Tribology

Prof. Dr. Harish Hirani
Department of Mechanical Engineering
Indian Institute of Technology, Delhi

Module No. # 06

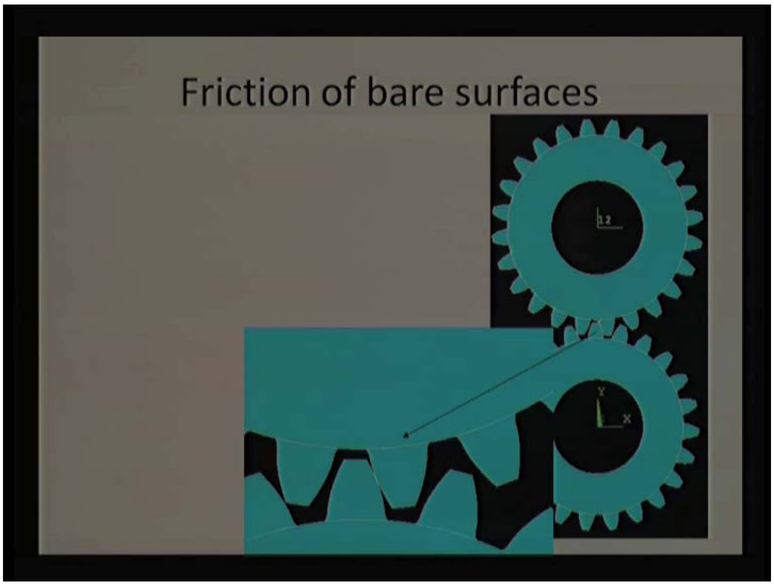
Lecture No. # 36

Friction and Lubrication of Gears

Hello and welcome to 36 lecture of video course on tribology, topic of the present lecture is a Friction and Lubrication of Gears. In previous lecture we started gears, we defined gears, defined profile of the gears and mentioned there will be some sliding, other than other pitch points, and that sliding causes a friction. In addition, as we use lubricant to reduce a friction, but there will be another disadvantage or that will cause a disadvantage, that will be churning effect.

Only one or two gear teeth will be in match, remaining gear teeth will be without machine, so they start splashing a lubricant and there will be some loses, frictional loses, and energy loses, because of that. We are going to discuss this topic in a present lecture.

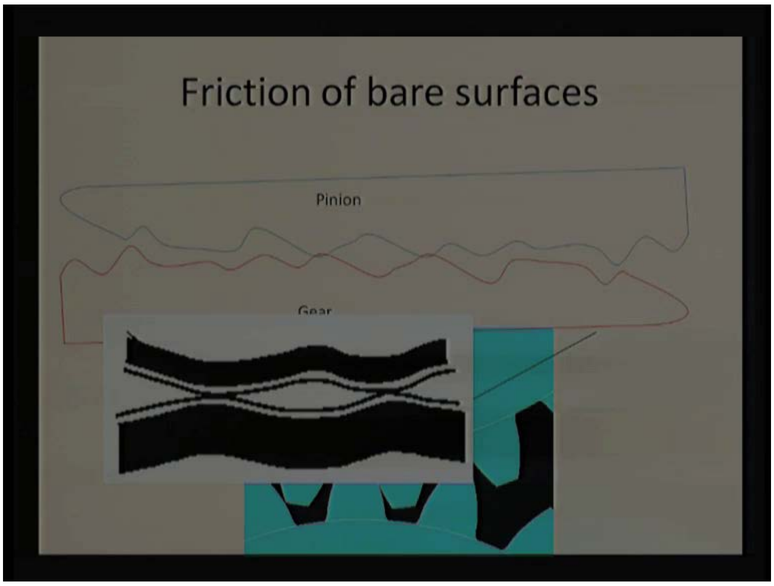
(Refer Slide Time: 01:25)



Say that, we can start thinking about the bare surfaces without any lubricant. When we are discussing about the friction at the different troposphere's, this was the first what we call as a dry friction. And to illustrate it, to demonstrate it, we have shown in this gear pair in the match, and we are able to see, there is a matching at this point there is a matching, if I zoom it we are able to see the match, and we are able to see some sort of elastic deformation on the surface.

If surfaces are without any lubricant and they need to rotate, naturally there will be high friction, coefficient of friction may even reached 0.6, 0.7, if surface finishes are not very good, surfaces are not harder they deflect easily then, that will not be desirable, because of gears are used for the power transmission, not for power losing effect. We want to transmit the power as much as possible, efficiency should be 100 percent or target will be always efficiency should be 100 percent and if we take care properly we can reach to 99 percent, 98 percent; reaching beyond that will be just impossible unless we change the technology, from mechanical point of view that will be pretty difficult.

(Refer Slide Time: 03:06)



Now, this can be demonstrated the why there is a friction, you are able to see there are some asperities, whenever there is a mechanical contact, under load condition there is a possibility of deformation of the surface and because of the asperities, there will be inter locking, and this inter locking may be based on just abrasion or may be based on adhesion of the cool welding.

In particularly dry surfaces, cool welding is predominantly phenomenal, coefficient of friction majorly occurs, because of the adhesion, because of the cool weld and the asperities; and we need to reduce as a low as possible, that is how we say this figure shows this this asperity (()) to penetrate on another surface it will be difficult to remove the pinion over the gear.

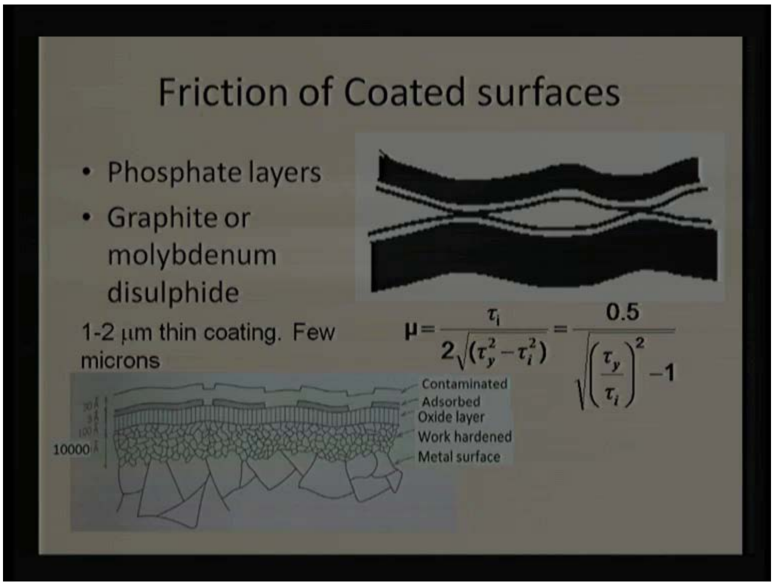
However, we know very well that, in normal atmosphere condition there will be always a one contaminated layer on the metallic surfaces, while we are discussing over here, particularly for the metallic surfaces and mostly gears are manufactured having metal surface, even though we know very well that polymers have, started of the polymers have been used to manufacture gears, but for the low load applications.

