Effect of Groove Angle on the Performance of a Four-Lobe Pressure Dam Bearing with Rigid and Flexible Rotors

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Abstract: *It is observed that among multi-lobe pressure dam bearings, a four-lobe bearing pressure dam possesses good stability characteristics. 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 effect of groove angle on the performance of four-lobe pressure dam bearing has been analyzed in this work. The static and dynamic characteristics have been studied at various groove angles of the bearing. The results indicate that the stability of the bearing decreases with an increase in groove angle.*

1. 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 number, $\frac{\mu NLD}{W}\left(\frac{R}{c}\right)^2$
V	: peripheral speed of journal
W	: bearing external load
x, z	: coordinates for bearing surface (x-peripheral, z-along shaft axis)
ϕ	: attitude angle
α	: whirl rate ratio, $\alpha = \dot{\phi}/\omega$
β	: squeeze rate ratio, $\beta = \dot{\varepsilon}/\omega$
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
ν	: dimensionless threshold speed, $\omega(c/g)^{1/2}$
g	: gravitational acceleration constant

2. 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 bearings [6]-[9] are found to be very stable. The factors affecting the bearing stability are load, L/D ratio, viscosity of the fluid, clearance between the journal and the bearing, type of fluid (compressible or incompressible), rotor unbalance, flow regime (laminar or turbulent), ellipticity ratio, misalignment, friction, shaft and liner flexibility, groove angle etc.[10]-[12]. Groove angle is one of the parameters that affects the stability of a bearing. The effect of grooving and bore shape on the stability of circular, lemon, offset-halves and three-lobe bearings was discussed by Akkok and Ettles [13]. Thermo-hydrodynamic performance of grooved oil journal bearing was studied by Roy [14]. The effect of groove angle on the performance of a four-lobe pressure dam bearing has not been investigated so far. The present study is undertaken to investigate the effect of groove on the performance of a four-lobe pressure dam bearing supporting rigid and flexible rotors.

3. BEARING GEOMETRY

The geometry of a four-lobe pressure dam bearing is shown in Figure 1. A four-lobe pressure dam bearing comprises four lobes whose centers of curvatures are not in the geometrical center of the bearing. Thus, though the individual lobes are circular, the geometrical configuration as a whole is not. A rectangular dam or step of depth S_d and width L_d is cut circumferentially in each of the lobes 1 and 4. The dam starts after the oil hole and subtend arc of θ_s degrees at the center. Circumferential relief-tracks or grooves of certain depth and width L_t are also cut centrally in the lobes 2 and 3 of the bearing. The relief-tracks are assumed to be so deep that their hydrodynamic effects are neglected. For concentric position of the rotor, there are two reference clearances of the bearing: a major clearance c given by a circle circumscribed by the lobe radius and a minor clearance c_m given by an inscribed circle. Thus the center of each lobe is shifted by a distance $e_p = (c - c_m)$ known as ellipticity of the bearing. The various eccentricity and ellipticity are non-dimensionalized by dividing the major clearance c.

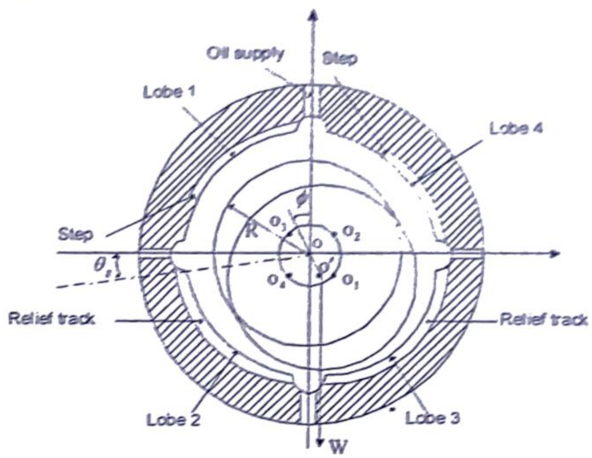

Fig. 1: A four-lobe pressure dam bearing.

