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Effect of L/D Ratio on the Performance of a Four-Lobe Pressure Dam Bearing

G. Bhushan, S. S. Rattan, and N. P. Mehta

Abstract**—**A four-lobe pressure dam bearing which is produced by cutting two pressure dams on the upper two lobes and two relief-tracks on the lower two lobes of an ordinary four-lobe bearing is found to be more stable than a conventional four-lobe bearing. In this paper a four-lobe pressure dam bearing supporting rigid and flexible rotors is analytically investigated to determine its performance when L/D ratio is varied in the range 0.75 to 1.5. The static and dynamic characteristics are studied at various L/D ratios. The results show that the stability of a four-lobe pressure dam bearing increases with decrease in L/D ratios both for rigid as well as flexible rotors.

Keywords**—**Four-lobe pressure dam bearing, finite-element method, L/D ratio.

I. INTRODUCTION

THE present trend in the industry is to run the turbomachines at high speeds. The ordinary circular bearings, which are the most common type of the bearings, are found to be unstable at high speeds. It is found that the stability of these bearings can be increased by the use of multilobes and the incorporation of pressure dams in the lobes. The analysis of multi-lobe bearings was first published by Pinkus [1]. It was followed by Lund and Thomson [2] and Malik [3] et al., who gave some design data which included both static and dynamic characteristics for laminar, as well as turbulent flow regimes. The experimental stability analysis of such types of bearing [4]-[5] showed that the analytical stability analysis reflects the general trends in experimental data. The analytical dynamic analysis has shown that non-cylindrical pressure dam bearingsare found to be very stable. L/D ratio is one of the important [6]-[9] parameters that affects the stability of a bearing. The effect of L/D ratio on the stability of circular bearings was discussed by Lund [10], Badgley [11] et al. and Hori[12]. The effect of L/D ratio on the performance of two-lobe and three-lobe pressure dam bearings was studied by Mehta [13] and Rattan [14] respectively. The present study is undertaken to investigate the effect of L/D ratio on the performance of a four-lobe pressure dam bearing supporting rigid and flexible rotors.

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II. NOMENCLATURE

c	: radial clearance
c_m	: minimum film thickness for a centered shaft
$C_{xx}, C_{xy}, C_{yx}, C_{yy}$	: oil-film damping coefficients
$\bar{C}_{xx}, \bar{C}_{xy}, \bar{C}_{yx}, \bar{C}_{yy}$	: dimensionless oil-film damping coefficients, $\bar{C}_{xx} = C_{xx}(\omega c / W)$
C_0, C_1, C_2, C_3, C_4	: coefficients of the characteristic equation
D	: diameter
e	: eccentricity
F	: dimensionless shaft flexibility, W / ck
h	: oil-film thickness, $c(1 + \varepsilon \cos\theta)$
$\bar{h}$	: dimensionless oil-film thickness, $h / 2c$
$2k$	: shaft stiffness
$K_{xx}, K_{xy}, K_{yx}, K_{yy}$	: oil-film stiffness coefficients
$\bar{K}_{xx}, \bar{K}_{xy}, \bar{K}_{yx}, \bar{K}_{yy}$	: dimensionless oil-film stiffness coefficients, $\bar{K}_{xx} = K_{xx}(c / W)$
L	: bearing length
N	: journal rotational speed
O_i	: lobe center of lobe i (i = 1,2,3,4)
p	: oil-film pressure
R	: journal radius
S	: Sommerfeld no., $\frac{\mu NLD}{W}\left(\frac{R}{c}\right)^2$
V	: peripheral speed of journal
W	: bearing external load
x, z (x-	: coordinates for bearing surface peripheral, z-along shaft axis)
ϕ	: attitude angle
$\dot{\alpha}$	: whirl rate ratio, $\alpha = \dot{\phi} / \omega$

$\dot{\beta}$	: squeeze rate ratio, $\dot{\beta} = \dot{\varepsilon}/\omega$
ε	: eccentricity ratio, e/c
δ	: ellipticity ratio, $(1 - c_m/c)$
θ	: angle measured from the line of centers in the direction of rotation
θ_g	: oil-groove angle
ρ	: fluid density
μ	: average fluid viscosity
ω	: rotational speed
v	: dimensionless threshold speed, $\omega(c/g)^{1/2}$
g	: gravitational acceleration const.

