

## EXPERIMENTAL INVESTIGATION AND ANALYSIS OF DIFFERENT BEARINGS THROUGH FEM FOR INCREASING ITS USAGE LIFE

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**Abstract.** *Bearings are rotating machine elements that limit the detrimental effects of whirl, centrifugal force on rotating shafts and journals used in machines in the industry. The bearings considerably subjugate the acrid effects of phenomenon of mechanical vibration. In everyday use in the industry, the wear or condition of bearing surface is checked manually through visual inspection, which quite often results in faulty interpretation of bearing wear. In this paper we have initially analyzed the data of the bearings on our own setup for limited number of hours and then verified with nearby industries and then extrapolated it to our requirement, related to three different types of bearings and generated mathematical models for each type relating the rating life to three factors i.e. radial load on the bearing, speed of journal, the nature of lubrication used. Later on FEM analysis has been done through CATIA software to investigate the points of maximum initial displacement and to develop a new model for each bearing to project the portion of maximum vulnerability for each bearing. CATIA uses static frequency analysis to analyze the structure by breaking down the bearing into meshes of suitable sizes through the generative structural analysis module in CATIA. The analysis is based on von mises criterion of stresses and FEM is used wholesomely in the analysis. We have considered the initial static displacement or indentation as the area of resulting maximum wear. The next step involves the process of relating the maximum initial static displacement and the coordinate vector of maximum vulnerable area to the load, speed, and viscosity of lubrication used. Developing the above mentioned will assist in prolonging the real life much closer to the ideal rating life by paying the necessary heed to the critical areas on the bearing's surface. This is because we can predetermine the vulnerable points and thus take the recommended steps to reduce wear or displacement at those points. Few recommended special lubricants have been given which can be used to treat the vulnerable areas timely to reduce the maximum indentations and thus allow the bearing to function for a greater number of hours than the usual.*

## 1 INTRODUCTION

The Bearings are an indispensable part of all rotor shaft systems and every such mechanical system where we need to reduce the friction wear or constrain the relative motion between two surfaces. Above all the protective uses of bearings one must note the fact that bearings themselves deteriorate from constant wear and tear in their daily

The Bearings are generally made of chrome steel, stainless steels and some are even manufactured from ceramics and plastics, all depending on the type of use and expenditure the customer/company can sustain. In the industry bearing wear is noted through visual inspection by the operator only after the bearing has been used for a massive percentage of its theoretical rating life. Thus the bearing at its most vulnerable part/area has undergone severe damage which cannot be treated or stitched in time to prolong the use of that bearing closer to its theoretical rating life.

## 2 LITERATURE REVIEW

The wear on the bearing surface during its application period can be gauged by the translational displacement (radially or axially) when a given load is applied at a constant speed in a given temperature field. The wear or displacement is generally of the order of  $10^{-4}$  mm, which is seemingly marginal. But then how does this allegedly negligible speck become the genesis of so many bearing failures? Micro pitting is one of the major causes of bearing wear and eventual premature failure. Pitting occurs and leaves visible damage on the bearing surface in due time as the small crevices on bearing skin develop in ugly and severe troughs on the surface of the roller or journal bearing. This happens due to lack of attention to the most vulnerable areas on the bearing surfaces which are left unattended throughout the running operation, whereas such areas as found later on in this paper, demand and deserve special attention and precautionary measures. C N Ko and Ioannides [1] have blamed debris accumulation as the factor that sets in and aggravates wear and becomes a cause of premature bearing failure. Razor sharp debris from surroundings, material from the wear of shaft, pieces of broken oil seal get embedded in the fine grooves and thus with the duration of time these end up magnifying the groove or an initial scratch into a visibly effective slot across the bearing surface or skin. Fredrezzi and et al [2] have highlighted the relevance of special lubrication or coating, which ends playing a vital role in reducing bearing wear considerably. Savaskan and et al [3] analyzed sliding wear restricted to zinc based alloy bearings in static conditions. We have also carried out the analysis considering static conditions for simplicity and accuracy of results. Takeo Koyama [4] investigated and they propagated the application of Finite Element Method analysis in bearing design. Our analysis is a more precise and accurate break up through FEM on CATIA. Serdar Tumkork [5] also mentioned the usability and validity of CAD models available online for projects and research work. DellaCorte and et al [6] focus on the effect of lubricant in controlling bearing wear. So we thoroughly investigated the most distinguishing property of lubricants i.e. their viscosity. A third mathematical model which relates the wear to speed, load on bearing and viscosity has been formulated using the initial data. Solid lubricants which are relatively expensive but effective can be applied at the critical areas/rings on bearing skin [7], [8]. Robinson and Shackelford [9] have explained regarding decreasing the Stress in Ball Bearings which can be achieved by constructions in which one or both of the race members are supported only at the end portions, so as to be capable of deflecting in the manner of beam, in distinction to conventional bearings wherein the race members are supported over their entire circumferential area and are compressed in effect as a solid block or column.

