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EFFECT OF GEOMETRY OF LOBES AND L/D RATIO ON THE STATIC PERFORMANCE OF 2-LOBE HYBRID JOURNAL BEARING

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Abstract

The two-lobe journal bearings are used in the rotor of turbo generators. These bearings can be designed with different profile of lower and upper lobes. The static and dynamic characteristics of the bearing would be affected with the profile of upper and lower lobes. In this paper effect on the static characteristic of the bearing has been carried out for two lobe elliptical and offset profile. The steady state form of Reynolds equation in two dimensions is solved numerically using finite difference methods.

INTRODUCTION

The characteristic of the multi lobe bearing is the non cylindrical bearing bore. This deviates from conventional, cylindrical, hydrodynamic bearings by having two or more lobes. The lobe radius (R) is larger than the shaft radius (r) by a specific amount. This difference in the radius of the shaft and the lobe results in the formation of a wedge gap in each arc. This gap begins at the oil inlet groove, axially positioned at the widest point of the respective arc. As a rule, the narrowest point of the gap lies in the centre of the lobe. When the shaft begins rotating, basic theory dictates that the lubricant's adhesive effect on the shaft and lobes acts to pull the lubricant into this gap, which narrows in the direction of rotation. Peak pressure develops between the shaft and the bearing. Once this pressure reaches a certain level, it lifts the shaft off the bearing.

Thus, the shaft and the bearing are separated by the lubricant gap. In other words, the shaft operates hydro dynamically with no metal to metal contact. The multilobe bearing theories say that, there are as many individual hydrodynamic carrying forces directed at the center of the shaft as there are lobes. The strength of the individual hydrodynamic force is, among other things, dependent on the width of the wedge gap. The vector total of all the individual carrying forces represents the effective load capacity of the multilobe bearing towards the outside. This results in a strong centering effect being applied to the shaft which produces good concentricity and generates a defined shaft position. By matching the lubricant viscosity to the shaft peripheral speed and the wedge gap shape, the degree of the hydrodynamic carrying force and the bearing friction can be varied to meet individual requirements.

Table1 gives the idea of peripheral speed, surface pressure and Somerfield number for different type of bearing. This table is very useful for designer.

Table1. Bearing bore configuration table

Type of bearing	Peripheral speed	Surface pressure	Somerfield number
Cylindrical	35 m/sec	5 MPa	0-10
Two lobe elliptical	80 m/sec	4 MPa	0-1.5
Two lobe offset bearing	100 m/sec	3.5 MPa	0-1.0

Multilobe bearings are being successfully used in turbochargers, boiler feed pumps, large electric motors, refrigeration turbines, water turbines, centrifugal test rigs, turbine test rigs, noise test rigs, precision drills and lathes, transfer machines, precision borers, grinders.

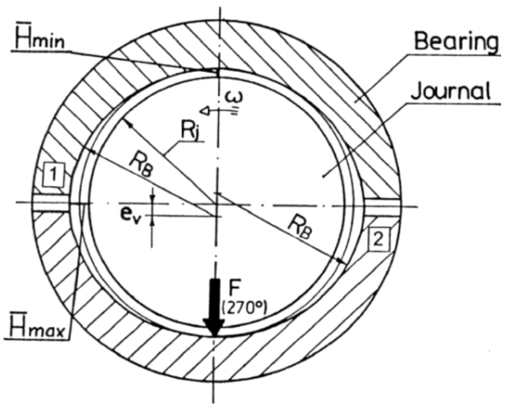

Figure 1. Geometry of two lobe elliptical bearing

In this paper concentration of work is given on fully flooded two lobes elliptical and offset and other combination of upper and lower lobe bearings. This paper gives attention on lobing (Fig.1) and lobing with offset (Fig.2).