III. Analysis

The Reynolds Equation for the laminar flow is:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right) = 6R\omega\frac{\partial h}{\partial x} + 12\varepsilon\dot{\phi}\sin\theta + 12\dot{e}\cos\theta \quad (1)$$

The above equation is non-dimensionalized by making the following substitutions:

$$\bar{x} = \frac{x}{D}, \bar{z} = \frac{z}{L}, \bar{h} = \frac{h}{2c}, \bar{p} = \frac{2\pi p}{\mu\omega}\left(\frac{c}{R}\right)^2,$$
$$\theta = \frac{x}{R} = \frac{2x}{D} = 2\bar{x}, \varepsilon = \frac{\dot{e}}{c}, \dot{\alpha} = \frac{\dot{\phi}}{\omega} \text{ and } \dot{\beta} = \frac{\dot{\varepsilon}}{\omega} \quad (2)$$

The non-dimensionalized equation thus obtained is:

$$\frac{\partial}{\partial \bar{x}}\left(\frac{\bar{h}^3}{12}\frac{\partial \bar{p}}{\partial \bar{x}}\right) + \left(\frac{D}{L}\right)^2\frac{\partial}{\partial \bar{z}}\left(\frac{\bar{h}^3}{12}\frac{\partial \bar{p}}{\partial \bar{z}}\right) = \frac{\pi}{2}\frac{\partial \bar{h}}{\partial \bar{x}} +$$
$$\pi\varepsilon\dot{\alpha}\sin 2\bar{x} + \pi\dot{\beta}\cos 2\bar{x} \quad (3)$$

The various assumptions made in the derivation of the Reynolds Equation are that the fluid is Newtonian, no slip occurs at the bearing surface, inertia terms are neglected, oil viscosity is constant and curvature is negligible. The Reynolds equation is analysed for a pressure profile using the finite element method[15]. The geometry of a four-lobe pressure dam bearing is shown in Fig. 1.

A four-lobe pressure dam bearing is produced by cutting two pressure dams on the upper two lobes and two relief-tracks on the lower two lobes of an ordinary four-lobe bearing. The various eccentricity ratios and attitude angles of the lobes of the four-lobe pressure dam bearing are given by:

$$\varepsilon_1{}^2 = \varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos(\pi/4 - \phi)$$
$$\varepsilon_2{}^2 = \varepsilon^2 + \delta^2 + 2\varepsilon\delta\sin(\pi/4 - \phi)$$
$$\varepsilon_3{}^2 = \varepsilon^2 + \delta^2 + 2\varepsilon\delta\sin(\pi/4 + \phi)$$
$$\varepsilon_4{}^2 = \varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos(\pi/4 + \phi)$$

$$\phi_1 = 5\pi/4 + \gamma + \sin^{-1}(\varepsilon(\sin\pi/4 - \phi)/\varepsilon_1)$$
$$\phi_2 = 7\pi/4 + \gamma + \sin^{-1}(\varepsilon(\cos\pi/4 - \phi)/\varepsilon_2)$$
$$\phi_3 = \pi/4 + \gamma - \sin^{-1}(\varepsilon(\cos\pi/4 + \phi)/\varepsilon_3)$$
$$\phi_4 = 3\pi/4 + \gamma - \sin^{-1}(\varepsilon(\sin\pi/4 + \phi)/\varepsilon_4)$$

