Design of Machine Elements – I

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Lecture No - 19

Design of Power Screw

dear student let us begin lectures on machine design part one

this is lecture number nineteen and the topic is design of power screws

now in lecture number eighteen you were taught the kinematic and ah the statistic analysis of a power screw

in this lecture you will learn how to design such a power screw

in order to elucidate the design principles we take a simple example an an example of a C clamp

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now here ah let me draw a C clamp first [noise] this is a C clamp this is the screw and this is the clamp

now it is held fixed to the ground that is it is held by some device and the purpose of the C clamp is to keep a pressure a compressive force there on the surface now we have to look into details of this portions of the C clamp

now you see whenever you want to give a pressure we have to turn this screw and the whole body turns but we do not want this surface to turn so abruptly

so therefore we need to stop this turning motions when it gets transferred to here

and this is done by means of what is known as a collar or thread collar

so this is kind of thread uh thrust bearing which we use here ah and the detailed picture will be if you consider this to be the clamp

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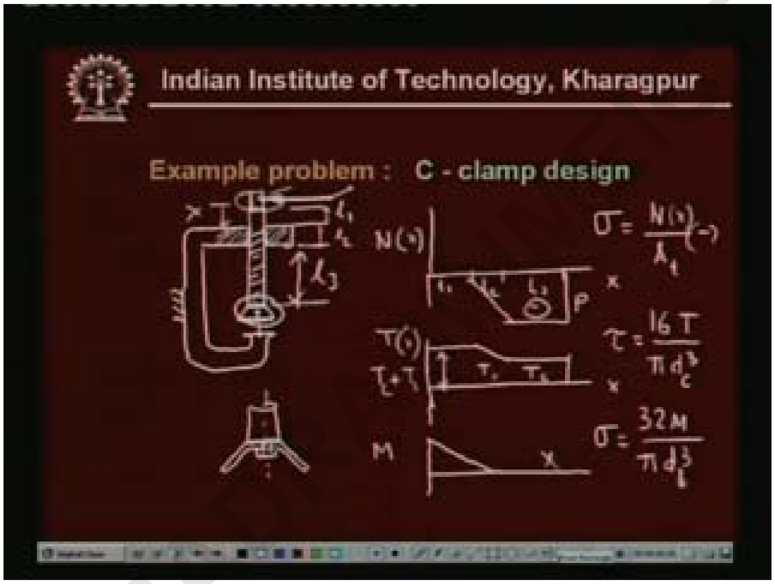

there is a little hole over there in which a fastener is connected so therefore when it rotates then it drops along the surface and ah because of the frictions we need to overcome this frictions and this is done by applying the torque

so the torque ah is required in order to overcome the friction over this area as well as the friction here

now whenever we want to design then we have to concentrate on three portions as it is very much clear that this portions which let us say we call l one the length this is the knot the length of the knot let us say l two

and this part which we designate l three

now there will be several kind of stresses

first of all if we designate x from this distance then there will be the compressive stress the compressive stress distribution is somewhat like this is the N x that is the normal force and here are this things that it is l one this distance l one l two and l three

now here there will be fewer compressive stress and the magnitude is

so this is negative the magnitude is the p the {app} (00:04:55 min) point of ah the load of over here and then it gradually goes to zero and here this part is not under any compressions

so therefore the [noise] the normal force distribution will be something like this and this is purely negative

now with the normal stress ah that is the normal force there will be the normal stress and the normal stress is given by this formula which is very well known that is N x divided by A times or this is the area of the threaded sections

so this is a critical sections of the of the ah screw and that will be the route of the thread

now this is the normal stress and this is compressive so this is negative stress

then of course we have the torque let me draw the torque distribution

now here up to this {poi} (00:05:55 min) point there will be the only the torque which is to overcome the collar friction here that is as we have mentioned before there will be friction so we want to {avoi} (00:06:04 min) uh overcome this frictions and thus remains constant here

then we have to add the ah the torque that is the torsion due to the friction in the thread and that varies almost linearly

and beyond this length that is l one is subjected to both the addition of this two that is Tc the collar torque and Ts this part

so therefore this will be Tc plus Ts this the length so this is the torque distribution

and we know that whenever there is a torque we have the shear stress this is equal to sixteen times the torque divided by pi dcq

so this is the shear stress and this is the normal stress

sometimes of course depending upon the application of the load of the of this ah um this torque we may have bending stress

for example let us consider not a direct application of the torque but {app} (00:07:14 min) with a handle we apply some force over here so if we consider the handle to be a ah rigid body then the equivalent force system will be a torque just like this and in addition there will be one force

so this force causes bending and this bending stress will be very high over here of course we do not consider any bending stress beyond this because we consider this to be clamped

so therefore the bending stress will be something like this and this is the bending moment distribution and the bending stress all we know this is again normal stress [noise] and this will be thirty-two M by pi d dcq

so there are different kinds of stress available here

now you have to find out you have to use the failure criteria and ah get the correct value of dc and that is the route diameter of the thread

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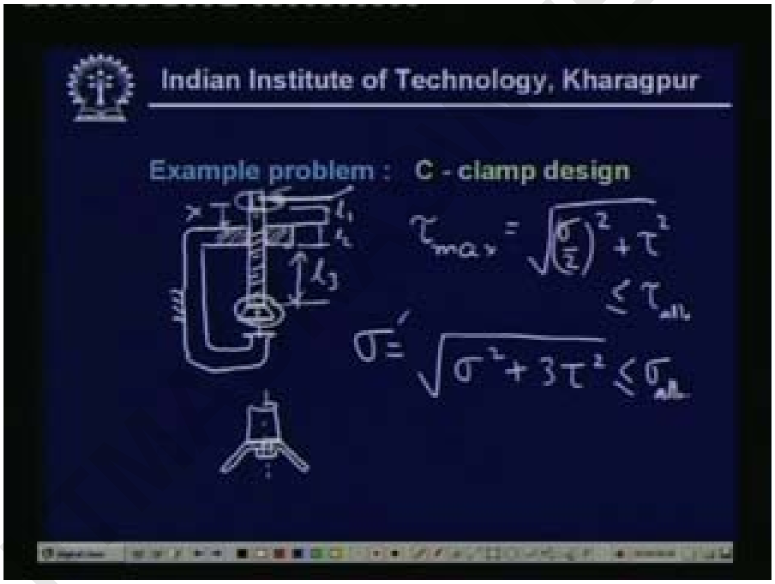

now there are few failure criteria's which you have say we have tau max this is the failure by maximum shear stress and this will be equal to sigma

now we have to consider the overall sigma

so that is depending upon the sections here you have the bending stress here you have the compressive stress and of course torsional stress will be present always

so the sigma tau max will be sigma by two square plus tau square and that must be less than equal to tau allowable

now this is the criteria with the maximum shear stress in some materials of course in most of the materials the better criteria is the Von Mises stress which is given this way this is Von Mises equivalent stress and this is given by sigma square plus thrice tau square and this must be less than equal to sigma allowable

so you know all this

now using this failure criteria you will have to find out the value of dc

then once the diameter is selected we have to consider one more thing that is all though it doesn't belong to the theory of failure but ah there is one instability problem with for which the material may fail and that is known as buckling