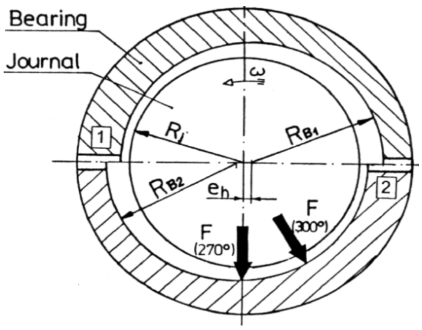

Figure 2. Geometry of two lobe offset bearing

LITERATURE SURVEY

Lubrication theory only gave its first step at the end of the nineteenth century with works such as those of Hirn (1854) and Petrov (1883) which verified in his experiments that drag was actually caused by the shear rate within the fluid rather than by direct interaction between the two surfaces in relative motion. Moreover, Petrov discovered that viscosity was the physical property responsible for drag and ultimately hydrodynamic lift. By the same time Beauchamp Tower (1883) was the first to study experimentally the relation between hydrodynamic pressure generation and the load capacity of rail road journal bearing. But properly it may be stated that hydrodynamic lubrication theory began with the theoretical achievement of Sir Osborne Reynolds. Li et al [1] used Fast Fourier transform analysis to carry out the linear and nonlinear transient analyses of rigid rotors in elliptical, offset half, three-lobe and four-lobe journal bearings. Allaire et al. [2] found the elliptical bearing exhibits the most and the offset half bearing exhibiting the least amount of sub-synchronous vibration among the four analyzed bearings (elliptical, offset half, three-lobe and four-lobe). Mehta et al. [3-8] have done the stability analysis of the multilobe pressure dam bearing for a Newtonian fluid under both steady and turbulent flows lubrication conditions. To the best of author's knowledge, no one has emphasized the effect of lobing geometry on the static performance of bearing.

The present study is mainly concerned with the effect of lobing on the static and dynamic performance of hybrid bearing. This analysis is done for two, three, four lobe bearing. The positive pressure zone for a finite journal bearing satisfying the Reynolds boundary condition is established and non-dimensional pressure is obtained using the Finite Difference Method (FDM). The non-dimensional pressure so obtained is used to evaluate various bearing characteristics (i.e. eccentricity, attitude angle and minimum film thickness)

ANALYSIS

Flow field equations

The Reynolds equation which governs the flow of lubricating oil in the clearance space of a hydrodynamic journal bearing is given by [7].

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\mu}\frac{\partial \bar{p}}{\partial x}\right]+\frac{\partial}{\partial y}\left[\frac{h^3}{\mu}\frac{\partial \bar{p}}{\partial y}\right]=6r\omega\frac{\partial h}{\partial x}+ 12\varepsilon\varphi\sin\theta + 12e\cos\theta \quad (1)$$

The above equation is written in non-dimensionalized by making following substitution:

$$\bar{x} = \frac{x}{D}, \bar{y} = \frac{y}{L}, \bar{h} = \frac{h}{2c}, p = \frac{2\pi p}{\mu\omega}\left(\frac{c}{R}\right)^2$$

$$\theta = \frac{x}{R} = \frac{2x}{D} = 2\bar{x}, \varepsilon = \frac{e}{c}, \alpha = \frac{\varphi}{\omega}, \beta = \frac{\varepsilon}{\omega}$$

The non- dimensionalized equation thus obtained is:

$$\frac{\partial}{\partial \bar{x}}\left[\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{p}}{\partial \bar{x}}\right] + \frac{\partial}{\partial \bar{y}}\left(\frac{D}{L}\right)^2\left[\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{p}}{\partial \bar{y}}\right] = \frac{\pi}{2}\frac{\partial \bar{h}}{\partial \bar{x}} + \pi\varepsilon\alpha \sin 2\bar{x} + \pi\beta \cos 2\bar{x} \quad (2)$$

Fluid film thickness

For an aligned lobe bearing the fluid film thickness of the n^{th} lobe is expressed as [8]:

$$h^n = 1 - (X_j - X_L^n)\cos\theta - (Z_j - Z_L^n)\sin\theta \quad (3)$$

Where (X_j, Z_j), represents the coordinates of the journal centre and (X_L^n, Z_L^n) represents the centers of the lobes.