Even in this case or those cases also there will be frictional loses, what we are talking here only for metallic surfaces and we say the metals, if kept in environment, it will absorb few layers may be (()) to the nanometer level thick layer, but there will be some kind of a layers on a surface and that is shown with this white colour layer on the surfaces, we are showing that, the both the surfaces may be the pinion surface and gear surface.

Both will have a some sort of contamination layer, without any lubricant we are talking that it is a without any lubricant and these layers are generally having a lesser shear strength, that is why the relative sliding will be simpler compared to the parent material, compared to the bare materials, without any contamination layer on that.

And this has been studied in our friction module lectures, may be there is in 3rd, 4th lecture which was discussed in detail that, if there is a bare surface and kept environment there will be some sort of oxidation and that oxidation itself, will generate low shear strength, lubricant layer on a surface, but still that can be treated as dry lubrication.

(Refer Slide Time: 05:55)



Now, as this is being shown that, it has there is a contamination layer on that, the question comes, can we really go ahead, can we design this lubricant layer our own using some sort of solid lubricant, we have studied solid lubricants, we know about the molybdenum disulphide, we know about the graphite surfaces, even we know the (()) coatings, those are the solid lubricant layers. It can be designed as per the requirement, while what we discuss in to the natural layer, natural layer we do not have much control, it will change with the environment, we are operating at the 30 degree centigrade, though there will be lubricant thickness (()) on a surface.

Now, we start operating 70 degree centigrade there is a possibility, that the layer get thicker or there is a possibility that the oxidation rate increases, but there is a possibility also that they will start softening and the because of the soften effect there will be more and more cold welding, their hardness will reduce. So, coefficient of friction will continuously change as the temperature is going to change similarly, in environment if there is more and more moisture, coefficient of friction will change, because of the thickness of the layer.

To avoid that kind of uncertainties, we can think about solid lubricants, we can coed these surfaces with solid lubricants, one of the very popular solid lubricants used for the solid, for the gears is a phosphate. So, that, we use a phosphate layers on a solids as a solid surface solid lubricant with a gear surface. Now, there is our relation this shows if thickness, if the shear strength of coating is a lower, then coefficient of friction will be lesser. Naturally, when we are talking about the shear strength of the layer, it is a relative term, is the relative to parent material tau y is a parent material or we say there are gear and pinion of the they are made of the same material, then we are talking we can take any tau y, but the gear pinion is softer material compared to the gear or the hardness is a loser lesser or shear strength is lesser then tau y will be the pinion material layer, we say the shear strength of pinion material.

Now, this tau y is interface layer whatever you say interface layer meet, if we are using solid lubricant naturally the shear strength of the solid lubricant will come, if this ratio is the large we say may be tau y is 99 times of tau i. In that case, it will be very huge number and coefficient of friction will decrease drastically, but if we say now tau y is just 20 percent higher than tau i naturally this factor will be 1.44 minus 1 so square root of this and that will factor still will be very high the coefficient of friction will be high. So, depends what is this ratio, if tau i is much lower than tau y coefficient of friction will be much lesser, and we do think solid lubricant from that point of view.

And this is diagram shows that, how if we kept, if we keep gear surface in environment there is a possibility of some contaminated layer on top of surface, the contaminant layer is a oxide layer some sort of a work hardening layer and they are (()). And what we discuss is that, instead of relying on this layers which will change their thickness as per the environment we better design lubricant layer as per the requirement, we think about the phosphate layer, we can think about the graphite layer, we can think about the molybdenum disulphide layer. And whatever is required, we can design accordingly most often these solid lubricants are we used in a liquid lubricant.

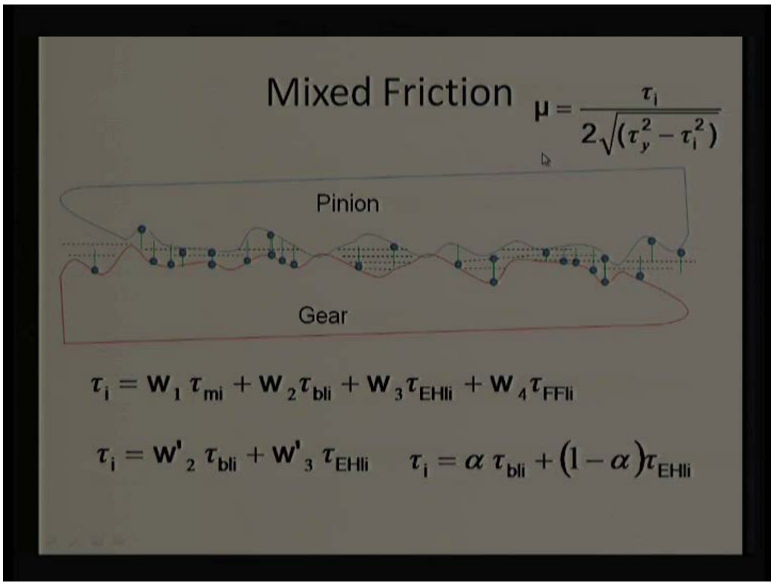
We know very well, if there is a solid lubricant deposited on the surface what will happen, under rubbing condition that thickness layer will slowly reduce it will not remain same or in other word, we are going to avoid the wear of gear material, but we cannot avoid the wear of solid material or solid lubricant material; if we are using Teflon, naturally Teflon powder form will come out or we say will be born out the from the coating surface.

Similarly, if we are using the molybdenum disulphide and there has been condition naturally molybdenum disulphide will be simply slowly slowly removed from the surface, same thing with the graphite sulphide also, so in gears particularly we cannot avoid the wear, we can only divert the wear from the gear surface or gear material to the solid lubricant material. We are not avoiding, we are not making it 0, where wear will be there, but just we are changing the wear from gear material, so that it can be diverted to the solid lubricant material.

Now, if there is a liquid lubricant what will the liquid lubricant do it can carry those powder from along with it and it can be again again a replenish or can be use again and again. So, there will be always some constant thickness layer depends on equilibrium, physical equilibrium, chemical equilibrium there will be layer and that is advantages.

Often we say 1 to 2 micron thin solid lubricant layer is sufficient, you do not require more than that, that is a more then sufficient but, we know very well there will be some sort of wear out obvious lubricant layer and we do not want any time the lubricant layer turn out to be 0, so we generally keep few micron it can be 10 micron, 15 micron depends on the micron of the components and depends whether we are going to use liquid lubricant along with along with the solid lubricant on that. If you are not using any liquid lubricant naturally we need to keep slightly thicker layer, so that component can be refurnish after certain number of hours or certain number of operating hours.