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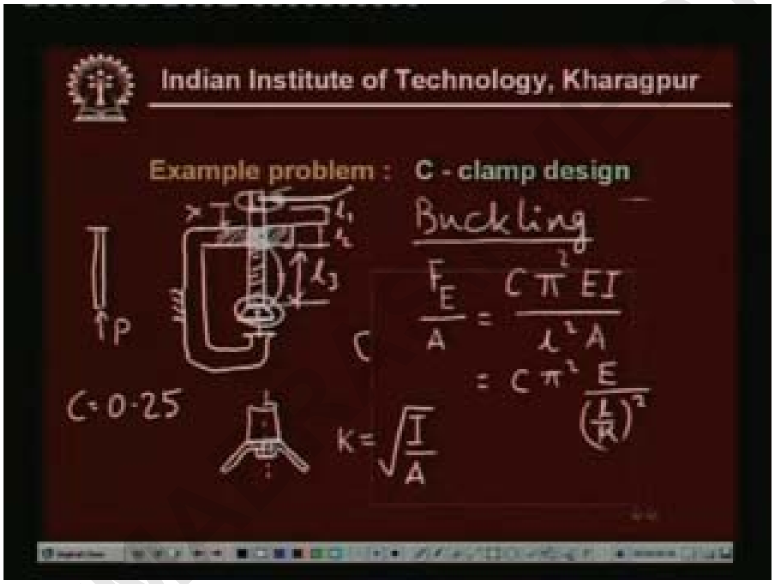

so the buckling is one of the important criteria which has to be considered here because this part is subjected to the compressive stress

one once a material is subjected to the compressive stress then all of a sudden the material will become unstable

the whole structure may become unstable and it may actually bend like this and whenever it bends then a large amount of bending stress is developed over here which ultimately leads to failure [noise]

so we will have to check for the buckling

now all of you know that the [noise] the critical load for buckling the very simplified ah analysis of buckling critical load is given by Euler formula and that is equal to C times which is C is a constant i i shall i come to that later C times pi square EI divided by l square A which you can write C times pi square E by L by k square

now this is the critical load for ah Euler's equations but uh where C will be a constant and that depends upon the boundary conditions depending upon the boundary conditions the failure may start at different region

now here one reasonable case of boundary condition will be let us assume that this part is clamped that is here we are considering a buckling of a beam which is fixed at one end and free at other end and subjected to pressure P

then of course the ah value of C will be in that case point two five and then we have the Euler ah critical load for buckling but if l is very very small that compared to the k now i didn't mention this k is the radius of (( )) (00:12:18 min) and this is nothing but ah I I A

so this is the radius of (( )) (00:12:40 min) and we have this formula C pi square E by l by K

now whenever this l is very small compared to K then we can see that if E could be very high but in fact it doesn't happen because before it reaches very high level then the material will be crushed so it will be smashed and we do not allow that

so therefore we need some correction whenever this factor is very very small and the one of such correction formula was given by professor um P A Johnson and this is known as Johnson's formula

now if you consider this loop if you remember this formula then the Johnson's formula will be given by this one

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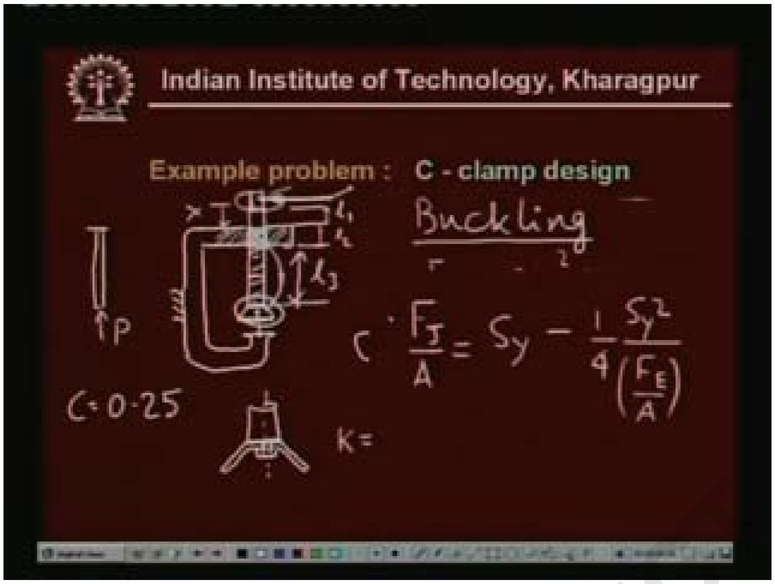

where F Johnson divided by A will be equal to the ill strength minus one-fourth ill strength square so this is a parabolic formula and this is divided by FE by A

so now this is a parabolic formula given by Johnson and we have to use this formula in order to check whether the given load that is once we have decided for the for the diameter the chord diameter then we have to decide whether the given load is more or less than F J

definitely if the load is more than FJ then we'll have to increase the diameter [noise] by some amount and that then again we have to check this formula

now this is about the buckling

now ah we have known sigma we have known tau and we have calculated the diameter but in order to calculate the shear stress we must know the collar frictions

let us talk a little bit about this collar frictions [noise]

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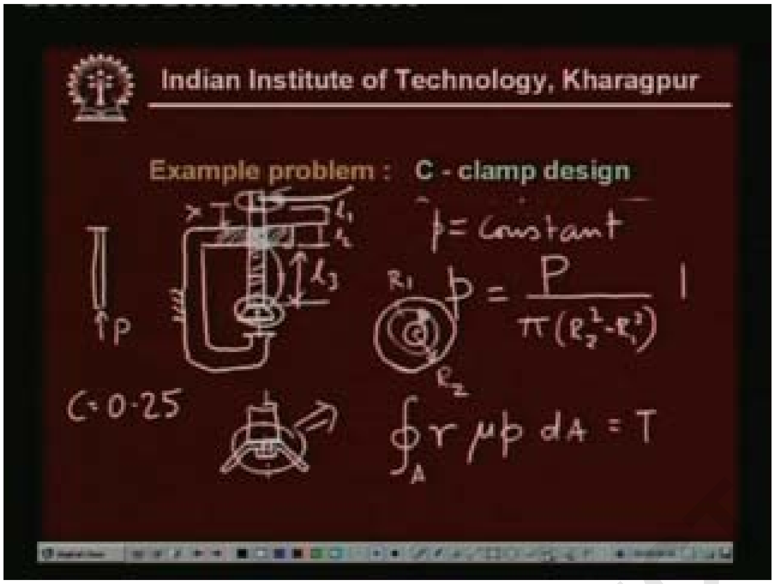

now the collar frictions if you look little bit closely here this thing then there is a collar and this is the recess

let us say the outer radius and the inner radius this this one is R two and the inner radius is R one say so this is R one

and then whenever we start from a of a from a smooth {sur} (00:15:22 min) of a new surface then definitely the first reasonable assumption will be that the pressure distribution is uniform

so this is subjected to pressure and this uniformly distributed

therefore we have P which is a constant

how to get the value of this {con} (00:15:41 min) constant

this comes definitely from P equal to then the total force divided by the area that is pi R two square minus R one square

and then how to calculate the torque from it

we know given a small area d A then the frictional force will be mu times the {for} (00:16:14 min) the pressure on it

so the net force net normal force on the this element area d A will be P and d A

now this is again the normal force is P times d A so therefore the friction force will be mu times P times d A and then we have to find out the torque

now the direction is always opposing the motions so the direction will be always tangential here therefore in order to calculate the torque net torque then we'll have to multiply it by R the radius of this elemental area from the centre and then we have to integrate over the entire A here

and this will be the torque now if you calculate then the torque becomes then this torque collar becomes two-third pi

