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Theoretical Investigation of Bearing Performance by Using Non- Newtonian Lubricant

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***Abstract**: In this paper, static performance characteristics of hydrodynamic journal bearing are analysed by using generalised Reynolds equations for lubricants with various non-linearity factors. Effect of increasing load have been studied for minimum fluid film thickness and maximum fluid film pressure. The non-Newtonian lubricant is assumed to follow the cubic shear stress law. It was observed that the value of maximum pressure increases while minimum fluid film thickness decreases with increase in load .*

1. Introduction

Hydrodynamic journal bearings are considered to be a vital component of all rotating machinery. The part which is enclosed by and rubs against the other is called journal and the part which encloses the journal is called bearing. The journal mainly consists of a stationary cylindrical body (sleeve) separated from a rotating shaft by a layer of lubricant [1-2]. Mostly the journal rotates in the fixed bearing but in a few cases both the journal and bearing are in motion, for example a crack pin and it's bearing in the connecting rod [3-4]. In some cases the journal is fixed and the bearing rotates as in a hoisting drum or a loose pulley [5]. Hydrodynamic journal bearing are found widely application in reciprocating machinery such as compressors, internal combustion engines and other industrial processing machinery. Heavy Load carrying capacity and less maintenance make them advantageous over other type of bearing. High speed of rotation causes the considerable rise in the temperature of the lubricant which significantly affects the performance of the bearing. Many researchers [2-8] investigated the bearing performance based on a thermo hydrodynamic (THD) analysis, which further requires simultaneous solution of the complex equations of flow of lubricant, the energy equation for the lubricant flow and the heat conduction equations in the bearing and the shaft. Previously, the researchers investigate the performance of the lubricant by solving the Reynolds Equation through Finite Difference Method approach [9-12].

In the present work, the surface of the bearing was modeled by taking journal diameter of 45 mm with 90mm width of bearing. The generalized Reynolds equation governing the flow of lubricant, has been used to determine static performance characteristics. The non-Newtonian lubricant is assumed to follow the cubic shear stress law. The maximum pressure and minimum fluid film thickness was measured for wide range of non-linearity factor. Newton-Rapson's method was used to measure shear stress. The non-Newtonian lubricant is assumed to follow the cubic shear stress law.

2. Theoretical Measurement

Most of the non-Newtonian oils follow the behavior, which is represented by cubic shear law. Many researchers have used the cubic shear law model and power law model for the analysis of non-Newtonian behavior of the lubricants in their studies [2,6,12]. The constitutive equation for cubic shear law is described in non-dimensional form as:

$$\bar{\tau} + \bar{K}\bar{\tau}^3 = \bar{\dot{\gamma}} \quad (1)$$

Here, $\bar{K}$ is known as non-linearity factor. The viscosity of non-Newtonian lubricant is described by the apparent viscosity ($\tilde{\mu}_a$) and is defined as the function of shear strain ($\bar{\dot{\gamma}}$).

$$\tilde{\mu}_a = \bar{\tau} / \bar{\dot{\gamma}}$$

The shear strain rate $\bar{\dot{\gamma}}$ is computed, and the corresponding equivalent shear stress $\bar{\tau}$ is obtained using Newton-Rapson's method. The apparent viscosity $\tilde{\mu}_a$ is computed by using above equation . The generalized Reynolds equation governing the laminar flow of incompressible lubricant between the clearance space of journal and bearing considering variable viscosity and usual assumptions is written as below:

$$\frac{\partial}{\partial \alpha}\left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \alpha}\right) + \frac{\partial}{\partial \beta}\left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \beta}\right) = \Omega\left[\frac{\partial}{\partial \alpha}\left\{\left(1 - \frac{\bar{F}_1}{\bar{F}_0}\right)\bar{h}\right\}\right] + \frac{\partial \bar{h}}{\partial \bar{t}}$$