(Refer Slide Time: 12:26)



So, this is a what a mix lubricant was more like a dry lubrication with a solid lubricant, if I say more appropriate word solid lubricant also can be treated as a boundary lubricants they are making a boundary layer. But, what we are talking when we use a liquid lubricant along with the solid lubricant, then situation will turn out to be mix lubrication what is the reason for that, here the load carrying capacity will be shared by liquid lubricant, solid lubricant, metal to metal contact and full film lubricants. So, all full all four are mean of the lubrication, all four mechanism of lubricant will be there or we can say this is mu coefficient of friction we are using the same relation, which we have we shown in previous slide, but this tau I can be calculated as a weighted summation of different shear strength.

So, shear strength or metal interface shear (()) strength or bond lubricant interface, shearing strength of elastohydrodynamic lubrication mechanism, shearing strength of fluid from lubrication. So, all lubrication mechanisms are present when we talk about the mix lubrication that is a name, we say that there will be solid lubrication and there will be liquid lubrication, so naturally it will going to be a mixture of that and we talk about the solid lubricant we know that there will be metal to metal interface, there is a possibility of boundary layer interface. When we talk about fluid from lubrication, we can talk about the elastohydrodynamic lubrication we talk about the full film lubrication with complete surfaces are separated by liquid lubricant and sufficient thick lubricant layer is there when there is a no deformation, there is no elastic deformation of the surfaces.

So, there is a possibility and when when we talk about the metal surface and we talk about the designing of those metallic surfaces particularly for the gears, naturally we will not tolerate this metal interface, we do not want any time gear material comes in a contact with other gear material, so this shear strength can be avoided. And of course, (()) use a W 1, W 2, W 3, W 4 summation of this W 1 plus W 2 plus W 3 plus W 4 will be equal to 1, we say that these are we are using in a normalization form, we say that these are the weighted these are the weights, should be given along with individual shear strength. Discussing about we cannot tolerate metal metal contact, so we try to avoid as far as possible in all situations, so this parameter may not come when we talking about the frictional force for gear pair this will not come similarly, this will not also come we cannot tolerate the full film lubrication.

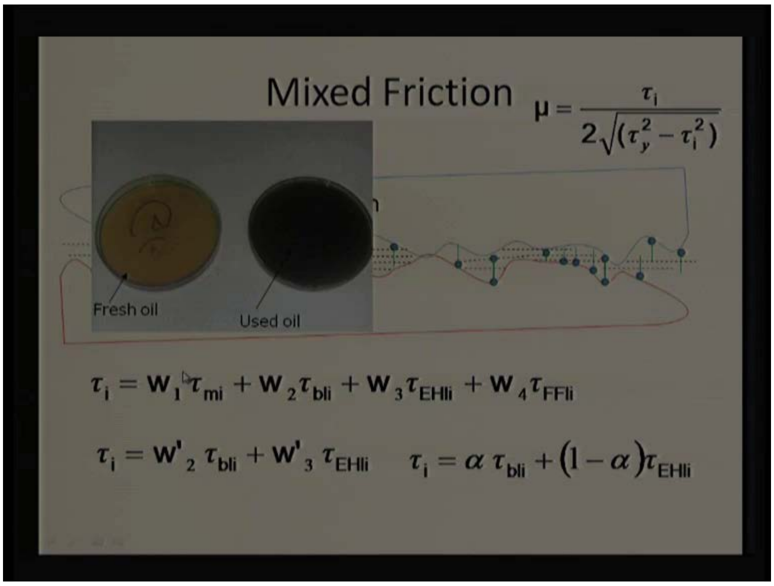
We cannot transfer the motion 100 percent without any we say that loss of the velocity using the F F l that is why this will not work this is not going to work only two terms we are going to work with bond lubrication and we are going to work with elastohydrodynamic lubrications or no word pressure or contact load will be immense in this situation that is going to cause a deformation deformation of surface, deformation of tooth surface and we will be using some sort of boundary additives, it will not be the only the liquid lubricant but, we need to use the boundary lubricants. So that, wear of the surface is minimum when there is extends to the elastohydrodynamic lubrication (()) even we say that there is a mechanism like that, they still the surface (()) come in a contact.

And to avoid that metal to metal surface we need to use boundary additives and this is a what we say, now if I you say that tau m i is not going to work tau F F l is not going to work, naturally this will be 0, this will be 0 and we have only two mechanism that is a bonded lubrication mechanism and elastohydrodynamic lubrication mechanism. Now, this two mechanisms (()) naturally this weights will be different now, initially may be we say this was a weight was the some value 0.1, 0.2, 0.3 and this value has a something like 0.1, 0.2, 0.3.

When we talking about the gear lubrication mechanism naturally this W 1 is 0, W 4 is 0 that is why we are not taking, we are not accounting tau m i and tau F l naturally this two values will differ. Now, there is a W prime 2, W prime 3 and this summation will be equal to 1 are in other way, so W prime 2 plus W prime 3 will be equal to 1.0, in other word if I say alpha is used as a weight weight factor, naturally other factor will be 1 minus alpha, so this is a proper shear in strength formula which can be directly substitute here and we can figure out what will be the relative coefficient of friction.

We are not talking about the absolute coefficient of friction, whenever we compare this is going to be good competitor, but if we talk about the absolute coefficient of friction it depends on many parameters, many factors and we cannot really estimate mathematically what will be exactly coefficient of friction on the interface, we need to do extends experiments for that. But, if we want to compare different material pair, if we want to compare different harnesses, if we want to compare different look lubricant, if we want to compare different profiles yes, this kind of relation can be used.

(Refer Slide Time: 18:31)

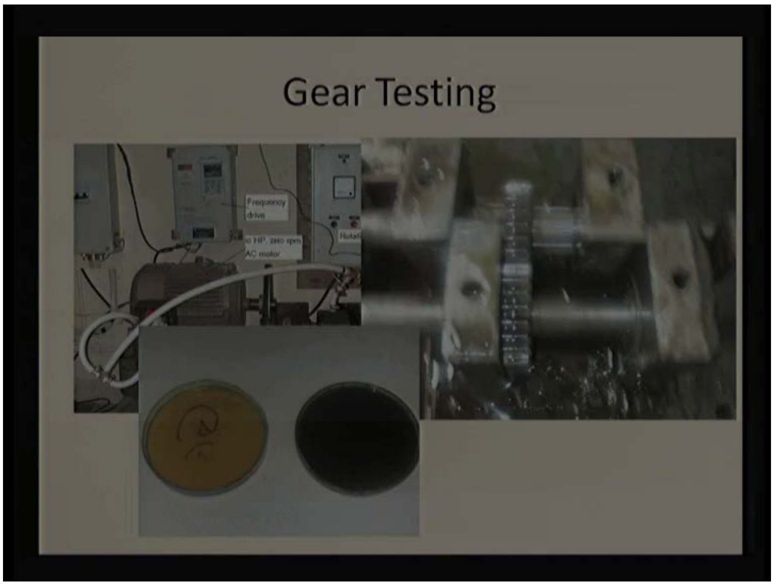


So, this is a relative parameter or we say this can be used as a competitor, but cannot be treated as a absolute value and this is what we are showing of figure (()) where the image of a liquid lubricant, I mean shown or we say that photograph of liquid lubricant may shown. What we did experiments on gear pair in our lab and what we found, when we were using initially this kind of the fresh oil, yellow oil and good oil (()) it shows that it is a liquid oil after when we say you doing a 100 hours operation, when we remove the oil from gear box we found this kind of a colour, this is a completely black and we could not find separate particles, but this liquid is filled with all the wear particles.

