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Effect of Turbulence on the Performance of Finite Offset-Halves Pressure Dam Bearings with Elastic Liner

Surjit Angra[1], Sushma Rani[2], Vinod Kumar[3]

[1,2,3]Department of Mechanical Engineering,
National Institute of Technology, Kurukshetra, India
[1]angrasurjit@yahoo.com

Abstract: *In the recent past the use of EHD lubrication theory is enlarged to include the study of hydrodynamic bearings as significant distortion of the bearing element were observed under the action of hydrodynamic pressures. The present paper describes the effect of turbulence on the performance of finite offset-halves pressure dam bearings with elastic liner. Solution of bearing considering the flexibility effects of liner involves the simultaneous solution of Reynolds equation in the lubricant and the elasticity equations in the bearing liner. Finite element method has been used to calculate the deformation of the bearing liner induced by the pressure on the fluid film which in turn is influenced by this deformation causing an increment in the oil film thickness. The range of turbulence selected will cover most of the cases of the practical interest. It is observed that stability of offset-halves pressure dam bearing increases both in terms of zone of infinite stability and minimum threshold speed with increase in turbulence. The flexibility of the liner also increases the stability of offset halves pressure dam bearings.*

Keywords: *Elastohydrodynamic, Pressure Dam Bearings*

1. INTRODUCTION

Elastohydrodynamic studies of some ordinary bearings have indicated that the flexibility of the bearing liner considerably affects the performance of the bearings. As the trend these days are towards high speeds, therefore, the effect of turbulence also becomes important in these bearings. The present paper describes the effect of turbulence on the performance of finite offset-halves pressure dam bearings with elastic liner. Mehta and Singh [1] presented the stability analysis of finite offset-halves pressure dam bearing and concluded that the threshold speed is increased as compared to elliptical pressure dam bearing.

The performance characteristics of some two-lobe pressure dam bearing is reported by Mehta [2], who also showed that pressure dam improves stability. Oh. and Huebner [3] studied the elastohydrodynamic effects of circular Bearing and found that the bearing performance is significantly affected by the elastic effect. Effect of the elastic deformation of a bearing liner on performance of finite offset-halves pressure dam bearing was considered by Angra et.al. [4]. Raju et.al. [5] studied the effect of turbulence on the stability of elliptical pressure dam bearings with elastic liner and concluded that turbulence improves the stability of these bearing. Mehta et.al [6] studied the stability of circular pressure dam bearing with couple stress lubricant.

2. BEARING GEOMETRY

Figure 1 shows the geometry of an offset-halves pressure dam bearings. The centre of each half is shifted by a distance d_h from the bearing centre O_B, along the split axis. d_h is called the horizontal preset. The centre of upper half O_U is on the right side of O_B, while the centre of lower half O_L, is on the left side of O_B for counter clockwise direction of rotation and will interchange for the clockwise direction of the journal. For a concentric shaft position there are two reference clearances: a minor clearance given by an inscribed circle denoted by c_m and a major clearance c given by a circle circumscribed on the bearing. For a given position of the journal centre O_J, there are three eccentricities: bearing eccentricity e, upper half eccentricity e_1 and lower half eccentricity e_2.

The various eccentricities and horizontal presets are non-dimensionalized by major radial clearance c as given in the following

$\varepsilon = e/c$, $\varepsilon_1 = e_1/c$, $\varepsilon_2 = e_2/c$ and $\delta_h = d_h/c$

Eccentricity ratios of the two halves of this bearing can be calculated from the following equations