Coe and et al [10] have made investigation and resulted that the torque and temperature of the bearings with the drilled balls (50% weight reduction) were significantly lower than those of solid bearings. Similarly, we have suggested some typical techniques for the three types of bearings by some considerations in lubrication.

### 3 OUR WORK

In our paper we have initially assimilated the average data of about 1200 pieces of each of the three different types of bearings being used prominently in the industry. The critical factors namely load, speed, nature of lubricant used (viscosity) have been related to bearing life through a number of mathematical models. At the given loads Finite Element Method analysis has been carried out through CATIA software using static analysis for a given temperature field. In our analysis we have restricted all but one degree of freedom of the outer race of each of the three types of bearings scrutinized. The bearings are allowed to slide in the direction orthogonal to radial load i.e. axial direction with the application of user defined constraint in the software. The load applied in the whole paper is the cumulative effect of the external load and the centrifugal force acting on the bearing. The FEM has been applied to make discrete meshes of the bearing surface and thus the maximum wear (translational displacement) has been noted and analyzes through different graphs. Throughout the paper we have considered the maximum initial displacement to represent the wear and tear i.e. crests and troughs at the latest stage of operation. Later on new mathematical models have been generated which ensure and depict the dependency of rating life on maximum wear, coordinate vector of area corresponding to the maximum wear and the initial three factors. This comprehensive model allows us to study and analyze the change in life corresponding to change in wear and lubrication. Suitable and feasible recommendations have been provided at the end. The application of regular coatings of these special recommended lubricants at the critical areas right from the beginning of the operation is the optimum solution to reduce the bearing wear, considering the cases of static load application. We must keep in mind that the groove/large crest formation on bearing surface happens due to the fact that fine grains/debris particles get trapped in the initial indentations/displacements. Gradually they enlarge the indentations converting them into sizeable troughs. So reduction of the size of the initial maximum displacement can be done by taking due care of those critical areas/points. Here in the Figure 1, the roller wear has been highlighted in this high detail picture. In Figure 2, Look at the wear/ displacement increasing towards the given direction in the inner race, this shows the difference in wear at different places. The initial miniscule displacement has ventured into the whole surface and formed an irreversible depression.



Figure 1



Figure 2

#### 3.1 Experimental Setup

Figure 3 represents the CADD image of the setup i.e. the milling machine where the cylindrical roller bearing was put to use. Figure 4 represents the cut section view of the cross section of the cylindrical roller bearing showing one of the rollers as labeled.

