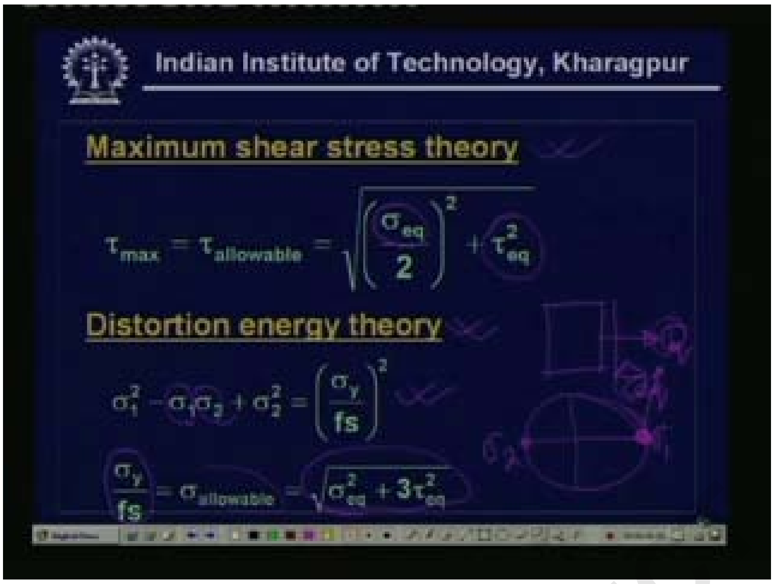

now [noise] if we come down to the next slide then we see that if we use a maximum shear stress theory then what we get the tau max equals to tau allowable nothing but the design stress and that with the usual expression is that sigma equivalent by two whole square plus tau equivalent whole square

so it is needless to mention that once again this tau equivalent will contain what it will contain all the terms ah of the stress amplitude the mean stress and the material properties

so this expression once we write down in the forms of tau equivalent and sigma equivalent will be the will be the terms containing all the design parameters what i told you just in the earlier slide

and just like the maximum shear stress theory if we try to look at the distortion energy theory then also we know this expression of distortion energy theory comes up this particular form and if we substitute the values of sigma one and sigma two then it can be shown that this expression sigma Y by factor of safety will be sigma allowable and that will come out to be as sigma equivalent square plus thrice of tau equivalent square

now here what is being done is that this sigma one and sigma two values are replaced in terms of the values of the shear stress and ah shear stress and the normal stress acting onto the element means what is the idea is that if suppose we are having an element we know that if it is acted upon by an sigma it is acted upon by an tau for which we can find out from the Moore's circle okay that means if we plot for the Moore's circle what will be this sigma one and this is the sigma two so this sigma one contains this terms of sigma and tau and that if we in this {parti} (00:09:04) present case we are expressing this as equivalent tau also as equivalent

so if we substitute these values in sigma ah eh in this particular expression or in other words that sigma one contains these values and so if we substitute sigma one and ultimately the form will come out to be of this nature

so in this case either we can use maximum shear stress theory or we can use a distortion energy theory for the design of the shaft

normally what happens that most of the designers opt for the maximum shear stress theory because as we have seen it is a ah more conservative compared to the distortion energy theory but anyway the choice lies for the both the theories

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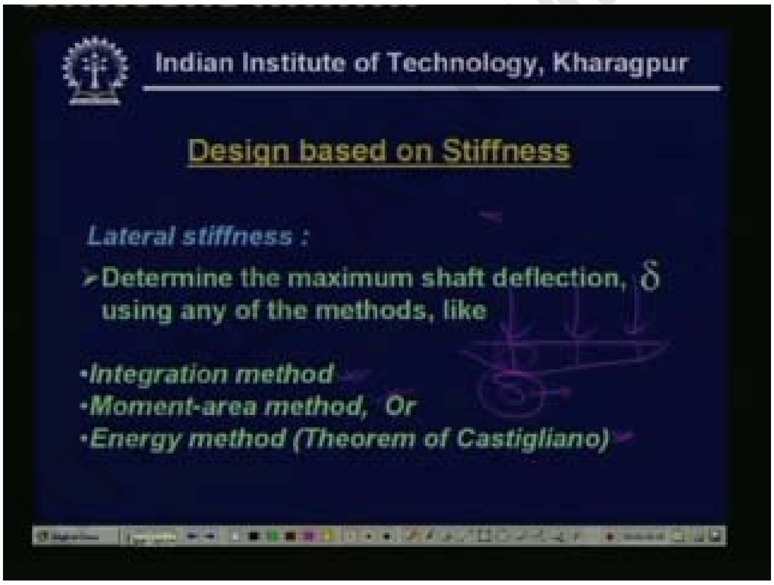

now this is a design based on the stiffness so prior to discussing that is design based on the stiffness ah let us make an concluding remark on the design of the strength

so as far as the design of the strength are concerned we have seen that uh you have only utilized the basic stress equations ah what we have learnt in the earlier lecturers and those strength theories were nothing one was ASME code design another was the design for fatigue for which we have used the Soderberg criteria so as long as the design of the shaft based on strength is concerned we will be treating the subject in two ways that means either we can use the ASME design code or we can use the just Soderberg criteria for the fatigue loading

it depends that how the design data are provided to a designer so based upon that one has a choice of choosing either of the two theories what we have just learnt

now we come down to the idea of the design based on stiffness now what is a design based on stiffness a simple situation ah we can conceive like this that suppose ah you are walking over the corridor of your institute and that is the first floor

now when all of you are walking over the first floor just imagine an condition that the entire floor is just swinging when you are putting your steps it is not breaking but it is just having an swing so how do you feel

so that is the idea that means what is happening by virtue of putting a load suppose i take up this particular one and that is the floor over which you are walking now if you are walking down and every time the floor swings means it is just having an flexibility so it is just swinging like this

so that is sometimes in the design is not desirable or in other words that when a load is there the two things are to be satisfied that one thing is that it should not break it should not break that particular part we have already covered that is the design based on strength

at the same time if the requirement is that it should be stiff enough so that the deflection is limited then we call the design is based on stiffness

so when we consider the design is based on stiffness two things comes into picture one is just by applying an bending load this particular beam or this particular machine member if i consider will bend but the bending should be limited to a certain given limit

if this machine member is applied upon with a twist then this machine member will of course twist but up to a certain limit so that limits are normally set by the designers and considering those limits one can set the dimension of the machine element and that is in short what we call as a design based on stiffness

