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AN ANALYTICAL APPROACH TO INVESTIGATE THE EFFECT OF GROOVED SURFACE ON SHORT JOURNAL BEARING'S PERFORMANCE

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ABSTRACT

In the present study, theoretical studies and approach for finite fluid film journal bearing used. It is a must to investigate the hydrodynamic performance of this kind of bearing due to its wide demand and application into different industries. In this work, Reynolds equation for the numerical part is applied. To investigate how they affect the performance of short journal bearing, an axial groove with various depth and length are considered. The oil film pressure, load carrying capacity and attitude angle for plain and grooved short bearing are calculated, compared. The presented results illustrate that applying one groove at inlet bearing surface, declines the performance of finite fluid film journal bearings.

Keywords: groove bearings, pressure distribution, load carrying capacity, attitude angle and finite journal bearings.

INTRODUCTION

Hydrodynamic journal bearings are widely used in industry, ranging from small application e.g. hard disk drive to the large one such as gas turbine in power plants. Fluid film journal bearings are the most common bearings from this category consisting three essential parts, a circular shaft and a bush, which are called journal and bearing (bushing), respectively and finally a lubricant, which should be a viscos fluid. Lubrication system in this kind of bearings is created by relative motion between journal and bearing to prepare necessarily pressure to separate these elements, make a smoother rotation away from wear and to enhance load carrying capacity of bearing. Meanwhile, to enhance the performance of lubrication, surface texturing can be employed. Many works were dedicated to the study of surface texture [1-3].

Brizmer *et al.* [2] used a number of numerical simulation and a technique named laser surface texturing (LST), to apply micro-dimples with different height, length on bearing surface of long and short (L/D=0.2) fluid film journal bearings. Their results show that regular micro-dimples in partial texturing situation, has this capability to enhance the load carrying capacity and attitude angle. Also, it has special influence on the load capacity of short journal bearing when eccentricity is less than 0.2.

Rahmani, *et al.* [4] employed an optimization procedure to find the best texturing parameters. They examined slider bearings under some different shapes of texture. They used Reynolds equation and their results shown that, some considerable amount of the load carrying capacity can be produced in comparison with the optimum configurations. In their work, it can be seen that increase in the number of textures boost the performance of textured bearing in pure hydrodynamic lubrication regime.

Moreover, Meng *et al.* [5] employed fluid structure interaction (FSI) method and numerical procedure based on Rayleigh – Plesset equation. Their studies have shown, the higher load capacity and the lower friction coefficient can be obtained with compound dimple because of hydrodynamics' behaviour in association with the simple dimple. Furthermore, the mentioned improvement is influenced by the geometry sizes of the compound dimple, the dimple interval, and working parameters of the bearing.

Tala-Ighil *et al.* [6] based on the numerical results and finite difference method (FDM), the most significant characteristics of the journal bearings can be enhanced through a suitable arrangement of the textured area on the contact surface. They considered various arrangements of texture as well and founded that the lubricant film thickness increases nearby to textured area. To improve the hydrodynamic characteristics in journal bearing, texturing for whole bearing surface cannot certainly be a positive procedure. However, through sufficient dimple size and rotational speeds, it can results to a remarkable progressive effects[7].

Additionally, the load carrying capacity has studied by Shen *et al.* [8] and the results clearly demonstrated that dimple structures has a remarkable impact on the load. Meanwhile, to produce more load, cylindrical dimples with rectangular cross-sectional can be better option in comparison with triangular profile. Furthermore, Sfyris *et al.* [9], studied pressure distribution of hydrodynamic journal bearing. To achieve a way for the lubrication of finite journal bearings it has been used the separation of variables to obtain the exact analytical solution of Reynolds equation. Besides, in the experimental study which is conducted by Lu and Khonsari [10], using chemical etching technique, it is shown that with appropriate dimples' dimension, the friction performance can be developed, especially for light viscos oils. Also, the results reveal that the friction performance with etched dimples in the entire bearing surface has a better result than half of bearing surface. Recently, Hamdavi *et al.* [11] showed that for a long

hydrodynamic journal bearings, applying a groove at bearing surface can make a significant enhancement on the static performance. He also compared the results with the static characteristics of plain hydrodynamic journal bearing.