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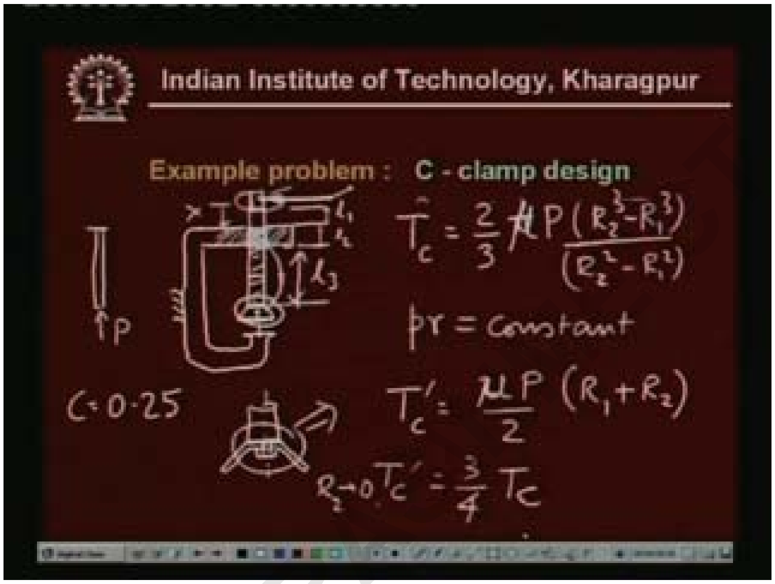

then we have P R two cube minus R one cube divided by R two square minus R one square

so this is the torque required to overcome the collar frictions

now once it has started working then after sufficient wear and tear then the the surface becomes smooth and then the pressure does not remain any more uniform and what the ah reasonable assumption will be that wear is uniform

then after some time is evolved then we assume that wear is uniform and definitely wear remains [noise] wear ah depends on ah on both the normal pressure as well as the travelled distance again the travelled distance is proportional to the r the radius from the centre

so therefore the wear constant implies P times r is constant and again if you do the same calculations what you get is this form that is for constant wear we have this is mu by mu times P by two times R one plus R two

now let me make a small correction here i wrote pi but this should be mu as you can very well see because it depends on the co-efficient of the frictions so therefore Tc prime that is the torque required in order to overcome the collar friction under the condition of uniform wear is given by this form

then it is ah very easy to check that if R two is very very small then what happens then Tc if R two is very small that is goes to zero then Tc prime is about three-fourth times of Tc

so therefore once it has worn out sufficiently then the friction becomes or the frictional torque becomes quite low at least the low compared to the the first case when the pressure was uniform

so this is how we could estimate the the shear ah the torsions here in the collar [noise]

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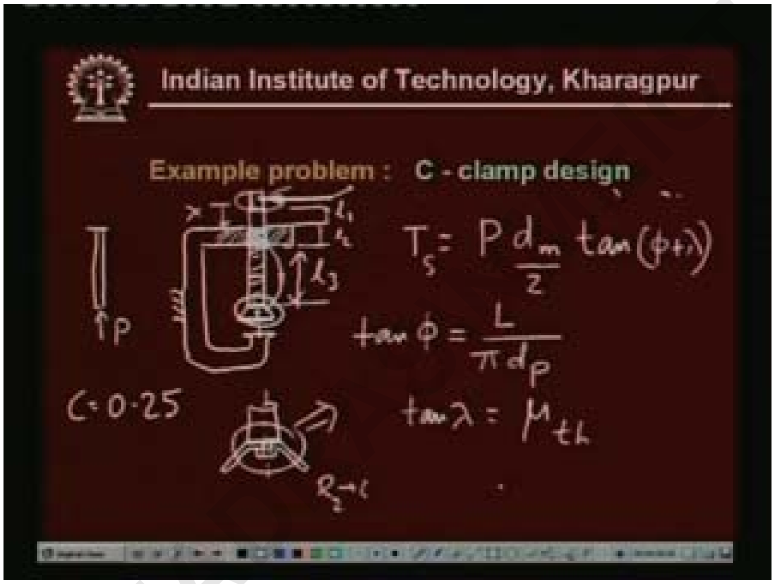

then of course the torsion in the screw that is torque in the screw that is torque to to overcome the friction in the thread will be given by the {same} (00:20:26 min) the known formula which i derived in last lecture and that was equal to P times mean radius of the thread and times tangent of phi plus lamda

where phi is the angle helix angle and this is nothing but lead divided by pi pitch diameter and ten lamda is nothing but mu at thread

so this is the friction co-efficient between the screw and the nut and um the friction force have appears in the thread

so we now know the individual values of Tc and Ts and from this we can calculate the ah the shear stress due to torsion now let us come to the region which is very very important namely the region of engagement between the {threa} (00:21:38 min) between the ah screw and the nut [noise]

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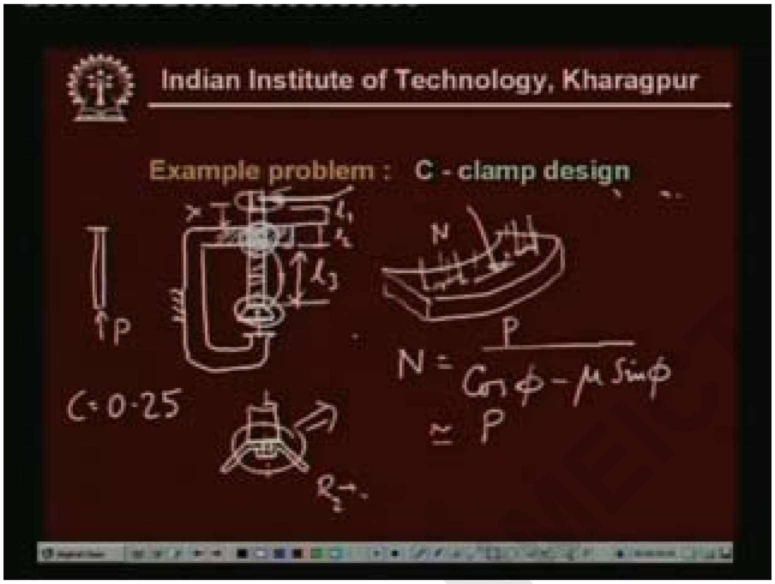

