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Effect of turbulence on the performance of an inverted three-lobe Pressure dam bearing

N. K. Batra, Gian Bhushan and N. P. Mehta

Abstract- This paper analyzes the effect of turbulence on the performance of an inverted three- lobe pressure dam bearing which is produced by incorporating a pressure dam in the upper lobe and two relief tracks in the lower two lobes of an ordinary inverted three- lobe bearing. For analysis purpose different values (0, 1500, 3000, and 6000) of Reynolds number are considered to study the effect of turbulence. The results shows that for an inverted three lobe bearing supporting rigid or flexible rotors, the zone of infinite stability as well as minimum threshold speed increases with increase in turbulence, thus indicating better stability at higher turbulence level.

Key Words: Inverted three- lobe pressure dam bearing, finite element method, multi-lobes

1 INTRODUCTION

THE analysis of multi-lobe bearings was first published by Pinkus[1]. This was followed by Lund and Thomson[2] and Malik, et al [3], who gave some design data which included both static and dynamic characteristics for laminar as well as turbulent flow regimes.

The analytical dynamic analysis Nicholas and Allaire[4,7] and Mehta, et al[9,11] and Mehta and Singh[10] has shown that cylindrical pressure dam bearings are found to be very stable. Also an experimental stability analysis of such types of bearings [5] showed that the analytical stability analysis reflects the general trends in the experimental data.

The effect of turbulence on the stability was first investigated by Constantinescu [6] who concluded that the stable zone decreases with increase in Reynolds number for plain journal bearing. Nicholas and Allaire[7] found that turbulence increases the stability of finite pressure dam bearings. Soni, et al [8] found that turbulence improves the stability of non circular bearings. Mehta [9-13] found that the threshold speed increases where zone of stability decreases with increase in turbulence in pressure dam bearings. The present study is undertaken to investigate the effect of turbulence on the performance of inverted three-lobe pressure dam bearing supporting either rigid or flexible rotor.

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2 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:

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

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

The non- dimensionalized equation thus obtained is:

$$\frac{\partial}{\partial\bar{x}}\left(\frac{\bar{h}^3}{k_x}\frac{\partial\bar{p}}{\partial\bar{x}}\right)+\left(\frac{D}{L}\right)^2\frac{\partial}{\partial\bar{z}}\left(\frac{\bar{h}^3}{k_z}\frac{\partial\bar{p}}{\partial\bar{z}}\right)=\frac{\pi}{2}\frac{\partial\bar{h}}{\partial\bar{x}}+$$

$$\pi\varepsilon\dot{\alpha}\sin 2\bar{x}+\pi\dot{\beta}\cos 2\bar{x} \quad (3)$$

Where $k_x=k_z$ =12 for laminar flow (4)

$$\left.\begin{aligned} k_x &= 12 + 0.0136\ \mathrm{Re}\ 0.9 \\ k_z &= 12+0.0043\ \mathrm{Re}\ 0.96 \end{aligned}\right\}\ \text{for turbulent flow} \quad (5)$$

These correction factors for the turbulent flow were suggested by Constantinescu and Galetuse [14]. The various assumption made in the derivation of Reynolds equation are that the fluid is Newtonian, no slip occurs at bearing surface, inertia forces are neglected , oil viscosity is constant and curvature is negligible as the height of the fluid film is very small as compared to span and length. The Reynolds equation is analysed for a pressure profile using finite element method [15]. The value of this Reynolds number based on major radial clearance is

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defined as

$$\mathrm{Re}_c = \frac{\rho V c}{\mu} \qquad (6)$$

Local Reynolds number at any nodal point Re is

$$\mathrm{Re} = \frac{\rho V h}{\mu} = 2\,\mathrm{Re}_c\,\bar{h} \qquad (7)$$

k_x and k_z are calculated by calculating Re from equation (7) for each point in the finite element mesh and turbulence correction factors are applied at the nodal point level.

3 Bearing Geometry

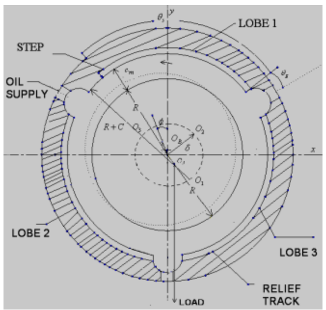

Fig.1 An inverted three–lobe pressure dam bearing.

