An Investigation on Grease as Lubricant for Journal Bearing Operating Under Mixed/Boundry Regime

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***Abstract*— The purpose of this paper is to examine the performance parameters examine the tribological characteristics and issues related to journal bearings operating under boundary and mixed lubrication conditions.Journal bearings are commonly used support a rotating shaft. It is inevitable that during the start-up and shutdown stages of the rotating shaft, boundary and mixed lubrication conditions will produced and allow direct contact between the bearing and journal. It is necessary to study the influence of speed, viscosity, load, temperature, frictional torque and coefficient of friction on the performance of bearing operating under such conditions. The wear is considerably more in boundary/ mixed lubrication regime as compared to bearing operating under hydrodynamic conditions. Hence it is necessary to measure the wear of journal bearing operating in boundary/mixed lubrication regimes experimentally and this is to be validated by theoretical wear model developed by Archards equation in combination with the bayer's equation. And also comparative study of Theoretical Model And Experimental Model.**

***Index Terms*— Mixed/ Boundary lubrication, Journal Bearing.**

I. Introduction

Journal bearings are widely applied in different rotating machineries. These bearings allow for transmission of large loads at mean speed of rotation. Hydrodynamic journal bearings are considered to be a vital component of all the rotating machinery. It is used to support radial loads under high speed operating conditions. They are frequently used in applications involving high loads and/or high speeds between two surfaces that have relative motion. Journal bearings are specific to surfaces that mate cylindrically with the applied load in the radial direction. In the study of journal bearings many aspects of engineering are present. Stress analysis, fluid dynamics, instrumentation, vibration, material properties, thermodynamics, and heat transfer are some of the common subjects encountered in understanding hydrodynamic bearings. [10]

There are two types of bearing rolling element type and fluid film type. Fluid film bearing further also classified as follows.

1. Based on type of load carried.
2. Based on type of lubrication.
3. Based on lubrication mechanism.

When a journal bearing operates under boundary lubrication, the sliding surfaces of the bearing and shaft are practically in direct contact and friction is at its highest level. Lower friction levels are achieved through the use of mixed lubrication, where the sliding surfaces are partially separated by the lubricant, and of hydrodynamic lubrication, where the sliding surfaces are completely separated by the lubricant. [5]

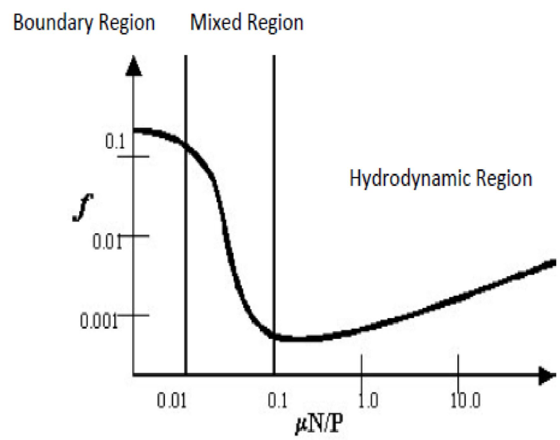


Fig.1 : Stribeck Curve

Corresponding to these regions Stribeck Curve is shown in Figure. The curve represents the minimum value of friction between full fluid separation and direct asperity contact of two surfaces. The friction is plotted as a function of a lubrication parameter ηN/P, where η is the dynamic viscosity, N is the shaft speed and P is the external load. The highest friction condition occurs in the boundary lubrication region, which represents significant or complete asperity contact between the two surfaces. On the other hand, the mixed lubrication region represents partial load support from the lubricating fluid and partial load support from asperity contact. Finally, the hydrodynamic lubrication region represents a load fully supported by the lubricating fluid with no asperity contact.

Significant wear of journal bearings can occur during boundary and mixed lubrication conditions when there is not enough pressure generated in the lubricant to carry the load. These conditions occur during startup, shutdown, and low speeds of shaft rotation. Excessive wear of journal bearings will degrade their performance over time and can result in bearing failure. Failure of a journal bearing can result in significant production losses and maintenance costs to companies that rely on them within their machinery. Research indicates that among other factors, bearing wear rate is dependent upon frequency of starts and stops, surface velocity, load, and material hardness. [6]

Journal Bearing

A journal bearing is a journal (such as a shaft) which rotates within a supporting sleeve or shell. Journal bearings are used in the rotation of the journal to pressurize a lubricant which is supplied to the bearing to eliminate surface-to-surface contact and bear the external load. The common design methodology is based on empirical data and has worked very well historically because the market and governments have accepted that bearings in industrial equipment need frequent lubrication and exchange of worn parts. Legal and market requirements will demand soon in lower environmental impact and increased machine efficiency. These requirements call for better methods for grease lubricated journal bearings. The goal of the outlined work is to develop better method for grease lubricated journal bearing used in heavy duty industrial equipment's or machinery operating on the low speed in order to prolong life and lubrication intervals.