$$\varepsilon_1^2 = \varepsilon^2 + \delta_h^2 - 2\varepsilon\delta_h \sin\phi$$

$$\varepsilon_2^2 = \varepsilon^2 + \delta_h^2 + 2\varepsilon\delta_h \sin\phi$$

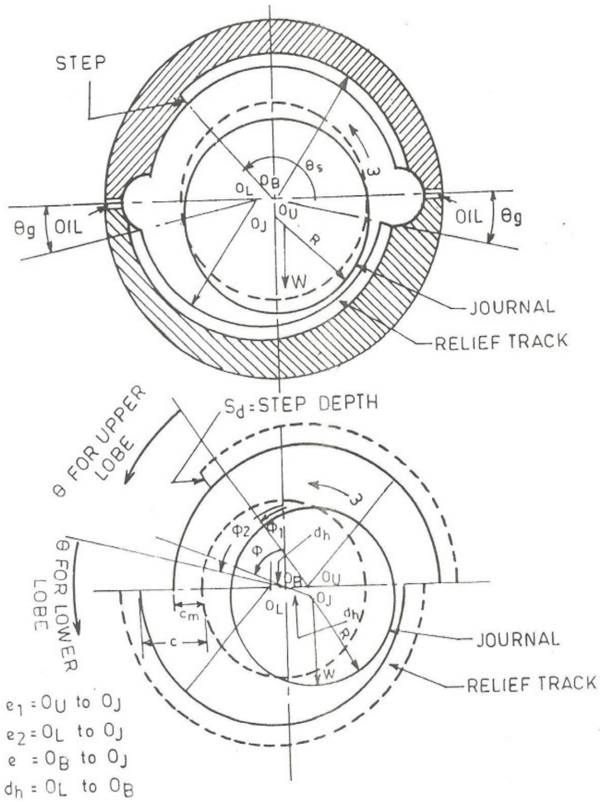

Fig. 1. Offset-Halves Pressure Dam Bearing

Angles ϕ_1 and ϕ_2 obtained from the fig.1 are given in degrees by

$$\phi_1 = \tan^{-1}\left(\frac{\varepsilon \sin\phi - \delta_h}{\varepsilon \sin\phi}\right) \times 180/\pi$$

$$\phi_2 = \tan^{-1}\left(\frac{\varepsilon \sin\phi + \delta_h}{\varepsilon \cos\phi}\right) \times 180/\pi$$

3. ANALYSIS

The method of solving the Reynolds equation for dynamic loading and stability analysis are discussed in ref. 4.

4. RESULTS AND DISCUSSION

Figure 2 shows the circumferential variation of fluid film pressures at the centre line, termed as centre-line pressures for finite offset-halves pressure dam bearings with elastic liner. These figures give the film pressure from 0° to 180° in the upper lobe and 180° to 360° in the lower lobe. For the upper lobe extent of pressure distribution remains the same, whereas, peak pressure increases with the increase in turbulence. In the lower lobe there is redistribution of pressures. Peak pressure increases from 11 at REC=0 to 18.8 at REC=2500.

It can be seen from figures 3, 4 & 5 that eccentricity ratios ε, ε_1 & ε_2 increase as the Reynolds number is increased for any value of the flexibility of the liner. From fig. 6, it can be seen that as Reynolds number increases the attitude angle also increases. The range of attitude angle becomes small for higher value of Reynolds number. For a particular Reynolds number as the Sommerfeld number increases attitude angle also increases. The effect of turbulence on the dimensionless flow & friction coefficient is shown in fig. 7 & 8 respectively. It is observed that the oil flow & friction coefficient increase with the increase in Reynolds number.