Now, this is still surface damage happening and we can say that is a clear indication that whenever we operate gear pair, they are going to work with a mix lubrication mechanism, they are not going to work with a full film lubrication they are not only going to work with E H l.

So, this two lubrication mechanisms similar, may be last one can be rejected this can be considered but, there will be boundary lubrication and if there is no sufficient boundary additives available, then metal interface will occur, but we now we are more careful about the gear pair. We know manufacturing cause on a higher side and they are very sophisticated components, they are very important components naturally will be using some sort of boundary additives, and whatever the boundary layer is made will be it is made with a chemicals or with the chemist chemical layer is made of the boundary lubricant layer is not on the solid lubricant then this kind of a lubricant, this kind of colour of the liquid lubricant will emerge that is obvious. So, even the chemistry if the happening or we say that chemical action is a happening or when we are talking about some sort of chemical layer on the gear pair, naturally some metal is going to act along with the oxide are getting oxidized and when it is removed naturally it is going to make a colour like this.

(Refer Slide Time: 20:57)

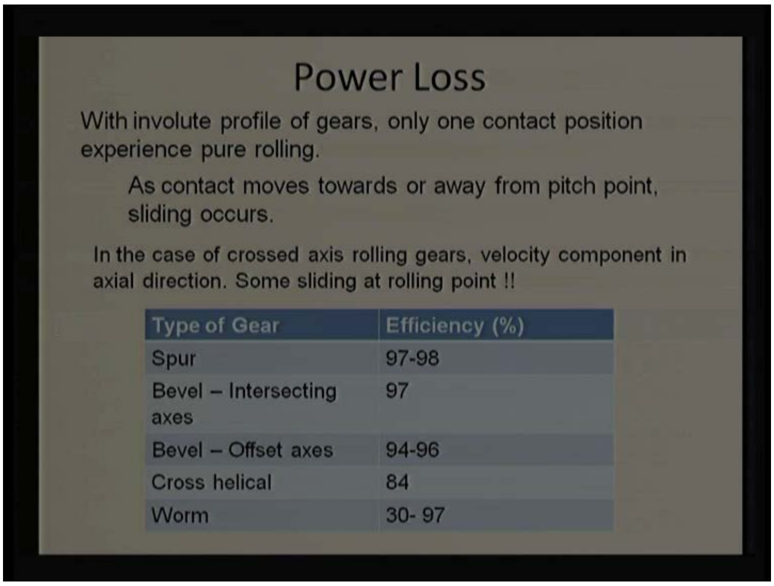


So, that is important to consider, what I was talking about what we had done in the lab, so this is a set up which is just demonstrating the, what we did is this is a 10 hp motor this is a belt for drive and this is a gear box. Naturally gear box need to be loaded that is why there are it is connected with a spinal coupling and this is (()) and that is a dynamometer.

This dynamometer was made in house, we could apply up to 20 Newton meter torque but, completely controlled condition when we operated for the 100 hours what we got, when we opened the gear box what we are getting is completely filled with a black colour oil and even this particle you can see that the kind of the colour of the oil here. If we use a fresh oil it will not be this kind of thing, it is generally yellow colour oil but, here it is turning out to be slightly blackish and these are the gear teeth, this is also having some sort of the black oil on the surfaces of gear teeth and this a what I (()) that it is a accumulation of some ware particle in this oil or it is completely (()) on this particles.

So, when we do experiments we 100 percent know that, it is not going to be a full film lubrication mechanisms and we are using liquid lubricant naturally it is, if there is a wear where will happen, because either directly removal of material or some oxidation oxidant oxidants of that matter.

(Refer Slide Time: 22:37)



Now, that clearly indicates when you talk about gear pair there will be wear of a surface, but this wear can be reduced when we talk about the power loss or when we talk about the friction loss, we can say the two surfaces they come in a contact they need to have a lesser friction resistant, but there is a problem we require intimate counting between two surfaces, so that relative velocity is 0. I would say relative velocity is 0 on that is a what we say the efficiency, we want complete velocity to be transferred from one surface to the other surface and that is basically with the friction mechanism, because we have used a invalid profile or the positive drive profile for that, but still that it is a problem.

We want minimum friction and we want good friction also, so we come out in the some sort of trade of, when you talk about the friction force and we know very well that friction force is going to work between the gear teeth, but we cannot term a as it is reason being that, there is velocity which is remain does not remain constant or we say the sliding velocities does not remain constant is going to be constant or a 0 sliding velocity will be 0 at the only on the pitch point.

Away from the pitch point velocity is going to change and if of course, in some gear pair even the there will be sliding at a pitch point, which we have studied in previous lecture and I am just repeating this slide with a few observation from the previous lecture, while we say that with invalid profile of the gears. Only one contact position experiences pure load, pure loading happening only at the one point, now as contact moves towards or away from the pitch point sliding is going to occur or sliding is a bound, so that sliding will continuously increase, if we move away from the pitch point.

Then another example was k 1, we say in the case of crossed axis rolling gears where the axis are not parallel, they are not intersecting, they are not coplanar and they are out of the plane, then surely there will be a some component in the third plane or third direction also which is going to create additional sliding. That is why you see, in the case of the cross access rolling gears velocity companion in axial direction will occur and that is going to increase the sliding, over all sliding will increase in this situations, that is why when we try to compute different gear pair or we say that different gears, what we get, so it type of the gears, efficiency this is stable choose that how efficiency of what kind of efficiency we can achieve.

Simplest way to manufacture, easiest to analyze is spur gear, we say that shaft axis are parallel and length of was a bearing gear width is also parallel to the axis; efficiency is pretty high it is 97 to 98 percent. Of course, we are talking about the good condition, we are not talking that we are using, that we are not going use any oil or something like that we are talking about a good conditions, all good conditions; this spur gear shows 97 to 98 percent efficiency in good conditions. Coming to the bevel side, is a in previous lecture we mention there is a spur gear works on cylinder as a cylinder, while bevel gear work as a cone on cone they role on each other.