Tribology spans several scientific fields including mechanical engineering, material science, physics and chemistry but its importance in everyday life is often overlooked. In order to study of the journal bearing we have to focus mainly on the contact between the surfaces and region in which the contact occurs such as boundary, mixed and hydrodynamic regimes.

Lubricants

As with material science, lubricant technology has progressed dramatically since the original development of mineral oil based lubricants in the 19th century. In the face of new developments such as custom synthesized lubricants and additives, many industries continue to operate with lubricants made to old specifications. New understanding of chemical processes provides the potential to tailor make lubricants for optimum performance in specific applications thus improving efficiency and machine safety. Equally important to developing and selecting the optimum lubricant for a given application is adapting the operation of the machine to best match the lubricant's characteristics. Opportunities to dramatically change journal bearing performance by changing lubricant have not been previously available thus a firm understanding of the differences that can be expected from application of new lubricants needs to be developed.

Typical lubricants available and used today are based either on classic high quality petroleum base stocks or some form of synthetic base stock such as ester, poly-alkaline-glycol, or poly-alpha-olefin. Additionally, any numbers of additives are blended into the base oil to improve specific characteristics such as anti-oxidation additives which extend lubricant life, extreme pressure additives to protect surfaces in high pressure contacts oranti-foaming additives to reduce the lubricant's tendency to trap air bubbles. Most important to this work are additives which improve the viscosity index(VI) of the lubricant. A lubricant's VI describes how the viscosity of a lubricant changes with variations in temperature.

- It increases stiffness of asphalt& mortar matrix.
- It helps to reduce drain-down in the mix which improves the longevity of the mix by using required amount of asphalt in the mix.
- It maintains adequate amount of void in the mix.

I. EXPERIMENTAL PROCEDURE

Experimental Set-Up

This provides a modern investigation of journal bearing design, testing, and application. Countless simulations have shown the potential of changes in bearing technology, but for any change in design or operation to be adopted by a greater audience, strong support and proof of viability based in experiments are necessary. Because full scale experiments are expensive and inflexible, experiments should be focused at the small scale.

Fig.2 : Journal Bearing Test Rig

Hence in order to develop the journal bearing test rig we have to focus on the various parameters such as ,

Selection Of Grease

The selection of grease for a specific application depends on five factors: speed, load, size, temperature range, and any grease feed system. For average conditions of speed, load, and size with no feed system, NLGI no. 2 grease would be the normal choice, and such grease with a mineral-grease base is sometimes known as multipurpose grease.

Speed - For high speeds, stiffer grease, NLGI no. 3, should be used except in plain bearings, where no. 2 would usually be hard enough. For lower speeds, softer grease such as no. 1 or no. 0 should be used.

Load - For high loads, it may be advantageous to use EP additives or molybdenum disulfide. Because higher loads will lead to higher power consumption and therefore higher temperature, a stiffer grease such as no. 3 or a synthetic-base grease may help.

Size - For large systems, use a stiffer grease, no. 3 or no. 4. For very small systems, use softer grease, such as no. 1 or 0.

Temperature range - The drop point should be higher than the maximum predicted operating temperature. For sustained operation at higher temperatures, synthetic-base grease may be necessary. For very high temperatures, about 230°C, one of the very expensive fluorocarbon greases may be required.
Feed systems- If the grease is to be supplied through a centralized system, usually it is desirable to use one grade softer than would otherwise be chosen (i.e., use a no. 0 instead of a no. 1 or a no. 00 instead of a no. 0). Occasionally a particular grease will be found unsuitable for a centralized feed because separation occurs and the lines become plugged with thickener, but this problem is now becoming less common.

Selection of motor

The selection of motor is mainly based on the speed of the journal required and the load applied on the bearing. By using this parameters we can calculate the power required and from this we can do the selection of motor.

$$P = \frac{2\pi NT}{60 \times 1000}$$

Consider speed varies from 10 – 50 rpm, radius of journal 0.020m and Load 500N

Table 1:Operating Conditions

Load	Speed
20	10
25	12
30	14
35	16
40	19

From the above parameters we can select 1.25hp motor.

1. L/D ratio

The L/D ratio is selected

As the length and diameter of the bearing is same and using these we can select the values of eccentricity, Somerfield number.