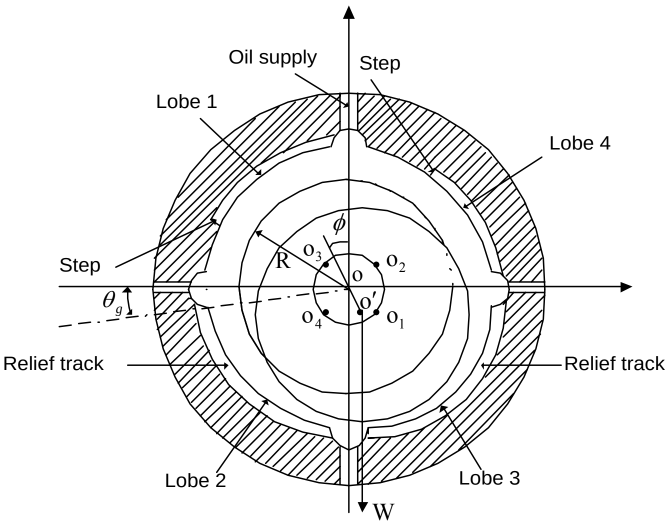

Fig. 1 A four-lobe pressure dam bearing

Each lobe of the bearing is analysed separately. Since the pressure profile has to be symmetrical about the bearing centre line, only half of the lobe is taken for analysis. Fluid pressures at nodal points are taken by applying the Reynolds boundary conditions. The mesh size used is 21 x 5 for each lobe. The resulting matrix is stored in a banded form. Then it is solved by the Gauss-elimination method. Stiffness and damping coefficients are determined separately for each lobe and then added. The values of these stiffness and damping coefficients, shaft flexibility, and dimensionless speed are then used to evaluate the coefficients of the characteristic equation, which is a polynomial of the 6th order for flexible rotors. This characteristic equation has been taken from Hahn [16] and is obtained from the general case of an eccentrically mounted rotor on a flexible shaft.

The characteristic equation is:

$$s^6F^2v^4C_0 + s^5v^4\left(F^2C_1 + FC_2\right) +$$
$$s^4v^2\left(v^2F^2C_3 + 2FC_0 + v^2 + v^2FC_4\right)$$
$$+ s^3v^2\left(2FC_1 + C_2\right) + s^2\left(2Fv^3C_3 + v^2C_4 + C_0\right) +$$
$$sC_1 + C_3 = 0 \quad (4)$$

where

$$C_o = \bar{C}_{xx}\,\bar{C}_{yy} - \bar{C}_{xy}\,\bar{C}_{yx}$$

$$C_1 = \bar{K}_{xx}\,\bar{C}_{yy} + \bar{K}_{yy}\,\bar{C}_{xx} - \bar{K}_{xy}\,\bar{C}_{yx} - \bar{K}_{yx}\,\bar{C}_{xy}$$

$$C_2 = \bar{C}_{xx} + \bar{C}_{yy}$$

$$C_3 = \bar{K}_{xx}\,\bar{K}_{yy} - \bar{K}_{xy}\,\bar{K}_{yx}$$

$$C_4 = \bar{K}_{xx} + \bar{K}_{yy} \tag{5}$$

For a rigid rotor, the value of F (dimensionless flexibility) is taken as zero. The value of F for most of the practical rotors vary from 0.5 to 4.0 and the same has been considered for the case of flexible rotors. The system is considered as stable if the real part of all roots is negative. For a particular bearing geometry and eccentricity ratio, the values of dimensionless speed are increased until the system becomes unstable. The maximum value of speed for which the bearing is stable is then adopted as the dimensionless threshold speed.

The present analysis has been done for the bearing with the following parameters which have been optimized for the best stability[17].

$$\bar{S}_d = 1.0, \bar{L}_d = 0.8, \bar{L}_t = 0.25, \theta_s = 55°, \theta_g = 10°$$

The ellipticity ratio (δ) = 0.5 is selected for the present study. The value of L/D ratio is varied from 0.75 to 1.5 and the bearing is investigated for its static and dynamic characteristics.

IV. Results and Discussion

The effect of L/D ratio on the static characteristics of a four-lobe pressure dam bearing is shown in Figs. 2 to 6. The values of L/D ratios considered for this purpose are 0.75, 1.0 and 1.5. It is observed from the Figs. 2 and 3 that with the increase in L/D ratio, eccentricity ratio decreases whereas, the attitude angle increases for a particular value of Sommerfeld number. The minimum film thickness is observed to increase with an increase in L/D ratio, when considered for a particular value of Sommerfeld number (Fig. 4). Figs. 5 and 6 show the effect of L/D ratio on oil-flow and friction coefficients. There is a considerable fall in the oil-flow, whereas there is no significant change in the friction coefficient with the increase in L/D ratio when considered for a particular value of Sommerfeld number. The effect of L/D ratio on the stability of a four-lobe bearing supporting a rigid rotor is shown in Figure 7. The plots show that both the zone of infinite stability and the minimum threshold speed increase with decrease in L/D ratio. The zone of infinite stability increases from 0.26 to 1.02 and the minimum threshold speed from 9.3 to 18.65 when L/D ratio decreases from 1.5 to 0.75. These effects are due to reduction in load carrying capacity of the bearing.