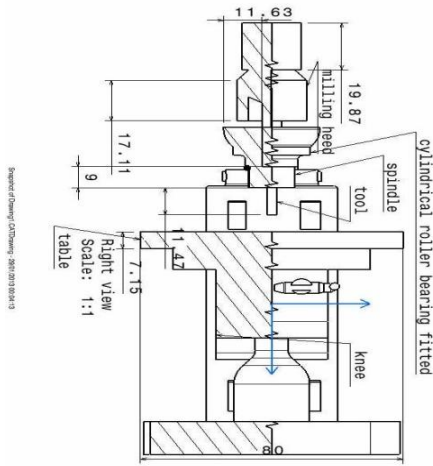


Figure 3

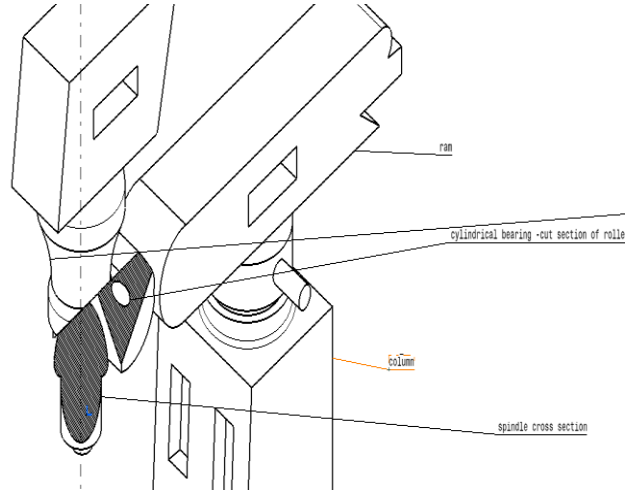


Figure 4

### 3.2 Initial Mathematical Model

We have started with making mathematical model relating Life to Force, Speed and Viscosity as described in Eq. (1).

$$L = aF^b N^c \mu^d \quad (1)$$

We have taken three different cases as we described earlier. Firstly we will proceed by Cylindrical Roller Bearings analysis. The type of Cylindrical Roller bearings is NU and Bore diameter is 240 mm and outer diameter is 360 mm. Next we have taken Deep groove ball bearing in which bearing type is open. The bore diameter is 75 mm and Outer Diameter is 115 mm and thirdly we have taken Thrust Ball Bearings in which the Bearing type is with grooved raceway. The bore diameter is 70 mm and Outer Diameter is 95 mm. The standard lubrication for grease has been taken as .113kg/m-s and that for oil as .054 kg/m-s. The data set obtained for each is given in Table 1 to Table 3.

| Sr. No. | Force(N) | Speed (rpm) | Lubrication(kg/m-s) | Life(hours) |
|---------|----------|-------------|---------------------|-------------|
| 1       | 40000    | 800         | Grease              | 109280      |
| 2       | 35000    | 1800        | Oil                 | 72500       |
| 3       | 45000    | 1000        | Oil                 | 61400       |
| 4       | 30000    | 2000        | Oil                 | 103600      |

Table 1: Data set for Cylindrical Roller Bearings.

| Sr. No. | Force(N) | Speed (rpm) | Lubrication(kg/m-s) | Life(hours) |
|---------|----------|-------------|---------------------|-------------|
| 1       | 1500     | 6200        | Oil                 | 49500       |
| 2       | 1200     | 6500        | Oil                 | 92150       |
| 3       | 1800     | 4500        | Grease              | 39450       |
| 4       | 1600     | 5700        | Grease              | 44300       |

Table 2: Data set for Deep Groove Ball Bearings.

| Sr. No. | Force(N) | Thrust Force (N) | Equivalent Load(N) | Speed (rpm) | Lubrication (kg/m-s) | Life (hours) |
|---------|----------|------------------|--------------------|-------------|----------------------|--------------|
| 1       | 1937     | 500              | 2000               | 2500        | Oil                  | 50682        |
| 2       | 1743     | 450              | 1800               | 3000        | Oil                  | 57840        |

|   |      |      |      |      |        |       |
|---|------|------|------|------|--------|-------|
| 3 | 1959 | 1000 | 2200 | 1500 | Grease | 63350 |
| 4 | 2291 | 1000 | 2500 | 900  | Grease | 71960 |

Table 3: Data set for Thrust Bearings.