Figure 1 shows the geometry of an inverted three- 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. The film area of the hydrodynamic oil film to be analysed by finite element method is divided in to mesh of finite elements. Elements involving linear variation of pressure are simplest for dynamic analysis of two dimensional hydrodynamic problems and the same has been considered for the present analysis.

An analysis based on variational principle is used to derive a set of simultaneous algebraic equations with nodal film pressure unknown. Fluid pressure at nodal points is taken by applying the Reynolds boundary conditions. The resulting matrix is stored in the banded form. Stiffness and damping coefficients are determined separately for each lobe and then added [16]. The values of these stiffness and damping coefficients, shaft flexibility, and dimensionless speed are then used to evaluate the coefficients of the characteristic equation [17], which is a polynomial of the 6th order for flexible rotors as given below(for a rigid rotor, F=0):

This characteristic equation has been given by

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

Where

$$\begin{aligned}
C_0 &= \bar{C}_{xx}\bar{C}_{yy} - \bar{C}_{xy}\bar{C}_{yx}\\
C1 &= \bar{K}_{xx}\bar{C}_{yy} + \bar{K}_{yy}\bar{C}_{xx} - \bar{K}_{xy}\bar{C}_{yx} - \bar{K}_{yx}\bar{C}_{xy}\\
C_2 &= \bar{C}_{xx} + \bar{C}_{yy}\\
C_3 &= \bar{K}_{xx}\bar{K}_{yy} - \bar{K}_{xy}\bar{K}_{yx}\\
C_4 &= \bar{K}_{xx} + \bar{K}_{yy}
\end{aligned} \qquad (8)$$

For a rigid rotor, the value of F (dimensionless flexibility) 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 curves divide any figure into two major zones. The zone above this curve is unstable, whereas the zone below is stable. The minimum value of this curve is termed as the minimum threshold speed. Mostly, the curve has a vertical line, towards the left side of which the bearing is stable at all speed. This portion is called the zone of infinite stability.

The present analysis has been done for the bearing with the following parameters

L/D=1.0, $\bar{S}_d$=1.5 $\bar{L}_d$ = 0.8,

$\bar{L}_t$ =0.25, θ_s =85^0 θ_g =10^0

The value of ellipticity ratio (δ) = 0.5 is selected for present study. The effect of turbulence on attitude angle, eccentricity ratio, oil film coefficient has been also observed by varying the Reynolds numbers in steps (0, 1500, 3000, and 6000). The effect of flexibility of rotor on the stability of bearing is considered by using non zero values of F (0.5 and 2.0) in the characteristics equation.

4 Results and Discussion

The effect of turbulence on the performance of inverted three lobe pressure dam bearing is shown in figures 2 to 9. The static characteristics are shown in figure 2 to 6 where's dynamic characteristics are shown from figure 7 to 9. The various relations to calculate minimum oil film thickness, oil flow and friction coefficients are given by Mehta [9].

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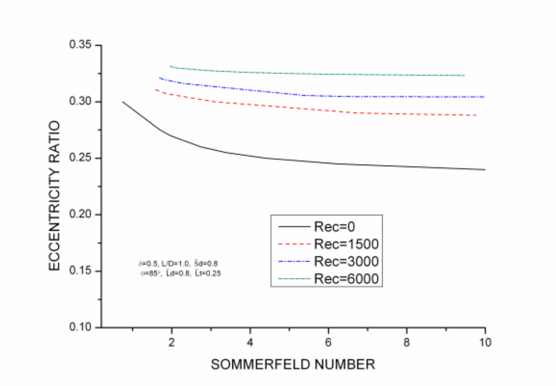