2. Loading Conditions
3. Operating Speed
4. Selection of Thermocouples

Six thermocouples are used to measure the temperature in the range of 40^0C to 150^0 C and these are at 60^0 on the bearing surface.

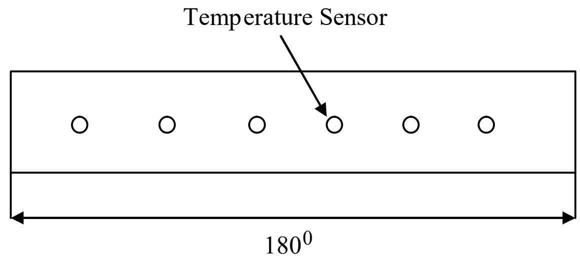


Fig.3 : Temperature Sensors Location

5. Torque Measurement

Torque meter is to be used for the torque measurement.

6. Sommerfeld Number

Using L/D ratio sommerfeld number selected is 0.001 to 0.0026

Corresponding ε is 0.99

Viscosity 0.10025pa.s

Clearance is 0.25 which is 0.1% of journal radius

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N}{p}\right)$$

Load is 2891Using scale in for 27rpm load is 373N And hence S is 0.0068.

Typically, a test rig consists of a frame, a bearing unit and loading, drive, lubrication, control and measuring systems:-

Support Frame:

The main components of the test rig are installed in the frame. The frame is generally relatively rigid to avoid disturbing deformations and vibrations.

Bearing Unit:

The bearing unit consists of the bearing, housing, shaft and supporting bearings. A large high precision roller bearing with both radial and axial load carrying capacity is a common supporting bearing type used in low and medium speed applications. The material used for the bearing is Tin Bronze. The shaft is typically supported on both sides of the bearing, but single side support solutions have also been used. The experimental device consists of a 100 mm diameter journal bearing, whose radial clearance and length was varied as a function of the cases considered. The shaft is supported by an assembly consisting of high precision rolling element bearings, driven by available speed DC motor, the latter being controlled by means of a Lab view program and National Instrument data acquisition boards.

Loading System:

The loading system generates the bearing load. There are test rigs with dynamic and static loading systems. Hydraulic and pneumatic cylinders are widely used in loading systems. A test rig with dead weights is used for bearing loading in tests on journal bearings lubricated with grease. A high-speed data acquisition system measured the start-up torque with a sampling rate of 1000 Hz. This high sampling rate permitted a maximum torque value C_{max} to be measured at startup. This is converted to a friction coefficient value f using the following formula: [1]

$$f = \frac{C_{max}}{WR}$$

Where W is the static applied load and R the radius of the bearing.

Drive System:

The drive system drives the shaft and consists of a power unit and a transmission unit. Because of relatively low power losses in the bearings, high output from the driving unit is seldom required. An electric motor is a common power unit type because of its high applicability to a laboratory environment. Belts and gearboxes are used for the transmission.

Lubrication System:

The lubrication system lubricates the bearing. Supporting bearings are lubricated separately by oil or grease.

Control System:

The control system controls the main operations of the test rig. The measuring system measures and records the data required for the control and analysis of the case that is being studied.

OBTAINING EXPERIMENTAL OPERATING CONDITIONS

The operating condition of the journal bearing is to be decided for heavily loaded slow speed journal bearing. The sommerfeld number of such heavily loaded applications is in the range of 0.0010-0.0026. The eccentricity ratio is 0.99. SM120 is the commonly used lubricant for such heavily loaded and slow speed applications. A radial clearance is maintained at 0.25mm (0.1% of journal radius) but practically it is not possible hence it is increased to 0.6.

Dimensions obtained from the journal bearing used in sugar factory application is mentioned as below

d = 380mm
r = 190mm
l = 495mm
c = 200×10^{-3}mm
W = 2.89×10^{9}N
μ = 78Cst = 0.078×10^{-6} m^2/s
n_s = 6 rpm

$$S = \left(\frac{r}{c}\right)^2 \times \frac{\mu n_s}{p}$$

Where,

r is the radius of the journal = 30mm
c is the clearance = 0.6
μ is the viscosity of the lubricant
n_s is speed of the journal
p is the bearing pressure = $W/_{A}$, N/m^2

Using above equation sommerfeld number obtained is

S=0.001695

Which is in the range of sommerfeld number for heavily loaded applications.

