VARIATION OF PRESSURE WITH RESPECT TO SHAFT SPEED FOR NEWTONIAN AND NON NEWTONIAN FLUIDS IN JOURNAL BEARING

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ABSTRACT

Journal bearing are cylindrical or ring-shaped bearings designed to carry radial loads. Hydrodynamic journal bearings are critical power transmission components that are carrying increasingly high load because of the increasing power density in various machines. The journal bearing has several advantages over other types of bearing, providing it has a constant supply of clean high-grade motor oil. Journal bearings with their inherent advantages are also used in other high-load, high-velocity applications, such as machines and turbines.

In this paper, the work deals with steady state analysis for lubrication in journal bearing. The effect of non-Newtonian fluid behavior has been studied by assuming Ree-Eyring model to describe the lubricant rheology. The present analysis has been carried out under the assumption of isothermal condition for the sake of simplicity.

Keyword : *Journal Bearing, Newtonian Fluid, Non Newtonian Fluid.*

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1. INTRODUCTION

The necessity of supporting a load in the presence of relative motion presents one of the most important of design problems. The motion may be either translatory or rotary and load may be radial, axial or both radial and axial .The machine part used for this purpose is known is bearing. Thus a bearing may be defined as a machine member whose function is to support and retain a moving member. The most important criterion to classify the bearing is the type of friction between the shaft and bearing surface. Depending upon the type of friction, bearing are classified into two main group sliding contact bearing and rolling contact bearing. Sliding contact bearings are also called plain bearings, journal bearing or sleeve bearing.

1.1 JOURNAL BEARING

These are cylindrical or ring-shaped bearings designed to carry radial loads.. The terms sleeve and journal are used more or less synonymously since sleeve refers to the general configuration while journal pertains to any portion of a shaft supported by a bearing. In another sense, however, the term journal may be reserved for two-piece bearings used to support the journals of an engine crankshaft.

The simplest and most widely used types of sleeve bearings are cast-bronze and porous-bronze (powdered-metal) cylindrical bearings. Cast-bronze bearings are oil-, or grease-lubricated. Porous bearings are impregnated with oil and often have an oil reservoir in the housing.

Plastic bearings are being used increasingly in place of metal. Originally, plastic was used only in small, lightly loaded bearings where cost saving were the primary objective. More recently, plastics are being used because of functional advantages, including resistance to abrasion, and they are being made in large sizes.

2. GOVERNING EQUATION

The basic equation for journal bearing which governs the generation of pressure in lubricating films is known as Reynolds equation and it forms the foundation of hydrodynamic lubrication analysis. It was derived for a Newtonian fluid by neglecting the effects due to curvature of fluid film. This assumption is well justified as the effective radius of bearing components is generally very large compared with the film thickness. This enables the analysis to consider an equivalent curved surface near a plane.

The derivation of Reynolds equation involves the application of the basic equation of motion and continuity to the lubricant.

Assumptions

1. Thin film geometry

2. The fluid is Newtonian and the coefficient of viscosity is constant.

3. Compressibility of the fluid is negligible.

4. There is no slip between the fluid and the solid surface.

5. Fluid pressure does not change across the film thickness.

6. The rate of change of the velocity u and w in the x direction and z direction is negligible compared with the rate of change in the y direction.

The basic theory of Reynolds equation can be understand by using the above assumption and with the help on the axes of rectangular coordinates x, y, and z are taken as shown in the figure. The x and y axes are on the lower surface and the z axis is perpendicular to it. The velocity of the fluid in the directions x, y, and z are denoted by u, v and w, respectively, and the velocity of the lower surface is similarly described by u_1, v_1, and w_1 and that of the upper surface by u_2, v_2, and w_2. In many practical cases, the lower surface and the upper surface perform a straight translational motion relative to each other. In this case, if the x axis is in the translational direction, then we have W1 = W2 = 0 and so the equations can be simplified.

Let the gap between the two surfaces, or the thickness of the liquid film, be denoted by h(x, z, t), with t being time. Let the coincident of viscosity of the fluid be η.

