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Effects of Surface Roughness in Squeeze Film Lubrication of Spherical Bearings

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Abstract

This paper describes a theoretical analysis of the effects of surface roughness in squeeze film lubrication of spherical bearings. The analysis is based on a deterministic theory of hydrodynamic lubrication. The important squeeze film parameters are obtained as simple closed form expressions for roughness types. The generalized Reynolds equation accounting for the surface roughness is considered and it is applied to study the effects of roughness in squeeze film lubrication of spherical bearings. In order to get the expression for pressure, the generalized Reynolds equation is solved. Then by making use of this expression we obtain the expression for load carrying capacity, which in turn is used to find the expression for response time. These expressions are numerically computed and the results are presented graphically. From the numerical computations of the results, it is found that, the load capacity and squeezing time increases with an increase of peripheral viscosity of the lubricant. It is observed that in case of transverse roughness pattern the load capacity and squeezing time increases where as it decreases for longitudinal roughness pattern. Also load capacity increases with an increase in squeeze velocity.

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Keywords: Reynolds equation; Surface roughness; Squeeze film lubrication; Load capacity; Squeezing time.

1. Introduction

Hydro dynamically lubricated spherical bearings are widely used in a variety of applications involving severe operating conditions of speeds, loads etc. In general the bearing surfaces are rough in nature. When the film thickness between the surfaces is very small, the surface asperities begin to interact with the lubricant. Then the

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effect of surface roughness must be taken into account to study the behaviour of lubrication systems, where the flow takes place in narrow recesses. Hence, the effect of surface roughness plays a significant role in the development of the science and technology of lubrication.

In recent years the study of lubrication of rough surfaces has gained much attention, by deterministic and stochastic approaches. In the deterministic approach, the effect of surface roughness is taken into account in the usual Reynolds equation by considering that the film thickness is a function of surface roughness which may represent by a series of sine and cosine waves. Dowson and Whomes [1] have applied this procedure to study the bearing characteristics of rollers, spiral groove bearings etc. Shukla [2] gave a new deterministic theory to study the effects of surface roughness when the mean height of the asperities is same order of magnitude as the minimum film thickness.

The phenomenon of two lubricated surfaces approaching each other with a normal velocity is known as squeeze film lubrication. The thin film of lubricant present between the two surfaces acts as a cushion and it prevents the surfaces from making instantaneous contact. The time required to squeeze out the lubricant depends upon surface configuration, fluid properties and the load applied. In general, the relation between the load carrying capacity and the rate of approach is studied in the most squeeze film analysis. Although squeeze film lubrication has been generally understood for some time, the importance of its applications leads to draw the attention of many workers. Pan et al. [3] studied characteristics of squeeze film bearings. Gould [4] investigated high pressure squeeze films for circular plates and considered the variation of viscosity with temperature and pressure. Beck et al. [5] studied flat disk squeeze film bearing including the effects of supported mass motion. Ramanaiah and Dubey [6] studied squeeze films and thrust bearing using micro polar fluid. Murti [7] studied squeeze films in porous bearings. Shukla et al. [8] studied various squeeze films using power law fluid. Rao et al. [9, 10] studied the effects of velocity-slip and viscosity variation in squeeze film lubrication of two circular plates and spherical bearings.

In Kumar and Rao [11], we have developed a generalized Reynolds equation for two surfaces by considering surface roughness at the bearing surfaces. In this paper, it is applied to study the combined effects of surface roughness and squeeze film lubrication of spherical bearings.

2. Basic Equations

Consider the squeeze film lubrication between two eccentric spherical surfaces of radii r_1 and R_1 which are approaching each other with a velocity V as shown in Fig. 1.