On calculating with the data set we have according to the mathematical model, we have founded the values of all the variables and written in Table 4.

| Variable | Cylindrical Roller Bearings | Deep groove ball Bearings | Thrust Bearings         |
|----------|-----------------------------|---------------------------|-------------------------|
| a        | $5.6110 \times 10^{-21}$    | $1.0330 \times 10^{18}$   | $7.8145 \times 10^{17}$ |
| b        | -3.0002                     | -2.9997                   | -2.9730                 |
| c        | -1.0002                     | -1.0001                   | -0.9935                 |
| d        | $-6.8455 \times 10^{-5}$    | $-1.3064 \times 10^{-4}$  | -0.0014                 |

Table 4: Values obtained for variables.

We can analyze from the results that Life increases with decrease in any of the parameters from Load, Speed and Viscosity as all the coefficients are negative. Let us proceed further and analyze the three different types of bearings using FEM. We need to have a look at the manner in which the initial static displacement occurs at the discretized level.

### 3.3 FEM Analysis of the Bearings

The need to discretize arises because the initial static displacement happens at a very miniscule echelon, and thus the bearing surface and the whole body has been broken down into triangular meshes which helps mirroring the initial defect which goes on to become the cause of concern after a certain period of time. The FEM has been applied keeping the mesh size as low as 1.12 mm to achieve higher accuracy and precision even though the time taken to compute with a smaller global mesh size increases considerably.

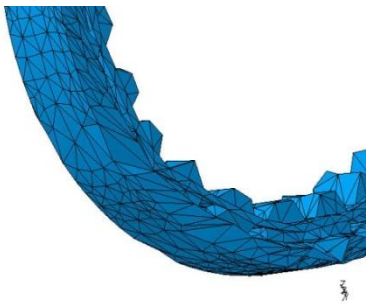


Figure 5

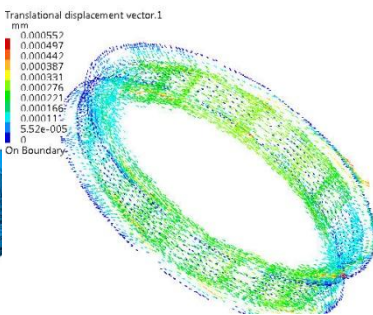


Figure 6

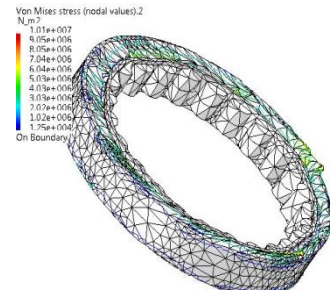


Figure 7

The Figure 5 represents the cylindrical roller bearing which has been broken into triangular meshes, here the discretized model is shown keeping in mind the deformation suffered by each point on the surface. It is very interesting to note the change in shape suffered by the rollers. They have changed from cylindrical to nearly trapezoids. Now let us look at the results obtained statistically as well as pictorially when the bearing is subjected to different bearing loads. The photos representing the bearing subjected to the corresponding loads have been included to show the displacement and von mises stress at every point i.e. every corner of each triangle of the meshing. The load applied in the whole paper is the cumulative effect of the external load and the centrifugal force acting on the bearing. The following is from the screenshots of the Finite Element Method analysis done for specific loads from our data tables, individually for each bearing. Above lie the screenshots of the analysis for a combined load of

40kn is being demonstrated. For 40000 N, in Figure 6 the displacement is represented by different colors from red to dark blue. Red color marks the hotspot or the place of maximum displacement. The picture also clearly shows the shape formed by the bearing surface if the corresponding displacements are put in effect. In Figure 7 the discretized bearing surface has been laced with the corresponding stresses at different points on the surface.

Similarly for Deep Groove Ball Bearings and Thrust Bearings same procedure is followed.

