Tribology

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Lecture No. # 32

Friction of Rolling Element Bearing

Welcome to thirty second lecture of video course on Tribology. Today's topic is very interesting. What we are going to discuss, the friction of rolling element bearing. Why do I say it is interesting? I will describe on a second slide or the next slide. In previous lecture we discuss about, how to select a proper bearing based on load consideration, based on number of rotation concentrations. What is the expected life is it expect a life with 30,000 hours, 40,000 hours, 50,000 hours we can select bearing accordingly.

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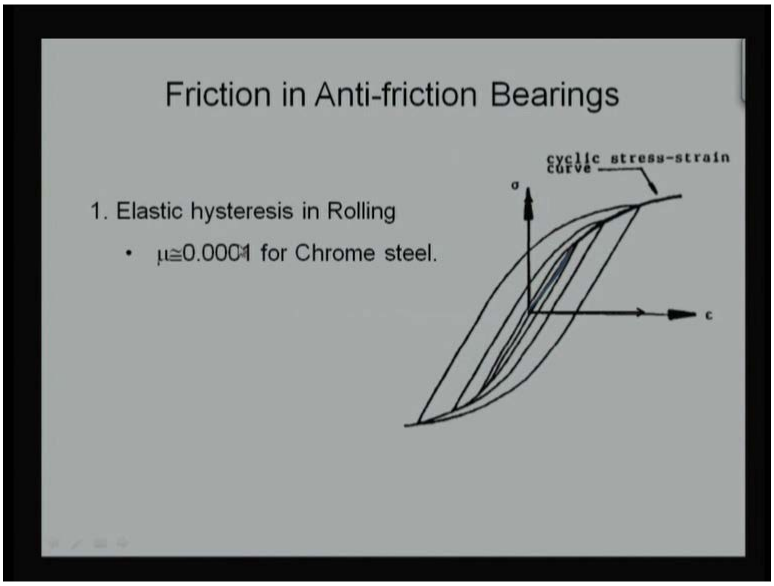

When come to the interesting part. You say that friction in anti friction bearing. Name itself sounds as a quiet interesting thing. We are talking about anti friction bearing. What is the meaning of that is, that there is no friction or negative friction, a bearing which can resist or show there will not be any friction in bearings. Just opposite of the friction and we are going to discuss about the friction in that anti friction bearing. In number of books it has been referred and defection bearing that is why I am again and again I am using the word the anti friction bearing, but in true sense, Tribology sense these bearings are not anti friction bearings they are rolling element bearings show lesser friction compared to sliding friction; particularly they start condition not even in running condition. Sliding bearing show much better performance during running condition if there is a proper lubrication

However, we call antifriction bearing a rolling element bearing as an anti friction bearing. The reason behind is these bearing show more or less same co efficient of friction and a steady condition and kinetic conditions. That not much variation in co efficient of friction during running condition. So, is much more stable, it shows much better performance, when I think from a power consumption point of view, whatever the power consumption may be one thousand RPMs slightly more than that. By and large it will remain same or much more stable.

Now, what are the causes of friction in rolling element bearing? One of this cause we can think about is hysteresis losses. The hysteresis loss is generally happen when there is loading and unloading of any mechanical element and that is shown over here in this case. You can see there is a graph in epsilon and sigma. Sigmais representing the stress, sigma is representing the strain and there is a general way to find how much deflection will occur during the loading condition and unloading condition. What is been shown here as a stress level is increasing, strain is also increasing initially, to some extent there is some sort of non-linearity in this. But when it is unloaded I am just showing this line when the first the loading is happening there is a no deflection for certain duration. And I mean certain value of sigma after this getting loaded and deflection is starts when a strain is starts. It reaches to max value after that we are reducing the stress.

When you are reducing the stress we are able to find out the some sort of the loop. It is not a perfectly elastic or we say the some energy is getting lost. If the loop area is 0 then energy conservation happens or we say that 100 percent energy is returning back to the system. But whenever there is a loop like this that means, there is a some consumption of energy and some consumption of energy means there is a some sort of friction in mechanical elements. That is why we say there is a elastic hysteresis in rolling element bearings, when the one rolling element is getting placed against the other rolling element or it is getting placed against resist. Interesting thing is that the co efficient of friction particularly for a elastic hysteresis is very low, particularly for some good material. We are discussing about the common steel which is being commonly used in ball bearings.

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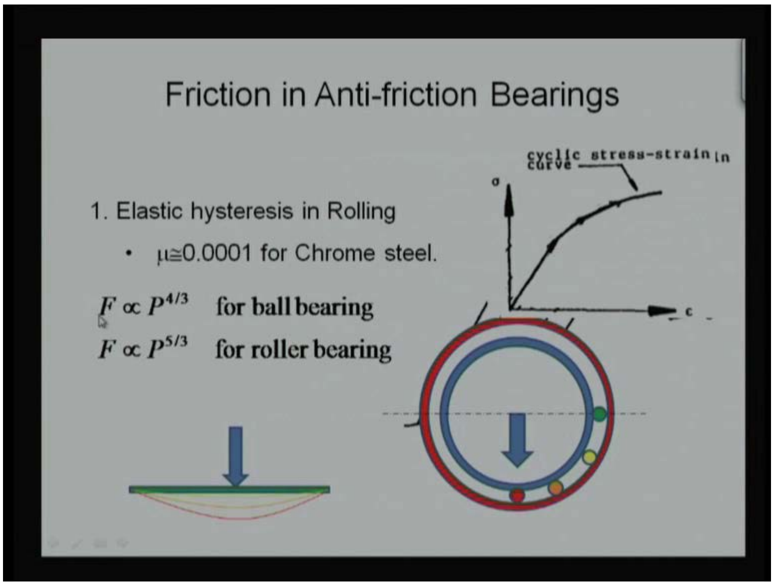

Now, this figure shows load is being applied and maximum load will occur at this ball. I am assuming this is a ball bearing, there is a inner ring, there is a outer ring, this arrow is showing a force direction and this force direction when is a loading the ball. After certain rotation I mean to say few degree rotations this ball will be shifted to this side. If I am assuming the clockwise rotation ball will be shifted to this side and other ball will come into picture. That means, there is a still a distribution of a load on this within balls, but load will decrease on this red ball and load will increase on this ball. That means, there is a continuous variation of a load on the ball. This is a happening over here when we are increasing the stresses that means, there is a more and more load been applied and more and more load means more and more deflection.