Where $\bar{F}_0$, $\bar{F}_1$, and $\bar{F}_2$ are the cross film viscosity integrals and given by the following relations:

$$\bar{F}_0 = \int_0^1 \frac{1}{\bar{\mu}} d\bar{z}, \quad \bar{F}_1 = \int_0^1 \frac{\bar{z}}{\bar{\mu}} d\bar{z},$$

$$\bar{F}_2 = \int_0^1 \frac{\bar{z}}{\bar{\mu}}\left(\bar{z} - \frac{\bar{F}_1}{\bar{F}_0}\right) d\bar{z}$$

The values of function $\bar{F}_0$, $\bar{F}_1$ and $\bar{F}_2$ are obtained using Numerical integration (Simpson's rule). The boundary conditions used for the lubricant flow field are assumed that the external boundary of the bearing have zero relative pressure with respect to atmospheric pressure.

The journal bearing operating under load is required to maintain an appropriate minimum fluid-film thickness to minimize the chances of metal to metal contact. For a rigid journal bearing system, operating under static conditions, the fluid film thickness expression is given as

$$\bar{h} = 1 - \bar{X}_J \cos\alpha - \bar{Z}_J \sin\alpha$$

Where, $\bar{h}$ is the fluid-film thickness, when the journal center is in static equilibrium position.

3. Range of Parameters

Static performance characteristics of hydrodynamic journal bearing are analysed by using generalised Reynolds equations for lubricants with various non-linearity factors. Bearing geometry consist of journal diameter with 45 mm and width of bearing was 90mm. Load is varying from the range of 0.25 -1.25Kg. Non linearity factor ( ) for lubricant was taken in the range of 0-1.

4. Results and Discussion

The two static performance characteristics of hydrodynamic journal bearing considered for analysis namely as maximum fluid film pressure ( ) and minimum fluid film thickness.

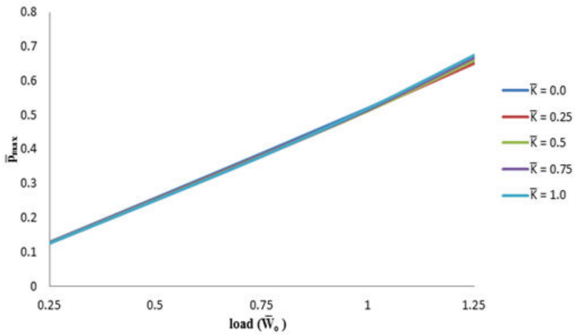

Figure 1: variation of ($\bar{p}_{max}$) with increase in load ($\bar{W}_o$)

The variation of maximum fluid film pressure ($\bar{p}_{max}$) with increase in external load $\bar{W}_o$ for lubricant with various non-linearity factors ($\bar{K}$) is shown in Figure 1. It is observed that for a given value of non-linearity factors ($\bar{K}$), the value of maximum pressure ($\bar{p}_{max}$) increases with increase in value of external load $\bar{W}_o$. For a given value of load $\bar{W}_o$ the variation in value of maximum pressure ($\bar{p}_{max}$) is marginal for both Newtonian and non-Newtonian.

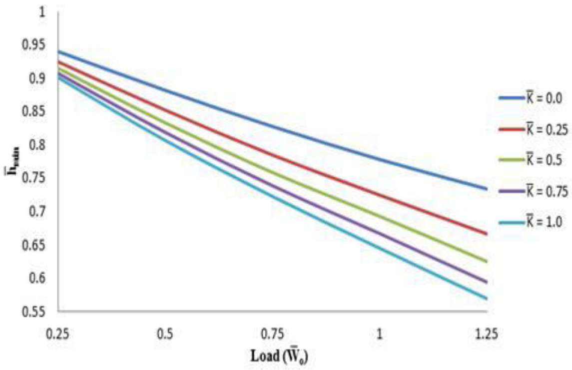

Figure 2: variation of ($\bar{h}_{min}$)with increase in load ($\bar{W}_o$)