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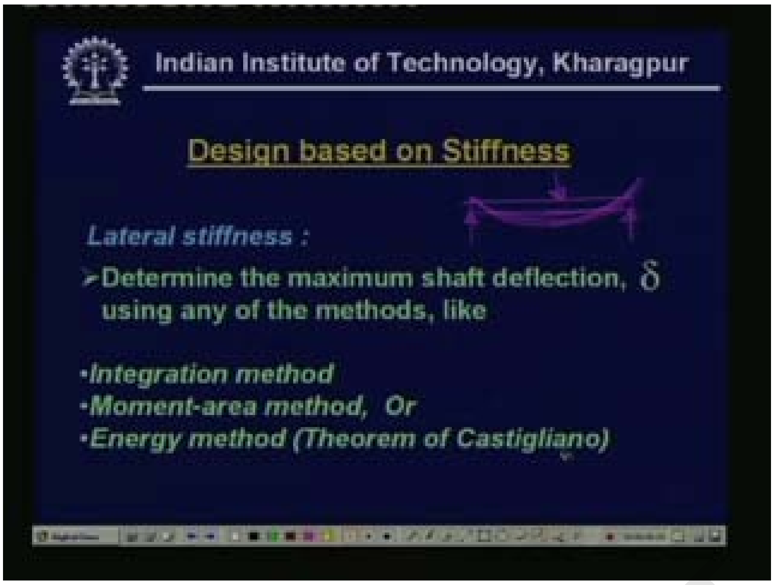

so here we first consider the fact that is called lateral stiffness so what is a lateral stiffness that means if we are considering some machine member

if we considering some some machine member okay so it is acted upon by some support and a load is there then it will have a natural tendency to bend by the action of the load and this particular deflection

well just let me draw a clear one so that means if this is the machine member it has got an deflection like that under the action of load say p then one has to limit this deflection so that is the stiffness criteria

so the requirement for such design is this that one has to determine the maximum shaft deflection delta and how using any of the methods that are listed below

one is an simple integration method one is an moment area method another is energy method you have ah you uh you are quite conversant with these three methods in your earlier course and so i do not go to details of these methods

however the main idea is that using either of these three suitable method one has to find out the deflection of the shaft and normally what we do if there are series of loads acting onto a shaft thus creating the lateral deflection then what you will do you will try to search out for the maximum deflection occurring onto the shaft due to the various kinds of load which is responsible for creating the lateral displacement that means ah shaft acted upon by various loads will have say a deflection pattern very exaggerated so here it is like that this is like that this is like that so we understand that this is the maximum deflection which is occurring

so if we can find out these deflection and design within certain limit then of course it will be satisfying all other conditions

now ah in this case we consider the design in summary like this

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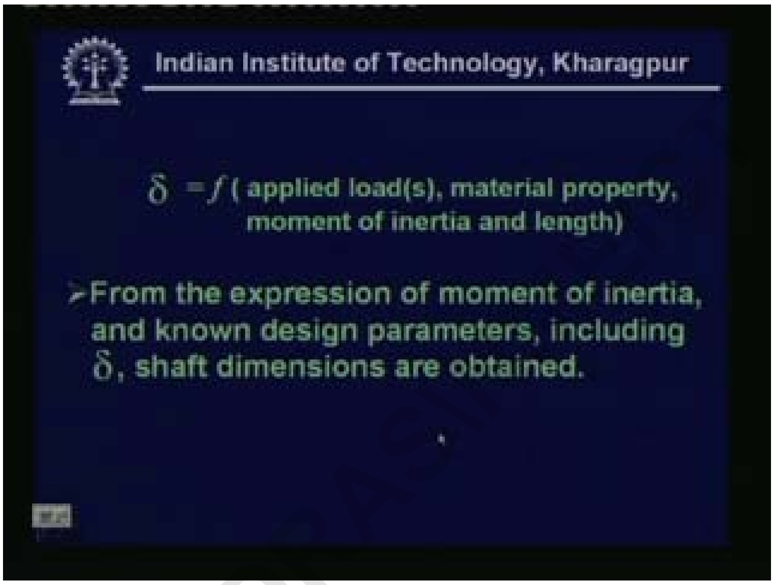

delta is a function of applied load or loads material property moment of inertia and length of the machine element from this expression of moment of inertia and known design parameters including the deflection delta and shaft shaft dimensions can be obtained

that means this is a basis of the design that means you find out this deflection as we have discussed earlier now you know the moment of inertia will come into picture moment of inertia means the geometrical dimensions will come into picture and other design parameters like load etcetera will come into picture and the shaft dimensions thus can be obtained if we choose a suitable limit for delta

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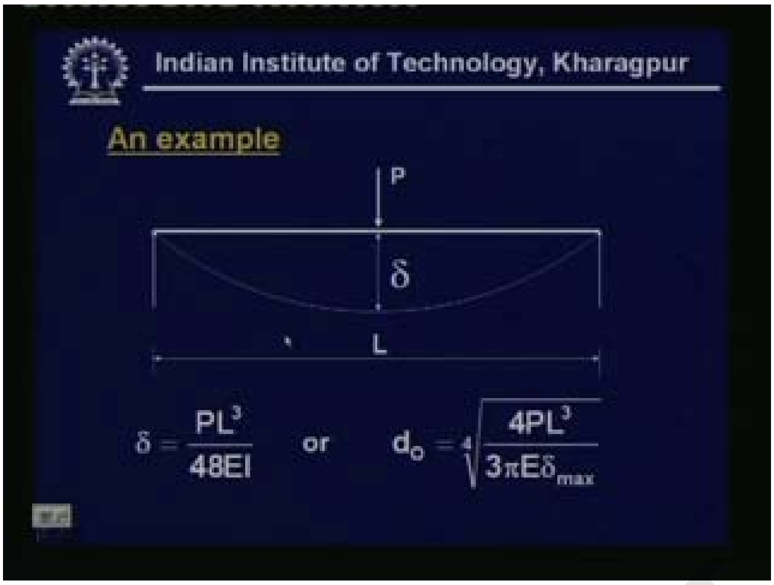

now coming down to the next slide we find an example what is this example ah simple beam acted upon by an load P at the central location so here we can find out that what is the maximum deflection

so obviously the maximum deflection will be occurring over here and and when we find out this maximum deflection the we know the basic equation for this deflection delta is PL cube by forty-eight EL

so that we find out this value of DI ah where you know the I has been this I has been replaced properly by the value of what you know that value of I will be in case of a circular bar

here we are considering an circular bar by root to the power four by sixty-four and that if we consider over here then you get the desired diameter

well [Noise] if it is an rectangular section obviously the dimension or the cross section area of the rectangular section could have been found out in the similar manner

so you can see this deflection this particular dimension of the shaft depends upon the load depends upon the length depends upon the material property and this is the delta max what you should prescribe

so according to your design you set a value of delta max so that with all other known parameters a shaft dimension can be obtained