The expression for maximum film thickness of offset bearing is written as

$$h_{o,max} = \frac{\Delta R_B + e_h}{C_R} \quad (4)$$

$h_{o,\,max}$= Gap height ratio, $\Delta R_B = R_B - R_J$, R_B= bore arc radius, R_J= journal arc radius, C_R= Radial bearing clearance, e_h= offset

so $R_B = R_J + e_h + C_R$,

$$\Delta R_B = C_R + e_h \quad (5)$$

Put values of Eq. (b) in Eq. (a)

$$h_{o,max} = 1 + \frac{2e_h}{C_R}$$

$$h_{o,max} = 1 + \frac{2e_h}{R_B - R_J - e_h} \quad (6)$$

Boundary condition

The boundary conditions pertinent to the problem are given as

(i) $\bar{p}$ = 0, at $\alpha = \alpha_1$ (ii) $\bar{p}$ = 0, at $\alpha = \alpha_2$

(iii) $\frac{\partial \bar{p}}{\partial \alpha} = 0$ at $\alpha = \alpha_2$

(iv) $\bar{p}$ = 0, at $\beta = \pm\frac{L}{D} = \pm\lambda$ (7)

Load carrying capacity

The load carrying capacity of journal bearing is found by integrating the pressure over the positive pressure zone.

$$\bar{F_x} = \frac{F_x C^2}{\mu_r \omega_r R^4} = \int_{-\lambda}^{+\lambda} \bar{p} \cos\alpha \, d\alpha \, d\beta$$

And

$$\bar{F_z} = \frac{F_z c^2}{\mu_r \omega_r R^4} = \int_{-\lambda}^{+\lambda} \bar{p} \sin\alpha \, d\alpha \, d\beta \qquad (8)$$

Equation (7) is integrated numerically to obtain using gauss-Legendre formula over the positive pressure zone.

The various eccentricity ratios and attitude angle for two lobe bearing is given by:

For two lobe elliptical bearing

$$\varepsilon_1^2 = \varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos\varphi$$

$$\varepsilon_1^2 = \varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos\varphi$$

$$\varphi_1 = \frac{5\pi}{4} + \gamma + \sin^{-1}(\varepsilon\cos\varphi/\varepsilon_1)$$

$$\varphi_1 = \frac{5\pi}{4} + \gamma - \sin^{-1}(\varepsilon\cos\varphi/\varepsilon_1)$$

where, ellipticity ratio= $\delta = \left(1 - \frac{c_m}{c}\right)$,]

ε_1 = relative eccentricity for upper lobe,

ε_2 = relative eccentricity for upper lobe,

φ_1= attitude angle for upper lobe,

φ_2= attitude angle for upper lobe.

RESULT AND DISCUSSION

The effect of L/D ratio on the static characteristic of a 2 lobe bearing is shown in Fig. 3 to Fig. 8. The values of L/d ratios used for the analysis are 0.75, 1.0 and 1.5.