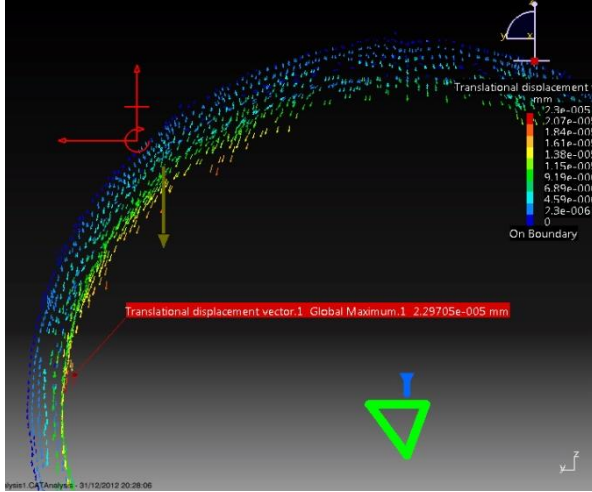


Figure 8

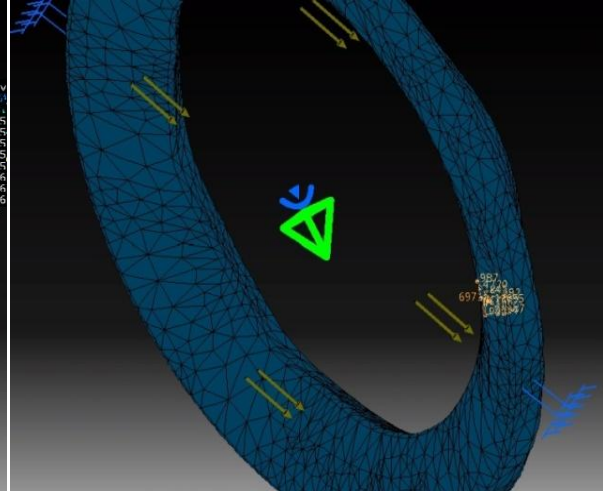


Figure 9

In Figure 8, the displacements of all the points have been shown including the one with maximum displacement. The red arrows (north & west region of the screenshot) represent the constraints placed on the outer race and the yellow arrow represents the radial load applied on the inner race. In Figure 9, FEM analysis with a temperature field of 47 degree Celsius has been done. So the finite element analysis of all the Bearings has given us the maximum displacement and maximum stress developed on the bearing surface. This also provides us with the coordinate vector of the point/circular ring on circumference/face of the bearing which has the maximum displacement. We will now relate the coordinate vector along with the three initial factors with the bearing life.

### 3.4 Improved Mathematical Models

These mathematical models are formed by incorporating position of maximum wear and then the maximum displacement. We will be taking into consideration the coordinate vector of the point/circular ring on bearing surface and forming a new mathematical model.

$$L = a_1 F^{b_1} N^{c_1} \mu^{d_1} r^{e_1} \quad (2)$$

We will also see the dependency of life when displacement is incorporated in the mathematical model in place of coordinate vector, in the second model.

$$L = a_2 F^{b_2} N^{c_2} \mu^{d_2} x^{f_2} \quad (3)$$

| Sr. No. | Force (N) | Speed (rpm) | Lubrication (kg/m-s) | Life (hours) | Radial Vector (mm) | Displacement (mm 10 <sup>-4</sup> ) |
|---------|-----------|-------------|----------------------|--------------|--------------------|-------------------------------------|
| 1       | 40000     | 800         | Grease               | 109280       | 217                | 5.6                                 |
| 2       | 45000     | 1000        | Oil                  | 61400        | 223                | 6.21                                |
| 3       | 30000     | 2000        | Oil                  | 103600       | 203.2              | 4.5                                 |

|   |       |      |        |       |       |      |
|---|-------|------|--------|-------|-------|------|
| 4 | 50000 | 600  | Grease | 74600 | 229.7 | 6.9  |
| 5 | 35000 | 1800 | Oil    | 72500 | 212   | 4.84 |

Table 5: New data set for Cylindrical Bearings.