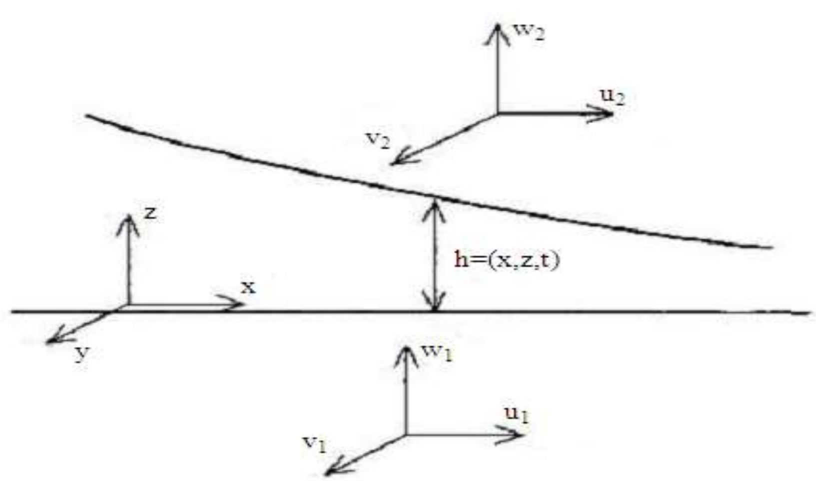

Fig. 2 Fluid film between two solid surfaces

The derivation of Reynolds equation involves the application of the basic equation of motion and continuity to the lubricant. The full equations of motion for a Newtonian fluid in Cartesian coordinates are:

$$\rho\frac{du}{dt} = \rho X - \frac{\partial p}{\partial x} + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial u}{\partial x} - \frac{\partial w}{\partial z}\right) + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial u}{\partial x} - \frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial y}\eta\left(\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial z}\eta\left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right) \quad (2.1a)$$

$$\rho\frac{dv}{dt} = \rho Y - \frac{\partial p}{\partial y} + \frac{2}{3}\frac{\partial}{\partial y}\eta\left(\frac{\partial v}{\partial y} - \frac{\partial u}{\partial x}\right) + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial v}{\partial y} - \frac{\partial w}{\partial z}\right) + \frac{\partial}{\partial z}\eta\left(\frac{\partial v}{\partial z} - \frac{\partial w}{\partial y}\right) + \frac{\partial}{\partial x}\eta\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \quad (2.1b)$$

$$\rho\frac{dw}{dt} = \rho Z - \frac{\partial p}{\partial z} + \frac{2}{3}\frac{\partial}{\partial z}\eta\left(\frac{\partial w}{\partial z} - \frac{\partial u}{\partial x}\right) + \frac{2}{3}\frac{\partial}{\partial z}\eta\left(\frac{\partial w}{\partial z} - \frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial x}\eta\left(\frac{\partial w}{\partial x} - \frac{\partial u}{\partial z}\right) + \frac{\partial}{\partial y}\eta\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right) \quad (2.1c)$$

The terms on the left hand side represent inertia effects and on the right hand side are the body force, pressure and viscous terms in that order. The inertia and body forces are negligible as compared to the viscous and inertia forces.

The equation of continuity representing conservation of mass is:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} = 0 \quad (2.2)$$

Using the above equations 2.1a, 2.1b, 2.1c and 2.2, general Reynolds equation is:

$$0 = \frac{\partial}{\partial x}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right) + \frac{\partial}{\partial x}\left(\frac{\rho h(u_a+u_b)}{2}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h(v_a+v_b)}{2}\right) + \rho(w_a - w_b) - \rho u_a\frac{\partial h}{\partial x} - \rho v_a\frac{\partial h}{\partial y} + h\frac{\partial \rho}{\partial t} \quad (2.3)$$

For only tangential motion, where

$$w_a = u_a\frac{\partial h}{\partial x} + v_a\frac{\partial h}{\partial y}$$ and

$$w_b = 0,$$

The Reynolds equation given in equation (2.3) becomes

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right) = 12\hat{u}\frac{\partial(\rho h)}{\partial x} + 12\hat{v}\frac{\partial(\rho h)}{\partial y} \qquad (2.4)$$

$$\hat{u} = \frac{u_a + u_b}{2} = \text{constant} \qquad \hat{v} = \frac{v_a + v_b}{2} = \text{constant}$$

Equation (2.4) is applicable for elastohydrodynamic lubrication.