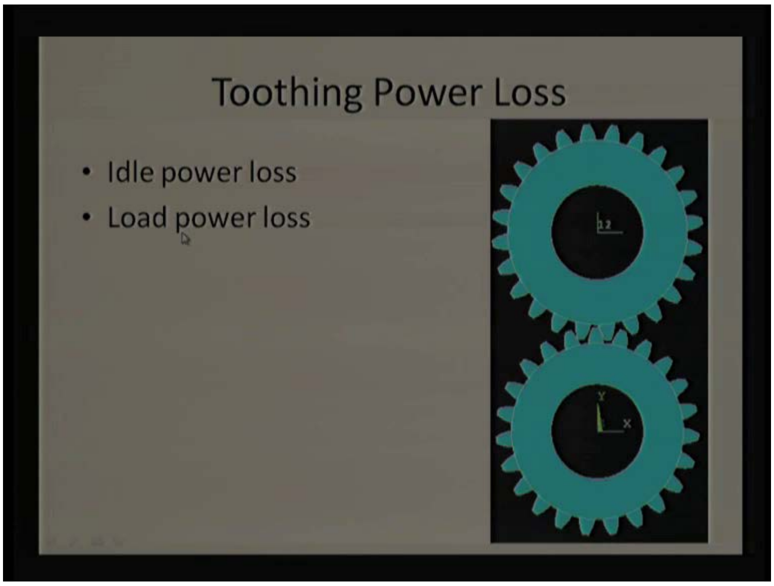
So, bevel gear when axis are intersecting may be 90 degree, 60 degree, 70 degree depends on kind of geometry, they show again an ideal condition, good operating condition efficiency of the 97 percent. However, if we offset it gear axis, there is some gap efficiency is slightly lower than 94 to 96 percent, we say that offset is very minute then efficiency will be high, offset on greater side then efficiency will be 94 percent.

Now, these are the all parallel axis related options when talk about the cross helical that the coplanar the surfaces are not there or the axis are not in a one plane, then efficiency because you can see is reducing drastically. Helical gears otherwise are good non gears or they work with a lesser noise, much better performance, when when we are talking about the cross, crossing of that or we say that instead of line contact they have point contact then efficiency is dropping from may be say 100 percent to 84 percent that is a substantial change.

Well when we work with a one gear, the torque ratio is almost 42 worm it is very very high torque ratio, but sliding also will be very very high in those situations, so efficiency goes down in this situations, when the sliding is very very high naturally efficiency is just 30 percent, when we are able to control it properly relative low sliding velocity the efficiency can reached with 97 percent, but of course, these are the very exceptional cases generally efficiency of one gear is a lesser then 90 percent.

So, this gives an idea that, this are having lesser efficiencies we say the efficiency having lesser than 100 percent, naturally there will be some sort of power loss and we can talk about the frictional loss and multiply with the (()) velocity but, that is not in our case or will be very very simple case. Here, the sliding velocity is going to change continuously as we move away from the pitch point and it always happened in one contact we say tooth contact, only one point will be at the pitch point all other points may be say there are 200 points, 300 points whatever the way we divide it, there will be different sliding speed and they where continuous point, there is change in the sliding velocity naturally we can have to talk in terms of power loss, we cannot talk just in terms of friction losses.

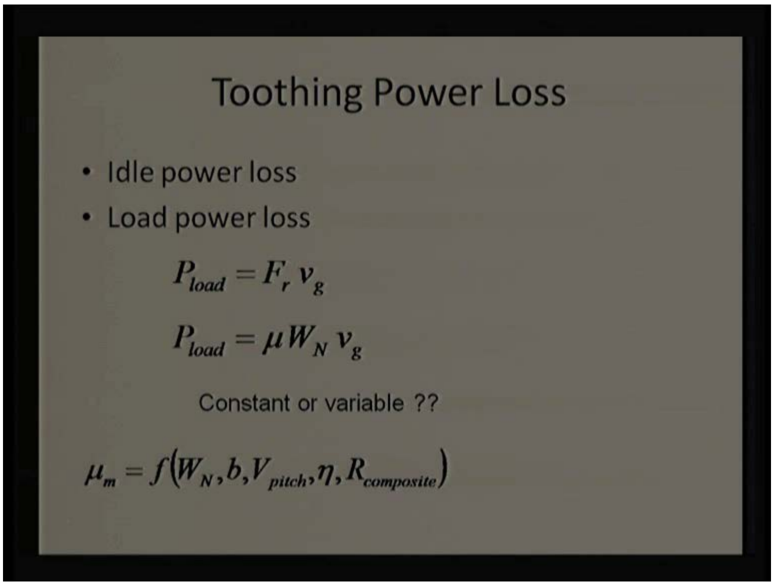
(Refer Slide Time: 29:08)



So, we say the one of the major factor in power loss when we talk about the gear box naturally there will be shaft, there will be bearings, and all that when we just talk about the gear pair, only the contact is happening at the tooth naturally there will be toothing power loss, however if we talk about the these surfaces, this tooth surfaces they are not under load. So, power loss due to the load will happen over here only, while power loss without load will happen on number of other teethes, as there will be some sort of dragging action this is the solid metal and if it is dipped in the liquid lubricant naturally there is a splashing will occur; and will try to splash lubricant out, you can see there is a discontinuity surface, if there is smooth surface then there will not be problem, but this surface, because of the massive requirement discontinuity.

Naturally, there will be more and more splashing loss, more and more this I can say idle power loss, we are not talking about the load transmitted through this systems or we can be talking about the churning effect, it is going to churn the like a lubricant or the or going to cause loss of the load. If viscosity of the liquid lubricant is as higher side naturally this will churning noise will also be on the higher side, so we have a two power loss we will talk about the toothing power loss is an idle power loss, when the load is not there and there will be load power loss. And naturally when we talk about the (()) load we talk about the elastohydrodynamic lubrication, we talk about the mix lubrication actually this component this is going to play major role.

(Refer Slide Time: 30:57)



And this can be given as F r frictional force into velocity into sliding velocity, to (()) precise side we say frictional force should be presented in terms of a normal load, and that is a mu (()) normal load is a mu W N, but the coefficient of friction is a mu over here, so it is a mu W N into v g this is going to change. Question comes, when mu be changing really or is going to be constant, will W N change or will remain constant, so these this two are important aspects are invite if you (()) my words mu cannot be constant.

We know loading points are going to change applied load is going to be change it may vary from 0 to the (()) Newton load as a load is such high variation on the point of contact; naturally mu is going to vary continuously. Talking with W N again the point of contact is continuously changing or line of contact is continuously changing, naturally this W N will not remain fix is also going to change.

