Effect of Frequency and Load on Friction coefficient in Gas Lubricated Journal Bearing

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ABSTRACT

Journal bearings are very common engineering components and are used in almost all types of machinery. Combustion engines and turbines virtually depend on journal bearings to obtain efficiency and reliability. A journal bearing consists of a shaft rotating within a stationary bush. The hydrodynamic film which supports the load is generated between the moving surfaces of the shaft and the bush. When a gas is used to lubricate the bearings then these bearings are known as aerostatic bearings. Aerostatic bearings are used in the precision machine tool industry, metrology, computer, peripheral devices and in dental drills, where the air used in the bearing also drives the drill. They are particularly useful for high speed applications and where precision is required since very thin film thicknesses are possible. In this paper, the analysis of one dimensional dynamic loading of gas lubricated journal bearing has been developed. The effect of different parameters i.e. frequency and load on friction coefficient has been studies. The isothermal conditions are taken into account during present analysis for the sake of simplicity.

Keyword: - *Gas Lubricated Journal Bearing, Friction Coefficient, Load, Frequency*

1. INTRODUCTION

Hydrodynamic lubrication or fluid film lubrication is obtained when two mating surfaces are completely separated by a cohesive film of lubricant. 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.

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1.1 Governing Equation

Reynolds equation governs the generation of pressure in lubricating films and it forms the foundation of hydrodynamic lubrication analysis. This equation was derived for a Newtonian fluid by neglecting the effects due to curvature of fluid film as the effective radius of bearing components is generally very large compared with the film thickness. This assumption 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. The flow is laminar.

2. The gravity and inertia forces acting on the fluid can be ignored compared with the viscous force.

3. Compressibility of the fluid is negligible.

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

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.

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

8. Thin film geometry

Reynolds' equation is an equation to obtain the pressure generated in a fluid film when two solid surfaces undergo relative motion. However, the Fluid Film must be sufficiently thick so that it can be analyzed by hydrodynamics, and at the same time it must be sufficiently thin so that Reynolds' assumptions described below will hold.

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:

$$\rho\frac{du}{dt} = \rho X - \frac{\partial p}{\partial x} + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial u}{\partial x} - \frac{\partial w}{\partial z}\right) + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial u}{\partial x} - \frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial y}\eta\left(\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial z}\eta\left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right) \tag{1.1}$$

$$\rho\frac{dv}{dt} = \rho Y - \frac{\partial p}{\partial y} + \frac{2}{3}\frac{\partial}{\partial y}\eta\left(\frac{\partial v}{\partial y} - \frac{\partial u}{\partial x}\right) + \frac{2}{3}\frac{\partial}{\partial x}\eta\left(\frac{\partial v}{\partial y} - \frac{\partial w}{\partial z}\right) + \frac{\partial}{\partial z}\eta\left(\frac{\partial v}{\partial z} - \frac{\partial w}{\partial y}\right) + \frac{\partial}{\partial x}\eta\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \tag{1.2}$$

$$\rho\frac{dw}{dt} = \rho Z - \frac{\partial p}{\partial z} + \frac{2}{3}\frac{\partial}{\partial z}\eta\left(\frac{\partial w}{\partial z} - \frac{\partial u}{\partial x}\right) + \frac{2}{3}\frac{\partial}{\partial z}\eta\left(\frac{\partial w}{\partial z} - \frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial x}\eta\left(\frac{\partial w}{\partial x} - \frac{\partial u}{\partial z}\right) + \frac{\partial}{\partial y}\eta\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right) \tag{1.3}$$

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

The continuity equation for conservation of mass is:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} = 0 \quad (1.4)$$

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

$$0 = \frac{\partial}{\partial x}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right) + \frac{\partial}{\partial x}\left(\frac{\rho h(u_a+u_b)}{2}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h(v_a+v_b)}{2}\right) + \rho(w_a - w_b) - \rho u_a\frac{\partial h}{\partial x} - \rho v_a\frac{\partial h}{\partial y} + h\frac{\partial \rho}{\partial t} \quad (1.5)$$

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

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right) = 12\hat{u}\frac{\partial(\rho h)}{\partial x} + 12\hat{v}\frac{\partial(\rho h)}{\partial y} \quad (1.6)$$

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

Equation (2.4) is applicable for elasto hydrodynamic lubrication.