| Sr. No. | Force (N) | Speed (rpm) | Lubrication (kg/m-s) | Life (hours) | Radial Vector (mm) | Displacement (mm $10^{-5}$ ) |
|---------|-----------|-------------|----------------------|--------------|--------------------|------------------------------|
| 1       | 1200      | 6500        | Oil                  | 92150        | 54                 | 3.1                          |
| 2       | 1800      | 4500        | Grease               | 39450        | 62.1               | 4.6                          |
| 3       | 1600      | 5700        | Grease               | 44300        | 60.3               | 4.08                         |
| 4       | 900       | 7000        | Oil                  | 202000       | 52.8               | 2.3                          |
| 5       | 2100      | 4000        | Grease               | 27950        | 64.5               | 5.4                          |

Table 6: New data set for Deep Groove Ball Bearings.

| Sr. No. | Equivalent Load(N) | Speed (rpm) | Lubrication (kb/m-s) | Life (hours) | Radial Vector (mm) | Displacement (mm) |
|---------|--------------------|-------------|----------------------|--------------|--------------------|-------------------|
| 1       | 1800               | 3000        | Oil                  | 57840        | 46                 | $5.47*10^{-4}$    |
| 2       | 2200               | 1500        | Grease               | 63350        | 47.5               | $1.31*10^{-3}$    |
| 3       | 2500               | 900         | Grease               | 71960        | 47.8               | $1.3*10^{-3}$     |
| 4       | 1500               | 3000        | Oil                  | 99340        | 45.4               | $5.62*10^{-4}$    |
| 5       | 2800               | 600         | Grease               | 76840        | 48.5               | $1.28*10^{-3}$    |

Table 7: New data set for Thrust Bearings.

We now have the new data set in Table 1 to Table 7 for all the bearings, whose values we have taken from CATIA FEM Analysis. We have found the values of variables to investigate more which has been given in Table 8 and Table 9.

| Variables      | Cylindrical Roller Bearings | Deep Groove Ball Bearings | Thrust Bearing   |
|----------------|-----------------------------|---------------------------|------------------|
| a <sub>1</sub> | $6.0610*10^{21}$            | $1.0497*10^{18}$          | $9.3316*10^{17}$ |
| b <sub>1</sub> | -2.9878                     | -3.0002                   | -2.9604          |
| c <sub>1</sub> | -0.9984                     | -1.0006                   | -0.9915          |
| d <sub>1</sub> | $9.5694*10^{-4}$            | $2.5082*10^{-4}$          | $2.1473*10^{-4}$ |
| e <sub>1</sub> | -0.0406                     | -0.0017                   | -0.0740          |

Table 8: Data set for Variables for Eq. (2).

| Variables      | Cylindrical Roller Bearings | Deep Groove Ball Bearings | Thrust Bearing    |
|----------------|-----------------------------|---------------------------|-------------------|
| a <sub>2</sub> | $5.5623*10^{21}$            | $6.0741*10^{17}$          | $7.3867*10^{17}$  |
| b <sub>2</sub> | -2.9999                     | -2.9691                   | -2.9677           |
| c <sub>2</sub> | -1.0005                     | -1.0001                   | -0.9918           |
| d <sub>2</sub> | $-2.6565*10^{-4}$           | $-8.7737*10^{-4}$         | $-2.5815*10^{-4}$ |
| f <sub>2</sub> | -0.0010                     | -0.0300                   | $-8.1357*10^{-4}$ |

Table 9: Data set for Variables for Eq. (3).

For Analysis Lets take Cylinder Bearings and we have seen that for a constant Force 40 KN and 800 rpm with Grease as the lubricant, variation of Maximum displacement and Life has been plotted in Figure 10. In Figure 11, combined effect of decreasing Maximum displacement and decreasing Viscosity has been shown with which Life increases.