now you see let me draw the part of the thread over here now this is a part of the thread and it is subjected to the pressure P and [noise] and the frictional force over here and if you remember the last classes lecture last lecture then the normal force that is the the normal force on it was P divided by cosine phi minus mu times sin phi

where again phi is the helix angle now {se} (00:22:46 min) for a normal thread phi is very very small therefore this term is very small and it is again multiplied by mu which is again a small number

so therefore the contribution of this term is very very small

and if phi is very small then cosine phi is almost equal to one so therefore it is very much equal to P [noise]

now this amount of normal force is distributed or acting uniformly over the thread

so it develops what is known as the bearing pressure on the thread

let us look at the bearing pressure now

now the as i said the pressure is uniformly distributed over the thread

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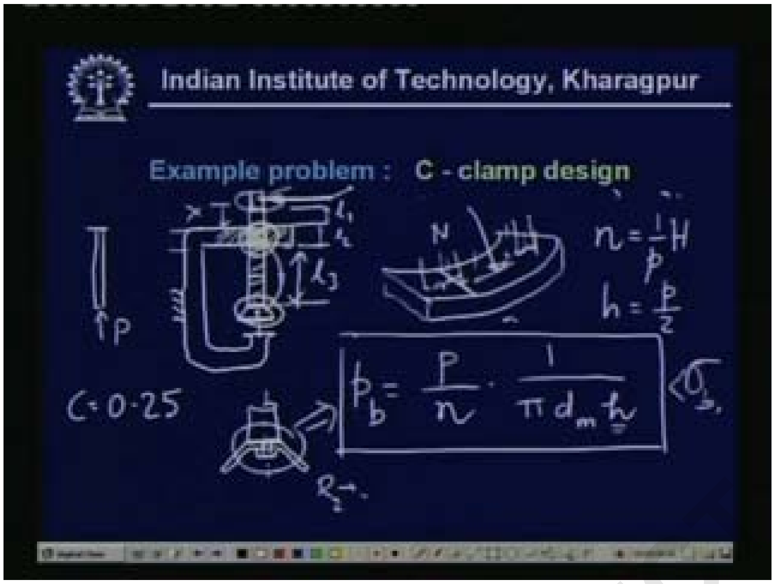

so if you consider a single thread then ah the total force on the single thread will be equal to P by n where n is the number of threads in engagement with the nut [noise] and it depends upon the pitch that is n will be equal to one upon pitch that is equal to number of thread per unit length and times the height of the nut that is this length

so this is the total number of threads in engagement hence the total force per thread will be here this divided by the area

and what will be the area

area will be definitely pi times d mean and times this length which is the h which is equal to the ah height of the thread and that is equal to P b that is the bearing pressure

now this bearing pressure must be low as as low as possible because depending upon the pressure the wear and tear will takes place and this ah um detrimental to the life of screw and the screw may gets damaged

so therefore P b is one thing which we have to control and that could be done by selecting the h again h is for a square thread h is nothing but P by two and hence we have to check for this P b such that this P b becomes less than some bearing allowable bearing pressure

there are other kinds of stress in the the thread

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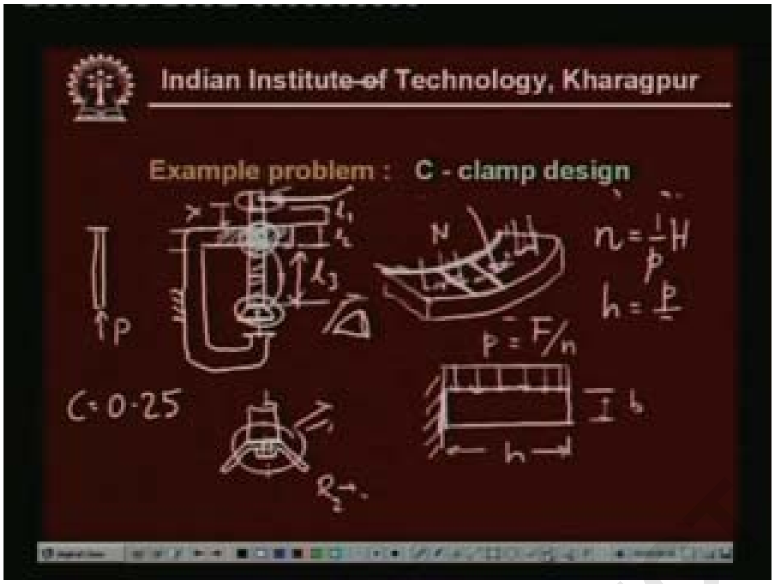

so if you consider a little part which is let us say unit distance apart and look from this side then what you see it looks like a cantilever beam because this side is formally grounded

so therefore this is cantilever beam whose height [noise] length is h and this length is b and it is subjected to uniform pressure again P which is equal to F by n

so therefore for this kind of force we we shall have both the shear stress here and the bending stress

what will be the value of shear stress [noise]

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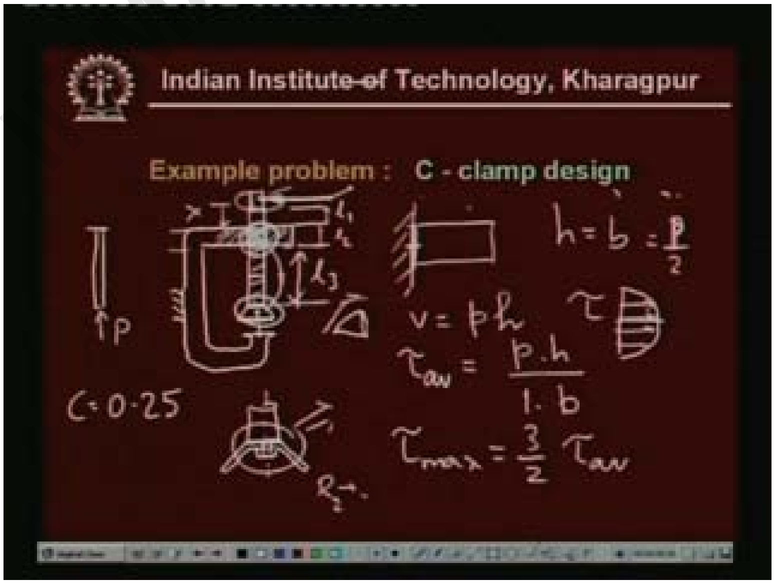

the value of the shear stress here the shear stress v at x equal to zero is nothing but P times h hence the this is the shear force therefore the average shear stress will be P times h divided by the area and area was unity the width and the height that is b

and for again for spare thread h is equal to b is equal to P by two where P is the pitch of the thread

now we can even go further because this is this has a ah uniform cross section then the shear stress distribution is not uniform in fact the shear stress is distributed parabolically and this is the roughly the distribution of shear stress

the maximum shear stress occurs somewhere here and the value tau max will be three by two tau average that is the maximum value is one point five times the tau average