That can be represented something like that we are assuming there is no load initially. This is a green color bar is here. Now if I am applying a load on this assuming slight load not at significantly high value, but finite value certain may be say whatever the load carrying capacity of the bearing and I am just giving a ten percent or twelve percent of that.

So, in that situation this yellow color line will appear and this is showing a deflection line. We increase a load, we are able to see another orange color line over here, further increase in the load we are able to see this red color line. I purposefully selected this three colors because this is a yellow colored ball, initially there is a green and when there is no load on the rolling element then yellow color line shows a some load and some deflection than orange color line over here. That shows a high deflection compared to yellow color line always will say that deflection which is causing, which is experienced by the this rolling element. Deflection in this case will be more than this.

And finally, is a maximum load over here as a red color and as this rotation is starts from red color it will shift to the orange color, from orange color it will shift to the yellow color and finally, it will come to the green color. That means, there is a continuous loading and there is a continuous unloading of an rolling element.

That is, if a material is not good then there will be more bigger loop available or there will be larger energy loss for that material or for that bearing. This is the hysteresis losses will be higher if we choose some cast iron, if we choose some other material where the energy loss is higher. While in the case of chrome steel which is being used in the rolling element bearing that has a lesser coefficient of friction.

And if I show only the one curve we say that we want a steel such a manner. If it is a loaded in direction it should return back in a same path. There should not be any area available in this curve. Loading and unloading should be giving the same curve. If we are getting the same curve, that means, there is a lesser or almost negligible hysteresis losses and that is a shown over here 0.0001 coefficient of friction particularly hysteresis condition or for this kind of a reason.

However, when we write coefficient of friction basic assumption behind this coefficient of friction is that friction force is proportional to the applied load. If applied load is a p on any element the frictional force will be proportional to this, but it has been experience, we have a curvature and this deflection is not proportional it is slightly more than proportional. And when this coefficient of friction cannot be directly multiplied with a p to get the results (( )) and this is shows that a ball bearing that is power is a 4 by 3 for the rolling element. For roller bearing there is a more larger length and deflection will be slightly more linear curve and will be getting p slightly or higher power of that.

However, this is a more on a research site and we should investigate whenever we required accurate results, but when you think about the comparable results the comparison among the results, comparison among the bearing and when we thinking of friction as one of the criteria of the bearing selection. Then we may not we may neglect this relation we say that F is proportional to p. It is not or the power is assumed to be 1 it is not 4 by 3, it is not a 5 by 3.

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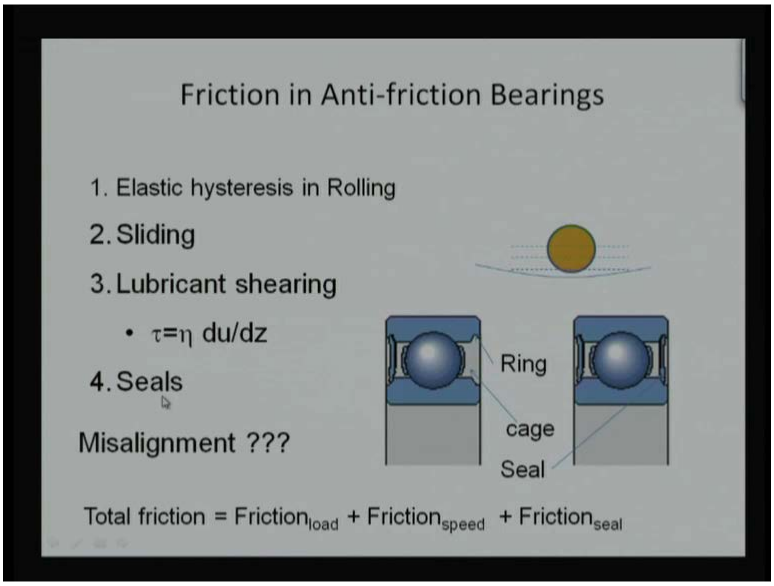

For time being we are assuming the F is proportional to p for our calculation purposes. So, this is being discussed last week hysteresis in rolling elements. The second one is a sliding even though there is a rolling element bearing name, but in those bearings sliding occurs. There will be some sort of relative velocity, there will be some sort of tangential velocity and that will not be rolling. This, there are number of reason we say that in rolling elements we know there will be kits to keep separation between the rolling elements, to keep separation between the balls, to keep separation between the rollers and that separation occurs with a some sort of a convict on the rolling element and there is a no rolling action out here. Even though there is a attach and by the rolling action will not be there due to more like a sliding action. Similarly, there is a inner ring and we know that to support this to some extent axial load we use a some sort of a curvature and again that is causing with slight change this side or slight change that side is going to introduce a some sort of a sliding. In pure rolling cases a pure radial load there may not be any sliding, but we know there will be some sort of actual force in which shifting and then pure rolling action will not be there some sort of sliding will occur. And last one is this and sliding case is this seals. We use seals to prevent oil leakage, to prevent grease leakage. So, if we are using the seals then what we are sacrificing that is a friction. With a seals coefficient of friction will increase or over all coefficient of friction will increase it will not be the same as without seals.

So, sliding is passed by the ring and ball interface particularly when there is other load reflexive action load and in this case you are talking about treadle bearing and we will say that if there is external load then this is a more sliding. When there will be any cage naturally there will be some sort of sliding between cage and rolling element. Then comes the seal, there will be a sliding because we are able to see there is a contact. We are assuming this is a last (( )).

This is embedded in outer ring and is touched with inner ring there is some sort of pressure, may be this spring tension or may be this is own stiffness spring whatever the seal material we are using that has a stiffness. So, it will be resisting leakage of oil, but that resistance is going to introduce a sliding between the seal material and inner ring and that is going to cause friction that is going to cause energy loss. So, that is a sliding. Now, to reduce a sliding we often use a lubricant. Interesting thing that lubricant otherwise is not required if there is a pure rolling action there will not be any contact. So, optional much area contacts or during and require any lubricant in this situation, but to reduce a sliding as well as to reduce a deflection of the surface, to reduce a seizure of two surfaces we use lubricants.

That will be detailed, there will be discuss in detail work in lubricant will be proper for rolling element during, but for time being we are discussing about the friction in anti friction bearing. So, we say that there will be some sort of lubricant shearing reason being there is a relative motion. If there is no relative motion we do not required any bearing. We require a bearing to provide a relative motion and if there is a relative motion surely there will be shearing of lubricant. If we are using some lubricant

(( )) To incorporate lubricant shearing is Newtonian law. Newtonian law says that shear hysteresis. Hysteresis developed in a fluid is proportional to the viscosity and velocity gradient. Velocity gradient I am assuming the z has a some, z is along the radial direction and there is a some gap between the rolling element and the inner ring is something like this.