Ellipticity ratio $(\delta) = (c - c_m)/c = 1 - c_m/c$

Eccentricity ratio $(\varepsilon) = e/c$

$$\varepsilon_1 = e_1/c, \varepsilon_2 = e_2/c, \varepsilon_3 = e_3/c, \varepsilon_4 = e_4/c$$

The various eccentricity ratios and attitude angles of the lobes of a four-lobe pressure dam bearing are given by:

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

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

4. ANALYSIS

The non-dimensionalized Reynolds Equation for the laminar flow is:

$$\frac{\partial}{\partial \bar{x}}\left(\frac{\bar{h}^3}{12}\frac{\partial \bar{p}}{\partial \bar{x}}\right) + \left(\frac{D}{L}\right)^2 \frac{\partial}{\partial \bar{z}}\left(\frac{\bar{h}^3}{12}\frac{\partial \bar{p}}{\partial \bar{z}}\right) = \frac{\pi}{2}\frac{\partial \bar{h}}{\partial \bar{x}} + \pi\varepsilon\alpha\sin 2\bar{x} + \pi\beta\cos 2\bar{x} \quad (1)$$

The solution of the equation (1) for pressure distribution in the finite element technique is obtained by minimizing the following variational integral [15] over the individual elements:

$$J_e(\bar{p}_e) = \iint_A \left[\frac{1}{2}\frac{h^3}{12}\left\{\left(\frac{\partial \bar{p}_e}{\partial x}\right)^2 + \left(\frac{D}{L}\right)^2\left(\frac{\partial \bar{p}_e}{\partial z}\right)^2\right\} + \frac{\pi}{2} h \frac{\partial \bar{p}_e}{\partial x} + \frac{\pi}{2}\varepsilon\alpha\cos 2x \frac{\partial \bar{p}_e}{\partial x} + \frac{\pi}{2}\beta\sin 2x \frac{\partial \bar{p}_e}{\partial x}\right] dA_e \qquad (2)$$

where $\bar{P}_e$ = dimensionless film pressure in the *e*th element.

Each lobe of the bearing is analysed separately. Since the pressure profiles for the lobes 1 and 4 are symmetrical about the center line of the bearing, only half of the lobe is considered for analysis. Each half is divided into 20 x 4 elements. The mesh size is reduced near the trailing edge where heavy pressures are produced. The resulting matrix in each case is stored in the banded form and is then solved by the Gauss-elimination method.

The Reynolds equation is an elliptical partial differential equation and, hence must be solved as a boundary value problem. According to MaCallion, *et al* [16], for a bearing having oil supplied at zero pressure, the largest possible extent of the positive pressure region is given by the boundary conditions that both pressure and pressure gradients are zero at the breakdown and build up boundaries of the oil-film. However, it has been shown [17] that even by setting the negative hydrodynamic pressures to zero as they occur in any iteration step, the results tend to satisfy the above mentioned boundary conditions in the limit. The latter approach has been followed in the present analysis. Stiffness and damping coefficients are calculated separately for each lobe and then totaled as described by Mehta, *et al* [6]. 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 [18] and is obtained from the general case of an eccentrically mounted rotor on a flexible shaft.

The characteristic equation is:

$$\begin{aligned} & s^6 F^2 v^4 C_0 + s^5 v^4 \left(F^2 C_1 + F C_2\right) + \\ & s^4 v^2 \left(v^2 F^2 C_3 + 2F C_0 + v^2 + v^2 F C_4\right) \\ & + s^3 v^2 \left(2F C_1 + C_2\right) + s^2 \left(2F v^2 C_3 + v^2 C_4 + C_0\right) + \\ & s C_1 + C_3 = 0 \end{aligned} \qquad (3)$$

For a rigid rotor, the value of F is taken as zero. 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 stability threshold curve divides any figure into major zones. The zone above this curve is unstable whereas the zone below this curve is stable. The minimum value of this curve is termed the minimum threshold speed. Mostly the curve has a vertical line, towards the left of which the bearing is stable at all speeds. This portion is called the zone of infinite stability. With an increase in minimum threshold speed the stability curve shifts upwards thus increasing the stable zone. Similarly with an increase in zone of infinite stability the stability threshold curve shifts towards right increasing the stable zone. Thus with increase in minimum threshold speed or zone of infinite stability or both the stability increases. The present analysis has been done for the bearing with the following parameters which have been optimized for the best stability [19].

L/D = 1, $\bar{S}_d$ = 1.0, $\bar{L}_d$ = 0.8, $\bar{L}_t$ = 0.25, θ_s = 55°, δ =0.5

To study the effect of groove angle on the performance of the four-lobe pressure dam bearing, the value of θ_g is varied from 10° to 30° and the bearing is investigated for its static and dynamic characteristics.