so this is about the shear stress it is maximum here minimum here and varies parabolically

what other stresses the other stress is the bending stress

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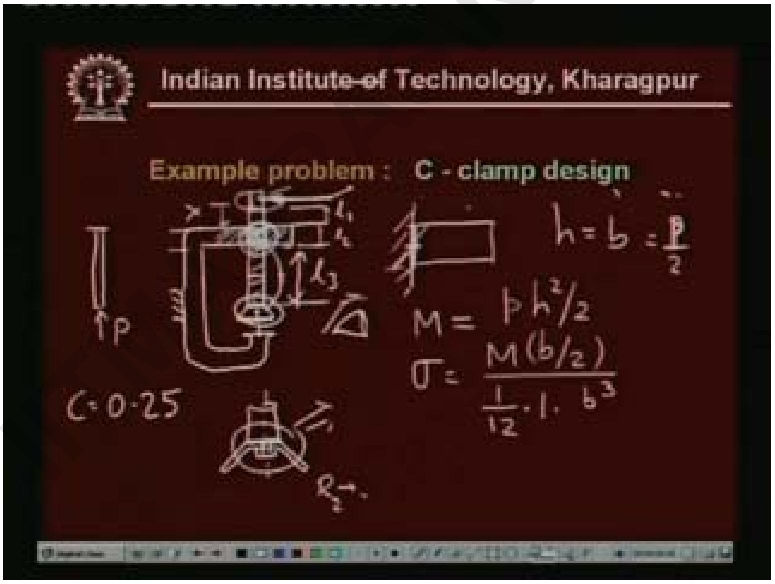

it is easy to calculate the bending stress first comes bending moment which is equal to P h square by two therefore the bending stress is M and here the maximum bending stress will be at b by two divided by i which is equal to twelve times one times b cube

so this is the formula for getting the bending stress

now we know how to calculate the shear stress and bending stress in the thread then different section of the thread will undergo ah different state of stress let me point out now one

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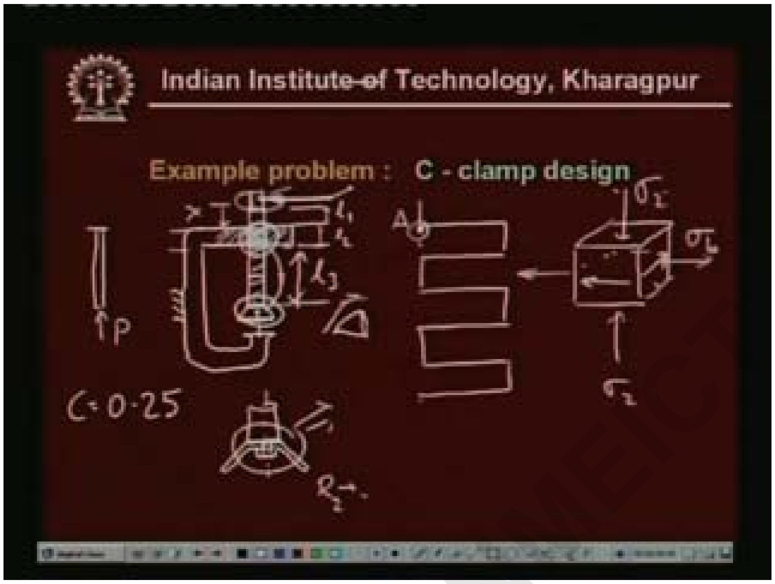

if you consider the cross section of the thread then this point will have the compressive stress

suppose it is here then the compressive stress is {maxim} (00:29:47 min) P there will be compressive stress so if you have one element at point A if you draw the state of stress there will be compressive stress

okay with sigma z sigma z say there will be the bending stress which is sigma b and the value which i have already calculated just now [noise]

then there will be the shear stress and the shear stress not from the thread thread pressure but from the external torque and that is appearing

here we have countering shear stress and this is the state of stress at this point

similarly if you want to draw the state of stress you can do it ah quite quickly [noise]

now once the state of stress is known then you can use those formula that is either the maximum ah shear stress formula or the Von Mises theory of failure criteria and find out the ah safe limit that is whether the thread will fail or not

now there is one problem in this very simple design that is [noise] here we have assumed that all the threads are ah having uniform pressure

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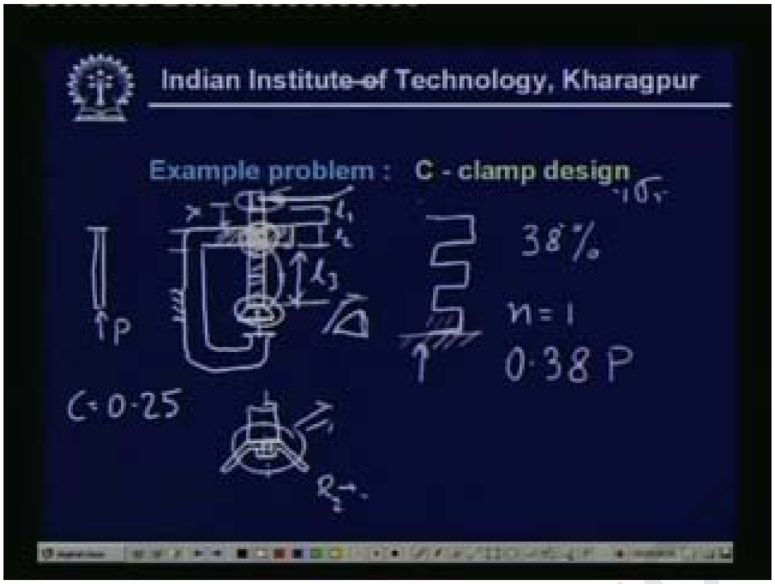

that is the engagement if there are n number of threads then all the threads will take the same load but this is not so and the problem of getting the stress this force distribution per tooth will be very complicated and it is ah in fact statically indeterminant problem

in order to calculate the stress distribution here we have to solve a different well a complicated {ela} (00:31:53 min) elastic problem

usually what is seen from experiment that the first few ah threads will take up the majority of the load

for example it was found out that the first thread which is engaged in the nut so this if this side is the ah is the portion of the nut which is directly first in contact with this thread then this thread will take up about thirty-eight percent

and usually as you go along then of course [noise] the ah the percentage of load carried by individual thread will be very very small

beyond seventh stage the threads do not carry any load practically at all

so therefore if we think that by increasing this height of this nut we can have the {min} (00:32:45 min) the low bearing pressure or low shear stress in the thread this is wrong assumptions because all the threads are not under uniform pressure

so one way of having a good design is that assuming n equal to one that is the number of thread in engagement is one and the load instead of taking the full load let it take point three eight of the load and then we calculate for ah these stresses and find out whether this is in safe or not and this is very conservative design

so this was roughly the design procedure of a C clamp

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now let us come make a summary[noise] what are the stresses in power screw

power screw will have different kinds of stresses one is axial compressive or tensile stress