The aim of this article is to study the effects of conducting a single groove with different groove length Θ_g and groove depth H_g on bearing surface of finite fluid film journal bearing Figure-1, on pressure distribution and load carrying capacity

METHOD OF ANALYSIS

In this work, the Dubois and Ocvirk [12] approximation for short bearing L<<D and the steady state operating condition (journal axes remains parallel to bearing axes) is assumed. The classical Reynolds equation and the half Sommerfeld boundary condition is applied. All the present results has been validated with the results of plain bearing under same operating condition of Rao [13]. A grooved journal bearing demonstrated through its schematic in Figure-1, which represents one groove surface. The groove length and depth are Θ_g and H_g, respectively. The film thickness *h* can be expressed as

$$h = c(1 + e\cos\theta) \tag{1}$$

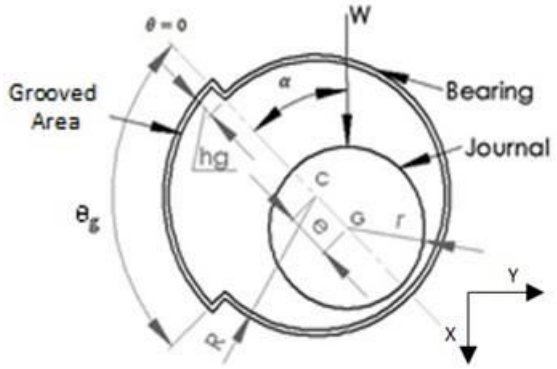


Figure-1. The schematic of grooved journal bearing.

With above assumption for finite journal bearing, the changes in pressure in circumferential direction is small in comparison with the axial pressure, so we can take Reynolds equation as

$$h^3 \frac{d^2p}{dz^2} = 6\mu\omega \frac{\partial h}{\partial \theta} \tag{2}$$

Integrating Equation. (2) twice with respect to z and applying boundary conditions, we have

$$p = 0 \text{at } z = \pm\frac{L}{2} \tag{3}$$

Pressure value is now obtainable as below

$$p = \frac{3\mu\omega}{h^3}\frac{dh}{d\theta}\left(z^2 - \frac{L^2}{4}\right) \tag{4}$$

Where the coordinate system is located midway along the bearing length L. Using the film thickness Equation. (1) and its derivative in respect to t and ignoring the pressure in the diverging wedge, the pressure profile for grooved area $0 \le \theta < \theta_g$ and plain area $\theta_g \le \theta < \pi$ are P_g and P_p, respectively, hence we have

$$P_g = \frac{3\mu\omega\varepsilon}{c^2}\left(\frac{L^2}{4} - z^2\right)\frac{\sin\theta}{(H_g + 1 + \varepsilon\cos\theta)^3} \tag{5a}$$

$$P_p = \frac{3\mu\omega\varepsilon}{c^2}\left(\frac{L^2}{4} - z^2\right)\frac{\sin\theta}{(1 + \varepsilon\cos\theta)^3} \tag{5b}$$

If we consider the equilibrium of the journal, we can obtain the load carrying capacity as below

$$W\cos\alpha = -\frac{\mu\omega r\varepsilon L^3}{2c^2}\left[\int_0^{\theta_g}\frac{\sin\theta\cos\theta}{(H_g + 1 + \varepsilon\cos\theta)^3}d\theta + \int_{\theta_g}^{\pi}\frac{\sin\theta\cos\theta}{(1 + \varepsilon\cos\theta)^3}d\theta\right] \tag{6a}$$

$$W\sin\alpha = \frac{\mu\omega r\varepsilon L^3}{2c^2}\left[\int_0^{\theta_g}\frac{\sin^2\theta}{(H_g + 1 + \varepsilon\cos\theta)^3}d\theta + \int_{\theta_g}^{\pi}\frac{\sin^2\theta}{(1 + \varepsilon\cos\theta)^3}d\theta\right] \tag{6b}$$