If the model, due to axis symmetry, treated as a one dimensional problem; then the Reynolds equation for an incompressible lubricant can be written as:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right) = \frac{u}{2}\frac{\partial h}{\partial x} + \frac{\partial h}{\partial t} \quad (1.7)$$

In the above 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 *x*.

1.2 Load Equilibrium Equation

In a quasi steady state problem the net radial force resulting from the fluid film pressure should always be equal to applied load. And the same quasi steady state problem has been assumed in this dissertation. The load carrying capacity of the oil film per unit length is:

$$\int_{x_i}^{x_a} p\,dx = W \quad (1.2.1)$$

Here, *W* is the load imposed

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

$$\int_{x_i}^{x_a} P\,dX = 1 \quad (1.2.2)$$

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

$$\Delta W = \sum_{j=2}^{N} C_j P_j - 1 = 0 \tag{1.2.3}$$

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

2. Solution Procedure

For the calculation of Friction Coefficient (μ) with respect to time some input values of different parameters like load and frequency are used. These values of load and frequency are used in the program to see their effect on μ with respect to time. The input values of these parameters are given in the following table.

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.

3. Results and Discussion

3.1 Effect of Frequency and Load

The table 3.1.1and 3.1.2 shows the percentage change in friction coefficient (μ) for percentage increase in frequency at constant load. And the same is represented in the figure 3.1.1.

Table 3.1.1 Percentage change in μ with constant load and changing frequency

Sr. No.	Value of Load	Percentage Increase in frequency	Decrease of μ at maximum loading	Increase of μ at minimum loading
1	4000	400%	1.03%	1.43%
2	5000	400%	1.52%	1.58%

Table 3.1.2 Percentage change in μ with constant frequency and changing load

Sr. No.	Value of Frequency	Percentage Increase in Load	Decrease of μ at maximum loading	Increase of μ at minimum loading
1	100	25%	15.06%	16.48%
2	500	25%	15.67%	16.05%

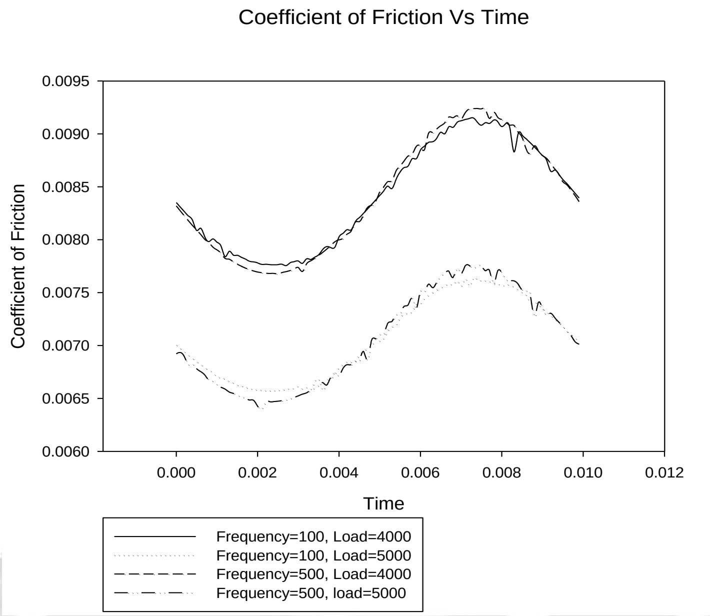

Fig 3.1.1 Friction Coefficient Vs Time (Frequency and Load)

4. CONCLUSIONS

The following conclusion has been made on the basis of the results drawn:

1. The value of coefficient of friction is decrease at maximum loading condition and increases at minimum loading condition.
2. The value of coefficient of friction is higher for constant value of frequency but by increasing the value of load.

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