The governing equation of flow of the lubricant in the case of squeeze film lubrication [11] is given by the equation

$$\frac{d}{dx}\left(F_2\frac{dP}{dx}\right)=-V \tag{1}$$

where

$$F_2=\frac{\left(M-Tanh(M)\right)}{M^3}\frac{\left(2h_s\right)^3}{2k\mu}+\frac{1}{4}\left(\left(h_n-2h_s\right)+2h_s\frac{Tanh(M)}{M}\right)^2 4h_s\frac{Tanh(2M)}{k\mu(2M)}+\frac{(h_n-2h_s)^3}{12\mu}$$

$$M=\alpha h_s\ ,\ \alpha=\sqrt{\frac{1}{k\phi}}$$

where k is the ratio of the viscosity of the peripheral layer to the middle layer, h_s is the height of roughness asperities, ϕ is the roughness parameter and μ the viscosity.

The nominal film thickness h_n is given by

$$h_n = c(1 - \varepsilon\, Cos\theta)$$

where $c = R_1 - r_1$ is the clearance width and $\varepsilon = \frac{e}{c}$ is the eccentricity ratio.

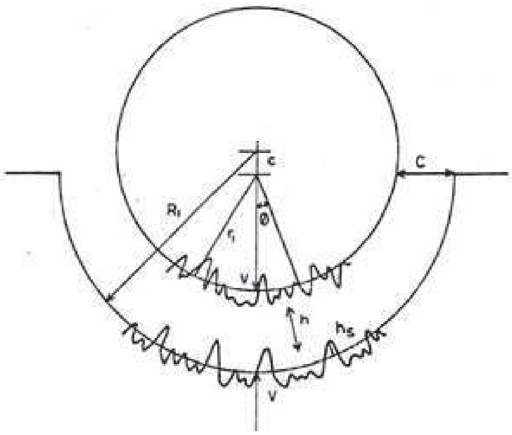

Fig.1 Squeezing film in spherical bearing

The flow flux, Q of the lubricant is given by equation (1) as

$$Q = \frac{F_2}{r}\frac{dP}{d\theta} 2\pi\, r_1 Sin\theta \quad (2)$$

The flux, Q obtained from the equation of continuity is

$$Q = 2\pi\, r_1^2\, V\, Sin^2\theta \quad (3)$$

Now from equations (2) and (3), we obtain

$$\frac{dP}{d\theta} = \frac{r_1^2\, V\, Sin\theta}{F_2} \quad (4)$$

where

$$F_2 = \frac{(M - Tanh(M))}{M^3}\frac{(2h_s)^3}{2k\mu} + \frac{1}{4}\left((c(1-\varepsilon\, Cos\theta) - 2h_s) + 2h_s\frac{Tanh\ (M)}{M}\right)^2 4h_s\frac{Tanh\ (2M)}{k\mu\ (2M)} + \frac{(c(1-\varepsilon\, Cos\theta) - 2h_s)^3}{12\mu} \tag{5}$$

$$M = \alpha\, h_s\ ,\ \alpha = \sqrt{\frac{1}{k\,\phi}}$$

The boundary condition for equation (4) is

$$P = 0 \qquad at \qquad \theta = \frac{\pi}{2}$$

Now integrating equation (4) and using the above condition, we get the expression for pressure distribution as

$$P(\theta) = \int_0^{\frac{\pi}{2}} \frac{r_1^2\ V\, Sin\theta}{F_2}\, d\theta \tag{6}$$

The load capacity, W is given by

$$W = 2\pi r_1^2 \int_0^{\frac{\pi}{2}} P Sin\theta\, Cos\theta\, d\theta \tag{7}$$

Using equations (6) and (7) we get the load capacity, W as

$$W = 2\pi r_1^4\, V \int_0^{\frac{\pi}{2}} \frac{Sin^3\theta}{F_2}\, d\theta \tag{8}$$

The squeezing time, T for the surfaces to approach from the initial eccentric position $(\varepsilon = 0)$ to a final eccentric position $(\varepsilon = \varepsilon_1)$ is given by

$$T = \frac{2\ \pi r_1^4}{W} \int_0^{\varepsilon_1} \int_0^{\frac{\pi}{2}} \frac{Sin^3\theta}{F_2}\, d\theta\, d\varepsilon \tag{9}$$

where F_2 is given in equation (5).

