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Experimental Analysis of Oil Film Pressure and Temperature on EN31 Alloy Steel Journal Bearing

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Abstract The design and analysis of hydrodynamic journal bearings has a great attention to the engineers. Hydrodynamic lubrication is the most common method of lubrication of journal bearing. Emphasis has been given to design those bearings so as to avoid boundary lubrication between the bearing surfaces. To design these elements, the characteristics like load-carrying capacity, maximum pressure, eccentricity, lubricant viscosity and so on are to be predicted accurately. These parameters can be determined if the pressure within the clearance space between contact surfaces is known. In this method as the journal rotates, it takes a slightly eccentric position relative to the bearing. The eccentric rotation of the journal in the bearing acts somewhat like a rotary pump and generates a relatively high hydrodynamic pressure in the converging zone. The hydrodynamic pressure for a properly designed bearing is responsible for supporting the journal without allowing it to come in contact with bearing. This study deals with the development of suitable laboratory test rig, which can be helpful in determining the load capacity, pressure distribution of journal bearing at different speed, location of maximum film pressure and effect of lubricants on bearing performance. This paper deals with an experimental study of oil film pressure and temperature responses for journal parameters on EN31 alloy steel journal. The study on journal bearing comes under engineering tribology and as known that small improvement in the field of Tribology leads to better usage of energy.

Keywords: *EN31 alloys steel journal, load carrying capacity, lubricant and journal speed*

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1. Introduction

Hydrodynamic journal bearings have received great attention from practical and analytical engineers during the past few decades. The rapid growth of journal bearing technology is mainly due to its wide range of engineering applications such as precision machine tools, high speed aircraft, nuclear reactors, textile spindles, pumps, compressors, fans, turbines and generators widely used in industries. A journal bearing is the most common hydrodynamic bearing in which, a circular shaft, called the journal, is made to rotate in a fixed sleeve is called the bearing. The bearing and the journal operates with a small radial clearance of the order of 1/1000th of the journal radius. A journal bearing is a journal (such as a shaft) which rotates within a supporting sleeve or shell [1]. Hydrodynamic journal bearings use 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 as seen in Figure 1. The relative motion between shaft and journal bearing results in a fluid film gap geometry allowing a hydrodynamic pressure build up. The resultant force F_h is in equilibrium with the external load F_e. Dependent on load, rotational speed and viscosity respectively temperature the operational point of a journal bearing can be situated in hydrodynamic, mixed or boundary friction regime. This relation can be visualized with the help of a Stribeck's curve, see Figure 2. 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 μN/P, where μ is the dynamic viscosity, N speed of journal and P is unit bearing pressure. 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 hydrodynamic lubrication region represents a load fully supported by the lubricating fluid with no asperity contact. Finally, the mixed lubrication region represents partial load support from the lubricating fluid and partial load support from 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 start up, shutdown, and low speeds of shaft rotation [1]. 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 [2].