And the radial load on the bearing and attitude angle can be found as

$$W = \sqrt{W^2\cos^2\alpha + W^2\sin^2\alpha} \tag{7}$$

$$\emptyset = \tan^{-1}\left(\frac{W\sin\alpha}{W\cos\alpha}\right) \tag{8}$$

RESULTS AND DISCUSSION

From theoretical point of view, it has been observed that applying groove on inlet of fluid film journal bearings' surface with e<<1, can leads to lower magnitudes of the pressure and load carrying capacity. Also, it results to lesser hydrodynamic performance.

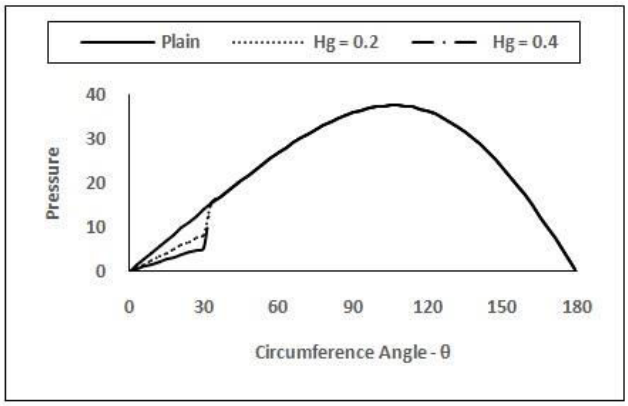
Effect of groove on pressure distribution

For the results presented below, it can be clearly seen that groove pattern decreases the pressure wave wherever groove length has implemented.

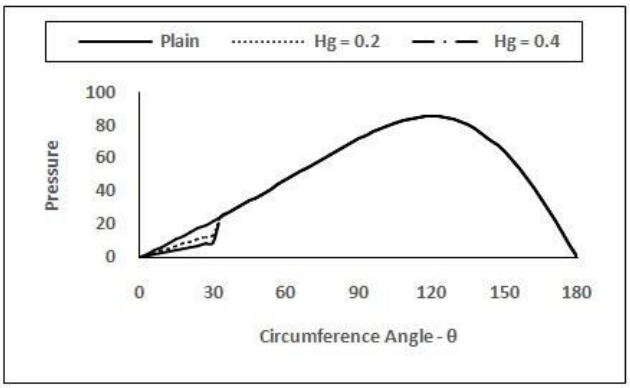
Figure-2(a) and (b), illustrates the effect of applying groove on bearing surface at $\Theta_g = 30^\circ$ and its effect on oil film pressure in different groove depth. In these figures half Sommerfeld is considered and the circumferential angle is from 0 to π. In each figure pressure distribution of plain bearing (Hg = 0) is compared with groove bearing. From figures it can visibly inferred that oil pressure in the groove area has a lower amount and increases to same value in comparison with plain bearing when the eccentricity ratio is 0.1.

Figure-2(a), reveals that the oil pressure in the groove area, dramatically falls by increasing in groove depth from plain to 0.4. However, if we consider all magnitudes of oil pressure in grooved bearing, it can be

seen that exactly after end of groove area their magnitudes increase to same magnitude of plain bearing.