There is a rolling element rollers or ball and this is a inner ring or outer ring particularly in this case when there is skidding in this form it will be inner ring and there is a lubricant. When the rolling or this rolling element comes closer to this and also rotate about its own axis, naturally what will do it will try to drag this lubricant in the space. The dragging action itself is going to cause some friction, is going to lose some energy and further this is having some velocity. Inner ring will be having different velocity. So, there will be difference in velocity of rolling element and the inner ring naturally that will cause the shearing of the lubricant and that is what we are trying to predict. We are trying to estimate, we are trying to find how much energy loss will occur because of the lubricant.

And last one is a seals, of course we mentioned in sliding the seal components, but it has been experienced the seal almost if there is seal it can show very high friction even more than thirty percent friction just because of the seals. And if there is a misalignment then the coefficient of friction due to seals will increase in an more than fifty percent of total friction. That is why they have a kept this seals in a separate category. Sliding is generally with in presence of the lubricant, it will change, that is why we go ahead with the three different components. We say that friction due to the loading and unloading or due to the load, we say load dependent component of the friction.

Second, we see the lubricant and sliding component of the friction, we have keeping this together we know very well the lubricant is going to reduce a sliding or it will reduce a friction between two surfaces. And lubricant itself is going to cause friction as going to cause a friction naturally it need to be combination. That is why we consider lubricant and sliding or the lubricant and velocity component separately, load dependent component separately and seal dependent component separately.

And this can seen, if there is a misalignment every component will get affected, but that will create we say that will be require slightly more understanding, more analysis the detail analysis and in this course we are not going to discuss that in detail. We will be touching little bit about the misalignment, but not to a very great extent. We will show some sort of elementary analysis to show the word is effect of the misalignment, but not to a detail analysis.

However, we can see if we consider these four components of the friction we say total friction will be because of friction due to the loads, friction due to the speed and friction due to the seals. Here when we are talking about the speed, we say that speed and the lubricant that will be better over to friction due to the speed and lubricant a combination. So, we will see how to estimate friction due to load, how to estimate friction due to speed and lubricant or lubrication and how to estimate friction due to seals. We will be using some sort of empirical formulas not a complete scientific finding, but has been done a number of experiment has been done, because it has lot of non-linear relationship in this and if we try to do detail analysis it will take complete one course on that.

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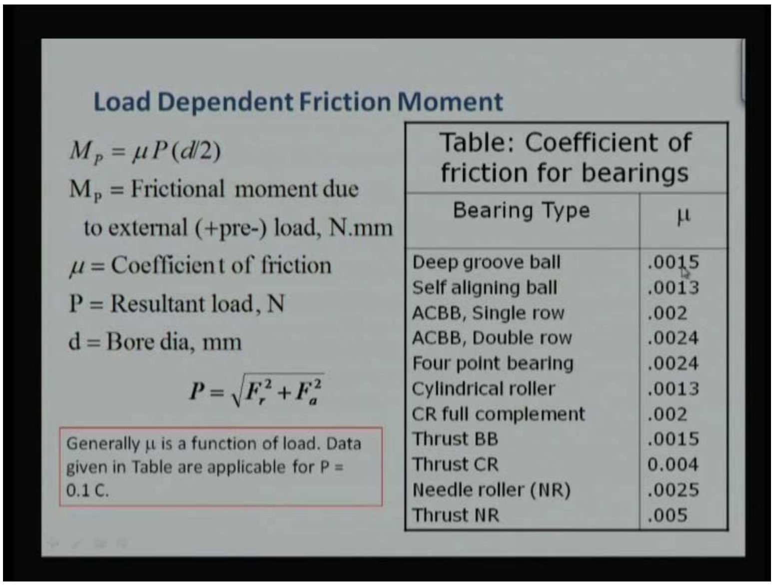

Let us take a first component; it is the load dependent friction movement. Again we are using the friction movement; we are not talking about the friction force, because this is the rolling action. And whenever there is a rolling action generally we use the word movement instead of using the friction force. So, we need to calculate, what is the friction movement due to the load components? Say, it will be proportional to the applied load and not taking nonlinearity due to the ball geometry, due to the roller geometry we are just taking proportional. This is a still an approximation, not to a very good level, but it gives a feel of the friction, give the results of the friction. We can give a good comprises or we say it can be a basis of good comparison.

Then there is a coefficient of friction and this is a radius, for which is the radius of more like a torque (( )) or movement (( )) and when we are talking about the force speed, we are assuming there is a external load. In addition, there is a possibility of preload, what is the meaning of preload? We say in bearing, if there is no clearance or there is a negative clearance, then there will be some load initiated or develop with in bearing that will be different than external loads.

How this kind of load will be generated? Generally when we mount bearing on the shaft or in some housing and if there is a interference fit, that interference fit will reduce a clearance or we say that it negate the clearance between the rolling element and inner ring. And there is a negative clearance between the rolling element and inner ring naturally there will be load or there will be deformation of the sorting elements as well as the inner ring. If there is a deformation initially before applying external load on that that is called a preload or that will be causing preload on a bearing.

The main chance of preload is a more than 5 to 6 times compared to external load, that is why we say the mounting of bearing is one of the important step if we are designing good bearing, we are selecting good bearing, but we do not know how to mount this proper bearing properly it may lead to failure of bearing within a few hours of operation. Even though we are designing bearing for the thousand or plus number of hours, but bearing may fail within a few hours.

So, this is a first component of the friction we say that friction due to the applied load. This is the coefficient of friction of the applied load and (( )) again this speed may be resultant of axial force and the radial force. It is not individual. So, that is why we say that P can be represent as F r is the radial load applied and axial load applied or may be axial load because of some sort of sampling or coming from some other components.

Interesting thing is that we are not equally, we are not calculating equivalent load in this case. We are just finding the resultant loads. These are being concentrating on only on the friction. We are not talking about the load carrying capacity. A load carrying capacity that will more involvement is required, more detail analysis require that is why we go out with the equivalent loads; while in this case we just take P as the resultant of F r and F a as a radial load and axial load.

Now, we required we can calculate P from problem itself, we can calculate d or we can estimate d from the data given to us, but coefficient of friction will be different, will be depending on the geometry and generally given by the manufacturer or can be figured out or can be dogged out from manufacturing catalogs. That is the one table is giving again this is not a complete table or it is not a detail table. If manufacturer is different than table may be different. So, always we should rely on the manufacturing data provided by the manufacturer, compared to this kind of thing, but this case is one sort of the some sort of the feel of coefficient of friction. We can think yeah this bearing is going to show much higher friction compared to second bearing. So, we do not worry about absolute values of this coefficient of friction, but relative value will be more or less same for all manufacturers.