If there is axisymmetry, so that the model can be treated as a one-dimensional problem; then the Reynolds equation for an incompressible lubricant can be written as:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \frac{u}{2}\frac{\partial h}{\partial x} + \frac{\partial h}{\partial t} \qquad (2.5)$$

In the equation *p* represents the film pressure, *η* denotes the lubricant viscosity, u is the sliding velocity of the lower surface, *h* is the film thickness depending upon the coordinate

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3. SOLUTION PROCEDURE

The steps involved in the overall solution scheme are given below:

1. The pressure distribution [P], minimum film thickness Ho and outlet boundary co-ordinate Xo were initialized to some reference values. Take Xin=0 and Xo=1.
2. Evaluated the fluid film thickness, H, at every node by using film thickness equation.
3. The residual vector [f] was calculated at each node.
4. The residual vector ΔW was calculated from the discretized load equilibrium equation.
5. The residual vectors calculated in the steps 3 and 4 were assembled in a single vector [F] to facilitate execution of Newton-Raphson scheme.
6. This was followed by computation of Jacobian coefficients.
7. The corrections to the system variables were computed by inverting the Jacobian matrix using Gauss elimination.
8. The corrections, calculated in step 7, were added to the corresponding system variables to get the new values of the pressure distribution *[P]* and minimum film thickness Ho.
9. The outlet boundary co-ordinate *Xo* was corrected by using an appropriate scheme.
10. The termination of the iterative loops required the fulfillment of the predefined convergence criteria to arrive at an accurate solution. In order to check the convergence of the pressure distribution, the sum of the nodal pressures corresponding to the current iteration (say n^{th}) was calculated. If the fractional difference between this value and that corresponding to the previous iteration was less than the prescribed tolerance TOL, the pressure distribution was assumed to have converged. Thus,

$$\frac{\left|\left[\sum_{i=1}^{N} P_i\right]_n - \left[\sum_{i=1}^{N} P_i\right]_{n-1}\right|}{\left|\left[\sum_{i=1}^{N} P_i\right]_{n-1}\right|} \le TOL$$

11. The minimum film thickness was assumed to converge if the fractional change in its value became less than the prescribed tolerance in successive iterations

$$\frac{\left|\left[H_o\right]_n - \left[H_o\right]_{n-1}\right|}{\left|\left[H_o\right]_{n-1}\right|} \le TOL$$

The value of TOL adopted in the analysis was 1×10^{-4} as it has been found that a lower value does not contribute to improve the accuracy of the solution. The iterative loop terminates and the current values were considered as the final solution only if all the relevant convergence criteria were satisfied simultaneously.

12. If any one or more of the relevant criteria were not satisfied, the next iteration began and the control was shifted back to the step 2.

13. Finally the values of pressure distribution, minimum film thickness, friction coefficient and attitude angle calculated using suitable formulae.

4. RESULT AND DISCUSSION

The results have been obtained for effect of shaft speed 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 radial load i.e W=150000N/m and radial clearance i.e. C=0.005μm have been taken constant for both Newtonian and non Newtonian fluids.

Figure 5.1 and 5.2 show the variation of pressure (p) with respect to angle (θ) for Newtonian and non Newtonian fluids. The solid and dotted curve line shows the results for Newtonian fluid, dashed line curve shows the results for first type of non-Newtonian fluid and dashed dotted curve shows 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 first type of non-Newtonian fluid, and greater than the second type non-Newtonian fluid

A quantitative comparison value of pressure with speed is given in the table 5.1. It can be observed from the table that for Newtonian fluid at an angle of 100^o there is an increase of 40% in the value of pressure when speed is increased from 1m/sec to 10m/sec i.e. by increasing the speed by 900%. Similarly for non Newtonian fluid1 at an angle of 100^o there is an increase of 40% in the value of pressure and for non Newtonian fluid 2 at an angle of 100^o there is an increase of 20% in the value of pressure when same values of speed is increased.