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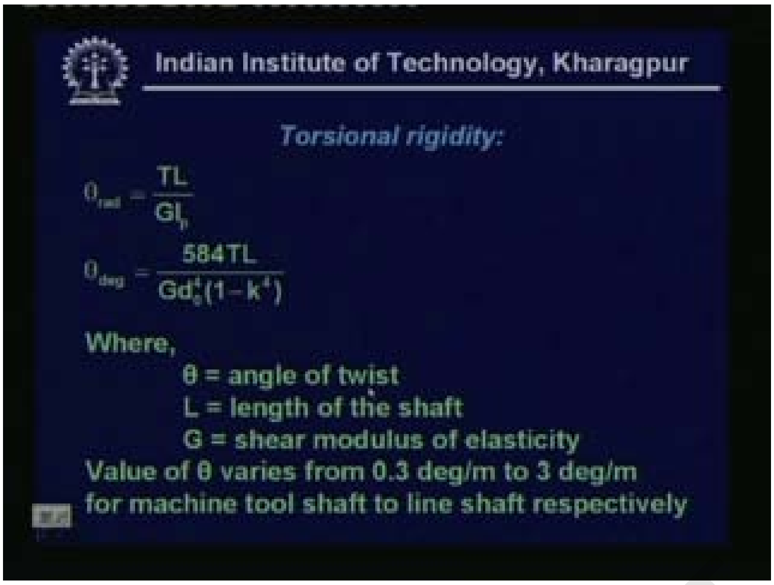

in the similar manner if we consider the torsional rigidity then we see that the torsional ah rigidity means what

it means basically we know the angle of twist what will happen to a machine member by application of a {loa} (00:20:07) load like torque we get ah angle of twist which is commonly given by this expression TL by GIp you know all these things

and if it is expressed in terms of degree then as form of light comes into this fashion here again the expression has been retained for a circular member well

so this Ip can be for a circular member can be for noncircular member ah like rectangular section etcetera but here just to have an continuity of the entire shaft design here the expression has been retained in the terms of ah in the terms of ah machine member having a circular cross section

so in the similar manner what happens that this ah this eh uh let me first give this um notations theta angle of twist as i told L length of the shaft and G is the shear modulus of elasticity

now coming back to the idea that what is this particular limiting value of theta

the limiting value of theta is normally point three degree per meter to around three degree per meter depending upon say for machine tool shaft to the line shaft

obviously you understand that the rigidity of a machine tool shaft is very important in considering to the line shaft because if a machine tool shaft is not rigid enough then what will happen by the application of the cutting tool then it will have obviously uh it will have a ah deflection and that deflection will cause what it will cause a material i mean the job which you are turning will be erroneous in as far as the dimensions are concerned

so that's the reason depending upon the application of the shaft where you are going to apply i mean where you are going to use the shaft one has to set for the value of the theta means the what is the allowable angle of twist one should one should tolerate

now this value of point three degree per meter to three degree per meter is just a guideline so you please note that this guideline ah has been provided to you to have an just an thumb rule idea but it is no way a fixed value and one has to change these values depending upon his requirement

so this actually ends the ideas of the shaft design based on the stiffness so based on the stiffness we see that we are just utilizing the bending equation ah and the equation of torsion ah in particular the angle of twist just to find out the design ah find out the shaft dimensions

so here we see that the design of shafts requires two ideas once again to just speak out one is based on strength another is based on rigidity so i think you are clear about these two design aspects

now once you have designed a shaft taken into account either the strength criteria or the stiffness criteria or may be both but anyway i would like to say that ah in the case of the stiffness the shaft dimensions obviously are going to be much higher compared to what you will get uh if you consider the strength criteria this is a general rule

now after a design the shaft ah one has to look into this aspect also

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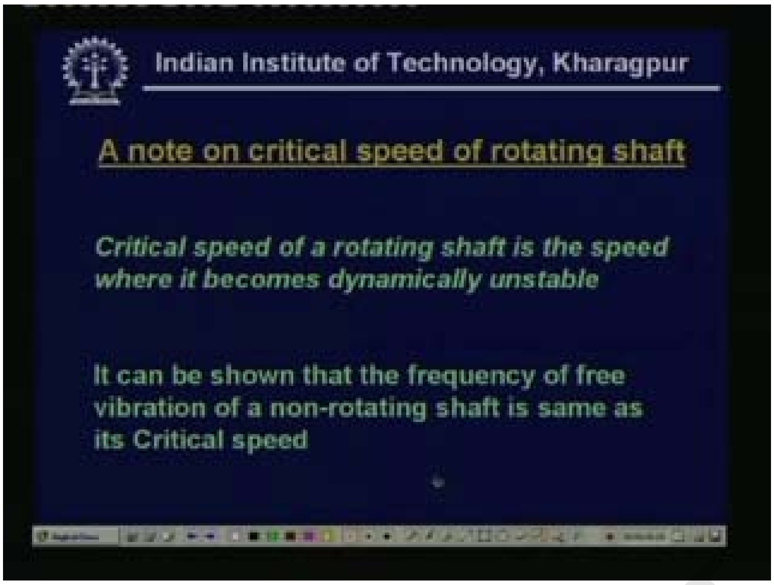

that is what has been given here as a note on critical speed of rotating shafts you know the what is the critical speed the critical speed of a rotating shaft is the speed where it becomes dynamically ah unstable

now what is that particular speed so in terms of frequency you can say that ah it is very common that means it can be shown that a frequency of free vibration of a non rotating shaft is same as it's critical speed

this particular part already you have done in your earlier vibration courses so i think we need not go into details of this particular aspect and moreover this is beyond the scope of this class

however we know the frequency of free vibration of a nonrotating shaft is the same as critical speed so we find out the frequency of free vibration and that will be the same as the critical speed