So, the deep groove ball bearing, where there is slightly lesser flexibility for the rolling element coefficient of friction is on a higher side. While a self aligning, there is a more flexibility coefficient of friction is slightly lesser same thing happens with the cylindrical roller bearing slight more flexibility, coefficient of friction is a lesser competitive groove ball bearing. Coming to the needle roller bearing in this a lesser flexibility that is why coefficient of friction is on a higher side and whenever we talk about the thrust bearing, thrust bearing generally have a higher coefficient of friction compared to the other bearings. That is a key factor wherever you find a thrust bearing we find greater coefficient of friction is more than the radial bearing.

We can use this data provided, applied load is a just 10 percent because this load has a data has been generated for the 10 percent of load, we say that 10 percent of dynamic capacity when the applied load or we say that applied resultant load, is our equivalent load is almost 10 percent of dynamic load. The load is a more load is a lesser than this affection force of coefficient friction will vary particularly whenever load is at the much higher sign, say equivalent loads P is more than ten percent of the dynamic load carrying capacity. This friction force or this component of friction or coefficient of friction will be much higher than this stated value or listed value.

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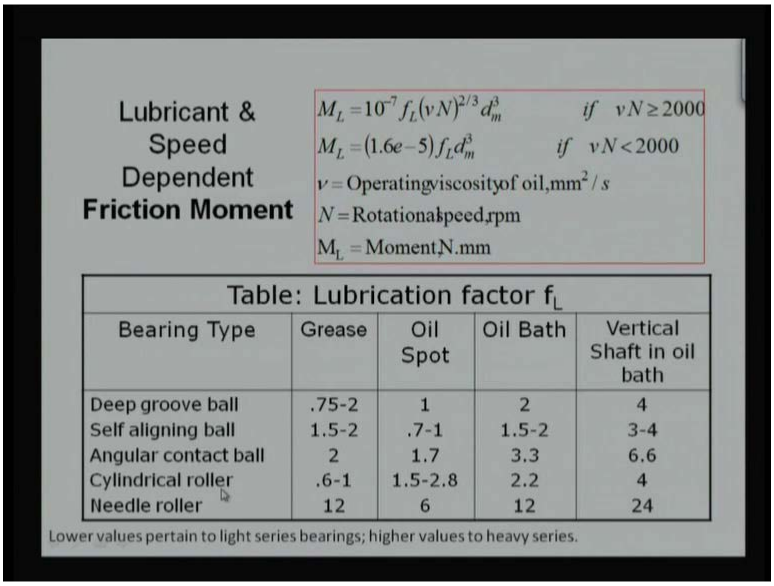

Then the second component is the lubricant and speed dependent component. In previous slide we discussed about friction force due to the load. While here, in this case we are talking about the friction force due to lubricant and speed dependent. However, changes depending on the speed of lubricant.

That is generally represented by the M L. M is movement, L is lubricant dependence. It has some steady and unsteady value, say value is a lesser than 2,000 which is product of kinematic viscosity and rpm, rotational speed is lesser than 2,000. This coefficient of friction or this friction movement will be steady. It is not going to change with speed, it is not going to change with viscosity. By and large value is constant, except it depends on the f L parameter, which is again depending on bearing type or geometry.

So, f L or geometry type and particularly when we are talking about kind of the lubrication mechanism. It depends on lubrication mechanism as well as the bearing geometry. Now, when this product kinematic viscosity and rpm is more than 2,000 or equivalent to 2,000, we have different relation. And interesting thing is that if I substitute mu into an equal to 2,000 in this relation. I will be getting this relation. So, there is continuity, there is a compatibility between these two equations.

Now, in this case whenever we substitute value of kinematic viscosity and rotating speed or rotational speed. We will be able to see that friction force increases with viscosity. If we are using thicker wall it will cause more friction, if we are using higher speed it will cause more friction, but if this product is a lesser than 2,000 it is not going to cause extra force. It is being by a larger same (( )) value of the friction movement due to the lubricant and speed. Now, finding the value of f L again we can, we have a number of tables available in catalogs and even in the books on the rolling element bearing. We can figure out, we can find out, we can use that value of f L for present later we are using this values. (( ))

We can see, for deep groove ball bearing this value is ranging between 0.75 to 2, question comes what is the meaning of 0.75 and what is value of the 2. why this range is being given? Whenever we talk about the light duty bearing when the diameter is smaller, when the width is smaller then we will choose a lower value. When we talk about extreme size of this, deep groove is on the largest size of the outer diameter and with then we will take higher value 2. So, this range is been given for the light and (( )) dimension are changing the larger dimension larger value for smaller dimension smaller value.

And we are able to see in case of the grease, oil spot, oil bath and particularly the vertical shaft. The reason for high value in this column, high value of f L in this column is because of the difficulty in lubrication will not be able to give sufficient lubricant for the bearing lubrication purpose. That is why f L is generally higher value whenever there is a vertical shaft and supported on bearing because we know that due to gravity lubricant will drip away, will move towards the downward direction. And we need to re lubricate it continuously or we required a continuous lubrication and in this case none of single lubrication mechanism is showing continuous lubrication of the rolling element bearing. (( )) some certain duration there will be oil is sprayed on the bearing in this case there is oil is completely in a bath. So, it is there will be some sort of heat generation that will be there also.

And is very difficult to maintain oil bath particularly for vertical shaft, that they will not be useful that kind of bearing arrangement will not be very useful. So, in that situation we required a higher value of f L or we say it is going to cause high energy loss. Vertical shaft generally going to cause high energy loss compared to the horizontal shaft. So, as far as possible we should go ahead horizontal arrangement of shaft. If we are designing some machine, we should give more emphasis on horizontal shaft compared to the vertical shaft.

Here we are able to see there is a changed in bearing, there is a change in a bearing there is a change in f L value for that. I can this case deep groove ball bearing as a angular contact bearing and for angular contact bearing we are able to see the higher value of f L. As in this is going to give us which it is going to cause a higher friction compared to the deep groove ball bearing. Whereas, case of the grease, even in the case of oil spot, even in the case of oil bath, in an case of vertical shaft

,in all the cases angular contact bearing is going to cause more particularly from friction point of view. We studied in previous lecture angular contact bearing has a high dynamic load carrying capacity and it is going to survive for more number of operation or service life of angular contact bearing will be more than the deep groove ball bearing.

