ANALYSIS OF LOAD IN HYDRODYNAMIC JOURNAL BEARINGS

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Abstract: In this paper, the work deals with steady state analysis for lubrication in journal bearing. The effect of non-Newtonian fluid has been studied by assuming Ree-Eyring model and it is used to describe the lubricant rheology. The current analysis has been done under the assumption of isothermal condition for the sake of simplicity. A first order perturbation approaches have been used to derive the Reynolds equation introducing the effect of non-Newtonian fluid behavior. It is discretized using finite difference scheme and a computational algorithm based upon Newton-Raphson technique has been developed. The deviations of lubrication characteristics in terms of film shape, minimum film thickness, attitude angle, fluid pressure distribution, coefficient of friction and studied with respect to shaft speed, load, and radial clearance.

Index Terms– *Hydrodynamics, Bearings, Lubrication.*

I. INTRODUCTION

Hydrodynamic journal bearing is the part of journal bearing which can be defined as a bearing operating with hydrodynamic lubrication in which the bearing surfaces separated from the journal surface by the lubricant film generated by journal rotation. Hydrodynamic journal bearings are critical power transmission components that are carrying increasingly high load because of the increasing power density in various machines.
Hence, observing the true operating conditions of hydrodynamic journal bearings becomes essential to machine design. Oil film pressure is a very important operating parameters which describes the operating conditions in hydrodynamic journal bearings. Measuring the oil film pressure in bearings has been demanding task and therefore the subject has been studied mainly by mathematical mean. Hydrodynamic lubrication requires thin, converging fluid films. Till they exhibit viscosity, these fluids cab be liquid or gas. In most of the computer components heads are supported with the help of hydrodynamic lubrication in which the fluid film is considered as the atmosphere. These films are of the order of micrometers. Their convergence forces them apart as it creates pressures normal to the surfaces they contact.

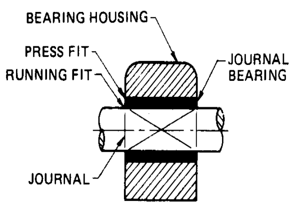

Fig. 1 Journal or sleeve bearings

II. LUBRICATION

The objective of lubrication is to reduce friction, wear and heating of machine parts that move relative to each other a modern dictionary define a lubricant as a result that will when imposed between moving parts of machinery make the surface slippery and reduce friction, eliminate asperities, and prevent cohesion. In the accepted terminology of engineering, lubrication results in the reduction of friction and wear

There are generally three types of lubrication can occur in bearing: full film, mixed film, and boundary lubrication. Full film lubrication is the one in which the bearing surfaces are fully separated by a film of lubricant, eliminating any contact. Full film lubrication can be hydrodynamic, hydrostatic, or elastohydrodynamic.

2.1 Problem Formulation

One of the primary objectives of the present work is related to hydrodynamic journal bearing to study effect of shaft speed, load and radial clearance for fluid pressure distribution, minimum film thickness, coefficient of friction and attitude angle for Newtonian and non-Newtonian fluids as well as for different type of film thicknesses. The perturbation method is used to obtain the Reynolds equation. These Reynolds equations can be simplified easily for the limiting cases of pure Newtonian and non-Newtonian fluids.

3. Analysis

3.1 Introduction

One of the primary objectives of the present work is related to hydrodynamic journal bearing to study effect of shaft speed, load and radial clearance for fluid pressure distribution, minimum film thickness, coefficient of friction and attitude angle for Newtonian and non-Newtonian fluids as well as for different type of film thicknesses. The modified Reynolds equation is obtained by perturbation method. These are the equations which can be reduced comfortably for the limiting cases of pure Newtonian and non-Newtonian fluids.

3.2 Analytical Formulation

3.2.1 Reynolds Equation for Newtonian fluid in non-dimensional form

As we know Reynolds equation for Newtonian fluid in dimensional form is given by:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)=\frac{u}{2}\frac{\partial h}{\partial x} \tag{3.1}$$

The oil film thickness can be written as a function of x:

$$h = h_o + s_h\left(1-\frac{x}{l}\right) \tag{3.2}$$

In dimensionless form equations 3.1 & 3.2 become

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right)=A_k\frac{\partial H}{\partial X} \tag{3.3}$$

$$H = H_o + 1 - X \tag{3.4}$$

Where, $P=\frac{p}{p_o}$ $\quad p_o=\frac{w}{l}$

$H=\frac{h}{s_h}$ $\quad H_o=\frac{h_o}{s_h}$

$X=\frac{x}{l}$ and $A_k=\frac{6\eta u l^2}{w s_h^2}$

Now, from equation 3.4

$$\frac{\partial H}{\partial X}=-1$$

Then equation 3.3 becomes

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right)+A_k=0 \qquad (3.5)$$

Now this equation 3.5 is used to develop a FORTRAN program to calculate pressure distribution in slider bearing along x axis for Newtonian lubrication behavior.