Now equations (8) and (9) are non-dimensionalised in the following manner:

$$\overline{h_s} = \frac{h_s}{c}, \quad \overline{\phi} = \frac{\phi}{c^2}, \quad \overline{F_2} = \frac{F_2}{\left(c^3 / 12\mu \right)}, \quad \overline{V} = \frac{V}{c}$$

Then, the non-dimensional load capacity, $\overline{W}$ is given by

$$\overline{W} = \frac{W c^2}{\pi r^4 \mu} = 24\overline{V} \int_0^{\frac{\pi}{2}} \frac{Sin^3\theta}{\overline{F_2}} d\theta \tag{10}$$

and the non-dimensional squeeze time, $\overline{T}$ is given by

$$\overline{T} = \frac{T W c^2}{24\mu \pi r_1^4} = \int_0^{\varepsilon_1} \int_0^{\frac{\pi}{2}} \frac{Sin^3\theta}{\overline{F_2}} d\theta \, d\varepsilon \tag{11}$$

where,

$$\overline{F_2} = k^{-1} \left[6 \, \frac{M - Tanh(M)}{M^3} \left(2\overline{h_s}\right)^3 + 6(2\overline{h_s}) \left[((1 - \varepsilon \, Cos\theta) - 2\overline{h_s}) + 2\overline{h_s} \, \frac{Tanh(M)}{M} \right]^2 \frac{Tanh(2M)}{(2M)} \right] + \left[\left((1 - \varepsilon \, Cos\theta) - 2\overline{h_s} \right)^3 \right]$$

$$M = \alpha \overline{h_s}, \qquad \alpha = \sqrt{\frac{1}{k\overline{\phi}}}$$

The equations (10) and (11) are analysed numerically and graphs have been plotted.

3. Results and Discussions

In Figs. 2 and 3, load capacity $\overline{W}$ and squeeze time $\overline{T}$ of spherical bearing are plotted with k for various $\overline{h_s}$, the mean height of roughness asperities. It observed that, the load capacity and squeezing time increases with an increase in the value of k which represents a ratio of the viscosity of the peripheral layer to the middle layer. Hence it can be seen that due to high viscosity in the peripheral layer the load capacity and squeezing time increases which is observed in many experiments [12, 13].

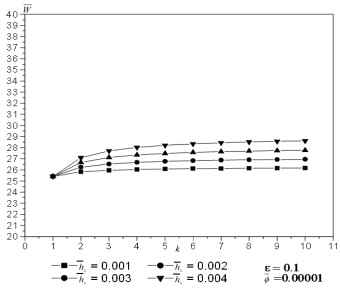

Fig.2. Variation of $\overline{W}$ with k for various $\overline{h_s}$.

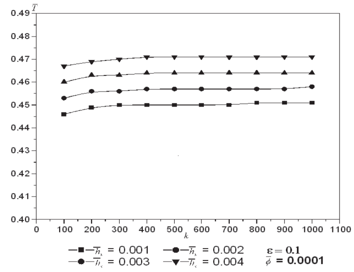

Fig.3. Variation of $\overline{T}$ with k for various $\overline{h_s}$.

In Figs. 4 and 5, $\overline{W}$ and $\overline{T}$ are plotted with $\overline{h_s}$ for various k . It is seen from the figures that the load capacity and squeezing time increases as $\overline{h_s}$ increases and the increase is more as k increases.

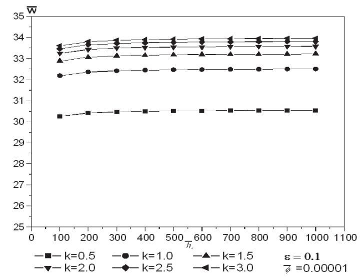

Fig.4. Variation of $\overline{W}$ with $\overline{h_s}$ for various k .