Now, if we consider friction also as a one of the objective function, then we may say in some cases angular contact ball bearing is going to give a better result. In some cases the deep groove ball bearing is going to give us better results. So, it will turn out to be the what is the main aim of that bearing, is it more supports or is a lesser friction whichever is the more dominating we choose a bearing accordingly.

Interesting thing is similar kind of competition can be met in a cylindrical roller and needle roller cylindrical the needle roller bearings. See when cylindrical roller bearing is used we know there will be lesser number of rollers and when you are talking about the needle roller bearing there will be more number of rollers. When there is more number of rollers naturally the dimension we say the load carrying capacity will increase for the same dimension.

And if we want to retain the same load carrying capacity dimension of the needles rollers will decrease, and that is going to save the space, and that is the main advantage of needle roller bearing. Needle roller bearings are required whenever there is a space constrain in radial direction. However, this is going to cause a friction, we are able to save the space, but we are going to lose energy. Say cylindrical roller bearing is f L factor is 0.6 to 1, 0.6 for the light series, 1 is for the heavy series or higher side series. However, the needle roller bearing (( )) is 12 and it is going to cause a 12 times high friction compared to the cylindrical roller bearing, when there is a grease lubrication.

Coming to the oil case, that difference is not that much significant; here the needle roller bearing are more comfortable with oil lubrication mechanism. This is a 6 times, again these are based on experiments in this maybe there will be some change if we shift from one catalog to other catalog.

Coming to the vertical shaft, needle bearing should not be used this is going to again cause high side very high friction movement for the needle roller bearing is almost 24 times, which is that 6 times compared to the cylindrical bearing or f L is at 24 in the situation. So, based on these data we can find out what will be the friction movement due to the lubricant and speed.

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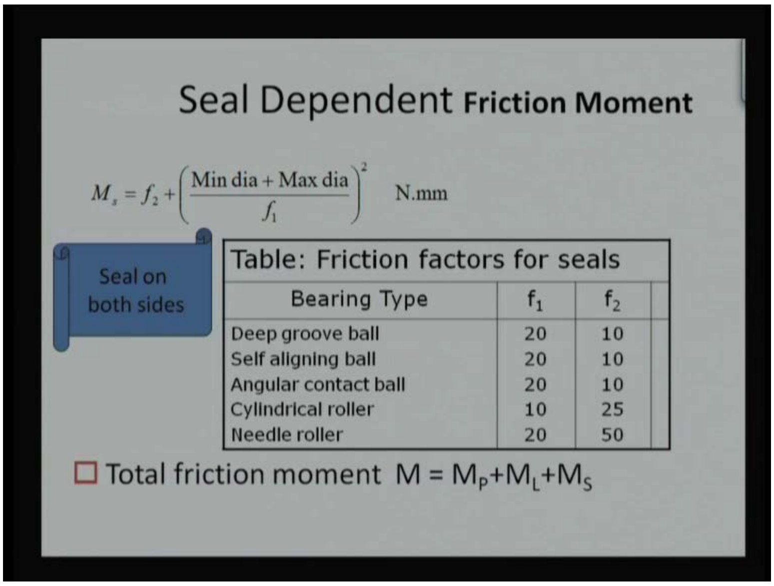

Last in the series is a seal dependent friction moment. Again in this case there are two components. One is a fixed when the geometry is fixed, this is fixed other case the depends on the geometry also, which will depends on the damager also. So, in the deep groove ball bearing with a 10 mm, 20 mm, 30 mm, 40 mm, 50 mm, 60 mm, 70 mm, 80 mm have to remains constant. That is remaining 10 Newton mm. However, as the geometry increases, how we say that value of diameters are increasing this factor is going to get domination. Let us take a minimum diameters at 10 mm of the bearing naturally outer diameter will be around twenty mm. So, 10 plus 20 is 30 and divides by 20, see it is 1.5 and 1.5 square of that one is going to add to this.

Now, if I increase diameter assume there is a 20 mm and this may be is a 30 mm or 34 mm. So, 20 plus 34 is 54 divide by 2. So, this factor is increasing with the dimension.

That means, it has a two components one is exclusively depending on the kind of bearings which we are using other depends on kind of bearing as well as damages.

So, based on this formula we can find out what will be the friction movement to the seals. Again, in this case basic assumption is that seals are provided with the both the sides, is a possibility the friction is seal is provided only on a one side. In that case this movement will be reduced to 50 percent. The seal is only one side friction movement will be reduced to 50 percent

Now, if I know M P if I know M L, if I know M S. I will be able to find out what is a total friction movement. That will be simply a summation of M P plus M L plus M S. To understand the various factors involved in this kind of calculation.

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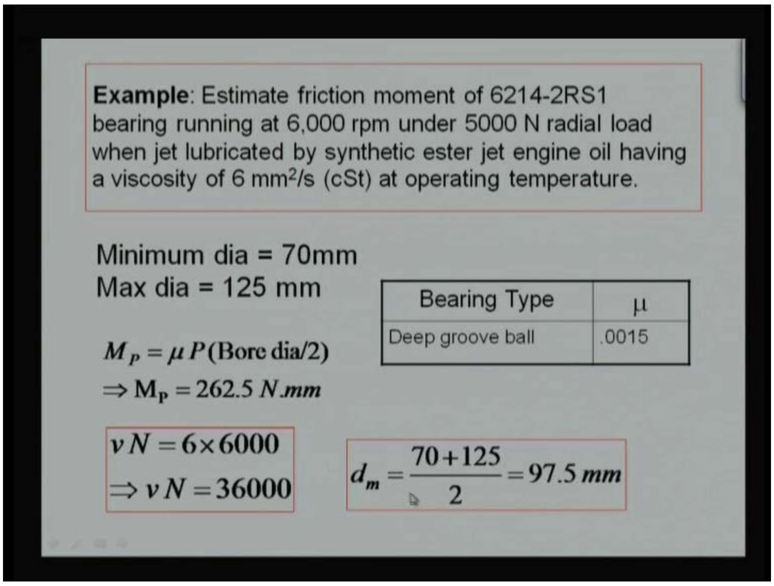

Let us take a one example, we say that estimate, again we are using the word estimate because we are there are number of assumptions lying in our theory and our calculations. So, we cannot say calculate friction movement. That is an estimation calculation with. I am 100 percent successful in the theory and I am 100 percent confident about my relations. Confident about the results, which I am going to get; however, in this case we are talking about the estimations, we are hundred percent confident.