3.3 Finite difference formulation

3.3.1 For Simple hydrodynamic journal bearing.

$$f_i = \varepsilon_{i+1/2}\frac{P_{i+1} - P_i}{\Delta X^2} - \varepsilon_{i-1/2}\frac{P_i - P_{i-1}}{\Delta X^2} + A_k = 0$$

Where,

$\varepsilon_i = H_i^3$, for Newtonian lubricant

$\varepsilon_i = H_i^3/\xi$, for non-Newtonian lubricant

$$\varepsilon_{i+1/2} = \left(\frac{\varepsilon_{i+1}+\varepsilon_i}{2}\right), \quad \varepsilon_{i-1/2} = \left(\frac{\varepsilon_{i-1}+\varepsilon_i}{2}\right)$$

3.4 Load Equilibrium Equation

$$\int_{x_i}^{x_a} p\,dx = W$$

This equation can also be expressed in non-dimensional form as given below.

$$\int_{x_i}^{x_a} P\,dX = 1$$

The Simpson's rule is used to derive this integral and then it can be written in the as given below:

$$\Delta W = \sum_{j=2}^{N} C_j P_j - 1 = 0$$

where, $$C_j = \begin{cases} \Delta X/3 & j=1 \\ 4\,\Delta X/3 & j=2,4,6\ldots \\ 2\,\Delta X/3 & j=3,5,7\ldots \end{cases}$$

3.5 Newton-Raphson Formulation

$$\begin{bmatrix} \frac{\partial f_2}{\partial P_2} & \frac{\partial f_2}{\partial P_3} & \cdots & \frac{\partial f_2}{\partial P_N} & \frac{\partial f_2}{\partial \bar{\varepsilon}} \\ \frac{\partial f_3}{\partial P_2} & \frac{\partial f_3}{\partial P_3} & \cdots & \frac{\partial f_3}{\partial P_N} & \frac{\partial f_3}{\partial \bar{\varepsilon}} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ \frac{\partial f_N}{\partial P_2} & \frac{\partial f_N}{\partial P_3} & \cdots & \frac{\partial f_N}{\partial P_N} & \frac{\partial f_N}{\partial \bar{\varepsilon}} \\ C_2 & C_3 & \cdots & C_N & 0 \end{bmatrix} \begin{bmatrix} \Delta P_2 \\ \Delta P_3 \\ \cdots \\ \cdots \\ \Delta P_N \\ \Delta \bar{\varepsilon} \end{bmatrix} = - \begin{bmatrix} f_2 \\ f_3 \\ \cdots \\ \cdots \\ f_N \\ \Delta W_0 \end{bmatrix}$$

3.6 Boundary Conditions:

Inlet boundary condition

$P = 0$ at $X = X_{in}$

Outlet Boundary Condition

$P = 0$ at $X = X_o$

The approach used to determine X_o is described subsequently.

4. RESULTS AND DISCUSSION

4.1 EFFECT OF LOAD:

The results have been obtained for effect of radial load on the fluid pressure distribution, film shapes, minimum film thickness, coefficient of friction and attitude angle for Newtonian and Non-Newtonian fluids. The value of shaft speed i.e., u_o =10 m/sec and radial clearance i.e., C=0.005 have been taken constant for both Newtonian and Non-Newtonian fluids.

Figure 4.1 shows the variation of pressure (p) with respect to angle (θ) for Newtonian and Non-Newtonian fluids. The solid and dotted curve represents the conclusions of Newtonian fluid, dashed line curve represents the results for first type of non-Newtonian fluid and dashed dotted curve represents the results for second type of non-Newtonian fluid. It can be observed from figures that the value of increase in pressure is almost similar for Newtonian fluid and for first type of non-Newtonian fluid, and greater than the second type non-Newtonian fluid.

Table 4.1 Effect of change in load on pressure at angle 100°

Fluid	Percentage change in load	Percentage Change in pressure at 100°
Newtonian fluid	Increase by 100%	Increase by 100%
Non-Newtonian fluid 1	Increase by 100%	Increase by 99%
Non-Newtonian fluid 2	Increase by 100%	Increase by38%