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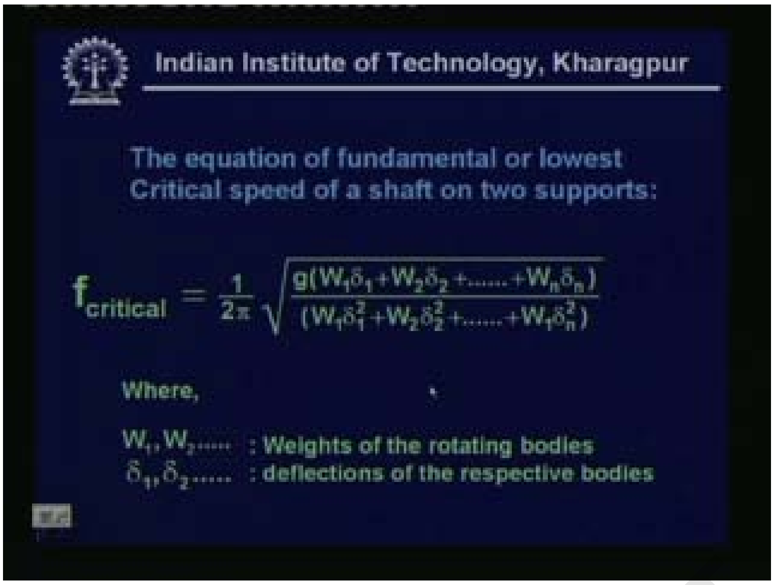

so if we look to this aspect the equation of fundamental or lowest critical speed of a shaft this is a typical case when it is placed on two supports is given as f critical okay this is the f critical

so here you please note that the expression has been represented in terms of frequency than the speed or the rotation

so this is here is the expression where this W one W two etcetera are the weights on the rotating bodies and delta one delta two are the deflections of the shaft due to the respective weights of W one W two etcetera

so what it specify ah what it implies in actual practice could be a gear could be a pulley and so on so forth

and corresponding to the gear you are having an deflection delta one corresponding to the pulley you are having an deflection delta two which are the known parameters and from there you can find out the critical frequency of the shaft by this formula

but one thing is there this particular formula has been derived using these expressions please have a look

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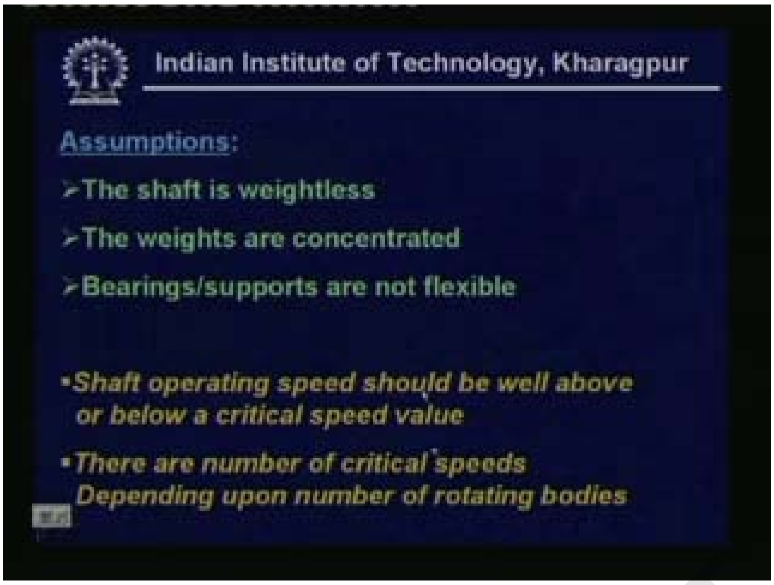

the assumptions are like this the shaft is weightless which is a very major assumption the weights are concentrated

bearings and supports are not flexible this is ah is very important because none of the support ah supports are really rigid they will be having an flexibility [noise] so that incorporates some additional deflections and with

ah with that particular point and again with our this two number one and number two points one need to get an accurate formulation other than what had has been displayed over here anyway ah as a first design aspect one can use this equation what has been given here very easily and find out the critical speed

look the shaft operating speed should be well above or below a critical speed value this is obvious otherwise you know the resonant frequency will occur and the shaft may have a failure

in the similar manner the number of critical speeds ah are there and that depends upon the number of rotating bodies as many as critical speeds will be ah uh [noise] will be coming into picture as many rotating bodies are there onto the shaft

so ah [noise] this requires the frequency will be normally that fundamental frequency is being considered but one can also look into the all other frequencies ah whether it is creating any sort of danger during design or not anyway the basic concept or the basic idea is something like this that ah check on the critical speed should be done to avoid any sort of resonance failure of a machine member [noise] so this one is a design check one can talk in this manner

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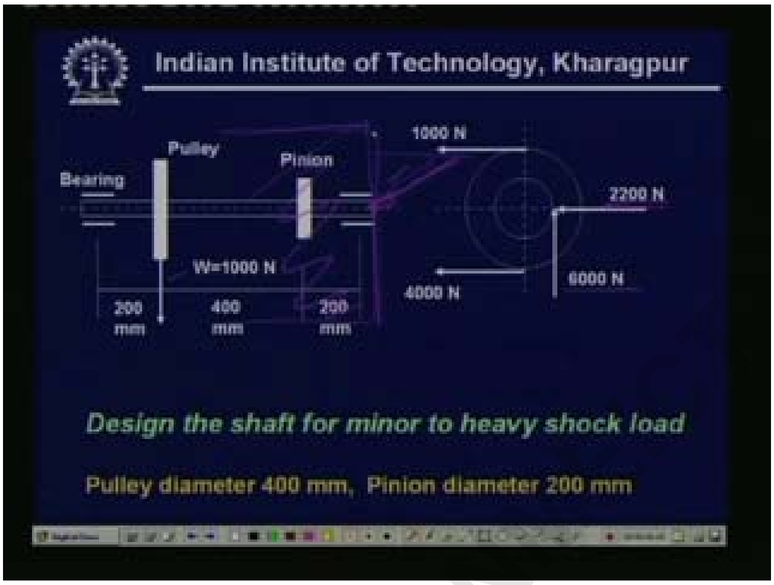

now let us consider a simple problem the problem is stated well if you look at ah you know this figure

so what is this one a shaft supported on supported on two bearings over here a pulley is there having an weight of thousand Newton pinion ah should have some weight but not specified ah and ah these dimensions of the shaft etcetera are given over here