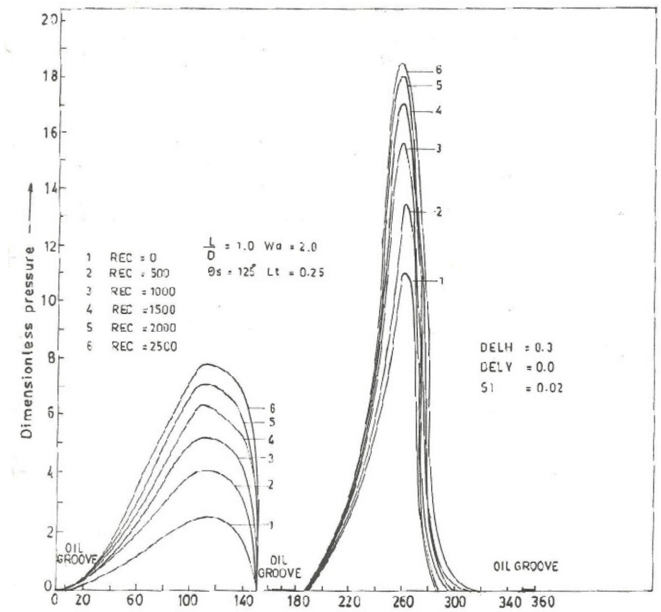

Fig. 2. Maximum Pressure over Each Lobe

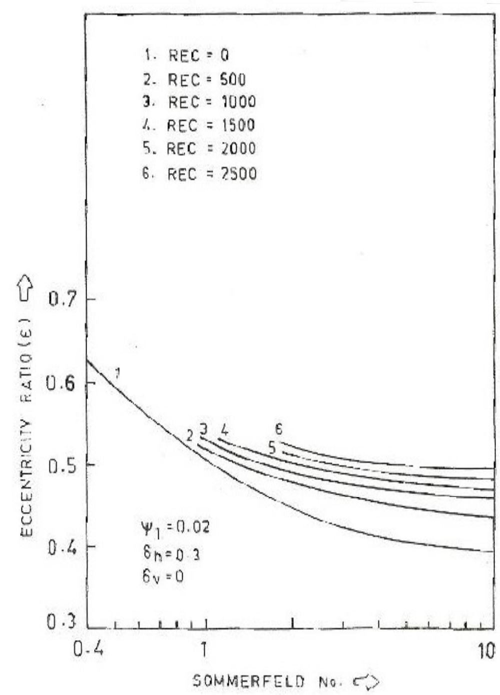

Fig. 3. Sommerfeld No. Vs. Eccentricity Ratio (ε)

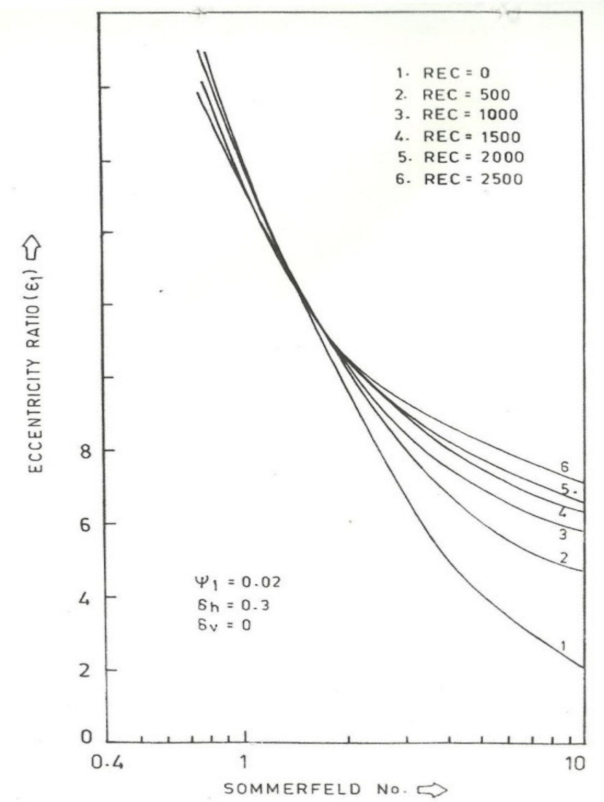

Fig. 4. Sommerfeld No. Vs. Eccentricity Ratio (ε_1)

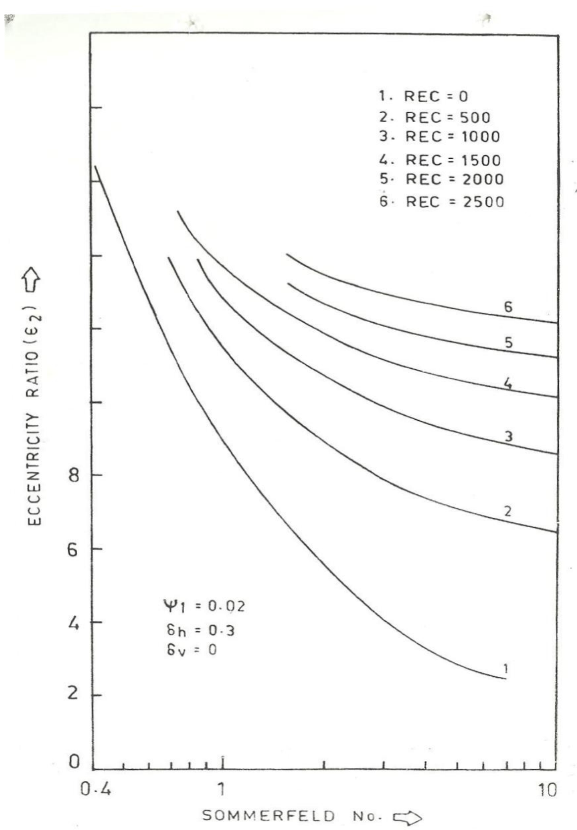

Fig. 5. Sommerfeld No. Vs. Eccentricity Ratio (ε_2)