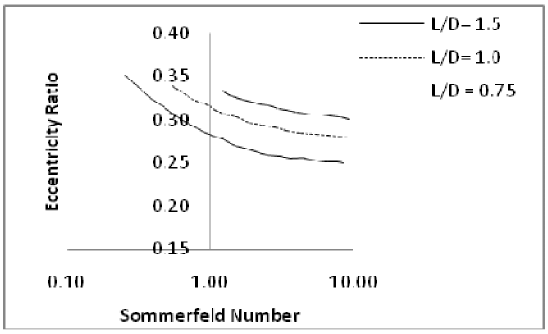

Figure 3. Effect of L/D ratio on eccentricity ratio

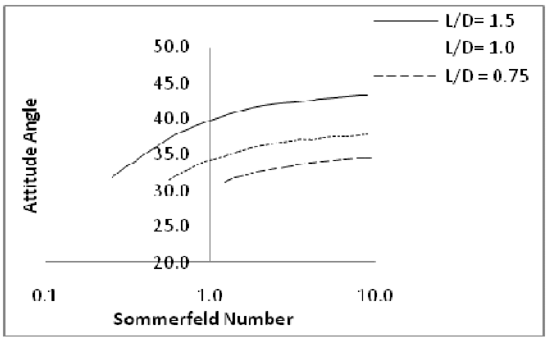

Figure 4. Effect of L/D ratio on attitude angle

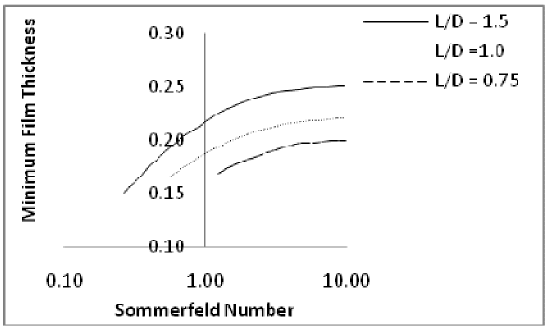

Figure 5. Effect of L/D ratio on minimum film thickness

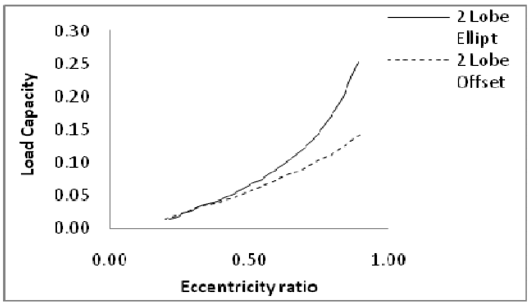

Figure 6. Load capacity Vs eccentricity ratio for L/D=0.5

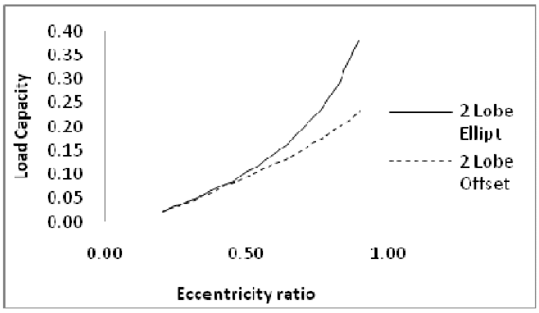

Figure 7. Load capacity Vs eccentricity ratio for L/D=0.75

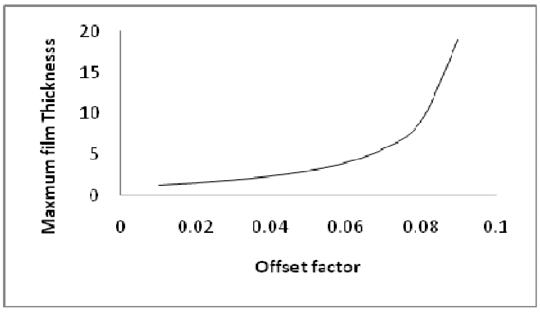

Figure 8. Effect of offset factor ratio on maximum film thickness

It is observed from Fig. 3 and Fig. 4 that with the increase in L/D ratio, eccentricity ratio decreases whereas, the attitude angle increases for exacting value of Somerfield number. The minimum film thickness is observed to increase with an increase in L/d ratio (Fig.6).

CONCLUSION

On the basis of analytical study of the effect of L/D ratio and lobe profile the following conclusion given below:

(1) The eccentricity ratio decreases while attitude angle increases with an increase in L/D ratio.

(2) The minimum oil film thickness increases with an increase in L/d ratio.

(3) The maximum film thickness increases with increases in offset factor (e_h).

(4) The load capacity of the elliptical bearing is better as compared with 2 lobe offset bearing with increase in eccentricity ratio and L/D ratio.

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