Hence selecting the maximum value of the sommerfeld number as 0.0026 for heavy load applications and maintaining length to diameter ratio as 1 and the viscosity of the lubricant SM120 is 64Cst that is 0.064×10^{-6} $m^2/s$and from that operating conditions obtained which are as below

Table 2 :Operating Conditions

Sommerfeld Number	Load	Speed
0.0026	20	10
0.0026	25	12
0.0026	30	14-15
0.0026	35	16-17
0.0026	40	19

THEROTICAL WEAR MODEL

The Wear factor of the bearing is to be calculated by using Pin on Disk Experiment

Pin on Disc Experimental Setup to Determine Specific Wear Rate K

Experimental set up which is available in S.N.D. College of Engineering, Nashik is as shown in following Figure Using Tribometer TR-20LE-PHM400 readings of wear and frictional force are taken.

Fig.4: Photograph of Pin on Disc Set-up

On the Controller the acquired test parameters like wear, frictional force, speed and temperature are displayed, the same values are displayed on PC screen with graph.

Obtaining Operating Conditions

In sugar cane mill bearing hydrostatic lubrication is used. Hydrostatic oil pressure varies from 775 psi to 900 psi depending on the load on mills. This hydraulic pressure directly acts on the piston of diameter 30 mm of the cylindrical chamber of hydrostatic oil pressure bearing.

So these pressure can be considered as directly acting on the test pins of Φ10mm.

Minimum Load on the test pin,

$P_{min} = \pi/_4 \times 10^2 \times 0.20$

$= 16.085$ kg

Maximum Load on the test pin,

$P_{max} = \pi/_4 \times 10^2 \times 0.23$

$= 18.6768$ kg

Therefore considering the masses available with the testing unit TR- 20LE, Maximum load to be applied on the test pins through the lever and the pulley arrangement are selected in the range of 16.75 kg i. e. the normal load of 164.31N to 18.758 kg i.e. the normal load of 184.02N is to be taken for the experimentation.

Sugar cane milling roller rotates slowly from 4.5 rpm to 6 rpm depending load on it. Therefore sliding velocity of journal of Φ380mm in hydrostatic bearing is:

Sliding Velocity V in m/sec = $\pi DN/_{600}$

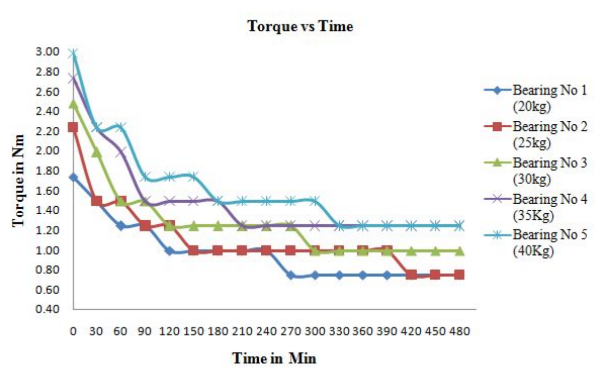
Where, D is track diameter

N is speed in rpm

$$V_{min} = \pi \times 380 \times 4.5/_{60000} = 0.09\ m/s$$

$$V_{max} = \pi \times 380 \times 6/_{60000} = 0.12\ m/s$$

So the minimum rotary speed of 3.5 rpm i. e. sliding velocity 0.09 m/s and maximum rotary speed of 6 rpm i. e. sliding velocity 0.12 m/s is to be taken for the experimentation. Hence operating conditions for pin on disc experiment as shown in Table



Graph 1:Torque Vs Time

II. EXPERIMENTAL RESULTS

Experimental results obtained by testing the bearing on journal bearing setup running for 08 hours for different loading conditions.

Table :Experimental Results

Sr. No.	**Load (kg)**	**Weight of Bearing Before Testing (gm)**	**Weight of Bearing After Testing (gm)**	**Wear (gm)**
1.	20	2251.35	2232.7	18.64
2.	25	2264.33	2235.7	28.63
3.	30	2299.48	2254.4	45.08
4.	35	2298.76	2242.7	56.06
5.	40	2307.20	2236.0	71.20

Therotical Wear Model Results

Therotical wear model for journal bearing operating in boundary/ mixed lubrication region is used to find out the Specific wear rate that is wear factor K based on pin on disc experiment and this value of k is used to determine the volume of wear and from that mass of wear obtained in Table Mat lab program is used to find out the mass of wear for different loading condition.