Table 5.1 Effect of change in speed on pressure at angle 100°

Fluid	Percentage change in shaft speed	Percentage Change in pressure at 100°
Newtonian fluid	Increase by 900%	Increase by 40%
Non Newtonian fluid 1	Increase by 900%	Increase by 40%
Non Newtonian fluid 2	Increase by 900%	Increase by 20%

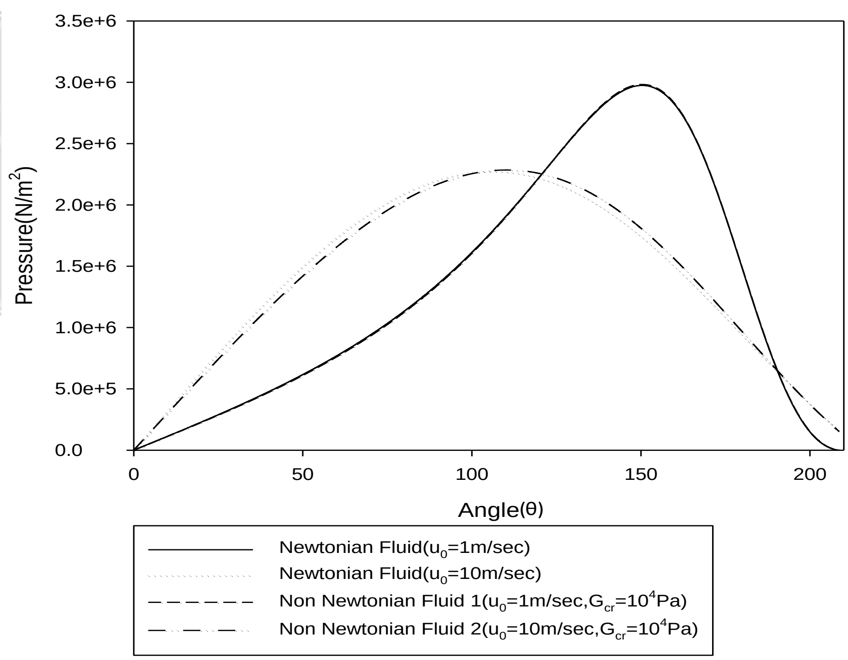

Fig. 5.1 Variation of pressure with angle(C=0.005 μm, W=150000N/m)

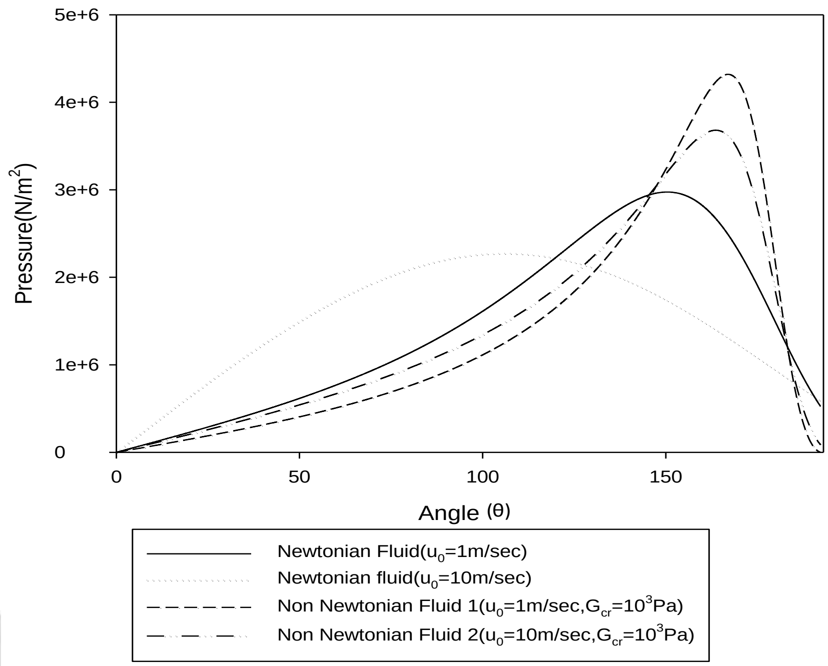

Fig. 5.2 Variation of pressure with angle(C=0.005 µm, W=150000N/m)

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