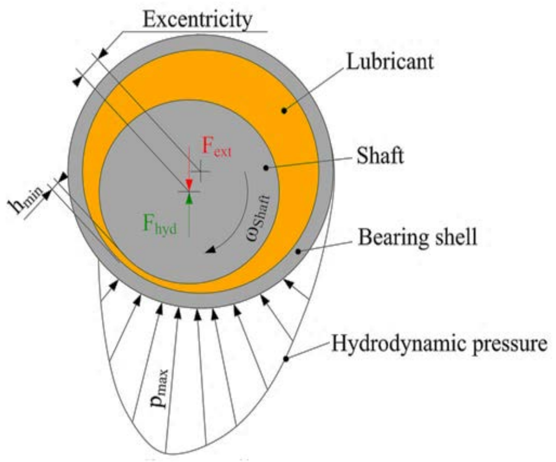

Figure 1. Hydrodynamic journal bearing

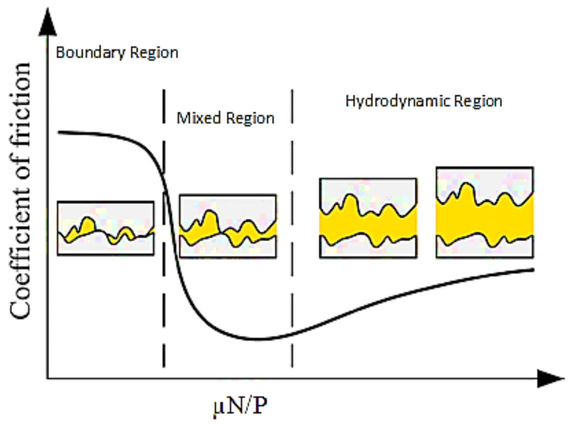

Figure 2. Principle Stribeck's curve

Actual developments, however, involve a reduction of the hydrodynamic carrying capacity resulting in lower fluid film thicknesses. Consequently surface asperities between shaft and bearing shell start to contact each other. In this case hydrodynamic journal bearings operate in mixed and boundary friction regime which is characterized by the coexistence of hydrodynamic and solid contact pressure. The consequences of solid contact are increased frictional losses and wear limiting life expectancy - making a numerical wear assessment necessary [3,4,5]. In most applications, journal bearing designs introduce lubrication fluid to decrease the friction between the two surfaces; however contact between the surfaces can still occur in the presence of lubrication [6]. The period of increased contact occurs most frequently during start-up, shut-down, and low speeds of the machine in which the bearing is used. As was previously discussed, these are known as boundary or mixed lubrication conditions.

When a bearing operates at high speed, the heat generated due to large shearing rates in the lubricant film raises its temperature which lowers the viscosity of the lubricant and in turn affects the performance characteristics. To obtain the realistic performance characteristics of the bearing, thermo-hydrodynamic (THD) analysis should be carried out. In literature, several THD studies have been reported. Most of these analyses used two dimensional energy equations to find the temperature distribution in the fluid film by neglecting the temperature variation in the axial direction and two dimensional Reynolds equations was used to obtain pressure distribution in the lubricant flow by neglecting the pressure variation across the film thickness. Kim Thomsen et al [7] gives a numerical simulation presented for the thermo-hydrodynamic self-lubrication aspect analysis of porous circular journal bearing of finite length with sealed ends. The results showed that the temperature influence on the journal bearings performance is important in some operating cases, and that a progressive reduction in the pressure distribution, in the load capacity and attitude angle is a consequence of the increasing permeability. Mukesh shahu et al [8] presented thermodynamic study of the 3 dimensional plain journals bearing using CFD. In this paper, author found out pressure distribution on journal surface not only circumferentially but also axially, with and without considering temperature effect. Amit Chauhan et al [9] have presented thermo-hydrodynamic analysis of plain journal bearing. During the analysis, deviation of pressure and temperature is considered on the fluid film. D. M. Nuruzzama et al [10] have calculated pressure distribution and load capacity of journal bearing by analytical method and finite element method. To check the validity, both the results were compared. During calculation isothermal analysis was considered. By comparing both the results it is identified that at low eccentricity ratio raises the dimensionless load steadily and rise with high eccentricity ratio. K. M. Panday et al [11] have done unsteady analysis for thin film lubricated journal bearing with different L/D ratios such as 0.25, 0.5, 1, 1.5, and 2. During the analysis, author observed maximum pressure present at minimum oil film thickness. Also they found out that shear stress on surface of bearing and journal is reduced with increase in L/D ratio, but the turbulent viscosity of lubricant rises with increase in L/D ratio.

The fluid film pressure and temperature distribution is one of the fundamental operating parameters to identify the operating conditions of journal bearing. The pressure distribution is crucial in load capacity estimation as well as dynamic analysis. In fluid film journal bearing, viscous shearing phenomenon occurs, that causes power loss and temperature rise. Rising temperatures lead to viscosity reduction of oil and bearing deformation. Hence it is needed to study pressure and temperature distribution in journal bearing. Journal-bearing performance characteristics, such as oil film pressure and temperature for both load and speed on EN31 alloy steel journal bearing is presented in the current work that comes under tribology and as known, these small improvements in tribology leads to better usage of energy.

2. Journal Theoretical Analysis

Lubrication theory for the dynamically loaded journal bearing is mathematically complex and, over the last few decades, several analytical approaches have been proposed. The multi grid techniques based on the Elrod algorithm [12] and the finite element methods [13] of analysis are among the most popular. The finite element methods are probably the most accurate and versatile, but tend to be very time consuming and require high level of knowledge, not accessible to the common designer and, so, remaining confined to research and development. Therefore, based on simplifying premises, engineers and designers prefer to use simpler and still accurate methods, such as the mobility method [14,15] and the impedance method [16,17]. In general, these approximate techniques, which belong to the category of rapid methods, are employed to perform analysis of simple journal bearings.

A. Governing equation:

The well-known Reynolds equation is used for finding the Pressure distribution in Journal Bearing. The non-dimensional form of the Reynolds equation for journal bearing considering Newtonian, laminar, incompressible fluid flow with no slip at boundaries and neglecting fluid inertia and curvature of bearing surfaces with pressure and viscosity assumed to be constant throughout the thickness of the film is expressed as

$$\frac{\partial}{\partial x}\left[h^3 \frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[h^3 \frac{\partial p}{\partial z}\right]=6\mu U \frac{dh}{dx}$$

Where h is the fluid film thickness, μ is the absolute fluid viscosity, p represents the film pressure, and U is the relative tangential velocity.