now depending upon the situations the part of the ah screw may have compressive stress or part of the screw may have tensile stress so we will have to calculate for the uh for both the cases

now whenever the screw is under compressions then in addition to checking the ah crushing failure we'll have to the buckling failure because this is very dangerous buckling occurs all of the sudden and suddenly you will see the material or the structure gets deformed and um a large bending stress develops and it fails

so therefore this is a very dangerous thing and has to be prevented this buckling

then comes the shear stress due to torsion

here we have two sources of ah shear stress that is one is ah the torsion required to overcome the friction resistance and and that occurs only in the thread and ah thread between nut and the screw

there is another source of the external torque which is equal to the frictional resistance at the collar or in the bearing then comes the bending stress again the bending stress doesn't ah happen always it happens only in those cases ah and depending upon which depends on the applications

then comes the bearing stress on threads so this is very important most of the power screws fail by this bearing stress and bearing stress would becomes very large then lots of wear and tear goes goes on and that has to be prevented

so this bearing stress is very important and has to be taken care off

then the shear stress in threads due to pressure we have seen that the shear stress developed in the threads that is because of the pressure then the bending stress in threads due to pressure

now all of the ah stresses due to this three last three [noise] criteria must be taken care of and we will have to find out the equivalent stress that is the ah um maybe when you use the Von Mises criteria you will have to find out equivalent a normal stress or if you want to use maximum shear stress then you will have to calculate the maximum stress

everything depends upon the state of stress at different points [noise]

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now let us come to the materials which are used for screws and nuts

the screw materials may be still without hardening that is we may use sometimes soft steel

then steel with hardening and ah what is the purpose of that hardening

now we know the lots of wear and tear will take place between the screw and the nut so in order to have the wear minimum we will have to make the surface of the of the thread wear resistance and that is that could be done by hardening

there are various methods of hardening but all of them introduce some sorts of deformations

we have to take care that sufficient wear resistance is developed as well as the deformation due to hardening is minimum because that will lead to inaccuracy in the thread profile [noise]

one such method of hardening which proves to be very very effective is nitriding and here you will see that the wear resistance gets improved as well as the ah work hardening become very very small

there are other methods using electromechanical hardening

the nut materials they are usually of softer material [noise] sometimes we take brass or bronze cast iron is also taken now cast irons are um used as nut material as it is whenever the load is small as well as the the speed that is the ah velocity rubbing velocity is small

if we want to {u} (00:37:54 min) increase the load then we have to ah we use the cast iron but the thread the surfaces are ((babited)) (00:38:04 min) by some alloying elements

so these are roughly the materials used for screws and the nuts

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Application	Material Screw	Material Nut	Pressure (MPa)	Rubbing Speed
Hand Press	Steel	Bronze	17.5-24.5	Low
Screw Jack	Steel	C.I.	12.6-17.5	<2.4m/min
	Steel	Bronze	11.2-17.5	<3 m/min
Hoisting screw	Steel	C.I.	4.2-7.0	6-12m/min
	Steel	Bronze	5.6-9.8	6-12m/min
Lead Screw	Steel	Bronze	1.05-1.7	>15m/min

now let us come to the bearing pressure of different types of ah threads that depends again on the applications to applications in the hands spray hands press you will see the materials are normally the screw material is steel and the nut material is bronze

the pressure is seventeen point five to twenty-four point five mega Pascal which is very very large but the rubbing speed here is quite low

and whenever we have such large bearing pressure there's this is the safe pressure for bearing and whenever the bearing load is very very high then there has to be lubricated properly otherwise it will wear it will wear out

then we have the screw jack the screw jack can be made of steel the screw is steel nut is cast iron here twelve point seven six to seventeen point five little lower than that the before

the speed is of course is less than two point four meter per minute

so this is quite small speed we have another steel when we use the bronze then of course the bearing pressure is is ah almost the same to eleven point two to seventeen point five but with bronze we can go upto three meter per minute rubbing speed

for the hoisting screw screw material is steel and the nut is cast iron here the bearing pressure is quite small four point two and seven point zero where as the rubbing speed here is quite high six to twelve meter per minute

see it is again ah very moderate speed not very ah not very high not very low we may have bronze

now with the bronze of course the pressure the bearing pressure could be little higher five point six to nine point eight mega Pascal

when you have lead screw now lead screw ah used for giving feed motions and uh that has to work very fast

so therefore the rubbing speed is very very large but now the pressure becomes quite small that is when we use the screw material as as steel and the nut material as bronze then the rubbing pressure becomes uh that is bearing pressure becomes one point zero five to one point seven mega Pascal

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Materials	Average friction coefficient	
	Starting	Running
Soft steel on C.I.	0.17	0.12
Hardened Steel on C.I.	0.15	0.09
Hardened Steel on bronze	0.08	0.06
Soft steel on bronze	0.10	0.08

then one more thing ah in order to design is the co-efficient of friction for thrust collar [noise] again it depends again the co-efficient of friction depends upon the material of contact that is whether the steel or cast iron or hardened steel on cast iron hardened steel on bronze etcetera

we will see the starting co-efficient of frictions that is the static co-efficient of frictions sometimes call it is quite higher than that of the running that is the kinetic co-efficient of frictions that is very much known to you [noise]

now soft steel on cast iron has a starting co-efficient of friction point one seven and the running co-efficient of friction is point one two

when you use hardened steel that is after hardening and again ah cast iron then the starting co-efficient of friction is point one five and the running co-efficient of friction is point zero nine

then the hardened steel on bronze now use the bronze which is the softer material then of course the friction becomes very very small

you see the when you use the hardened steel and bronze it becomes point zero eight in starting conditions that is the ah static co-efficient of frictions and the kinetic co-efficient of friction is point zero six

soft steel on bronze the starting co-efficient of friction is point one zero and the running co-efficient of friction is point zero eight

again these are representative values you should not take it very ah very rigidly so it is usual practice to take such values but one can differ ah depending upon the situation to situations and also the co-efficient of friction depends on the environmental conditions also that one must not forget

so let us see what we have learnt today in this lecture

we have designed we have taken a very small example a simple example of a C clamp the design is to calculate the the diameter of the thread of the the screw

now that is based on different forces the compressive stress as well as the torsion the compressive stress again compressive and tensile stress may appear

and the torsion which you have to take into account then comes the thread[noise] the thread is very ah important