So, estimate friction movement of a bearing that is 6 2 1 4 this a deep groove ball bearing. 14 is a 70 mm diameter if I multiply 14 with a 5 I will be getting 70 mm. It has a seals at the both the end that is why we are writing it 2 R S 1. If it is only 1RS then seal is only one side, if it is RZ then that we can say the seal on one side and the shield on other side.

All configurations are available in market. We can choose the bearing from a catalog or as per the our requirement. However, in this problem we are talking about the seals on the both the sides and is we are going to use the M S, full value of M S. we are not going to reduce to the 50 percent. If it is only RS then we will be reducing to 50 percent.

Now, in this case again we are saying the bearing is running on the 6000 rpm the inner ring is rotating under 5000 Newton load or 5 clone Newton radial load. There is a some sort of lubrication mechanism divided or described out here we say the when jet lubricated, is more like a oil spots. We have some sort of comprise gas and we push oil with a compressor in gas and only the some sort of droplets will come out, will be jet formation.

And we are using a synthetic ester jet oil, which has a viscosity of 6 centi stroke or in talk about the mk's unit we say mm square per unit second at operating temperature. What is the operating temperature we are not describing this. Whatever that operating temperature has a viscosity equal to 6 centi stroke. This word is important here because we know as operating temperature changes viscosity will change. It will not be same or discarding or it will not be same as a room temperature. It is a possibility of high heat generation in bearings and that will change the lubricant temperature.

Now, how to do calculation part is we see, we require dimensions from the bearing catalog and this is already given 14 mm is going to be a 70 mm or 14 series is going make a board diameter 75 into 14 is a 70 mm. Max diameter for this series is 6 2 1 4 is 125 mm diameter. So, based on these two dimensions I can find out, what will be the mean diameter which will be required particularly for calculating or with the estimating friction due to lubricant and velocity.

Now, for the first component, load dependent component we require coefficient of friction, for the deep groove ball bearing because we know the 6 2 1 4 is a deep groove ball bearing now for the utilize this coefficient of friction into radial load which is given a 5000 Newton and board diameter which is that is a movement (( )) movement (( )) can be calculated 70 mm divide by 2 that is a 35 mm.

Now, using this substitute this value 0.0015 into 5000 into 35 mm, we will get movement value as a 262.5 Newton mm movement value. So, this is a first component which is based on the load and the bearing geometry has been calculated or has been estimated. Second components require the product value product of kinematic viscosity and operating kinematic viscosity and operating speed.

Kinematic viscosity in present case is a 6 centi stroke and an rotational speed in this case is a 6000 the 6 into 6000 makes a 36000. Which is a greater than 2000, that really we need to use a fast relation, for fast relation we require dm and that is a dm in this case is 70 plus 125 divide by 2. The mean diameter and that and all to be 97.5 mm; the mean diameter is known now for us

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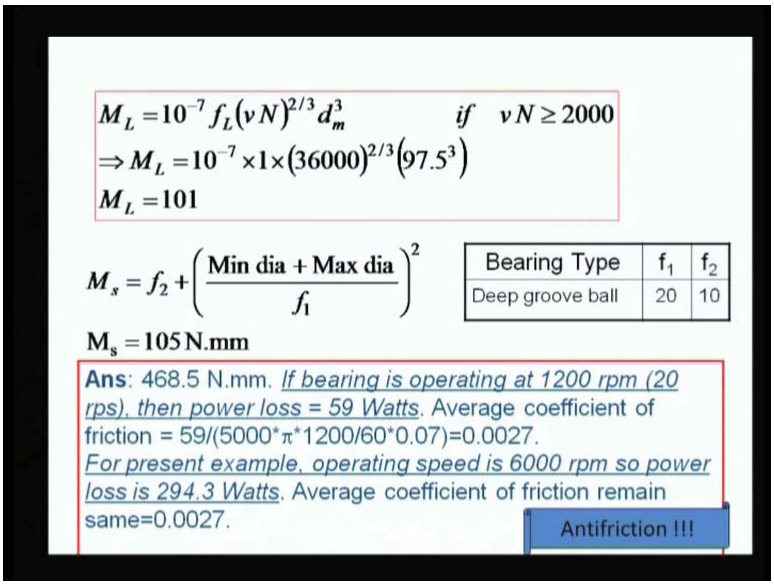

We knew and product is known to us, we can directly use the relation which we describe in previous lines that amble. Friction movement due to lubrication is 10 is to minus 7 into f L into mu into n in the power of the 2 by 3 and this is a heavy companion, is a highly sensitive companion. The dm we are using power 3, the 10 is to minus 7, f L in this case is equal to 1 because of the jet lubrication mechanism and this product is standing out to be more than is a 36,000 which is more than 2,000 that is why we are using this relation. And d m is a 97.5 and is a cubic so we are using cube of that.

This over all is standing out to be 1 0 1 Newton mm. It is not as high as what we have calculated for the load dependent components. In this case friction movement is not a very high may be say around 40 percent of the load dependent component. The third component is a seal because we are using 2 R S 1, then the seal is on both the sides and for the deep groove ball bearing. The deep groove ball bearing f 1 is given as a 20 and f 2 is given as 10. We will be utilizing this. So, 10 will be substituted here that means, whatever the dimension 10 by 10 meter Newton mm will be the friction movement, plus depends on dimension. We know minimum diameter is 70, maximum diameter is 125, f 1 was can be figured out from this table has a 20 and if we do the calculation it turn out to be friction due to seals is 105 Newton mm. We know we can add up all these movements and this is turning out to be 468.5 Newton mm. You get a feel of energy, you get a feel of power loss due to this front bearing. We can think about the rotational speed of the bearing which is may be assuming in this case2,000 which was given earlier in this case, 6000 rotational speed is given as 6000 rpm. We use a 6000 rpm than will be quite a lot of frictional loss. If I assume frictional loss or may be operating speed can be reduced to a slightly lesser level 1,200 rpm even in this case there is a cost of the 59 watts.

If I assume a 6000 rpm it will be 59 watt into 5. So, roughly 300 watt we are going to lose in this bearing which is significant. Almost half of the h p or slightly lesser than half h p is going to be a loss in the bearing that is a significantly high friction loss in the bearing.

And if we try to calculate there is only 1200 rpm. This movement what we say that coefficient of friction is turn out to be 0.0027. Now, why we are doing this? To differentiate whatever the friction movement coefficient of friction given in bearing catalog for the load dependent component it was only 0. 0015.