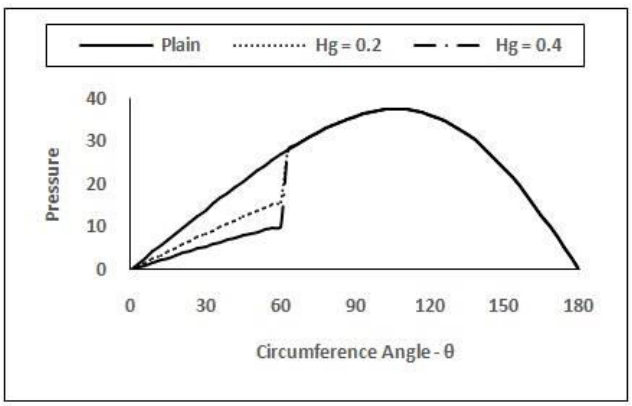
(a)



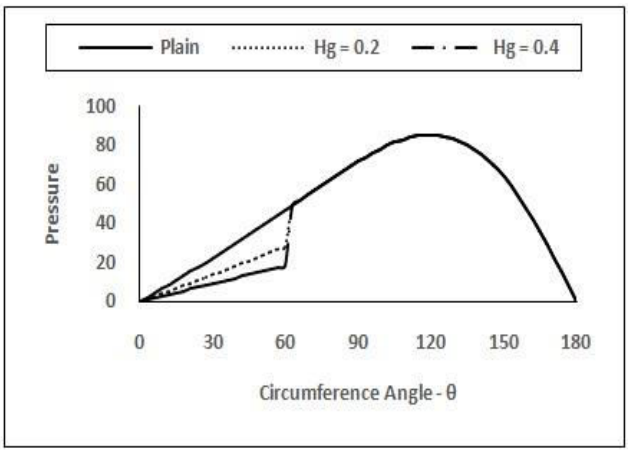
(b)

Figure-2. Pressure distribution at $\Theta_g = 30^\circ$ (a) ε = 0.1 and (b) ε = 0.2

Figure-3(a) and (b), provides the results obtained for pressure distribution when the groove length is considered as 60°. In other hand, the length of grooved area is increased to investigate the effect of groove length on pressure distribution. For a second time, it is obviously clear that applying a groove at inlet of bearing surface cannot produce higher pressure neither making it lengthier nor deeper.



(a)



(b)

Figure-3. Pressure distribution at $\Theta_g = 60^\circ$(a) ε = 0.1 and (b) ε = 0.2

Effect of groove on load carrying capacity

In this section, Figure-4, demonstrates how values of load carrying capacity changes while eccentricity ratio increases. All values are just commence from zero when there is no any eccentricity which means the centre of gravity of journal and bearing are located in same point.

As it can clearly be seen that, magnitudes of load capacity, declines slightly by increasing in depth of groove from 0.2 to 0.4, this is when in total view their values goes up by increasing in eccentricity ratio. The highest value for load in Figure-4 is when the journal bearing is operating on the steady state condition with no groove on its surface. On the other side, the lowest value of load belongs to Hg = 0.4.

Finally, it seems that applying one groove at inlet surface of bearing cannot make neither a remarkable improvement nor keeping the last values of load capacity and its magnitudes goes down. Also, increasing in groove length does not play a significant role to enhance the capability of carrying load.

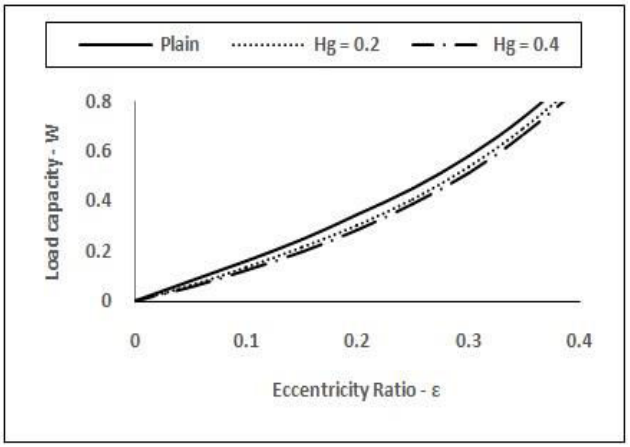


Figure-4. Load carrying capacity vs. eccentricity ratio at $\Theta_g = 90^\circ$.