B. Pressure boundary conditions:

Pressure at bearing ends are taken as zero. Positive pressure during calculation is identified and negative pressure is taken as zero.

C. Pressure distribution:

Eccentricity plays a key role in varying the pressure in the bearing. Varying pressure is directly proportional to varying eccentricity. The maximum possible eccentricity is the radial clearance of the bearing. So the ratio of eccentricity to the clearance gives the eccentricity ratio. Eccentricity ratio can vary from 0 to 1. If the ratio is zero, then the shaft is exactly in the centre of the bearing sleeve. Also this indicates that there is no pressure and in the bearing. And if the eccentricity ratio is one, then the load on the bearing is maximum and there is contact between the shaft and the sleeve. The pressure around the journal in bearing considering long bearing approximation is expressed as;

$$P=\frac{\mu U r}{c^2}\left[\frac{6\varepsilon(\sin\theta)(2+\varepsilon\cos\theta)}{\left(2+\varepsilon^2\right)(1+\varepsilon\cos\theta)^2}\right]$$

Where μ = viscosity of the lubricant; U = velocity of the shaft; r = radius of the shaft; θ = 0 – 180°; c = clearance of the bearing for minimum tolerance; e = eccentricity of the bearing; ε = eccentricity ratio

Table 1. Pressure Distribution

Cases	L/D	ε	Load (N)	Pressure (Pa)
1	1	0.2	30	233
2	1	0.4	40	311
3	1	0.6	50	388

D. Stress distribution of journal bearing:

The bearing stress distribution has been calculated by considering the journal speed and bearing eccentricity ratio. It has been observed that journal speed and bearing eccentricity ratio increases the stress distribution of journal bearing. The force on the journal bearing is expressed as

$$F=\frac{\pi^2 * D^2 * L * N * \mu}{30 * c}$$

Where; F = Force (N); D = Diameter of journal bearing; μ = Co-efficient of friction; L= Length of journal bearing; N = speed in rpm; c=clearance.

Table 2. Stress Distribution

L/D	ε	RPM	Force (N)	Stress (N/mm^2)
1	0.2	600	2742493	213
1	0.2	800	3656657.3	284
1	0.2	1000	4570821.7	355
1	0.4	600	5169269.9	404
1	0.4	800	6892359.9	535
1	0.4	1000	8615449.9	664

3. Materials and Methodology

In present work, laboratory setup was developed to determine the maximum fluid film pressure and temperature distribution in the journal bearing, under certain load conditions. The bearing is made up of acrylic material of inner diameter 64mm while the journal is made up from the EN31 alloy steel of length and diameter 63.5mm. The radial clearance provided was 0.5mm. Material EN31 is a quality high carbon alloy steel which offers a high degree of hardness with compressive strength and abrasion resistance. This EN31 alloy steel journal is to be tested using lubricant SAE40 oil having kinematic viscosity of 15cP. The variable frequency drive has provided to adjust the speed of journal and to measure the voltage and current of the DC motor. This motor shaft is connected to the journal using coupling and bearing is mounted on the journal using gaskets and side plate to avoid the leakages. Ball bearing is provided to support the journal bearing assembly.

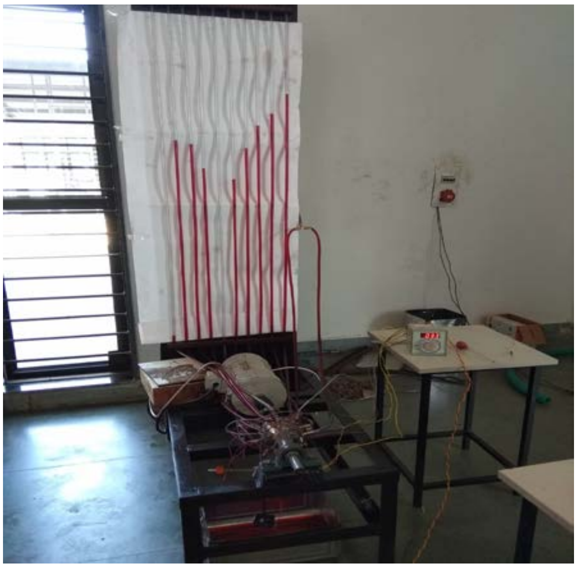

Figure 3. Experimental Test Rig Setup

Six K type thermocouples having outer diameter of 4 mm are provided on the bearing. One is attached at oil inlet and another five thermocouples are attached on hydrodynamic acrylic bearing at angle of 120 degree between three thermocouples (T_1,T_2,T_3) and 90 degree between two thermocouples (T_4,T_5) to measure the temperature.