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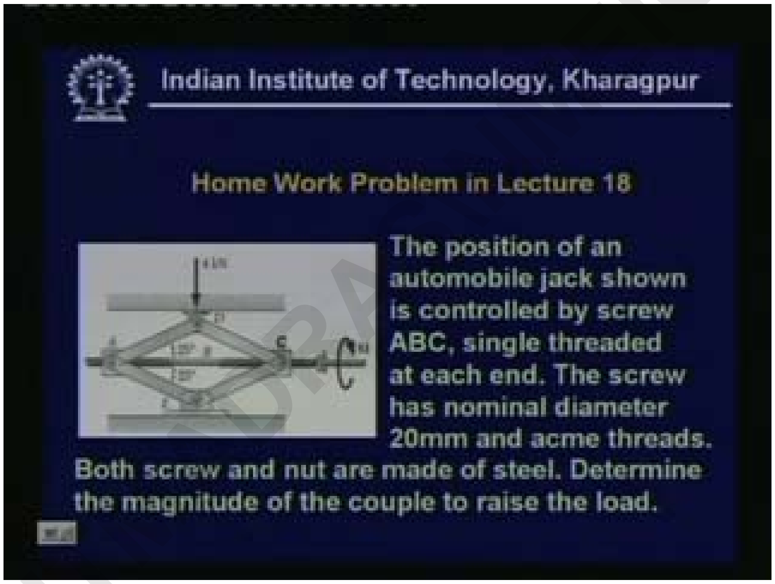

and we have to consider [noise] ah the bearing pressure in thread as well as the shear stress in the thread as well as bending stress in the thread and

depending upon everything so combining all we'll have to find out the safe diameter that is ah used for the screw

now let us come to the homework problem which was given in lecture eighteen [noise]

the problem was the position of an automobile jack shown is controlled by screw A B C single threaded at each end the screw has nominal diameter twenty millimeter the nominal diameter is twenty millimeter it was acme thread now we know that acme thread has profile this thread angle is twenty-nine degree

both screw and nut are made of steel determine the magnitude of the couple to raise the load

let us see the important thing is that it has acme thread then the material screw and nut

now remember i gave you one table let me reproduce the table

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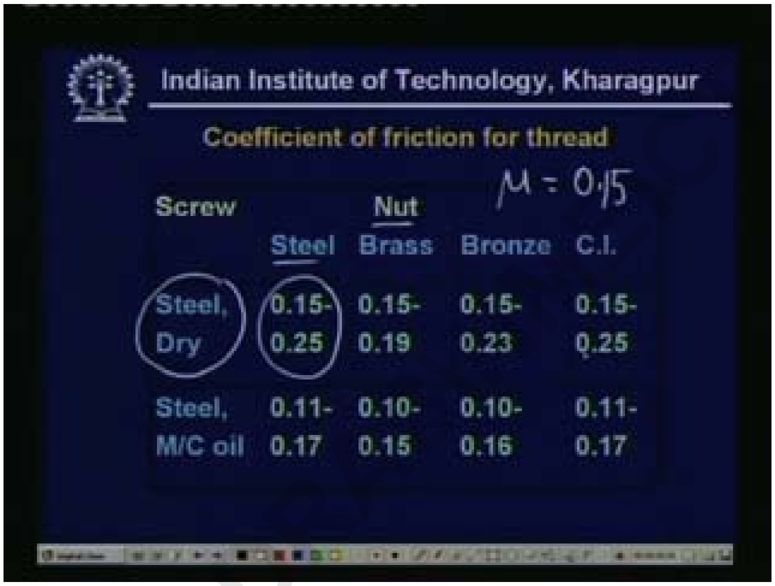

here is the table which i ah showed in the last lecture

the co-efficient of friction for thread the screw material and nut material let us assume the steel is ah steel dry and the nut is steel

so therefore we have to take the co-efficient of frictions from this data

we choose the minimum co-efficient of frictions that is mu is point one five and calculate

so this gives the minimum torque required [noise]

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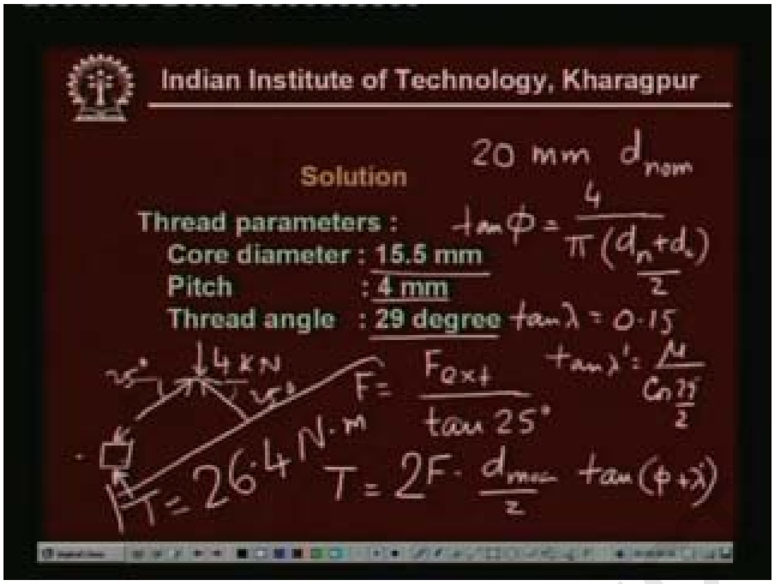

here comes the solution when you look up the table that is given in any design handbook you see that for twenty millimeter nominal diameter the core diameter is fifteen point five millimeter and the pitch diameter is or pitch is four millimeter and the thread angle is twenty-nine degrees

so therefore [noise] we know the phi that is tangent phi will be equal to pitch it is single threaded tangent phi is four divided by pi and we'll have to consider the pitch diameter and the

pitch diameter is nothing but ah the here for the acme thread pitch diameter will be equal to the nominal diameter plus the core diameter by two

so this is nothing but the mean diameter

so when you calculate you get the value of phi then we have to calculate the value of lamda that is tan lamda is equal to point one five

then from this given data we'll have to calculate the force on the nut

so that is again done by elementary static analysis if you consider four kilo Newton here then of course you get this was twenty-five degree [noise] and again this is twenty-five degree

so you have you have to calculate the force over here then this force is again transferred to this nut again from this side and we have to calculate the net force

so the net force will be here net force on each nut will be equal to external force divided by tangent of twenty-five degree

so this is the force which we have to which the ah thread has to withstand and when you have one thread one ah bolt then the torque is given by F times d again d n plus d c by two that is the d mean by two and tangent of of phi plus lamda dash

now for the acme thread you know lamda dash tan lamda dash is nothing but mu divided by cosine of[noise] theta by two twenty-nine degree by two

and when you calculate that you get the value of the torque but there are two threads which are opposite handed and therefore the total torque will be equal to double of that

so hence the answer is here and when you calculate everything what you get the torque required will be equal to about twenty-six point four Newton meter

so this is how we have to solve this kind of problem [noise]

so lets us now once again recapitulate ah in last two lectures we have learnt the kinematics and statics of thread of ah power screw and then today's lecture we have known how to design such a power screw

the most often used screws are ah with acme thread or square thread they have their respective advantages and disadvantages this screw materials are mostly steel hardened or soft the nut materials could be bronze brass or cast irons

so you know now after this lecture you know how to calculate how to design a safe power screw so thank you

Preview of next lecture

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Design of Machine Elements-I