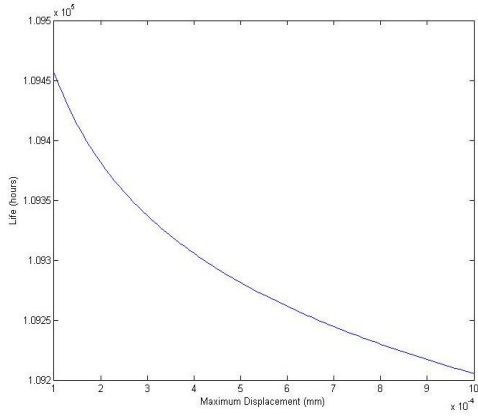


Figure 10

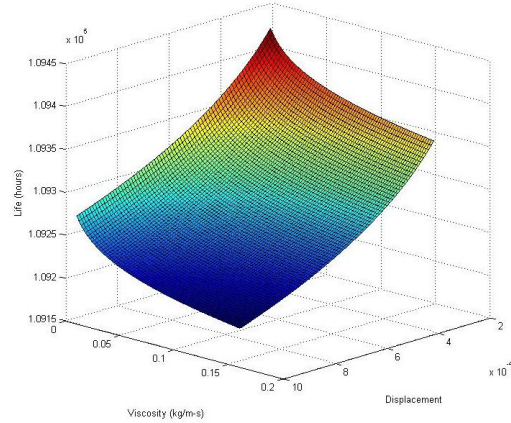


Figure 11

As expected the life of the bearing will increase if the maximum displacement is brought down. Thus we can conclude at this stage that the areas/points of maximum displacement are the ones that we need to focus on as the indentations on those points due to the static loading condition need to be reduced to a feasible extent. We require an optimal solution for our problem as there cannot be a single way out to improve the life of the bearing. For very less displacement and very less viscosity, Improvement in operating life can be done. We have to see how displacement can be decreased, for this we will make new mathematical model using the data from Table 1 to Table 7 which will show dependency of displacement on Force, Speed and viscosity as in Eq. (4) and Values of variables are founded and given in Table 10.

$$x = a_3 F^{b_3} N^{c_3} \mu^{d_3} \tag{4}$$

| Variables | Cylindrical Roller Bearings | Deep Groove Ball Bearings | Thrust Bearing           |
|-----------|-----------------------------|---------------------------|--------------------------|
| $a_3$     | $4.3976 \times 10^{-11}$    | $1.9808 \times 10^{-23}$  | $4.1820 \times 10^{-10}$ |
| $b_3$     | 1.3689                      | 2.9228                    | 1.8172                   |
| $c_3$     | 0.3361                      | 2.4634                    | 0.4698                   |
| $d_3$     | 0.1799                      | 0.1562                    | 1.1300                   |

Table 10: Data set for Variables for Eq. (4).

We will now plot some graphs to show effect of decreasing the viscosity on Maximum Displacement for three cases. Figure 12 is the case for Cylindrical Bearings, Figure 13 for Deep Grooves and Figure 14 for Thrust Bearings. The plots prove that maximum indentation depth/displacement decreases with Viscosity. So now we know with very less displacement, and very less viscous lubricant Life will be increased.