Initially an analytical calculation was carried out, in order to determine the pressure distribution with necessary assumptions. The journal performance was tested for load capacity of 30, 40 and 50N with journal speed of 600,800 and 1000rpm.

4. Result and Discussion

The pressure and the temperature of the oil film have been obtained for journal bearing at different load capacity of 30, 40 and 50N for the oil under study at various journal speeds. The theoretical pressure distribution along the journal circumference at different speed and load has been shown in Figure 4. The effect of load and speed on the experimental pressure distribution and temperature distribution of the lubricating oil has represented in Figure 5 and Figure 6. The significance of the film thickness provides the accurate variation of the pressure profile along the bearing. The maximum value of pressure would be occurred at the point of minimum film thickness. The similar variation has been obtained in the pressure plots obtained experimentally. It has also been observed that the range of positive pressure increases with the increase in journal load.

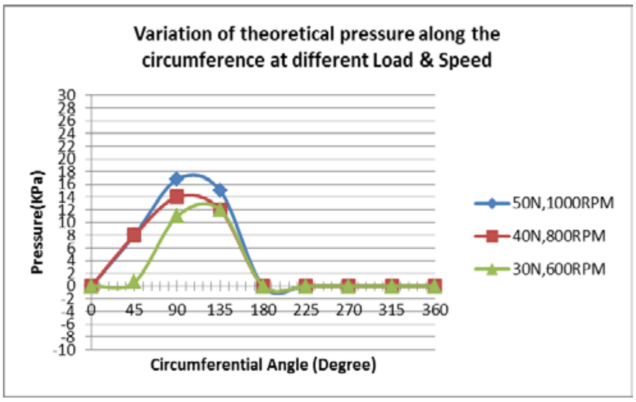

Figure 4. Theoretical pressure distribution along the circumference

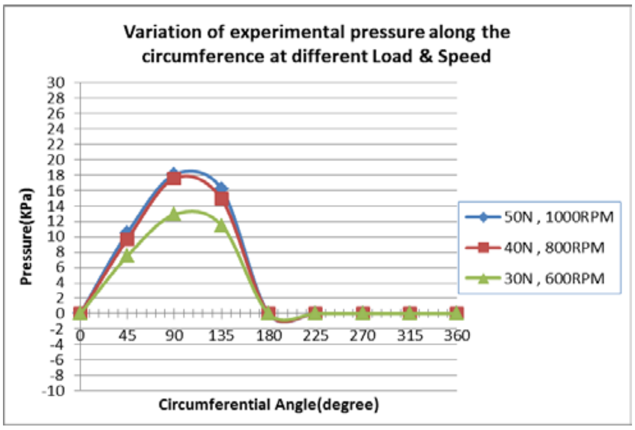

Figure 5. Experimental pressure distribution along the circumference

After testing the bearing for 6 hours temperature is to be measured at five different locations on the bearing and current is to be measured using VFD with an interval of 30 minutes. Figure 6 shows the oil temperature verses time, which shows that the temperature is gradually increasing with respect to time and then it remains constant at the end. As the load increases the temperature also increases.

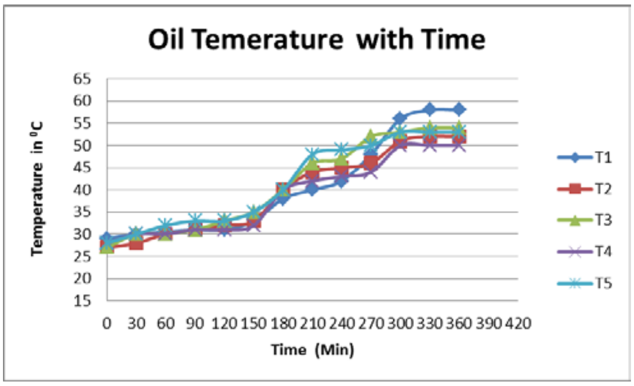

Figure 6. Oil temperature Vs Time

5. Conclusion

Experimental test setup has been developed to measure simultaneously both oil film pressure and temperature along the circumference of EN31 alloy steel journal bearing. The pressure and temperature has been measured with the direct contact type manometer and thermocouples fitted on the bearing. The following conclusions were made from the various conducted experiments during the study.

The thermal behaviour of journal bearing is affected significantly by speed and load. Frictional torque of the bearing shows that it is more at starting and then it decreases but after running the bearing at operating conditions for 6 hours it becomes 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. It has also been observed that the range of positive pressure increases with the increase in load. The friction resistance of the journal has been improved due to high degree of hardness for EN 31 alloy steel material used for journal bearing.

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