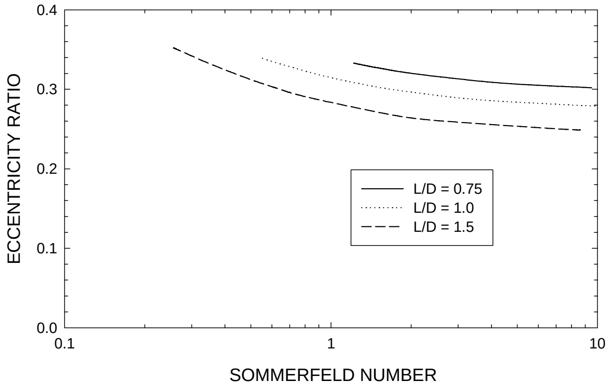

Fig. 2 Effect of L/D ratio on eccentricity ratio

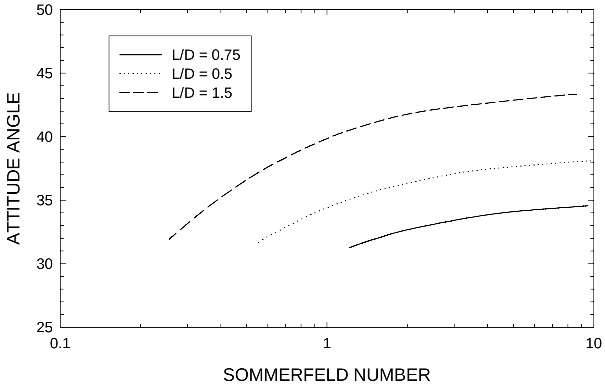

Fig. 3 Effect of L/D ratio on attitude angle

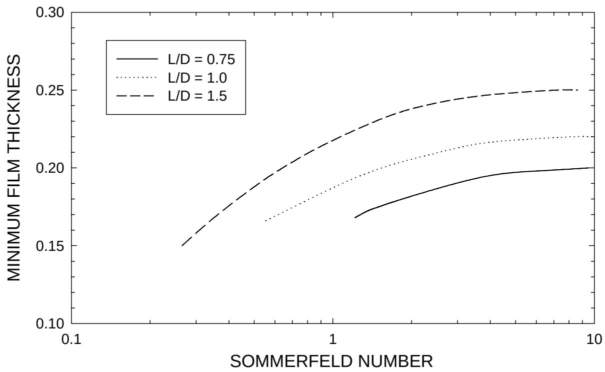

Fig. 4 Effect of L/D ratio on minimum film thickness

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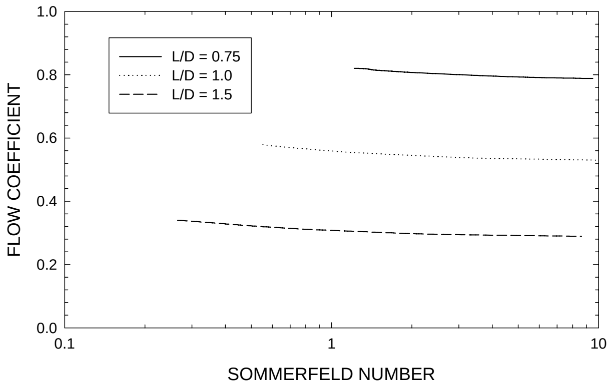