While when we are doing here, we are finding that coefficient of friction is going on higher side. That is turning out be 0.0027. Now even if I go ahead with 6000 rpm then multiply with a 5 in a numerator and then divide ah 5 again in denominator. 5, 5 will cancel out coefficient of friction is still will remain 0.0027.

So, this coefficient of friction should be counted, compared to coefficient of friction what we get from the catalog that is a load dependent component or that will be utilized only for load dependent components. This is a just two times of that coefficient of friction.

Many times we say this coefficient of friction is negligible which will not be worried about too much about this. Even the coefficient of friction about 0.01 is not a very high value, that is a tolerable and it will always happen we cannot do, we cannot improve bearing further. So, there is no need to worry that is a possibility we can keep that in our mind if the coefficient of friction is we say lesser or we say 0.01. We should not worry about the bearing, then bearing selection should be exclusively based on the load and speed on what we say that operating hours.

Anyway there is an interesting point is that we need to find out if there is a this much energy is generated, this much power loss is happening in a bearing, what is going to be the temperature of the bearing because the bearing temperature also will change. And change in the bearing temperature there is a possibility of change in clearance. That is possibility of extra preload in the bearing those need to be considered properly. So, to think about that we can think about the temperature however, there is one exclamation mark comes over here such this is a loss in a bearing and if we talk about the 6000 is a 5 times on that this is higher site. So, almost 295 watt has been lost in a bearing and still we are saying this bearing is a antifriction bearing.

So, we need to think whether should we call antifriction this kind of bearing anytime antifriction or we should not be calling this. We should call only on the rolling element bearing I prefer to call this has a rolling element bearing compared to antifriction bearing.

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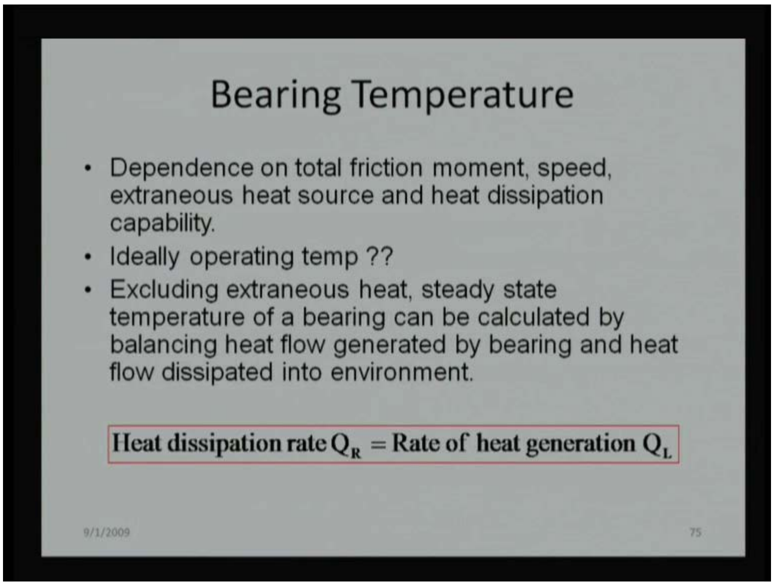

Now, as I mention if there is a heat generation naturally there is a possibility of some sort of temperature increase of the bearing surfaces. And if there is a temperature increases significant then what will happen there is a possibility of change in the dimension and if there is a change in dimension there is a possibility of loss in the clearance. There is a power possibility of interference rate. I mean the rolling element and the rings and if there is a interference rate then it will cause a more load more load is going to cause more temperature, more temperature is going to cause more load, it will be more like a catastrophic. (( ))

That is what the main times bearing fail instantaneously if they are not properly mounted, if there is a friction is not controlled properly, if the temperature is not controlled properly. So, we say that in this case what we need to find out, what is operating temperature? Ideally operating temperature should be within few degrees or we say that increase in temperature should be within few degrees or slightly more than the atmospheric temperature.

However, to find out that what we require, we require how much heat is generated how much heat has generated, how much heat has been dissipated or we required a balance between a heat generation and heat dissipation. We say that heat convicted or conducted or way to some other elements. That is why we require heat dissipation rate and heat generation rate that should be in equilibrium. If that is in equilibrium temperature will be more like an within tolerance, if there is a more heat generation compared to heat dissipation rate than naturally temperature rise will be very high.

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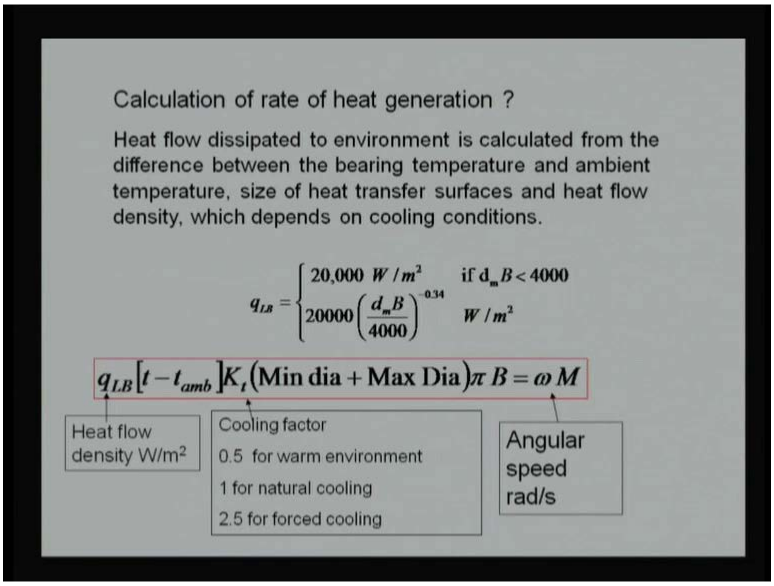

There will be always energy balance, but this is continuously going to change if heat generation is going to continuously increase. Let us take a simple case, we say that we know how to find out the heat generation rate using the previous like calculation, find out the M P, find out the M L find out the M S sum up find out what will be the operating speed multiplied with that and we will be able to figure out how much heat was generated or what is the power heat generation rate.

Coming to the heat dissipation we can make some sort of assumption. So, if we know ambient temperature. What we are looking for the temperature increase, temperature difference for that purpose what we required is something like the how much heat can be dissipated from the surface. That means, we require more or less surface dimensions. We require the length, we require the peripheral dimension, we say that circumference of the rolling elements and in addition we require heat flow density, which is generally given by manufacturer from the material property point of view, the arrangement point of view. And in addition if there is a cooling arrangement if there is a possibility of the fan running near during temperature will be different. So, we require a cooling factor, we required density factor, we require dimensions.