The variation of minimum fluid film thickness ($\bar{h}_{min}$) with increase in external load $\bar{W}_o$ for various non-linearity factors ($\bar{K}$) are shown in Figure 2. It is observed that for a given value of non-linearity factors ($\bar{K}$), the value of minimum fluid film thickness ($\bar{h}_{min}$) decreases with increase in value of external load $\bar{W}_o$. At external load $\bar{W}_o$ = 1.25, the value of minimum fluid film thickness ($\bar{h}_{min}$) is maximum for the lubricant having non-linearity factors ($\bar{K}$) = 0.0, while the value of minimum fluid film thickness ($\bar{h}_{min}$) is minimum for the lubricant having non-linearity factors ($\bar{K}$) = 1.0. It was also noticed that that value of minimum fluid film thickness ($\bar{h}_{min}$) is highly pronounced with lower load $\bar{W}_o$.

5. Conclusion

Based on the theoretical investigation on hydrodynamic journal bearing following conclusions are made :

- The value of minimum fluid film thickness ($\bar{h}_{min}$) is highly pronounced with lower load $\bar{W}_o$.

- It is observed that for a given value of non-linearity factors ($\overline{K}$), the value of maximum pressure ($\overline{p}_{max}$) increases with increase in value of external load.

References

1. Sheeja D. and Prabhu B.S. “Thermohydrodynamic lubrication of non-Newtonian journal bearings: theory and experiments”. Journal of Appl. Phys., 1992; 25; 1706-1712.

2. Rao Ramamohana A. and Mohanram P.V. “A study of mixed lubrication parameters of journal bearings”. Wear, 1993; 160; 111-118.

3. Ju Sheau-Ming and Weng Cheng-I “Thermohydrodynamic analysis of finite-width journal bearings with non-Newtonian lubricants”. Wear, 1994; 171; 41-49.

4. Rao A. Ramamohana and Mohanram P.V. “A study of wear characteristics of journal bearings operating under mixed-lubrication conditions”. Wear, 1994; 172; 11-22.

5. Nabhan M.B.W.,Ibrahim G.A. and Anabtawi M.Z. “Analysis of hydrodynamc journal bearing lubricated with a binary water-based lubricant”. Wear, 1997; 209; 13-20.

6. Qi An, Yinsheng Zhou and Yongxin Quan “Study on the viscosity properties of bubbly oil and the static characteristics of journal bearing lubricated with bubbly oil”. Wear, 1997; 213; 159-164.

7. Turaga Ram, Sekher A.S. and Majumdar B.C. “The effect of roughness parameter on the performance of hydrodynamic journal bearings with rough surfaces”. Tribology International, 1999; 32; 231–236.

8. Rho Byoung-Hoo and Kim Kyung-Woong “A study of the dynamic characteristics of synchronously controlled hydrodynamic journal bearings”. Tribology International, 2002; 35; 339–345.

9. Sun Jun and Changlin Gui “Hydrodynamic lubrication analysis of journal bearing considering misalignment causedby shaft deformation”. Tribology International, 2004;37; 841–848.

10. Singh U., Roy L. and Shau M. “Steady-state thermo-hydrodynamic analysis of cylindrical fluid film journal bearing with an axial groove”. Tribology International, 2008; 41; 1135– 1144.

11. Molka Attia Hili, Slim Bouaziz, Mohamed Maatar, Tahar Fakhfakh and Mohamed Haddar “Hydrodynamic and elastohydrodynamic studies of a cylinderical journal bearing”. Journal of hydrodynamics, 2010; 22(2); 155-163.

12. Garg H.C., Kumar Vijay and Sharda H.B. “Performance of slot-entry hybrid journal bearings considering combined influences of thermal effects and non-Newtonian behavior of lubricant”. Tribology International, 2010; 43; 1518–1531.