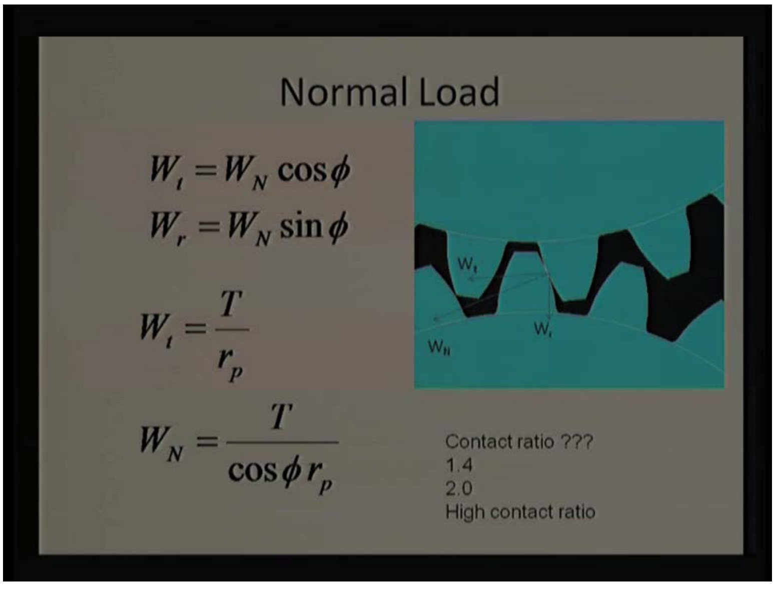
So, mu is going to change, W N is going to change and v g is also going to change naturally, we are talking about the power loss, which the average power loss and mean power loss and that is why we say that, these parameters are not constant they are going to vary continuously in magnitude in may be direction also. And interesting thing is that mu, itself is a function of number of parameters, that is why, is it in a mu m this is a mean value, average value of coefficient of friction it is the function of itself load is going to depend on the load itself, it is going to depend on the bearing or we say the gear length.

If the length is increasing there shall be this loss will be high or a mu will be higher side, is going to depend on a pitch velocities or summation of pitch velocities, is going to depend on the liquid lubricant viscosity (()) fluid viscosity and it is going to be a function of surface roughness a larger the surface roughness, larger the value of mu.

So, this is difficult to find out through the signs has been shown, if I take a constant value, if I change any of this parameter mu is going to change. Of course, sensitivity of the parameters is different, talking about this R the surface roughness; it is a composite surface roughness. Naturally, the gear frank surface roughness will be a content, if a gear frank or a pinion as well as the gear frank of the gear or say the both the both the surfaces will surface roughness of both the surfaces will be counted, and generally it is a RMS value which is be counted. So, we now mu of the mu even the mean value is going to be function of so many parameters or many times is a difficult to prove through the signs.

But, when do number of experiments we can find out using (()) equation we substitute we use this parameters 3, 4 level parameters do experiments and find out what will be the mu. So, it is going to be decided based on experiments or (()) equation with a some unknown constant which need to be determined through experiments.

(Refer Slide Time: 34:39)



Now, what we talking about the what we use a word a normal word W N, question comes what is this W N, see this is a gear pair now any surface perpendicular to that, that should give away normal load something like this, this is a W N question comes how to find out from where this figure will turn out, what is mechanism? Ya, we have a some sort of a pressure angle and we know pressure angle if I account a pressure angle I will be getting two components, we say W t (()) components which is going to drive the gear pair, and W r that is a radial load is going to push gear away from other gear or is going to be analyzing bearing, so there will be radial load on the bearing.

So, how to find out this W t and W r, that is another question, when we (()) we generally know what is a, what power to be transmitted when we know power to be transmitted and that to in a rotational direction; so we can figure out this however, before that we need to find out relation between W N, W t and W r and the see that W t will be W N into cos phi. Actually W N is going to be greater than W t and W r or we can say we can take square root of this is vector summation of W t and W r we can find out W n, so this W n will be maximum load, and W r which is there need to be minimized, so that bearing is less loaded or subjected to less load.

Now, we talk how to figure out W t or we say that any of this W once I know that any of w and I, if I know the phi value of course, this phi again will change the pitch circle diameter generally is this known, but away from the pitch circle diameter is generally not known we need to find out from the geometry.

Now, this W t can be given in terms of torque by pitch circle, we say the radius, pitch circle radius for pinion will be r p similarly, for gear it will be r g and generally r g is a lesser, r g is a more than r p that is why we are using this relation (()), we want on more on conservative side and that is why we are taking higher value of W t. So, W t is a torque, torque will remain same in the both the cases or and not remain the same it will be power will remain same, the torque will remain the same, so torque whatever torque on the pinion and divide by r p that is going to give us a relation. Now, once we know this we can figure what will be the W N and that is turning out to be W N is a torque divide the r p in divide by cos phi.

So, once we know W N, we should be able to find out the friction losses, the question comes, we are trying to find out this W N through using this relation, but am I doing right job, is it any possibility the only one pair in a contact and we know there will be some sort of clearance, if this gear tooth go in this spacing to avoid the jamming we keep some extra clearance; naturally that means when the one tooth pair engagement is over there will be some clearance and that is going to give a jerk.

Naturally, we need to design gear pair in a such a manner before loosing first contact completely other pair should start contacting it, than only the smooth transition of the movement will occur, through smooth transition of the power will occur from one shaft to other surface shaft power can be transmitted when there are more than one pair of the teeth in a contact.

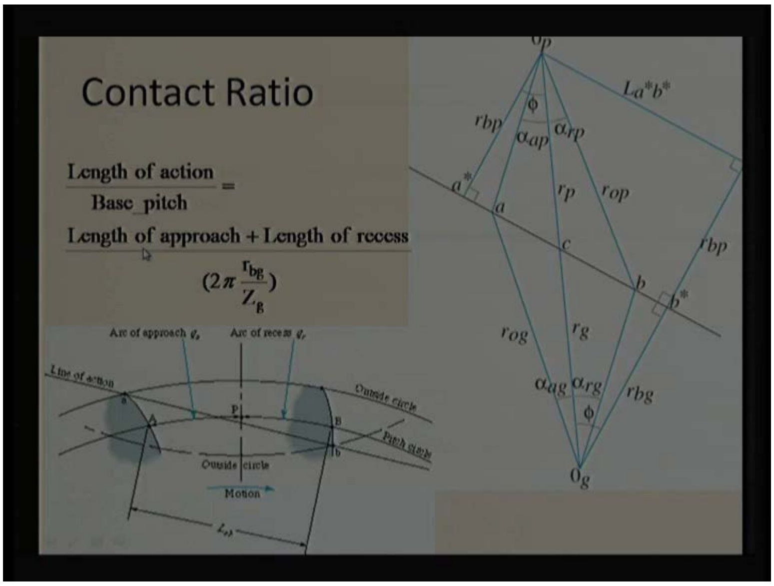
May be the pitch point there may be only one pair in a contact but, away from the pitch point may be just a start or the just end of that naturally there need to be more than one or two pair minimum to be in a contact, so that we can get a transmission. When it happens, when it happens naturally, this W N calculation is not going to be right, because there will be some sort of load sharing and if there is a load sharing we should account that load sharing effect.

And what we say, we should use a contact ratio and contact ratio is generally more than one to be designed and more than one means that is a load sharing at the extreme points at the pitch point yes maybe same load, but at the extreme points there will be different load. And we need to evaluate the power loss, we need to find out the load at a different points, because the sliding velocity also going to change, so we if you want to find out the power loss at the different points, we require force normal load at those points as well as, we required a sliding speed and that is we required all this calculations.