And there is a relation, again this is what we say the imperial relation. We say that bearing can dissipate up to 20,000 watt per meter square provided dimension, the mean dimension and length of the bearing is a lesser than 4,000.Which happens in most of the small size bearings, but if dimension is increasing is going beyond this, going beyond 4000 or equivalent to 4000 then we should use this relation. We can see the substitute value of the 4000 over here the numerator and denominator both will be 4,000. That means, this ratio is one and we will be getting same result of this.

However, if dimension is increasing, I should be happy if dimension increasing the surface area is increasing, but it is not happening surface area has have increasing, but heat distribution density is coming down. We know that material is not able to dissipate that much heat.

So, overall not very encouragement of the lot of encouragement to increase the dimensions; a larger dimension this density will be lesser compared to what we are getting 20,000.I mean to say that maximum value of density is 20,000 watt per meter square. Right now, if I know q LB that is the heat density and if I know the temperature, ambient temperature this is a cooling factor we say the (( )) cooling natural cooling and we are not using any constrain. We are not restricting bearing from all the sides then this factor will be equal to 1

If we are using extra force cooling we are using some sort of fan over there then this cooling factor will be on slightly higher side and means more heat dissipation can is possible from the bearing surfaces.

However, there is a possibility there is a no circulation there is no air circulation and then environment also remains slightly hotter side. Heat dissipation is not that much fast. So, this factor can be treated to 0.5. Again these are based on imperial relation factors; we can say that we will not be getting accurate result, we will be getting the feel of the temperature rise.

Now, minimum diameter can be figured out, maximum diameter for the given geometry will be knowing to us, length will be knowing to us. With the (( )) geometry is given angular speed is operating speed will be known to us, movement or friction movement due to load, due to the lubricant, due to the seals can be estimated. So, we will use this relation to find out what will be the operating temperature this t.

If operating temperature with few degrees we do not have to worry too much. Now, if we utilize this relation for the given bearing geometries we will be able to estimate what will be the operating temperature. That is given; we say that previous example whatever been mentioned. Let us add few more data to find out what will be the operating temperature.

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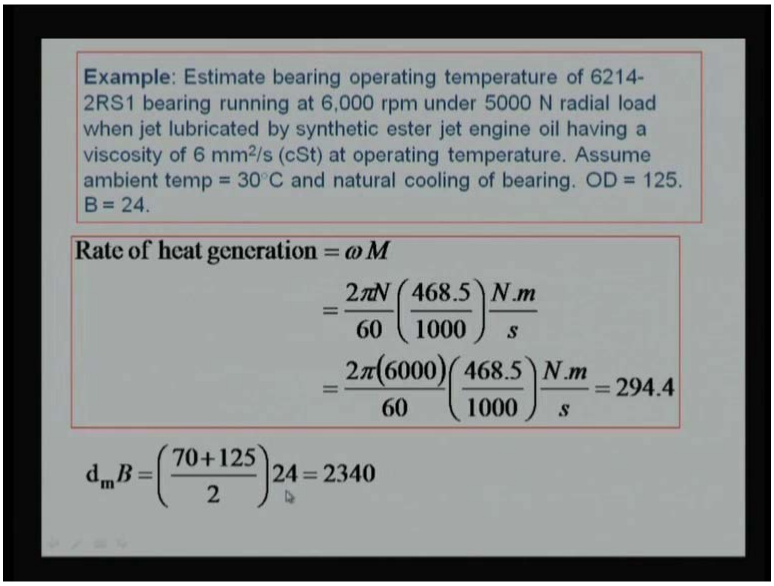

See in this case, assume the ambient temperature or t ambient is a 30 degree centigrade and natural cooling of the bearing, bearing is naturally cooled and this factor is equal to 1. O D is was already mention that is 125 mm value of the B is at 24 mm. Now, we require this data, rate of heat generation we already found in previous slide, previous to previous slide. That was a 468.5 Newton mm, if I divide by 1,000 it turn out to be Newton meter per second. This is given as 2 phi N and is rotationally speed as a 6000 in this case. When we find the rate of heat generation is standing out to be 294.4 watts.

Now, we require this dimension for heat dissipation and we say that this dm into B is 70 plus 125 divide by 2 into 24 that is standing out to be 2,340. Which is lesser than 4000. That is why we say that for most of the bearings.

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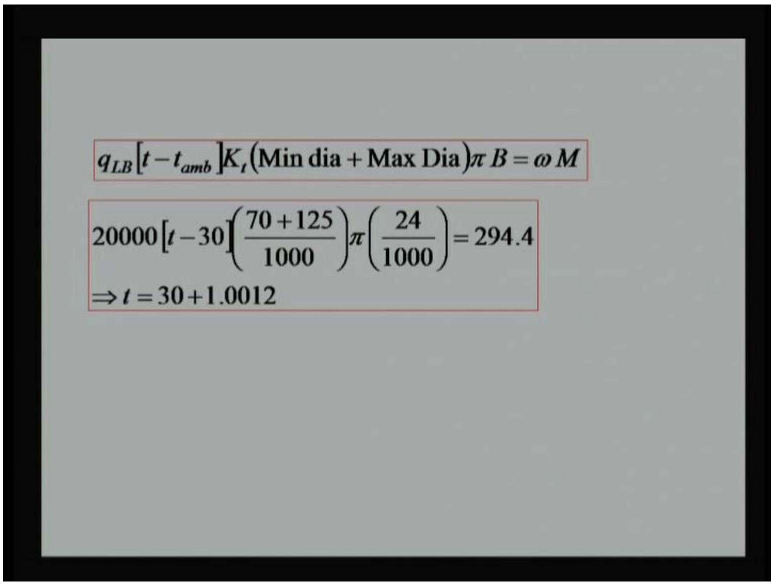

This product will be lesser than 4000 and that is why we can use directly heat dissipation density as a 20,000. This is a ambient temperature K t is in this case is 1 minimum diameter is known, maximum diameter is known, value of the B is also known to us. We already figured out and estimated this value omega into m that is turning out to be 294.4 watts.

We do calculation estimate we find the temperature is 30. what is the ambient temperature plus 1 degree. That is well within limit that is not a much problem we can choose this kind of bearing from a friction point of view, is not going to cost much loss or much energy loss, is not going to change the bearing damages significantly. In this case one degree temperature rise, that is tolerable we can go add with this, but there is a possibility. There is a possibility if there is a some sort of a misalignment then there is a problem. That may increase the temperature significantly. We will be studying or we will be discussing that aspect in our next lecture. Thank you, thank you for attention