5. RESULTS AND DISCUSSION

The effect of groove angle on the static characteristics of a four-lobe pressure dam bearing is shown in Figs. 2 to 6. The values of groove angles considered for this purpose are 10°, 20° and 30°. It is observed from the Fig. 2 that with an increase in groove angle from 10° to 20°, eccentricity ratio increases for a particular value of Sommerfeld number. But when the groove angle increases from 20° to 30°, the eccentricity ratio decreases except for the values of S<0.8. It is observed from the Figure 3 that attitude angle of a four-lobe pressure dam bearing increases with an increase in the groove angle for a particular value of Sommerfeld

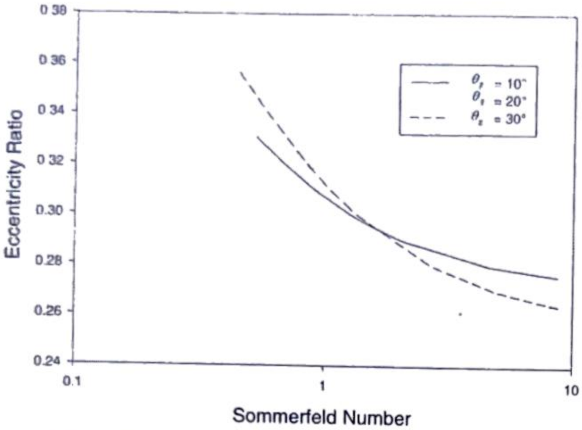

Fig. 2: Effect of groove angle on eccentricity ratio

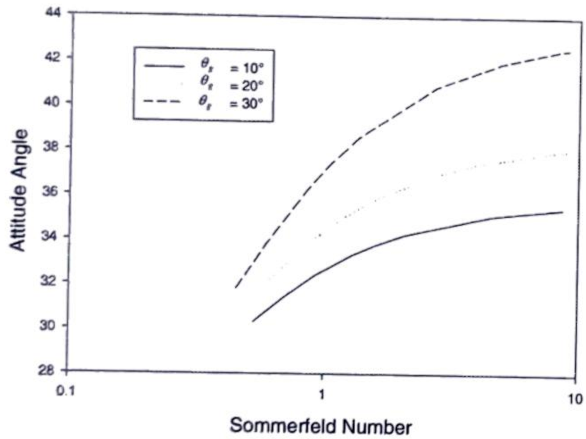

Fig. 3: Effect of groove angle on attitude angle

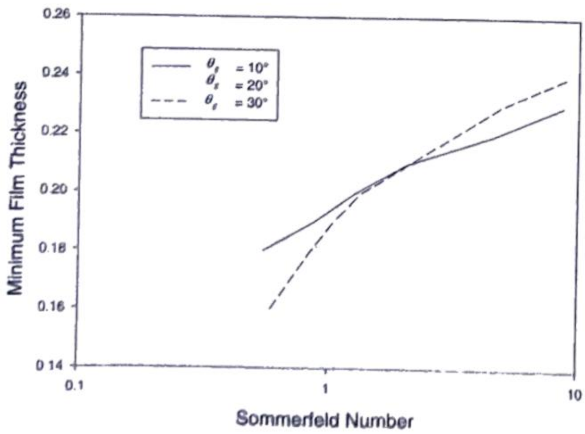

Fig. 4: Effect of groove angle on minimum film thickness

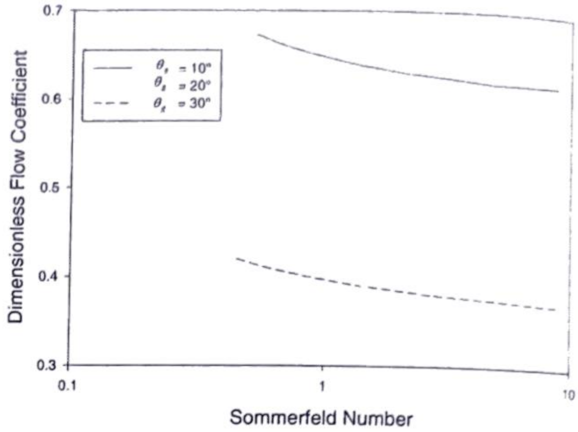

Fig. 5: Effect of groove angle on flow coefficient.