and this pulley is acted upon by belts whose tensions are thousand Newton and four thousand Newton respectively but now for normally ah for actual design one has to find out from the idea of belt drives that what should be the tensions actually acting onto the pulley

next ah this particular pulley will be um rotating in pinion and the pinion in turns will be rotating a gear so gear force is coming over the pinion has been given uh given here for simplicity and this are given as twenty-two hundred Newton and six thousand Newton respectively

so if you look into the particular problem you see one thing is that the this shaft this particular shaft is getting a loading one from this side another from this side okay that means one in this particular plane and another into the transverse plane in a perpendicular plane so ah we require to find out the what will be the bending moment onto these {partif} (00:31:16) while we consider the loading on this plane and similarly while we consider the loading onto this transverse plane so [noise] this particular requirement first has to be followed and then we find out the design

by the way ah just look at this pulley diameter ah is four hundred mm and pinion diameter is two hundred mm just this um distances at two hundred four hundred two hundred respectively

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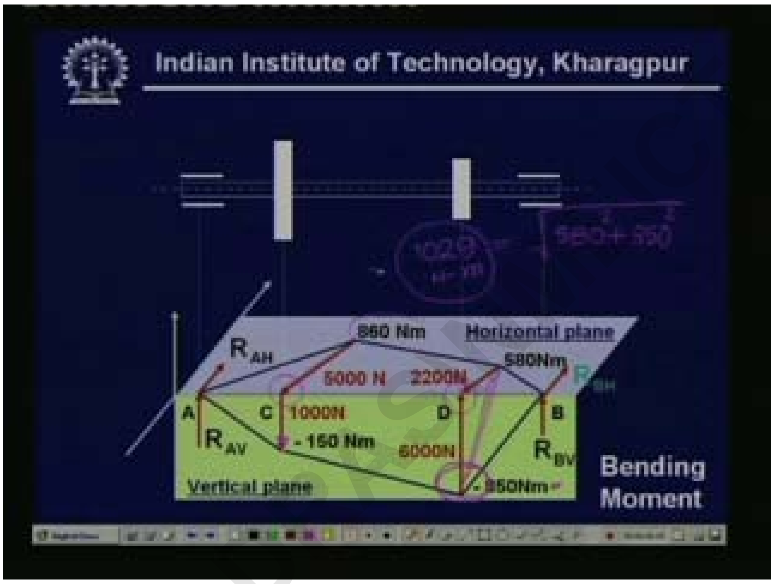

so this is the loading as well a s the bending moment diagram now ah if i ah just go back to this slide then please note that the torque applied onto this particular shaft will be obtained by the six thousand {nuu} (00:32:29) Newton multiplied by this radius ah this radius of this particular pinion or we can find out four thousand in minus one thousand multiplied by the radius of the pulley

so here ah we can use either of these two to find out the torque and incidentally the torque is constant over the shaft

now if we look into this diagram you can see very well that the pulley weight if ah the this is one is a vertical plane another is a horizontal plane well whatever is telling that is pulley weight of thousand Newton will be in the vertical plane and as it is given as thousand Newton directed downward and the gear force acting onto the pinion is an upward direction

so this is the loading as far as the vertical plane is concerned now if you consider the horizontal plane then you can see

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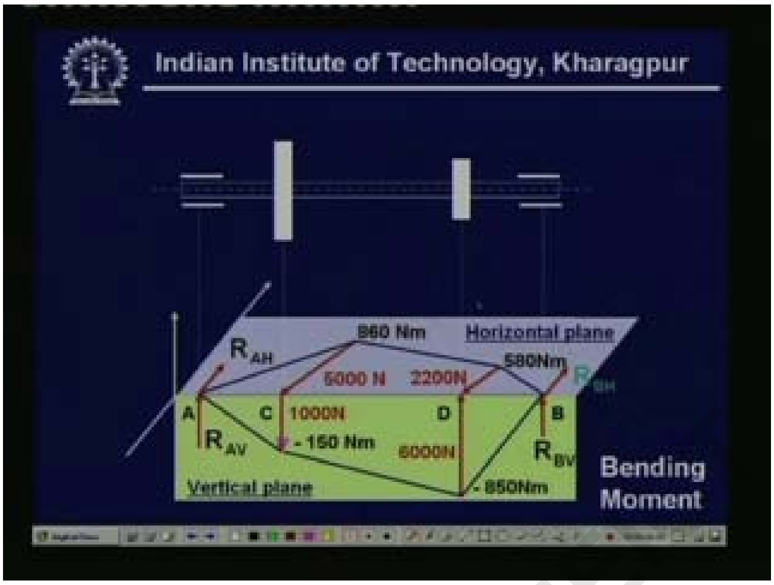

the total pulley force was thousand and four thousand so combine the five thousand is coming like this and the gear force onto the pinion in this direction was twenty-two hundred so both are coming as downward whereas here it is one downward and this is upward

so if we ah look into this one then in the vertical plane the bending moment diagram is given is this manner and where the maximum value of bending moment is occurring at the D and given by the value minus eight hundred fifty Newton meter

in the similar manner onto this particular horizontal plane the maximum bending moment is coming at C ah at as eight sixty Newton meter

but if we consider the resultant of the two bending moments of the vertical plane and the horizontal plane together then obviously by inspection we can find out that this resultant that means at the point D the bending moment will be more compared to the bending moment what we are getting at this particular [noise] point C

so if we consider that fact then what we can see is that the bending moment here will be how much it will be coming out to be ah i just use this this location to write so five eighty square plus eight fifty square and square root of that

so that comes out to be well uh i just giving you the value as one zero two nine Newton meter and ah this value will be utilizing as a maximum and that is the moment at D now what we can see for the given data ah we can see that the design of the shaft design the shaft for minor to heavy shock loads

so with a given data we think ah we we we can see that the design will be based on strength because nothing has been told about the stiffness and if we consider the design based on the strength then we find that ASME code will be the most suitable over here because we do not know anything about the fluctuating components

so if we consider the design based on ASME code for the minor to heavy shock load if you just look at the various factors to be utilized then you will be finding out that Cbm can be well taken as two and Ct the factor for torsion can be taken as one point five

so we come down to the this particular slide from where we know the bending moment was one zero two nine so let us solve the total problem