Now, here it is written contact ratio, so many questions marks, how to find out this contact ratio that may be the bigger expression, we will try to figure out and but, this value mostly first per gear sigma ranges from 1.4 to 1.7, so may be assume that lowest value should be 1.4 whenever we decide, it should be more than 1.4. When we talk about the helical gears the contact ratio is generally on a higher side that is 2.0, that is a reason that is a reason why helical gears works smoothly compared to spur gear. The engagement is gradual and almost of the time they two pairs are sharing the load and that is advantage.

Because, of this advantage many times people go had with spur gear geometry, because of a less complexity, but they try to make a high contact ratio gears they change profile to some manner, we say that they add addendum and dedendum a larger (()) addendum and dedendum, so that gear is modified to give us high contact ratio, contact is a greater than, contact ratio is greater than 2.0 in those situations. And generally that gives a smooth operations, that a give us the lesser load on per gear, gear teeth or gear tooth on the surface particularly for pinion and gear, now question comes how to figure out, how to estimate this contact ratio, so that can be done mathematically.

(Refer Slide Time: 41:35)



For first, we can give some sort of description, so you assume this is one sketch and this is a one point, this is another point, we say the beginning of point and the end of the point, and this arc this arc can be known as the angle of approach, when the gear teeth is start contacting, reaching to the pitch point and after that angle of recess, if their try to depart from that pair right. So, there is a angle of approach, angle of recess, if we know this angle or we say can we can estimate what is a arc length arc length and when they are coming under contact, and if we are able to estimate we will be try to, we will based on that we can figure out what will be the contact ratio.

Another word, we say contact ratio can be given as a length of action, the whole length of action and when when they are coming in a mechanical contact assume that whatever the curve which we have shown over here, if we know the length and (()) this is in polar coordinates, we can transfer to the retina coordinates, we say effetely, we can say that this curve if it is a developed in a straight line, that will give me a length of action. And divide by the base pitch, that the base from where it is started from where the gear teeth is started that goes, if that pitch is known to us we will be able to find out the contact ratio or we can say contact ratio is defined as a ratio of the length of action divided by base pitch; and this length of action can be divided in two parts, so length of approach and the length of recess and base pitch can be figured out by finding the (()) of pitch circle divide by number of teeth.

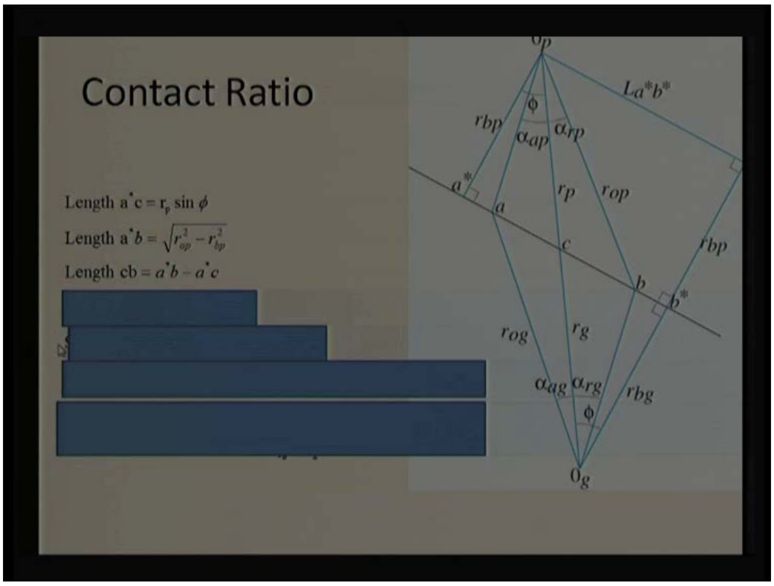
So, Z g here is a number of teeth, r b g is a base radius of the gear, so we are talking about the gear pair over here may be the we can think about the pinion also but, here it is a being used about gear f g. So, base pitch sorry, radius of pitch circle is r v g number of teeth on the gear is a Z g, and we are talking about the circumference we say that 2 pi r b g is a given circumference of the pitch circle divide by number of gear teeth; that is going to give as a base pitch, and that once we know the base pitch, we know the length of the approach length of recess, we can figure out what will be the contact ratio which is the important to evaluate what will be the normal load on per teeth or per tooth right.

So, to be on a surplus (()) we can draw this line sketch say assume this o p as a center of pinion o g as a center of k, now point c is a pitch point contact, so there is a straight line in this case, we know length of action will be at a some angle with this and this is a starting point I mean to say that a star is a point where the contact should start and b star is a point where the contact should end.

But, we know there will be lot of discontinuity, so what we do we provide some sort of flit at the start of contact and end of the contact, so the excess hysteresis are avoiding, that is why in reality contact will starts at the a, and ends at the b, it is not a star, it is not a b star, so actual contact is starts after the this base circle, it is not at the base circle point. We use some sort of flitting them, so this is a and end also, because of tooth we generally do not give very sharp edge, we give some sort of flit there, so truth also we will be departing what is the separation will occur before a b star, and that is assuming that b, so this is a to b is length of action and we need to find out this length itself exactly.

We already know there will be 2 pi r b g divide by z g, this this is generally geometry parameter known to us before operation, but this action is can be calculated based on the operating parameters, so we say we need to figure out what is the length of the approach, what is the length of recess and once we know that we can be able to will be able to find out what will be the contact ratio right.

(Refer Slide Time: 45:59)

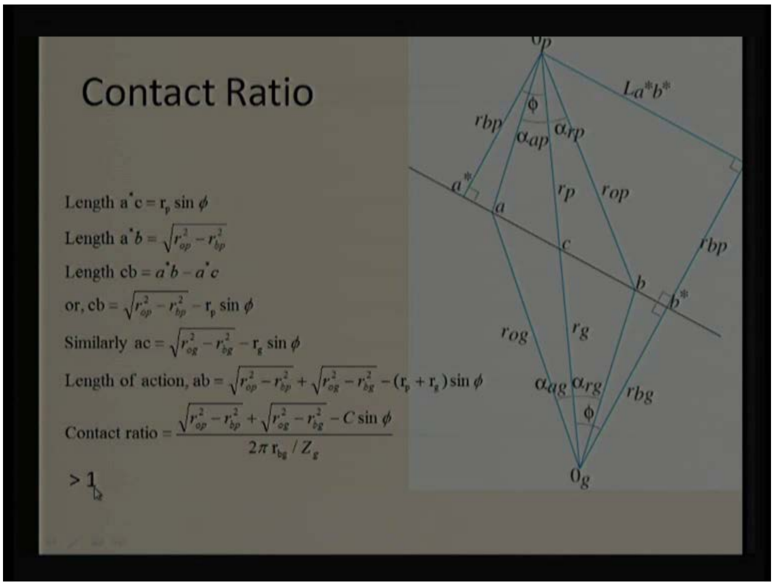


Now, to find out the contact ratio we can start finding the some (()) some lengths we say a star c, the length a star c can be figured out can be given in terms of r p, we know this is a right angle triangle, and r p is generally parameter known to us at the start particularly which from the manufacturing point of view, so r p is if it is known and this is a sin phi, this is a phi we know the, this attitude angle, so we take that and on the try the a c can be given (()) the a star c can be given in terms of r p and sin phi that is given over here.