Fig. 5 Effect of L/D ratio on flow coefficient

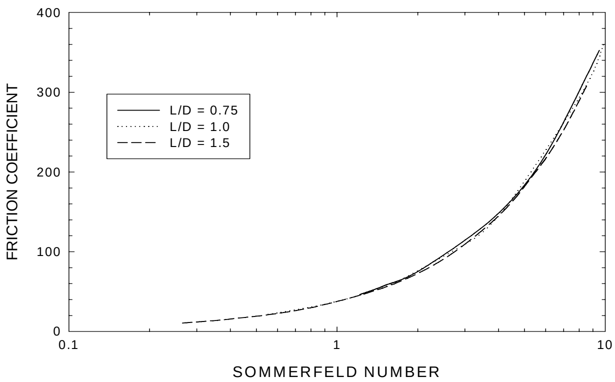

Fig. 6 Effect of L/D ratio on friction coefficient

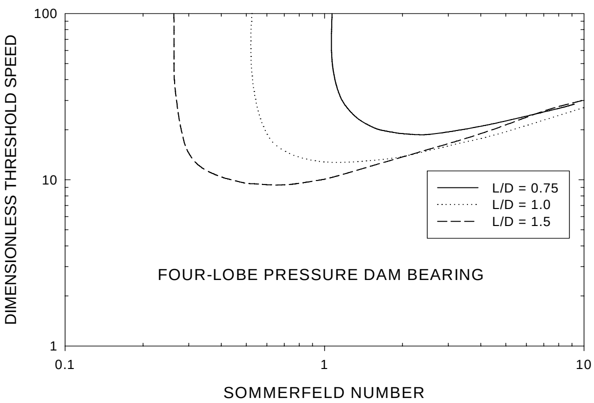

Fig. 7 Effect of L/D ratio on the stability of a four-lobe pressure dam bearing supporting a rigid rotor (F=0)

Figs. 8 and 9 show the effect of L/D ratio on the stability of a four-lobe bearing supporting flexible rotors. The results are found to be similar to that of the bearing supporting a rigid rotor. It is also observed from these plots of the stability that for a particular L/D ratio the minimum threshold speed is reduced with the increase in flexibility of the rotor while there is no change in the zone of infinite stability.

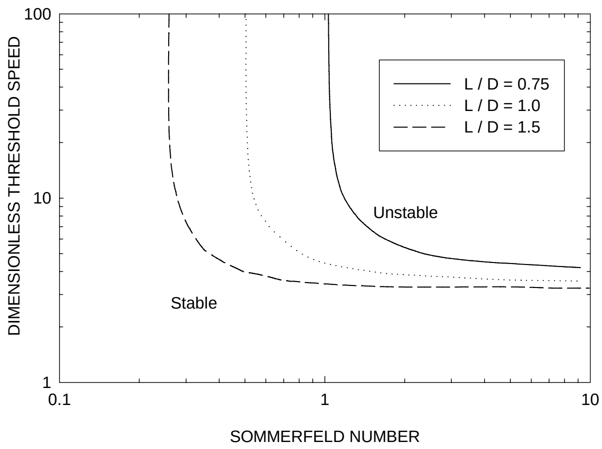

Fig. 8 Effect of L/D ratio on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=0.5)

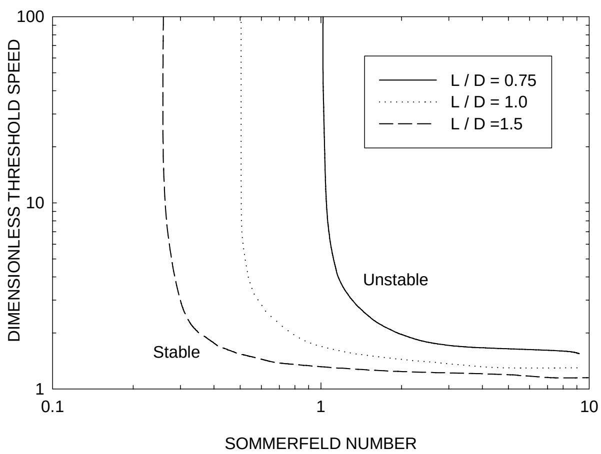

Fig. 9 Effect of L/D ratio on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=4.0)

V. Conclusion

1. The eccentricity ratio decreases while attitude increases with an increase in L/D ratio.
2. The minimum oil-film thickness increases with an increase in L/D ratio.
3. The oil flow coefficient decreases with an increase in L/D ratio.
4. The friction coefficient remains almost unchanged with an increase in L/D ratio.
5. Both the minimum threshold speed and the zone of infinite stability increase with decrease in L/D ratio for a four-lobe bearing supporting rigid rotor as well as flexible rotor. Therefore the stability of a four-lobe pressure dam bearing increases with decrease in L/D ratio.

6. For a particular L/D ratio the minimum threshold speed reduces with increase in the rotor flexibility while the zone of infinite stability remains unchanged.

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