EFFECT OF SHAFT SPEED ON FILM THICKNESS FOR NEWTONIAN AND NON NEWTONIAN FLUID FOR JOURNAL BEARING

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ABSTRACT

Radial load is carried by cylindrical or ring-shaped bearing known as Journal Bearing. Now to carry high load and for critical power transmission Hydrodynamic journal bearing are used in various machines. In machines and turbines journal bearing are used because of their advantages of carrying high load and high speed applications.

In this research paper, the work deals with steady state analysis of shaft speed on film thickness for Newtonian and non Newtonian fluid

A first order perturbation approach has been employed to derive the Reynolds equation incorporating 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 variation of lubrication characteristics in terms of fluid pressure distribution, film shape, minimum film thickness, coefficient of friction and attitude angle studied with respect to shaft speed, load and radial clearance

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

1. 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.

Therefore, knowing the true operating conditions of hydrodynamic journal bearings are essential to machine design. Oil film pressure is one of the key operating parameters describing 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

The journal bearing has several advantages over other types of bearing, providing it has a constant supply of clean high-grade motor oil. First, it handles high loads and velocities because metal to metal contact is minimal due to the oil film. Second, the journal bearing is remarkably durable and long lasting. Finally, because of the damping effects of the oil film, journal bearings help make engines quiet and smooth running. Journal bearings with their inherent advantages are also used in other high-load, high-velocity applications, such as machines and turbines.

1.1 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 describes a situation in which the bearing surfaces are fully separated by a film of lubricant, eliminating any contact. Full film lubrication can be hydrostatic, hydrodynamic, or elastohydrodynamic

1.2 HYDRAUDYNAMIC LUBRICATION

Hydrodynamic lubrication suggests that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal to metal contact, and the stability thus obtained can be explained by the laws of fluid mechanics. Hydrodynamic lubrication depends on the existence of an adequate supply of lubrication at all times rather than having lubrication under pressure. The film pressure is created by moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing

Hydrodynamic lubrication, also known as fluid film lubrication has essential elements:

1. The surfaces between which the fluid films move must be convergent.
2. Hydrodynamic flow behavior of fluid between bearing and journal.
3. A lubricant, which must be a viscous fluid.

Hydrodynamic (Full Film) Lubrication is obtained when two mating surfaces are completely separated by a cohesive film of lubricant.

The thickness of the film thus exceeds the combined roughness of the surfaces. The coefficient of friction is lower than with boundary-layer lubrication. Hydrodynamic lubrication prevents wear in moving parts, and metal to metal contact is prevented.

Hydrodynamic lubrication requires thin, converging fluid films. These fluids can be liquid or gas, so long as they exhibit viscosity. In computer components, like a hard disk, heads are supported by hydrodynamic lubrication in which the fluid film is the atmosphere.

The scales of these films are on the order of micrometers. Their convergence creates pressures normal to the surfaces they contact, forcing them apart.

2. ANALYSIS

One of the primary objectives of the present work is related to hydrodynamic journal bearing to study effect of shaft speed for minimum film thickness 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 equations can be reduced easily for the limiting cases of pure Newtonian and non-Newtonian fluids.

2.1 Analytical Formulation

2.1.2 Reynolds Equation for Newtonian fluid in non-dimensional form

As derived in the previous chapter Reynolds equation for Newtonian fluid in dimensional form is

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)=\frac{u}{2}\frac{\partial h}{\partial x} \tag{2.1}$$

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

$$h = h_o + s_h\left(1 - \frac{x}{l}\right) \tag{2.2}$$

In dimensionless form equation 3.1 & 3.2 become

$$\frac{\partial}{\partial X}\left(H^3 \frac{\partial P}{\partial X}\right) = A_k \frac{\partial H}{\partial X} \tag{2.3}$$

$$H = H_o + 1 - X \tag{2.4}$$

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

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

$X = \frac{x}{l}$ 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

$$\frac{\partial}{\partial X}\left(H^3 \frac{\partial P}{\partial X}\right) + A_k = 0 \tag{2.5}$$

Now this equation 2.5 is used to develop a FORTRAN program to calculate pressure distribution in slider bearing along x axis for Newtonian lubrication behavior.

3. Finite difference formulation

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

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

3.2 Load equilibrium equation

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

The above equation is expressed in non-dimensional form as follows

$$\int_{x_i}^{x_a} PdX = 1$$

The integral is calculated using Simpson's rule and it can be written in the following form:

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

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

4. CONCLUSIONS

The results have been obtained for effect of shaft speed on the minimum film thickness 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 4.1 and 4.2 show the variation of film thickness (h) with respect to angles (θ) for Newtonian and non Newtonian fluids. The solid and dotted curve 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 figure that the value of increase in film thickness is nearly same for Newtonian fluid and for first type of non-Newtonian fluid, and very large than the second type non-Newtonian fluid.