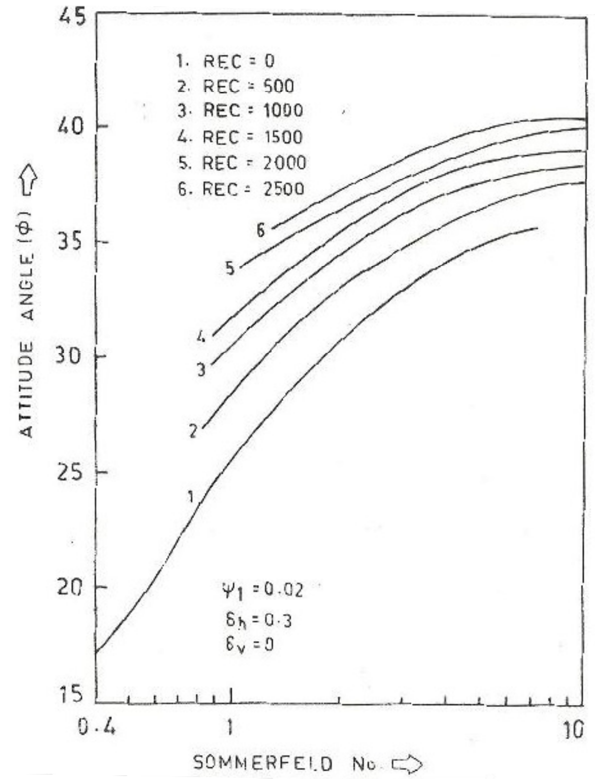

Fig. 6. Sommerfeld No. Vs. Attitude Angle

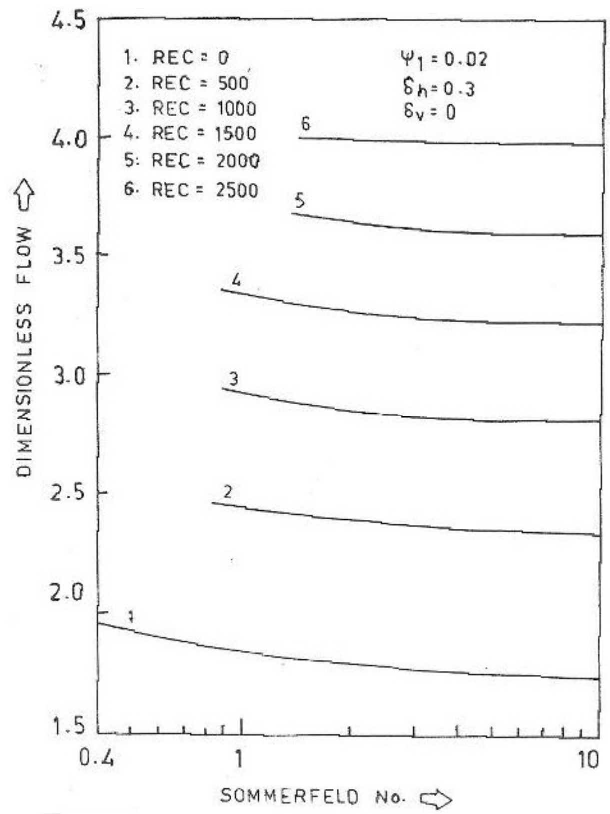

Fig. 7. Sommerfeld No. Vs. Oil Flow Coefficient

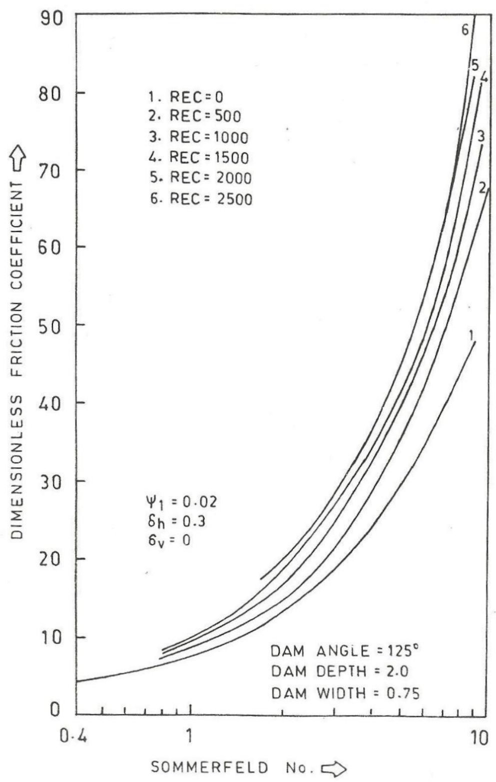

Fig. 8. Sommerfeld No. Vs. Friction Coefficient

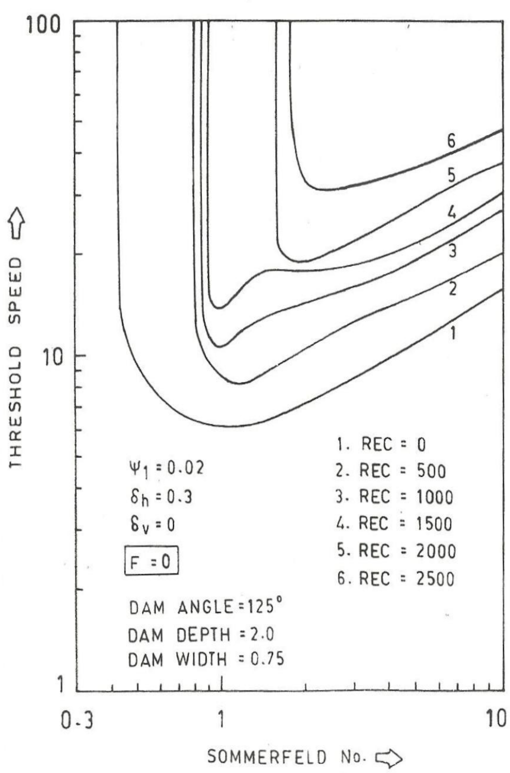

Fig. 9. Sommerfeld No. Vs. Threshold Speed

The effect of turbulence on the stability of this bearing supporting a rigid rotor is shown in fig. 9. It is observed that the zone of infinite stability increases with the increase in the value of Reynolds Number in the investigated range. The minimum threshold speed also increases with the increase in turbulence.