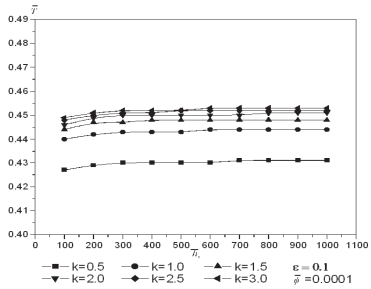

Fig.5. Variation of $\overline{T}$ with $\overline{h_s}$ for various k .

In Figs. 6 and 7, $\overline{W}$ and $\overline{T}$ are plotted with $\overline{h_s}$ for various $\overline{\phi}$, which represents roughness parameter. It is seen that, the load capacity and squeezing time increases for lower values of $\overline{\phi}$ with an increase of $\overline{h_s}$ but decreases for higher values of $\overline{\phi}$ with an increase in $\overline{h_s}$. From the nature of $\overline{\phi}$, it may view that lower values of $\overline{\phi}$ may represents transverse roughness and higher values of $\overline{\phi}$ may represents longitudinal roughness. Hence for transverse roughness, the load capacity and squeezing time increases as the mean height of the roughness parameter, $\overline{h_s}$ increases and for longitudinal roughness the load capacity and squeezing time decreases as the mean height of the roughness parameter, $\overline{h_s}$ increases.

From these figures, it is also observed that the effect of roughness is more pronounced in the case of transversal roughness. The results are consistent with stochastic theory.

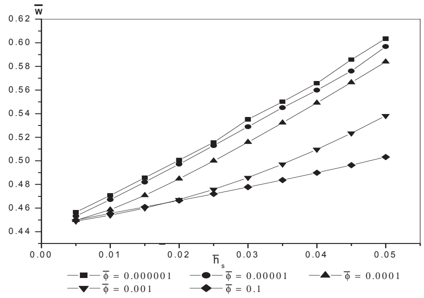

Fig.6. Variation of $\overline{W}$ with $\overline{h_s}$ for various $\overline{\phi}$.

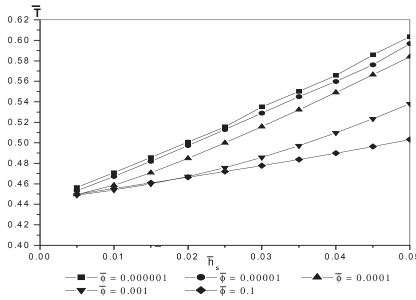

Fig.7. Variation of $\overline{T}$ with $\overline{h_s}$ for various $\overline{\phi}$.

In Fig. 8, the load capacity $\overline{W}$ is plotted with $\overline{V}$, the squeeze velocity for various k . The load capacity increases with an increase of squeeze velocity.

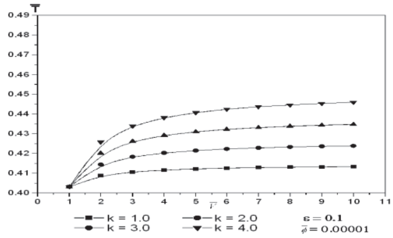

Fig.8. Variation of $\overline{T}$ with $\overline{V}$ for various k .

4. Conclusion

In this paper, a generalized form of Reynolds equation for two surfaces is taken by considering surface roughness at the bearing surfaces. This equation is applied to study the effects of surface roughness for the lubrication of squeeze film of spherical bearings. Expressions for the load capacity and squeezing time are obtained and studied theoretically for various parameters. The load capacity and squeeze time increases with an increase in the value of k which represent a ratio of the viscosity of the peripheral layer to the middle layer. In the case of transverse roughness the load capacity and squeezing time increases as the mean height of surface asperities increases and the load capacity and the squeezing time decreases as the mean height of surface asperities increases in the case of longitudinal roughness. Hence the effect of surface roughness is more pronounced in the case of transverse roughness.

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