Now, we can figure out this length also a star b again there is a right angle triangle, a star b can be figured out using this relation, we can directly we can take this r o p also in terms of that, and we can find out this using this relations. So, this a r o p given this is a length of o p b, that is a r o p is a outside diameter of the gear, if that is known and this is base radius is known to us by subtracting this we can figure out what will be the a stars b, so when this that is why says the length a star b is a square root of r o p square minus r b p square.

Now, what is interested to interest of our interest is a c length and c b length, we know we will first case we figure out what is a a star c and second case we figure out what is a a star b. Now, if we subtract this length from the complete length, naturally c b will be known to us, and that is given over here, the length c b is equal to a star b complete a star b minus a star c.

(Refer Slide Time: 47:58)



We can substitute this values, this is (()) in mathematical term c b can be given as a square root of r o p square minus r b square minus r p, this is a pitch circle radius (()) to sin phi, sin phi in the sin phi is known to us from a geometry point of view from manufacturing point of view r p will be known to us from manufacturing point of view this also will be known, so that I will be we will be able to figure out what will be c b.

Now, we do the reverse order we can figure out what will be the a c or we can start from here we can find out what will be the a c and that is the why we say this similarly, a c can be figured out from in terms of a r o g here what we figure out in terms of a pinion, here we are trying to figure out this value in terms of gear. So, r o g square minus r b g square both of the pinion is a outer diameter of pinion sorry, outer of diameter of gear this is a base diameter of gear. I will say radius in terms of radius this is a radius of outer radius of a gear and then base radius of gear minus again pitch circle diameter we are using, so pitch radius into sin phi assuming this the same pressure angle for the both the surfaces.

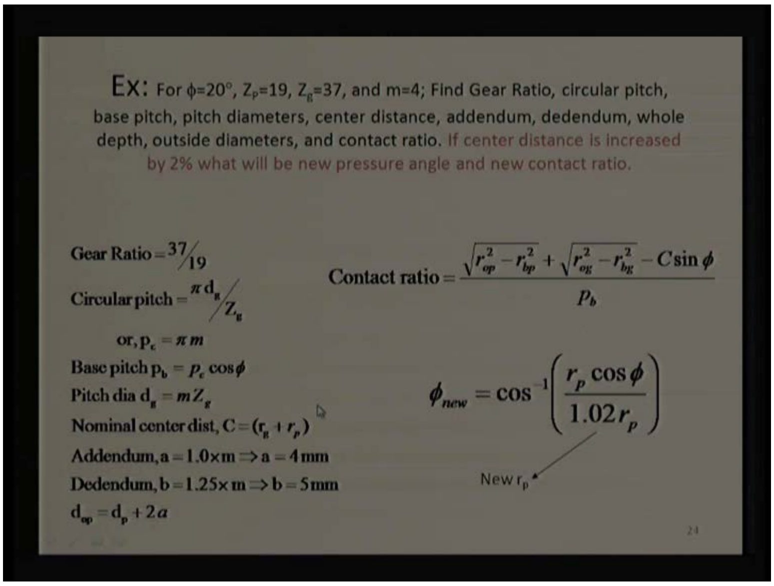
Using this what we say overall length will be c b plus a c and that is a what we say a b, the c is common and this will be turn out to be the first parameters related to the pinions second parameter related to gear, and this will be summation. Now, we can use through the find out the contact ratio, so contact ratio can be given relation that is a numerator this is a denominator.

Once we know, what is the center distance c is a center distance the summation of r p plus r g that once we know this values, we can figure out what will be the contact ratio based on the contact ratio, the contact ratio is one naturally we know that the friction losses will be tremendous, but contact ratio is a lesser the friction losses will come down, because W N will come down or will be reduced.

And it has been shown that contact ratio over the period of time will continuously decrease, initially when the gears are new contact ratio will be larger as if progresses or the wear out starts, then this contact ratio is going to decrease continuously that is why we say that spur gear, if I want 1.4 as a optimum value I should be able to keep it more slightly on a higher side, otherwise there will be some problem later may be say after 10000 hours or 15000 hours of operation, the gear will wear out and it start giving much lesser contact ratio that lesser contact ratio is going to reduce the stiffness of the system and the vibration will start and we say that there will be some sort of jack motion, there will be some sort of impact motion and the noise will start.

To illustrate this I have an example, I will just try to elaborate that and what this indication over here, contact ratio should be more than one always whenever we define.

(Refer Slide Time: 50:59)



Now, just for the elaboration, I am just giving one example, how to use this parameter here the manufacturing data are given that is phi is equal to 20 degree, number of teeth on a pinion is 19 number of teeth on a gear are 37 module is 4, so we find gear ratio the simple Z g divide by z p, circular pitch to be figure out base pitch is diameter center distance over the addendum these are standard terms, addendum whole depth outside diameter and contact ratio.

In addition this is given, if the center distance is increased by 2 percent what will be new pressure angle new contact ratio, so we say contact ratio is given by this relation which was shown in previous slide, and gear which will symbol number of gear and number of teeth on gear divide by number of teeth on pinion circular pitch. This is the 2 pi r g divide by number of teeth and when we want to find out the base pitch, that it will be multiplied by cos phi that is why this P b can be correctly substitute over here.

Whenever we have a module and the number of teeth we can find out what will be the pitch diameter it is just simple multiplication, module into number of teeth is going to give us the pitch diameter, and that is what is been given the pitch diameter of the gear is m Z g.

Similarly, pitch diameter of pinion will be m into Z p and once we know the pitch diameter we can figure out what will be the radius, pitch radius, summation of pitch radius will be equal to center distance, and we are using the word nominal pitch center distance, nominal center distance reason being that, this values are hypothetical may change during assembling. Addendum and dedendum they are generally manufactured with (()) and we say that standard thing is that there will be one m dimension above the pitch circle pitch radius, and below distance between the base circle and pitch circle at anytime thickness between this two will be 1.25 times of module.

So, a is given, b is given to us we can figure out all the parameters and we can figure out what will be the contact ratio, now if we find the contact ratio it will turn out to be say 1.62, when the second part in this case is given that it is center distance increased by 2 percent, if a center distance is increased by 2 percent then the c is increased by 2 percent and that is being given the base pitch also will be will remain same, but r p will change and that will increase 2 percent, and that is going to give us a new angle, new pressure angle, so this new pressure angle will be slightly greater than 20 degree.

What we will do will in a next lecture, we will continue with this example I will just repeat the slide and elaborate it and ensure some sort of the calculation with this kind of example, and will continue with a friction and lubrication of the gears thank you, thanks for your attention.