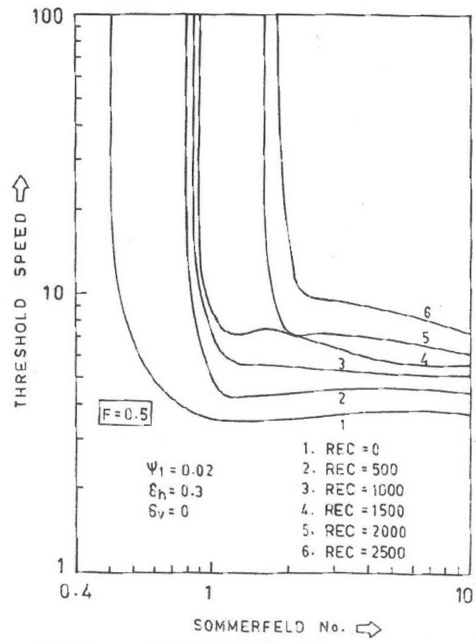

Fig. 10. Sommerfeld No. Vs. Threshold Speed

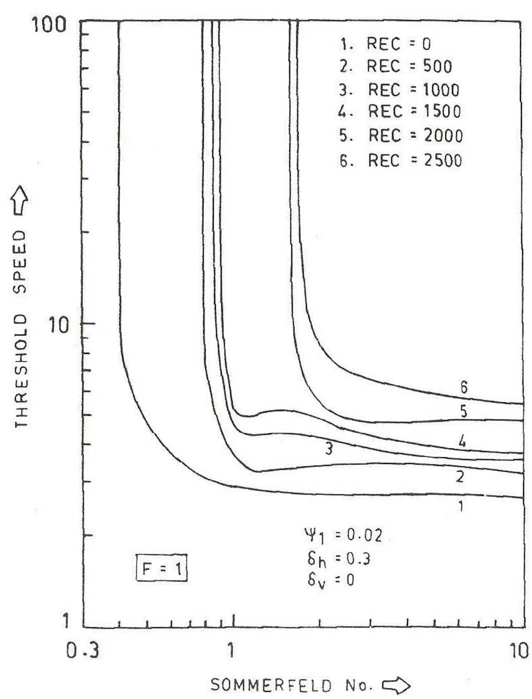

Fig. 11. Sommerfeld No. Vs. Threshold Speed

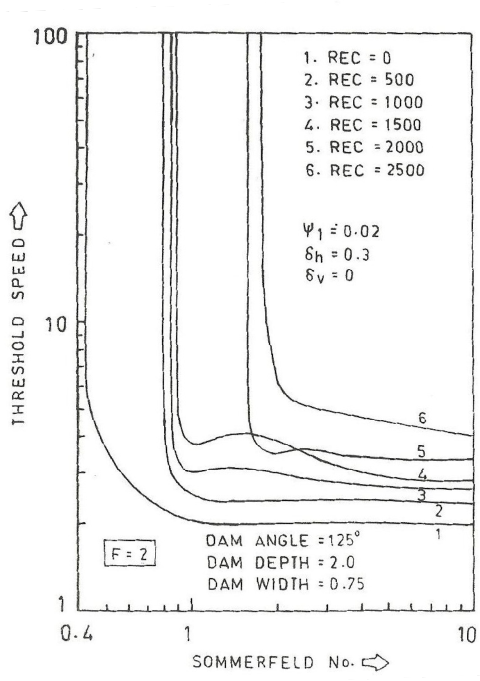

Fig. 12. Sommerfeld No. Vs. Threshold Speed

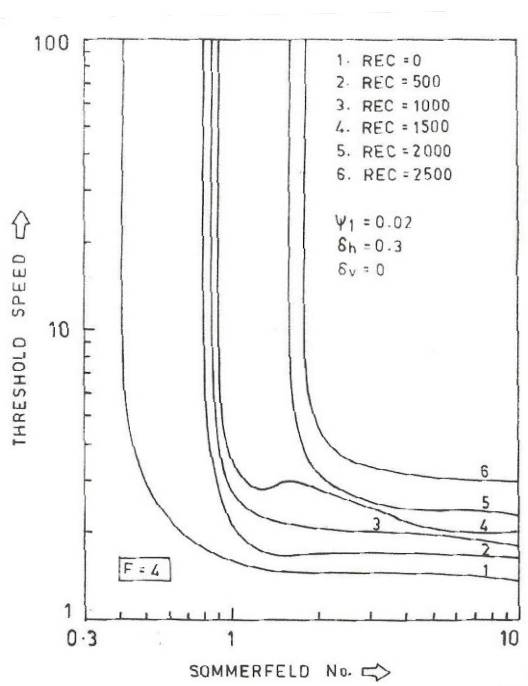

Fig. 13. Sommerfeld No. Vs. Threshold Speed

Figures 10-13 show the variation of Sommerfeld Number with dimensionless threshold speed for flexible rotors with different value of REC. It is observed from these figures that there is improvement in the zone of infinite stability with increase in the turbulence. The zone of infinite stability is not affected by rotor flexibility.

5. CONCLUSIONS

1. The peak pressure in both the lobes increase with turbulence for a particular value of deformation coefficient.
2. Eccentricity ratios, attitude angle, oil flow coefficient and friction coefficient increase with turbulence for a particular value of ε.
3. For rigid rotors the zone of infinite stability and minimum threshold speed increases with the increase in turbulence.
4. Rotor flexibility adversely affects the minimum threshold speed, but the zone of infinite stability remains unaffected.

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