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Lecture No. # 20

Shaft Couplings- I

dear student we are now going to begin lectures on machine design part one this is lecture number twenty and the topic is shaft coupling

so this is the first of lectures on {rest} (00:50:20 min) same topic

now couplings are normally used to connect two shafts axially now the shafts are usually manufactured in small pieces in order to avoid the difficulties or ah disadvantages of transport problem [noise]

ah and whenever they are connected they are connected by means of this couplings

now there are ah another device there is another device which serves the same purpose this is known as the clutch

the important difference between the clutch and the coupling is that the clutches are used in order to disengage and engage the shafts frequently whereas the couplings are used to fasten them together

now this is permanent connections and they are only opened up during replacement or repair or maintenance

so this is one of the importance difference of coupling

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now the couplings are used mainly to transmit power without any loss

the transmit this is [noise] now this is what we want to say the transmit power without any loss that is very much expected

and second provide for shaft misalignments now shafts are normally [noise] ah installed without any misalignments but because of various reasons there will be inevitable presence of misalignments

now the misalignments may be of different types let us see suppose we want [noise] this shafts to be connected this way

so this ideal situations two shafts axially connected in this fashions and there is no misalignments

there may be misalignments for example one shaft is here so this is this one is ideal there will be misalignment when one shaft is here and the other shaft the axis of that shaft is offset by some distance delta

this is one kind of misalignment

there will be another kind of misalignment that is there is one shaft then then the other shaft is acute little bit by an angle alpha this is one such misalignment

there may be a combination of these two that is a shaft may be there and the other shaft is both angularly misaligned there is angle alpha between this two axis and there is offset delta

so there are four kinds of misalignments and those are to be avoided

ah but normally this misalignment occurs in practice and it is difficult to avoid [noise]

then we have provisions that is couplings have to used in order to provide for end movements of the shafts [noise]

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now the types of shaft couplings

there are two types two principle types of shaft coupling one is rigid coupling and another is flexible coupling as the name suggest the rigid couplings are couplings without any flexible member

there are different kinds of rigid couplings we have to discuss four which are the principles types of rigid couplings

namely the first one is flange coupling this is the most commonly used coupling

the second one is sleeve or muff coupling

third one clamp coupling or sometimes it is called split muff coupling then compression coupling

flexible couplings are of two types one is coupling with kinematic flexibility that is we insert certain members

so which undergoes rigid body translation or rotation but that gives a little bit resilience or flexibility in the overall ah joint [noise]

the second type is coupling with resilient members

now here ah we introduce some members which have the flexibility

so this is inherently a flexible couplings we are going to discuss in the this lecture the principle types of rigid coupling

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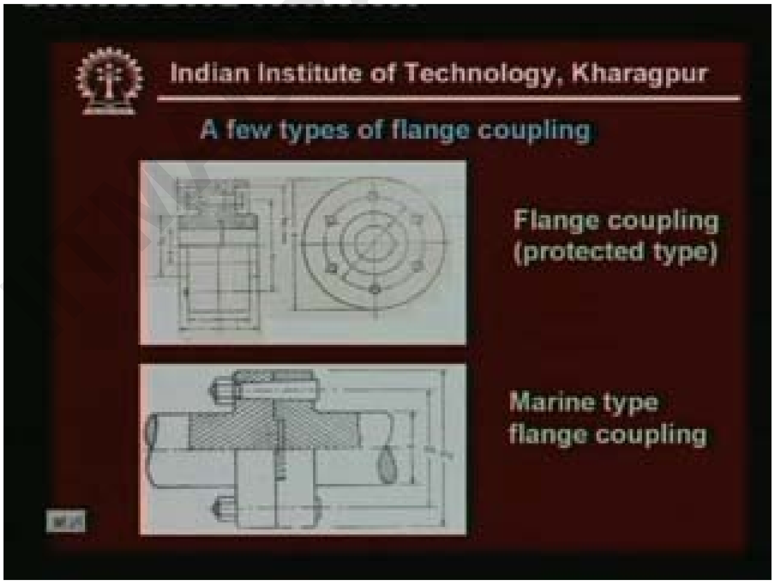

now let us come to few types of flange coupling now flange coupling you see ah these are the ah couplings again rigid couplings and the coupling is done by means of two flanges what you see here is the cross section of such a flange coupling here the shaft and on top of it this part which is known as the flange which has two sections one is here which is relatively ah thick part which is which is it is known as hub

and this part which is relatively narrow then this is known as the flange and these two flanges are connected by means of few bolts

parameters of flange couplings are given here [noise]

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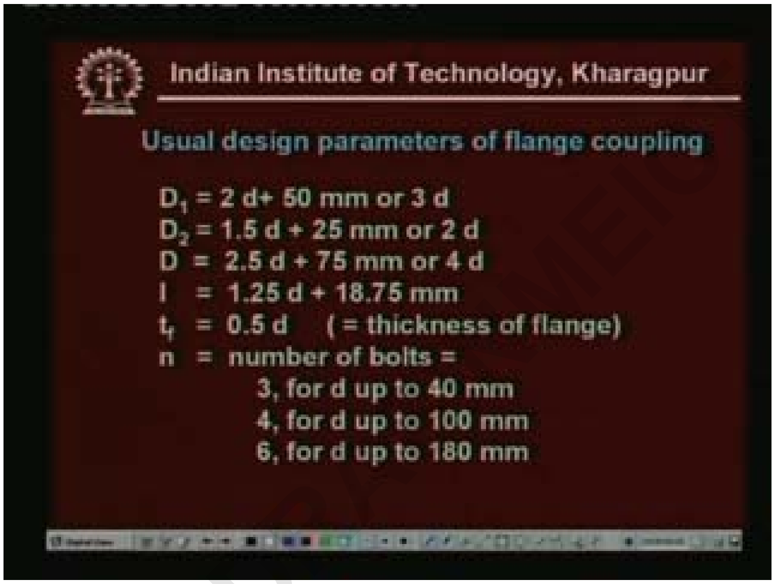

D one as i said which is equal to the bolt circle diameter this is roughly taken to be twice D small d that is the diameter of the shaft plus fifty in millimeter that is fifty millimeter or it may be three d

now why all these because the {cap} (00:56:26 min) couplings are now manufactured by different manufacturers and so we have to come to a standard these are normally the standard followed unless otherwise specified to the manufacturers

there are ah catalogues available from the manufacturers and we have to look up into the catalogues to find out exact diameter but what you see here are normally the usual diameters usual dimensions maintained in practice[noise]

D two which is the hub diameter is one point five times d plus twenty-five millimeter or it may be twice of diameter of the shaft

everything is mentioned in terms of diameter of the shaft the outer diameter is to two point five times d plus seventy-five millimeter or roughly four d

length which is quite important it is about one point two five times d plus eighteen point seven five millimeter you see this odd figure

but these are standards tf the width of the ah or the thickness of the flange which i have just i have told in the last slide is point five d and n that is the number of bolts {wui} (00:57:36 min) that may be three for d up to forty millimeter

four for d up to hundred millimeter six for d up to one eighty millimeter

so these are the usual dimensions ah which are maintained in practice

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now let us come to the design criteria of flange coupling so [noise]

so these are the two flanges which are connected by bolts

obviously the