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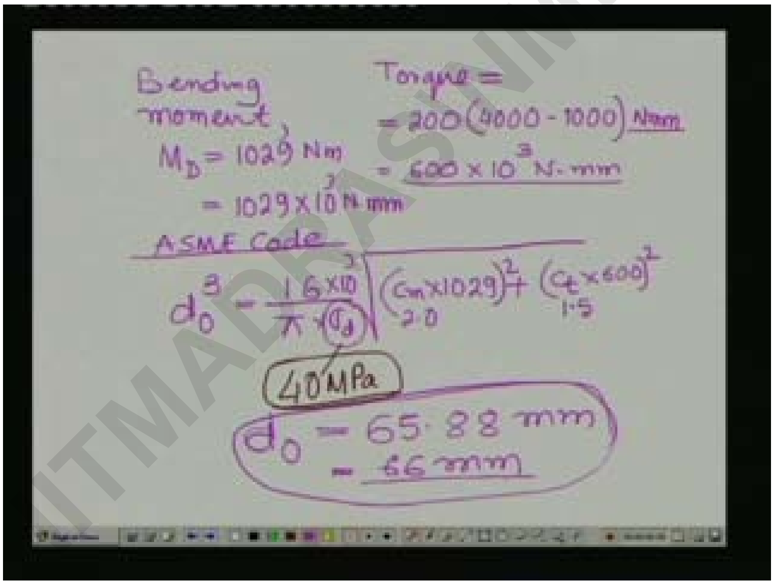

so what we find is that torque can be retained as how much it is two hundred multiplied by four thousand minus one thousand and the unit will be coming out to be Newton millimeter

so this comes out to be six hundred into ten to the power three Newton millimeter what is the bending moment bending moment which was maximum at MD came out to be one zero two nine Newton meter so that is equals to one zero two nine into ten to the power three Newton millimeter so if we use the ASME code then the design comes out to be ah i uh this is d zero cube will be equals to sixteen pi into sigma design means allowable or whatever you can call it in square root of Cm into M is one zero two nine square

so let us bring out this ten to power three outside ah plus Ct into six hundred square this Cm is two point zero and Ct is one point five what is this value of sigma design

this sigma design we can consider as forty mega Pascal why because ASME code suggests that ah it can be fifty-six MPa for the shafts without any shaft ah um ah key key ways etcetera and forty MPa if it has got a key ways

so of course the pulleys and pinions will be attached with the key ways so we consider this to be the value for the material property

so if we consider all these values then a simple calculation will be will show you that this value will be coming out to be sixty-five point five eighty eight MPa and that approximately sixty-six mm

why because if we look at the standard sizes of the shaft then ah this should be sixty-six mm so that it ah it specifies the standardized values

now in this case ah you see one thing that while doing this calculations once again to repeat that you always consider this ideas uh so before i just talk about so i i think we are ah we are confident that this value of d zero is coming out to be sixty-six mm so that our problem is solved

so for the given problem the shaft size should be sixty-six mm

anyway i have considered the simplest values like that the shafts are solid it is not hollow like that so many combinations could have been done in this particular um ah calculations it could have taken some other values Cm and Ct also

anyway ah whatever i was just trying to um uh talk about is that while doing these calculations

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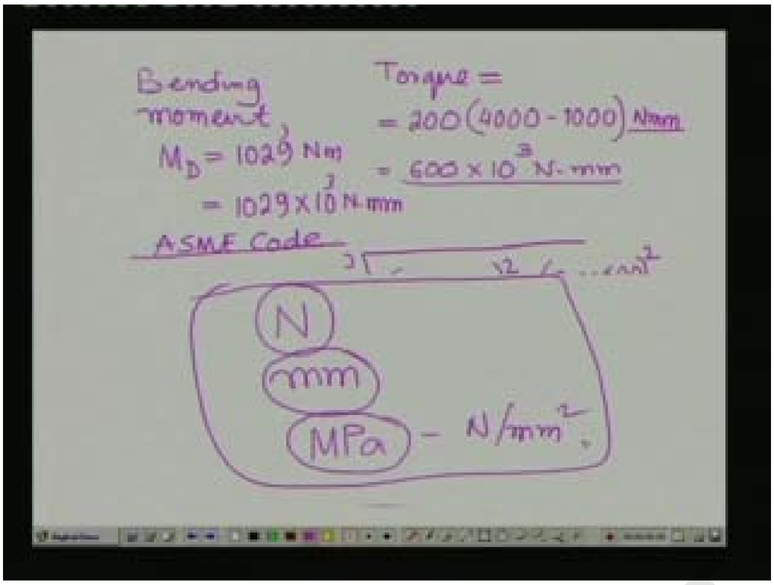

ah once again just to repeat it is very important that you always take the force in Newton unit dimensions in millimeter unit and stresses and related parameters in MPa that is Newton per millimeter square unit

so this will make your life very simple in the sense that you do not have to unnecessarily think of changing over to the units every time so uh this gives you the idea of how you are going to design a shaft

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now one of the important feature of the shaft in design is that the bearings now what we understand that the shafts will be mostly a rotating bodies as we have seen our in our earlier discussion and these rotations are possible through the bearings so as because the bearings and shafts are so closely related to each other one also should have some knowledge of the bearings that are normally utilized in the design so we are not considering any separate ah lecture course or a lecture chapter on bearing ah rather in this particular shaft we are just giving an brief overview of the bearings so this particular bearing ah ideas are the main ah main theme of this particular lecture

now if we consider this idea then we do have largely two types of bearings one are the fluid film bearing another are the rolling contact bearing the fluid film bearing categories are journal bearing thrust bearings slider bearing etcetera where the you will be having the entire load of load of the shaft are being carried by a fluid film and another category is called the rolling contact ah bearings which are categorically are the ball bearings and the roller bearings

now this are very common you have seen and also you must have seen the bearings like journal bearing and thrust bearing so ah these are the two types of bearing one normally use for the [noise] uh designs