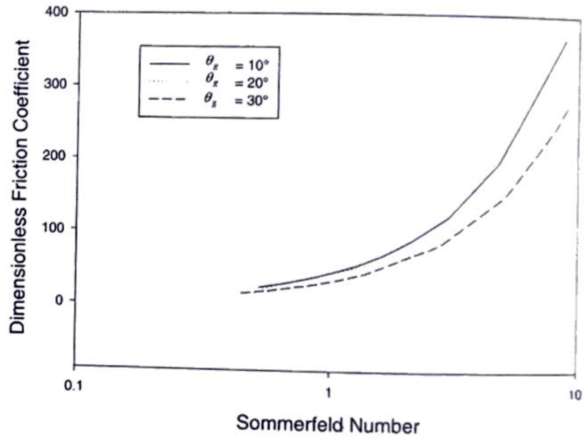

Fig. 6: Effect of groove angle on friction coefficient.

number. The Figure 4 depicts the variation of minimum film thickness with Sommerfeld number. The minimum film thickness is observed to decrease with an increase in groove angle from 10° to 20°, when considered for a particular value of Sommerfeld number. It is also observed that when the groove angle increases from 20° to 30°, the minimum film thickness increases except for the values of $S<0.8$, where it decreases. Figures 5 and 6 show the effect of groove angle on oil-flow and friction coefficients. There is a considerable fall in the oil-flow and a slight reduction in the friction coefficient with the increase in groove angle when considered for a particular value of Sommerfeld number.

The effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a rigid rotor is shown in Fig. 7. It is observed from the figure that both, the zone of infinite stability as well as the minimum threshold speed, decreases with an increase in groove angle. Also the decrease is minimal when the groove angle changes from 10° to 20°. The zone of infinite stability decreases from 0.53 to 0.51 with an increase in groove angle from 10° to 20°, while the decrease is from 0.51 to 0.45 when the groove angle increases from 20° to 30°. The value of minimum threshold speed are 16.15, 13.75 and 9.8 for groove angles 10°, 20° and 30° respectively.

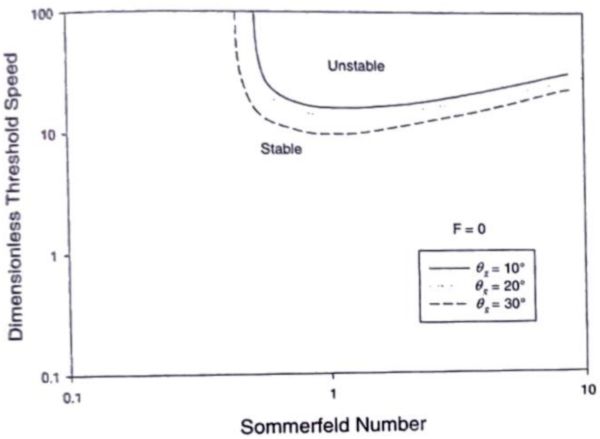

Fig. 7: Effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a rigid rotor (F=0).

Figures 8 to 11 show the effect of load orientation on the stability of a four-lobe pressure dam bearing supporting a flexible rotor. The effect of flexibility of a rotor on the stability of the bearing is considered by using non-zero values of F in the characteristic equation. The increase in the value of F indicates more flexibility of a rotor. The value of dimensionless flexibility F of most of the practical rotors may vary from 0.5 to 4 and the same range has been considered. The similar effects are observed for the bearing supporting a flexible rotor as for a rigid rotor. It is also observed from the plots that for a particular groove angle, as the flexibility of the rotor increases, the zone of infinity remains unchanged and the minimum threshold speed decreases. For groove angle of 30°, the value of zone of infinite stability remains 0.45 as the rotor flexibility F is varied from 0.5 to 4. For the same groove angle of 30°, the values of minimum threshold speed are 3.1, 2.2, 1.55 and 1.1 for the F = 0.5, 1, 2 and 4 respectively.