Figure 2 Effect of turbulence on eccentricity ratio

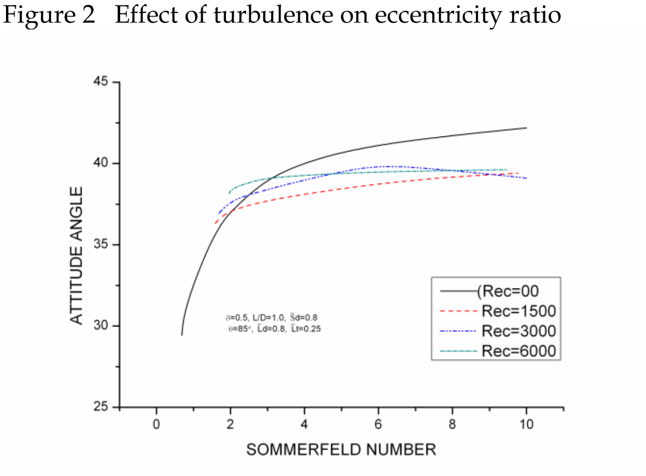

Figure 3 Effect of turbulence on attitude angle

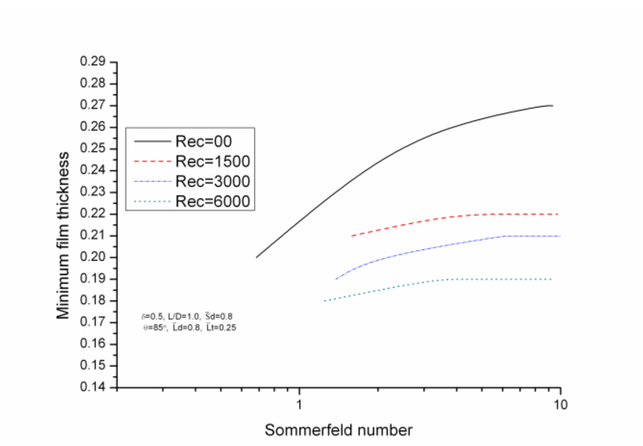

Figure 4 Effect of turbulence on minimum film thickness

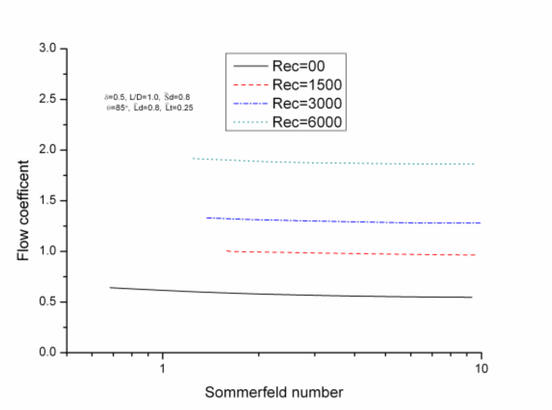

Figure 5 Effect of turbulence on oil flow coefficient

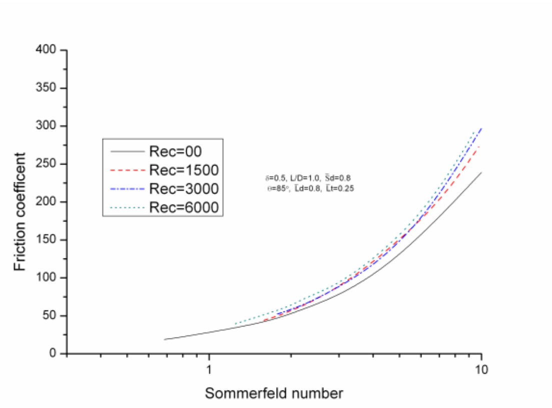

Figure 6 Effect of turbulence on friction coefficient

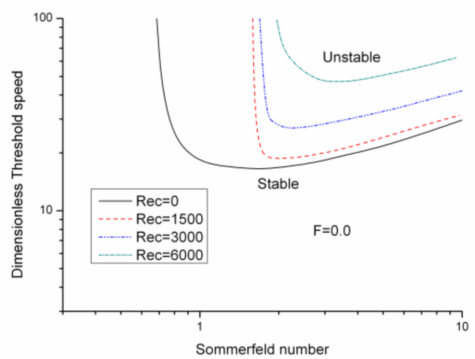

Figure 7 Effect of turbulence on the stability of inverted three lobe bearing supporting a rigid rotor (F=0)

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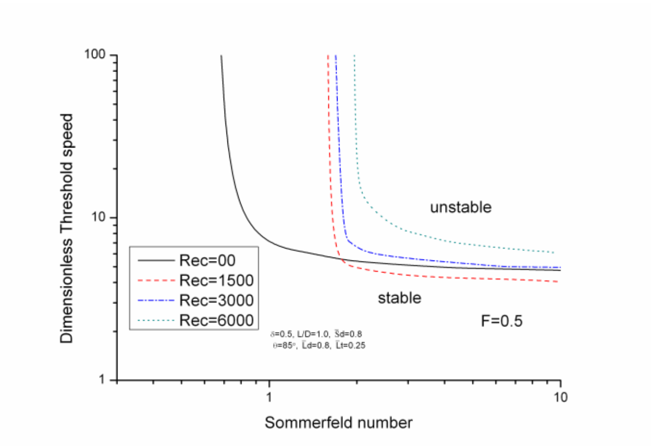