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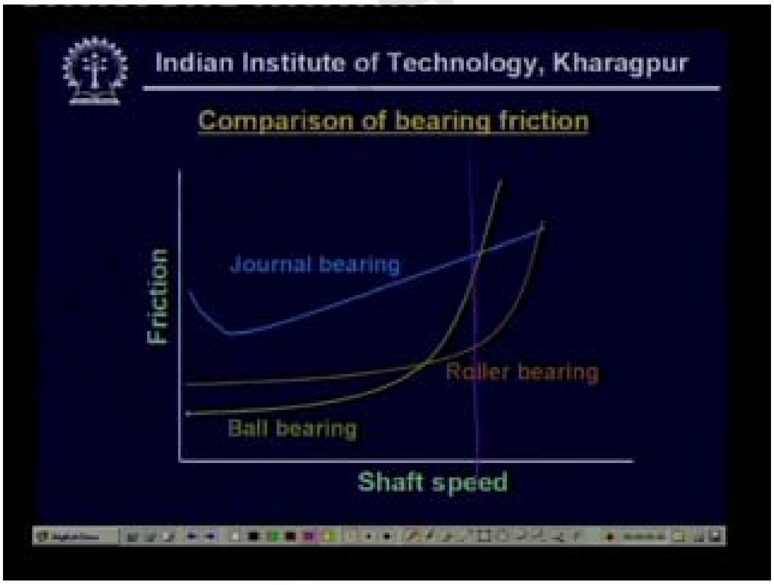

now first of all let us have a very quick comparison of bearing friction because friction is one which will be always there and one has to look to be ah friction uh to reduce the frictions in the bearings and so that the operations are much more smooth and power loss is less now if we see ah a plot of the friction verses this is the shaft speed then you can see for the lower shaft speed somewhere here lower shaft speed [noise] you can see journal bearing are having more friction compared to what you get the roller bearing and the ball bearing

that means ball bearings are do not have ah do do have a very small amount of friction then comes a roller bearing and then comes a journal bearing so that is sometimes the ball bearings and roller bearings are also sometimes called as antifriction bearing

but it is not exactly antifriction but ah still people call it ah anyway ah we'll be we'll be seeing the interesting part from this side when the shaft speed increases you see the this ball bearing and roller bearing frictions phonemically increasing but the journal bearing friction is relatively low in that particular case

so that means what is happening as because the shaft speed increases then obviously the journal bearings are more acceptable compared to what we get for the ball bearing or the roller bearing

now ah there are ah of course there are some advantages and disadvantages as far as the friction is concerned we need not discuss you can just see from this figure [noise]

and other situations something like that this ball and roller bearings are relatively costly it takes up ah little bit more space and ah eh it requires also the seats to be a manufactured over the shafts so that the bearings can be properly fitted over there

but ah this bearings had one of the ah good advantages that low friction at the low shaft speeds relatively low shaft speeds and ah the maintenance of these bearings are also relatively easier compared to journal bearings okay

now however the journal bearing can take up an huge load of course these also can take up huge loads but journal bearings are mostly suited for very high speed and high loads also and it takes lesser space and little compact

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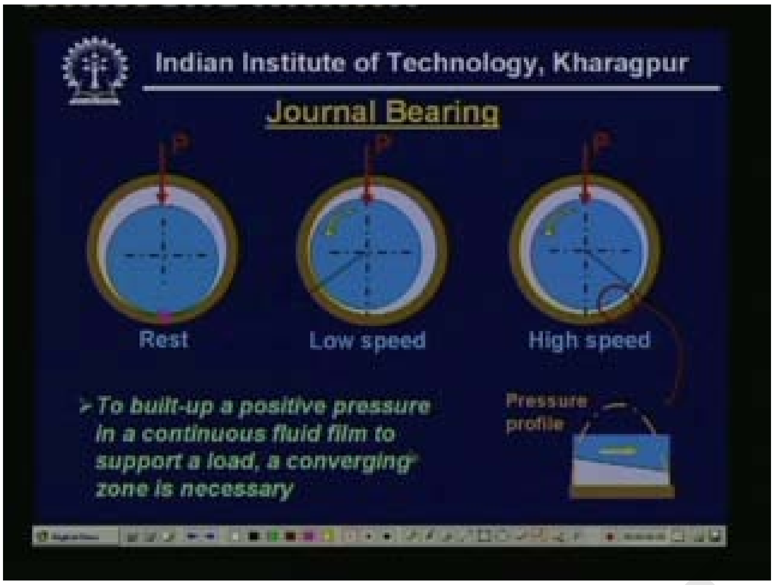

now we start with the idea of journal bearing so this we can consider to be that if you look at this one then this is the housing or the main bushing rather you can say and this is the shaft which is placed within this bush and this one is called actually the journal this blue one is sometimes it is called as the um this is the ah journal

now this one you can see is that when it is at rest then when it is at rest then you can see that the bearing load will keep the journal in contact with the bush at this point and this light blue color is nothing but the filled up oil or the lubricant what you can see

now whenever the journal starts rotating then what will happen at the very at the low speed conditions it is a low speed conditions what will happen that the with this load acting over the journal the bearing has got a tendency to just go over here because what will happen that if it is rotating onto this one onto the bearing a frictional force will come into this way

so the orientation of this journal with respect to the bush will come out to be in the some way so that these angle is basically these angle is basically the angle of friction

so it will be the angle of friction that means this particular friction force component of this ah load will be just balancing the friction force in this particular line so along this line there will be an balance of this friction force and the load component

and if there lies some amount of oil then this amount of oil won't be very thick in what i mean to say that in this particular case almost a metal to metal contact will be prevailing and ah some sort of oiliness or something will be there if it contains the oil as i told you if there is no oil then there will be a purely steel there will be a purely metal to metal contact and in case of the oil presence there will be a just an oiliness not the entire journal will be just lifted up by this oil

now when it goes to a higher speed at the high speed then what happens that this oil field this particular oil that is being over here will have a continuous fill frequency there will be a continuous fill which will be created over here and this continuous fluid fill will be generated in this zone and if you consider this zone from here to here you can see the passage is something like an converging one

so whenever there is this type of converging zone then there will be an oil film having an minimum thickness over here will be created and this oil film can take up an extensive load as it is being shown in that exploded view that if there is a converging zone and an oil is over here as i have shown you and if this particular one is moving with respect to this then you can see that it will be generating an pressure profile over here and it will create an load bearing capacity [noise]

so the summary of this idea is that to build up a positive pressure in a continuous fluid film to support a load a converging zone is necessary and that is what the concept of journal bearing so what is that one that if it is at rest it is just having an contact over here no fluid contact only metal to metal contact at this zone

there will be mostly an metal to metal contact ah very thin ah layer of oil something we can guess an oiliness is present over there at a low speed whenever the speed goes higher then you will see a continuous fluid film having an converging and the diverging zone

now as a matter of fact these diverging zone what you can see over here is actually will be responsible for creating a negative pressure

so sometimes these oil due to the negative pressure over here can be drawn into this passage ah due to this converging to diverging zone where it creates a negative pressure so this idea is given this idea is the underlying fact of the journal bearing