A quantitative comparison value of film shapes with speed is given in the table 4.1. It can be observed form the table that for Newtonian fluid at an angle of 100^o there is an increase of 12% in the value of film thickness 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 10% in the value of film thickness and for non Newtonian fluid 2 at an angle of 100^o there is an increase of 0.44% in the value of pressure when speed is increased from 1m/sec to 10m/sec i.e. by increasing the speed by 900%.

Table 4.1 Effect of change in speed on film thickness at angle 100°

Fluid	**Percentage change in shaft speed**	**Percentage Change in film shapes at 100°**
Newtonian fluid	Increase by 900%	Increase by 12%
Non Newtonian fluid 1	Increase by 900%	Increase by 10%
Non Newtonian fluid 2	Increase by 900%	Increase by 0.44%

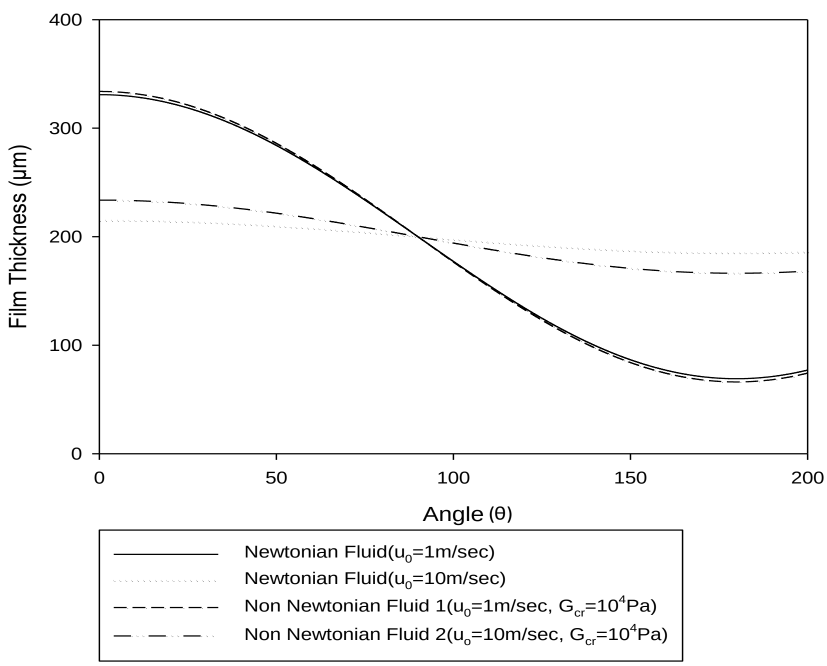

Fig. 4.1 Variation of film thickness with speed(C=0.005 μm, W=150000N/m)

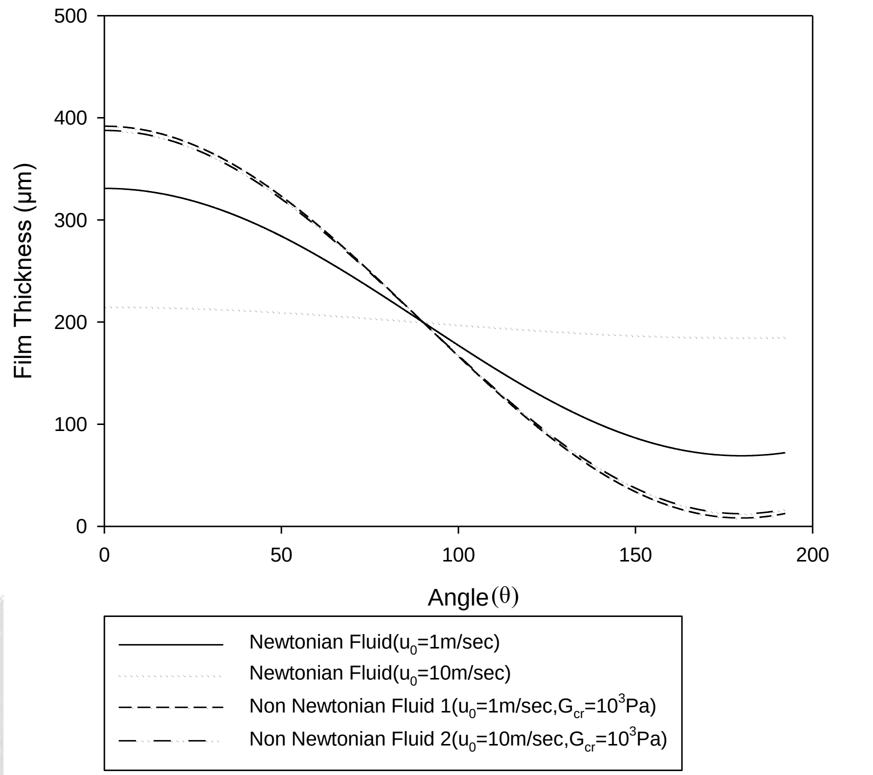

Fig. 4.2 Variation of film thickness with angle(C=0.005 μm, W=150000N/m)

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