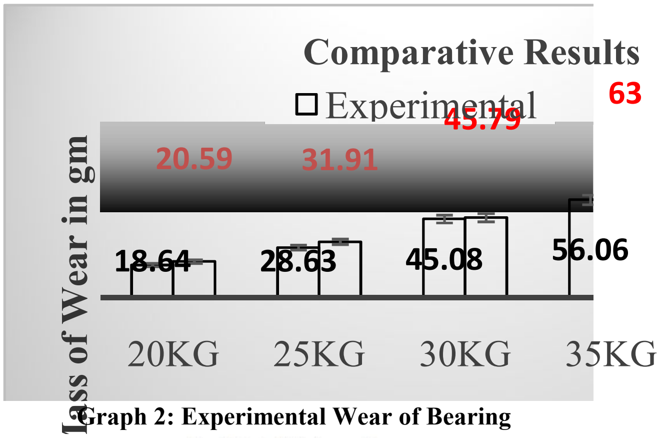
Table 3: Therotical Results

Sr. No.	Load (kg)	Therotical Wear Model Mass of Wear (gm)
1.	20	20.59
2.	25	31.91
3.	30	45.79
4.	35	63.00
5.	40	80.99

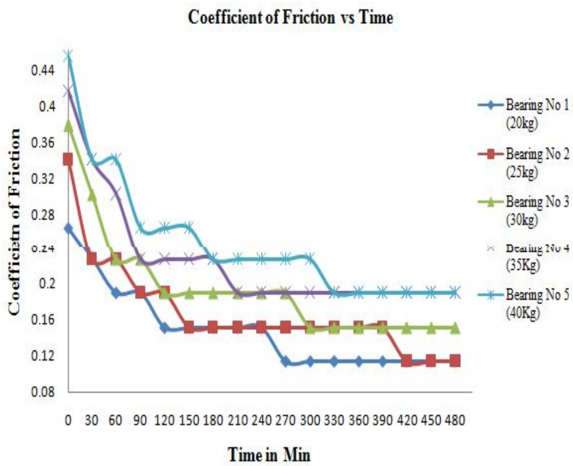
Table shows the comparison between theoretical and experimental (calculated by weight loss of the bearing from test rig) values of Mass of Wear and indicates the percentage of error in experimental and therotical wear model.

Table 4: Pin on Disc Operating Conditions

	Load (N)	**Sliding Velocity (m/s)**	**Speed (rpm)**	**Track Daimeter (mm)**
Pin 1	164.31	0.09	49.11	35
Pin 2	174.12	0.11	60.02	35
Pin 3	184.02	0.12	65.48	35
Pin 4	164.31	0.09	42.97	40
Pin 5	174.12	0.11	52.52	40
Pin 6	184.02	0.12	57.52	40
Pin 7	164.31	0.09	38.19	45
Pin 8	174.12	0.11	46.68	45
Pin 9	184.02	0.12	50.92	45



Graph 2: Experimental Wear of Bearing



Graph 3:Coefficient Of friction Vs Time

III. CONCLUSION

- Journal bearing experimental set up is constructed to test five bearings for five different loading conditions.
- Bearing is to be tested for evaluation of performance parameters such as temperature, coefficient of friction and frictional torque verses time.
- Coefficient of friction and frictional torque of the bearing shows that it is more at starting and then it decreases but after running the bearing as per operating conditions after 04 hours it remains constant. This may be due to the rise in temperature of lubricating oil which decreases the viscosity and coefficient of friction. As load increases coefficient of friction also increases.

REFERENCES

1. "Experimental measurement of the friction torque on hydrodynamic plain journal bearings during start-up" by J. Bouyer, M. Fillon in Tribology International 44 (2011) 772–781.
2. "Numerical Unsteady Analysis of Thin Film Lubricated Journal Bearing" by K. M. Panday, P. L. Choudhury, and N. P. Kumar Vol. 4, No. 2, April 2012.
3. "Synthetic lubricants in hydrodynamic journal bearings: experimental results" by G.F. Simmons and S.B. Glavatskih
4. "Design, testing and Analysis of Journal Bearings for Construction Equipement" by Henrik Strand
5. "Journal Bearing Design Lubrication and operation for Enhanced Performance" by Gregory Simmons
6. "Oil film pressure in hydrodynamic journal bearings" by Antti Valkonen in Helsinki University of Technology, Dec.2009.
7. "Tribology of Journal Bearings Subjected to Boundary and Mixed Lubrication" by Steve Pickering in Mechanics of Contact and Lubrication ME5656, 2011.
8. "Experimental Analysis of Pressure Distribution of Hydrodynamic Journal Bearing: A Parametric Study" by Chaitanya Desai and Dilip Patel in ICME05AM-30, 2005.
9. "Experimental Study of Temperature Profile in a Journal Bearing" by S. Kasolang, M. Ali Ahmad in 2012.
10. "CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant" by K.P. Gertzos, P.G. Nikolakopoulos, C.A. Papadopoulos in Tribology International 41 (2008).