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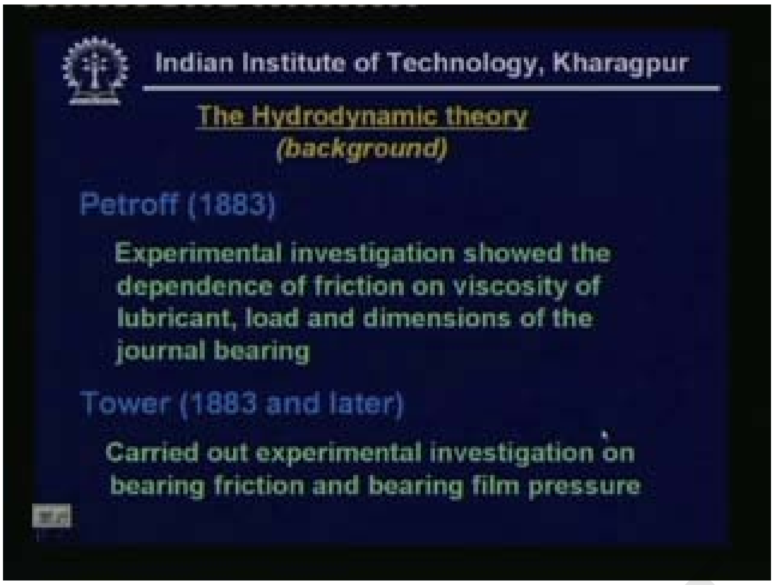

now this is best on what we call as the hydrodynamic theory of lubrication and this was ah first seen by Petroff in nineteen eighty-three where he did the experimental investigation and that showed the dependence of friction on viscosity of lubricant load and dimensions of the journal bearing

so this was a first man who conducted that experiments extensive experiments and Tower in the same year year nineteen eighty eighteen eighty-three and later ah also carried out an experimental investigations on bearing friction bearing film pressure

so ah he took out this experiment looking onto the some aspects of this particular um experiments by Petroff so the experimental investigations of Petroff and [noise] Tower are the background of development of the hydrodynamic theory

and which later was given a mathematical shape ah in the form of well known Reynolds equation due to Osbourne Reynolds ah in the later years who found out from the experimental investigations ah the interesting results in which she put into the form of a mathematical equation of the hydrodynamic theory

so we continue of more on the overview of the bearings in the next class

thank you

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good day we continue our lecture on design of shafts and this is lecture number thirty-five

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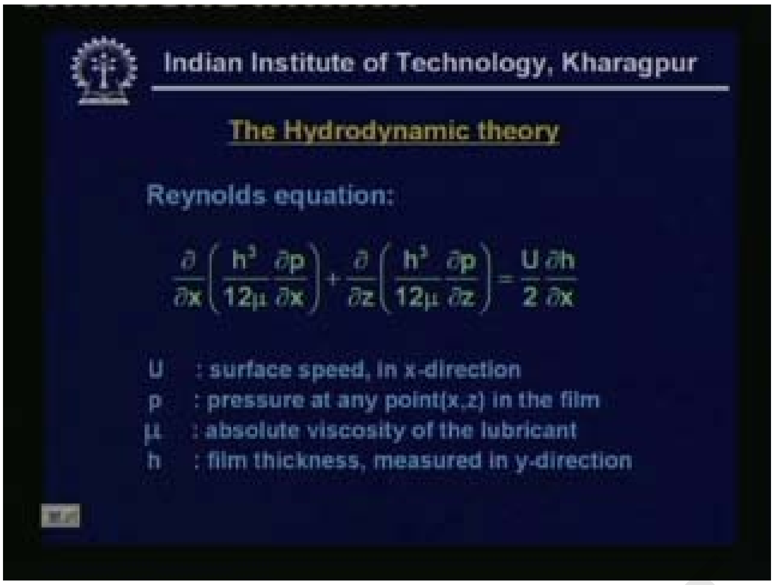

in the last class we have been discussing about the hydrodynamic theory of lubrication so what in short we have written hydrodynamic theory

and ah just a recapitulation that this particular hydrodynamic theory ah the historical background is that that from the experimental investigations of Petroff and Tower ah and looking at their interesting results ah what happened that Reynolds ah suggested ah general equation for the hydrodynamic theory

and this equation is given here as you can see

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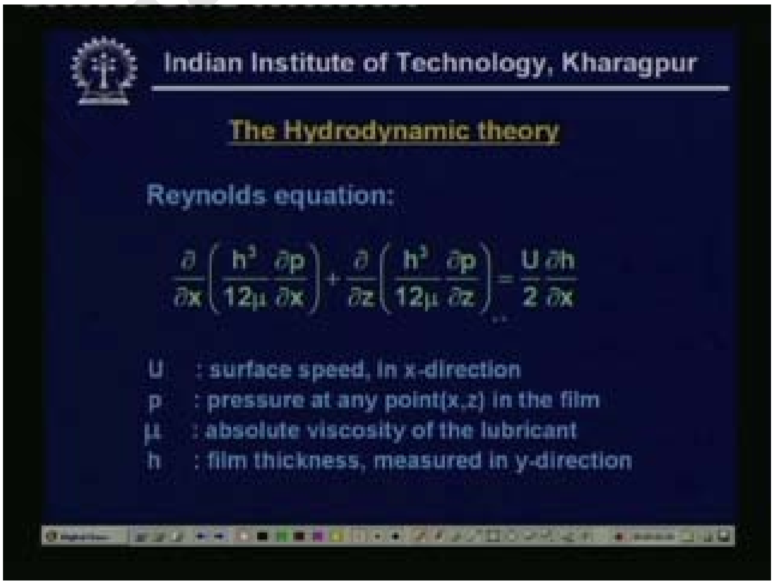

this equation is nothing but purely a flow equation under the pressure gradient and that is for the left hand side and the corresponding right hand side is ah some sort of device or some sort of pressure generation mechanisms that we will talk about

in this {expe} (00:58:15) in this particular analyze i mean equation what has been assumed is that the lubricant is incompressible and ah you know the wedge shaped what we have been talking about earlier has been assumed to be ah somewhat ah a straight profile okay this uh this is the wedge if you consider and this is the