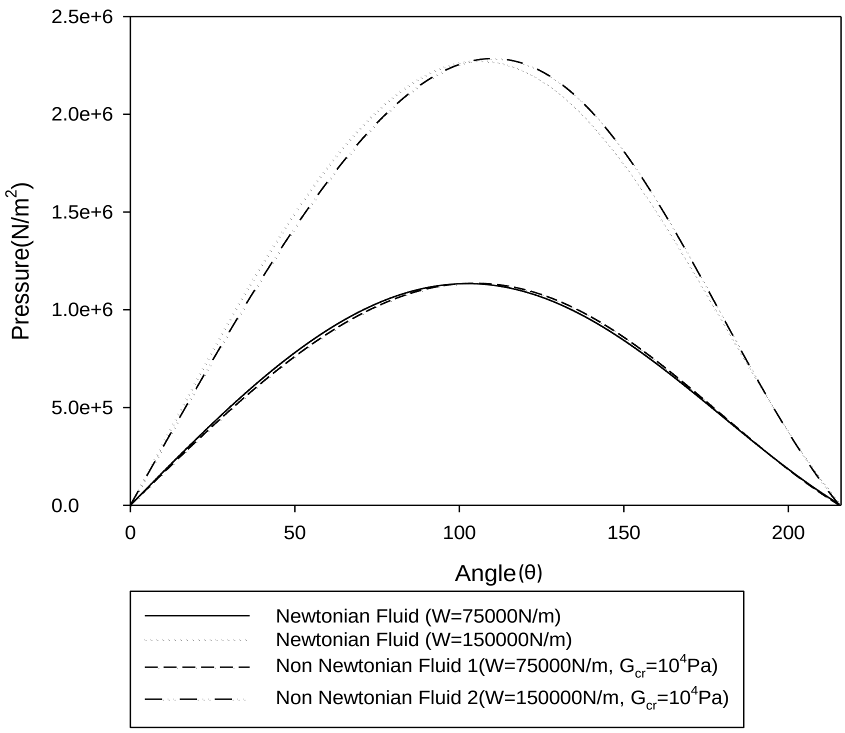

Fig. 4.1 Variation of pressure with angle (u_0=10m/sec, C=0.005µm)

CONCLUSION:

A quantitative comparison value of pressure with speed is given in the table 5.6. It can be found out form the table that for Newtonian fluid at an angle of 100° there is an increase of 100% in the value of pressure when load is increased from 75000N/m to 150000N/m i.e. by increasing the load by 100%. Similarly for first Non-Newtonian fluid at an angle of 100° there is an increase of 99% in the value of pressure and for non-Newtonian fluid 2 at an angle of 100° there is an increase of 38% in the value of pressure when the same value of load is increased.

REFERENCES

1. S.P.Tayal and R. Sinhasa “Analysis of Hydrodynamic Journal Bearings by a Finite Element Method”, University of Roorkee, India, 1981

2. Chien Hsin Chen and Cha’O-Kuang Chen “The Influence Of fluid inertia of finite journal bearing” in Department of Mechanical Engineering, National Cheng Kung University, (Taiwan),1988

3.D.Vijayarghavan and T.G.Keith “Effect of cavitation of a grooved misaligned journal bearing” in Department of Mechanical Engineering, University of Toledo, Toledo, OH (U.S.A.),1989

4 S.K. Guh “Steady-state of misaligned hydrodynamic journal bearings with isotropic roughness effect” in Department of Mechanical Engineering, Bengal Engineering College , Howrah,India,1999

5 P.L. Sah and Satish Sharma “Bearing flexibility on the performance of slot-entry journal bearing” in Department of Mechanical and Industrial Engineering, University of Roorkee, Roorkee, India,2000

6. . S. Das and A.K. Chattopadhyay “Misaligned hydrodynamic journal bearings lubricated with micro polar fluids”, Bengal Engineering College, Howrah, India, 2001

7. Qiuying Chang and Shizhu Wen “ Thermoelastohydrodynamic analysis of tilting-pad journal bearings” in Department of Mechanical Engineering, Qingdao Institute of Architecture and Engineering, Qingdao ,China,2001

8. Satish C. Sharma and T. Nagaraju “Hybrid journal bearing system considering combined influence of thermal and elastic effects” in Department of Mechanical and Industrial Engineering, IIT Roorkee, India, 2003

9. Jun Sun and Gui Changlin “Journal bearing misalignment caused by shaft deformation” in School of Mechanical and Automotive Engineering, Anhui , China,2004

10. Jaroslaw Sep “Analysis of a journal bearing with a two-component surface layer” in Mechanical Engineering and Aeronautics, Rzeszo´w University of Technology, Al. Powstan, Poland, 2004

11. Jie Peng and Ke-Qin Zhu “Effects of electric field on hydrodynamic characteristics of finite-length ER journal bearings” in Department of Engineering Mechanics, Tsinghua University, Beijing,China,2005

12. Slim Bouaziz and Mohamed Haddar “Journal bearings in presence of rotor spatial angular misalignment” Unit of Dynamic Mechanical Systems (UDSM), National Engineering School of Sfax (ENIS), Tunisia, 2006

13. D.Rh.Gwynllyw and T.N. Phillips “Influence of Oldroyd-B and PTT lubricants on moving journal bearing systems”, University of the West of England, UK, 2007

14. Nicoleta M. Ene and Theo G. Keith Jr “Analysis for a hydrodynamic three-wave journal bearing” University of Toledo at NASA Glenn Research Centre, Cleveland, USA,2007

15. K.P. Gertzos and C.A. Papadopoulos “CFD analysis of journal bearing by Bingham lubricant” in Department of Mechanical Engineering and Aeronautics, University of Patras, , Greece,2008

16.Mongko Mongkolwongrojn and Chatchai Aiumpronsin “Stability analysis of rough journal bearings under THE with non-Newtonian lubricants” in Department of Mechanical Engineering, King Mongkut’s Institute of Technology Ladkrabang, Bangkok, Thailand,2009

17. H.C. Garg and H.B. Shard “Journal bearings with combined influences of thermal effects and non-Newtonian behavior of lubricant in Department of Mechanical Engineering, Guru Jambheshwar University of Science and Technology, Hisar, India,2010