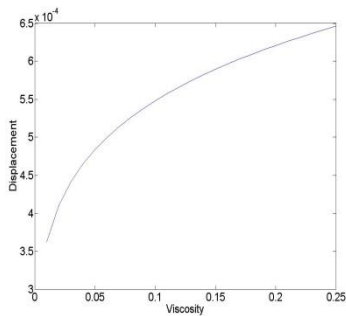


Figure 12

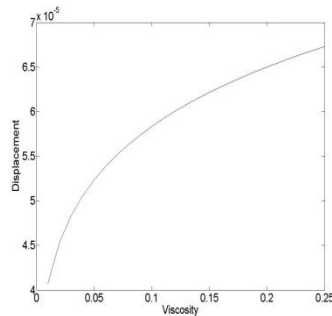


Figure 13

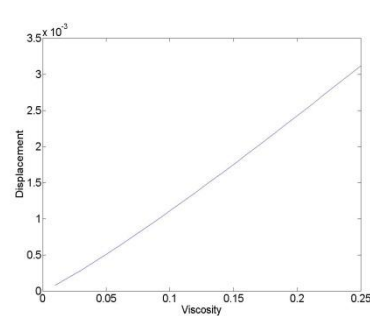


Figure 14



#### 4 CONCLUSIONS

- With very low maximum wear and very less viscosity high operating life can be achieved. The normal predicted rating life can be reached to greater extent even if we do not fully use the lubricants with low viscosity, but use them judiciously and carefully at critical instances and areas.
- With application of our new mathematical model, we can find the radial vector of the region with maximum wear/displacement. After some usage of time, proper consideration can be given to that region by using a lubricant of very low viscosity. As the oil with very less viscosity is generally expensive so it should be used for the region with maximum displacement to improve the general usage life.
- The graphs plotted for the final/third mathematical models for each type of the bearing have quite unanimously concluded that regardless of type of bearing, the bearing will have an increase in expected life with a lower viscosity lubricant. But what is also interesting to note is that the displacement increases almost linearly for the thrust bearings, whereas in ball bearings the decrease is quite sharp after a particular viscosity is reached (.002 kg/m-s). Thus, a critical point is formed beyond which a hypothetical scenario arises for the deep groove ball bearings.
- We can conclude firmly that the important factor/result to focus on remains the maximum displacement and not the stress induced as the maximum stress induced in every case is way below the yield strength for any general bearing material. Thus contradictory to conventional design for strength, here we design/choose the lubricant keeping in mind the maximum displacement as this is what causes high volume abrasion and deformation of substantial magnitude.

#### 5 RECOMMENDATIONS FROM OUR WORK

- For Cylindrical Bearings, We have three decent replacements/add on to the Lubricants. We have Carbon Tetrachloride  $\text{CCl}_4$  whose boiling point is 77 degree Celsius and its viscosity  $0.975 \times 10^{-3}$  kg/m-s. Others are Carbon disulphide and Bromine water. But carbon disulfide is weak as a lubricant, thus the main purpose of lubrication diminishes. Also it is very toxic and poisonous, thus any amount of exposure to it or its leakage can be fatal to the operators. The option of bromine water is highly impractical as it is difficult to apply in high load conditions.
- For thrust bearings, which generally bear axial loads Soyabean Oil layer can be coated to form a protection mechanism on the maximum displacement region. It is practically feasible and can also be used aplenty. Soyabean oil viscosity is 0.031kg/m-s
- Deep groove ball bearings with a working temperature between 40 and 60 degree Celsius, Boron nitride coating is a very good option for increasing the life of the bearings. Grease - MULTEMP SRL with viscosity .03 kg/m-s can also be used.

Now, Let us take an arbitrary case and examine the overall change on life of the bearing if the suggested lubricant is applied for the duration. For Deep groove ball bearing with case 1200 N and 6500 rpm, Viscosity 0.03 kg/m-s is used and using our model to find the maximum displacement we obtain reduced maximum displacement  $2.8283 \times 10^{-5}$  mm and Life comes out to be 92450 hours and Earlier Life was 92150. Increase is 300 Hours with an increase of .36%.

The above puts forward the fact that the change in overall rating life is around (.3% to 1%), thus the judicious use of special lubrication must be carried out. Hence, the usage of special low viscosity lubricants at maximum wear/displacement areas is a better option than completely replacing the original lubricant. So it is the optimal solution to put some special coating on those special areas, which helps us in prolonging the life of the bearing and is also economically and technically feasible. In case we use the suggested lubricants for the full life cycle we get an increase in life which is not colossal and also the properties like hydrodynamic film thickness are neglected to quite an extent. Thus the best option is to sequentially reduce the maximum displacement at the critical points/circular rings on the bearing skin. This will ensure that the bearing lives up to a large proportion of its rating life and also not much money and technical finiteness gets wasted.

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