Effect of groove on attitude angle

The results of attitude angle are presented in Figure-5, which corresponds the difference between values in short plain and grooved journal bearing.

It is crystal clear that in this chart, there is a positive effect on attitude angle when a groove applies at inlet of bearing surface. Once the groove length increase from $\Theta_g = 30^\circ$ to $\Theta_g = 60^\circ$ this remarkable influence is higher clear at low eccentricity ratios, where stability of plain journal bearing is lesser. This figure also reveals that all trends related to groove bearing have this tendency to have as same value as plain bearing when the groove length finish.

The result has been evaluated and compared with what Brizmer *et al.* [2] have done which shows that they are in good agreement.

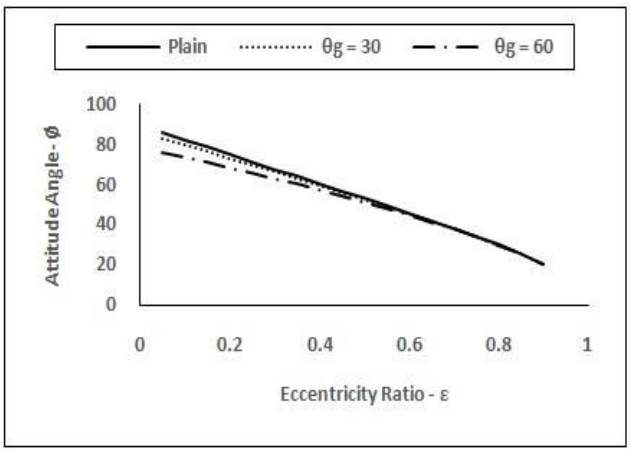


Figure-5. Attitude Angle vs. eccentricity ratio for short plain and grooved journal bearing at ε = 0.2.

CONCLUSIONS

In the present study, due to symmetry of the bearing, only one half of the journal bearing system is studied. In finite fluid film bearing, pressure magnitude cannot boost by using groove pattern at inlet of bearing surface. The most important results from provided figures are:

- Pressure magnitude as well as load carrying capacity has a lower values in grooved bearing in comparison with its magnitude in plain bearing.
- The lowest value of pressure in each related graphs belongs to grooved bearing with deeper groove of $H_g = 0.4$.
- The higher pressure and load magnitudes can be produced when the eccentricity ratio climbs from 0.1 to 0.2 for pressure graph and from 0 to 0.5 into load figure.
- Due to convergent wedge phenomena and approximately after the end of groove area, oil pressure followed by rapid increase up to plains' pressure values.
- Increase in depth and length of groove cannot be an effective way for short bearing to enhance its load carrying capacity.
- Attitude angle have lesser values at low eccentricity ratio for grooved bearing in compare to plain bearing, which it makes groove bearing more stable at low eccentricity. Also, all the attitude angle have almost same magnitudes at higher eccentricity ratio at both plain or groove journal bearings.

NOMENCLATURES

$B_{1,2}$ = Integration Constants
c = Radial clearance
D = Journal diameter
e = Eccentricity
h = Lubricant film thickness
H_g = Groove depth ($H_g = \frac{h}{c}$)
L = Length of the bearing (Z direction)
P1 = Fluid film pressure in grooved area
P2 = Fluid film pressure in plain area
R = Bearing radius
r = Journal radius
W = Bearing load carrying capacity
W_g = Load carrying capacity for grooved bearing
W_p = Load carrying capacity for plain bearing
X, Y = Cartesian coordinate axes
α = Attitude angle
μ = Dynamic viscosity
Ø = Attitude angle
ε = Eccentricity ratio ($n = \frac{e}{c}$)
ω = Angular velocity of the journal
Θ = Angle in the circumferential direction
Θ_g = Groove length

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