Figure 8 Effect of turbulence on the stability of inverted three lobe bearing supporting a flexible rotor (F=0.5)

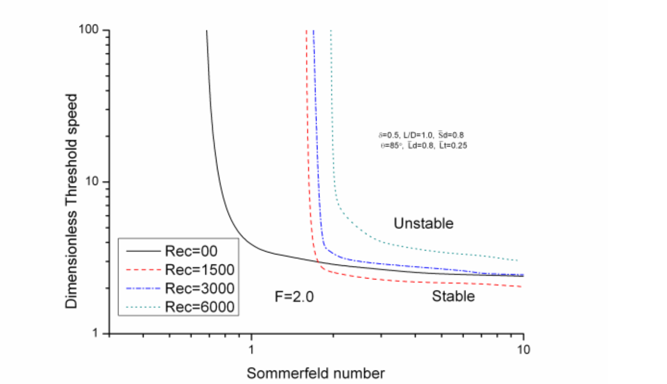

Figure 9 Effect of turbulence on the stability of inverted three lobe bearing supporting a flexible rotor (F=2.0)

The effect of turbuleance on the eccentricity ratio of the bearing is shown in the figure 2. It is observed from the plots that the eccentricity ratio increases with increase in the value of turbulence, for a particular value of sommerfeld number. As the bearings become more stable when operating at higher eccentricity ratio, the effect of turbulence indicates a better operation of the bearing. The variation of attitude angle vs sommerfeld number is shown in figure 3 with different turbulence levels. The figure depicts that with increase in Reynolds number the attitude angle the increases with low value of sommerfeld number and decreases with increase in the value of sommerfeld number. However for a particular value of Reynolds number, the attitude angle increases with increase in sommerfeld number, the increase being more at lower values of Reynolds number. The variation of minimum film thickness with sommerfeld number with different turbulence level is shown in figure 4. The minimum film thickness is observed to reduce with the increase in turbulence level for a particular value of the sommerfeld number.

Figure 5 and 6 shows the effect of turbulence on the oil – flow and friction coefficients, respectively. Both are observed to increase with increase in the turbulence level for a particular value of sommerfeld number.

Figure 7 shows the effect of turbulence on the stability of inverted three lobe pressure dam bearing supporting a rigid rotor. The zone of infinite stability as well as the value of minimum threshold speed is observe to be increase with the turbulence level ,thus indicating better stability of the bearing at higher turbulence levels. The values of zone of stability of Reynolds number 0, 1500, 3000and 6000 are up to S=0.684, 1.59, 1.786, and 1.094 while the values of minimum threshold speed are15.6, 18.3, 19.05, and 46.8 respectively.

The effect of turbulence on the stability of inverted three-lobe bearing supporting a flexible rotor is shown in figures 9 and 10. The same effects, as for the case of a rigid rotor, are observed for the bearing supporting a flexible rotor.

5 Conclusions

1. The values of eccentricity ratio, oil flow coefficients and friction coefficient increases with increase in turbulence.
2. The attitude angle increases with increase in turbulence value of the sommerfeld number less than 1.9 while it decreases for value of sommerfeld number greater than 1.9
3. Turbulence increases both the minimum threshold speed and zone of infinite stability of inverted three lobe pressure dam bearing supporting either rigid or flexible rotor, thus increasing its stability.

Notation

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 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

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$\bar{K}_{xx}, \bar{K}_{xy}, \bar{K}_{yx}, \bar{K}_{yy}$: Dimensionless oil- film stiffness coefficients, $\bar{K}_{xx} = K_{xx}\left(c/W\right)$

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 $\dfrac{\mu NLD}{W}\left(\dfrac{R}{c}\right)^2$

V : Peripheral speed of journal

W : bearing external load

x, z : coordinates for bearing surface (x-peripheral, z-along shaft axis)

ϕ : Attitude angle

$\dot{\alpha}$: whirl rate ratio, $\dot{\alpha} = \dot{\phi}/\omega$

$\dot{\beta}$: 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

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