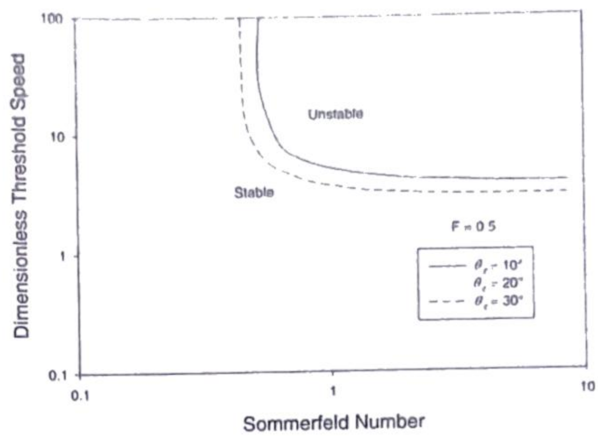

Fig. 8: Effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=0.5).

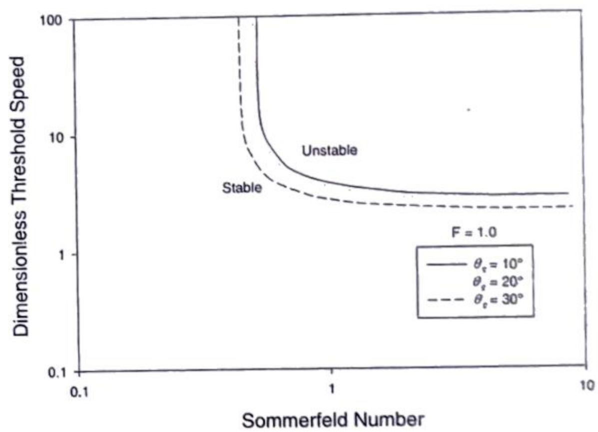

Fig. 9: Effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=1.0).

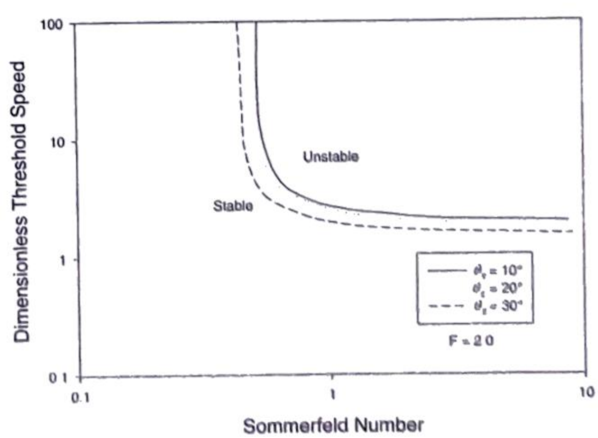

Fig. 10: Effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=2.0).

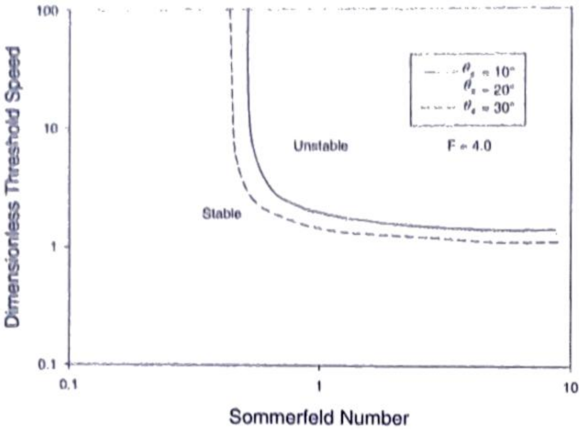

Fig. 11: Effect of groove angle on the stability of a four-lobe pressure dam bearing supporting a flexible rotor (F=4.0).

6. CONCLUSIONS

1. The eccentricity ratio of a four-lobe bearing increases with an increase in the groove angle from 10° to 20° but decreases when groove angles is increased to 30°.
2. Attitude angle of a four-lobe bearing increases with an increase in the groove angle for a particular value of Sommerfeld number.
3. Minimum film thickness of a four-lobe bearing decreases with an increase in the groove angle from 10° to 20° but increases when groove angles is increased to 30°.
4. Dimensionless oil-flow coefficient and friction coefficient decrease with an increase in groove angle.
5. For a four-lobe bearing supporting a rigid rotor as well as a flexible rotor, both the zone of infinite stability as well as minimum threshold speed increases with an increase in groove angle.
6. With an increase in the flexibility of the rotor, the zone of infinite stability remains unchanged while the minimum threshold speed decreases.

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