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RESEARCH

Performance Characteristics of a Misaligned Single Pad Externally Adjustable Fluid-Film Bearing

Externally adjustable fluid film bearing has been devised whereby the hydrodynamic conditions can be changed as required in a controlled manner. Principal feature of the bearing is the facility to adjust its radial clearance and circumferential film thickness gradient. Unlike a tilting pad or a conventional partial arc bearing, this bearing has a cantilevered externally adjustable bearing element or pad that can be given an independent and controlled position inputs along the radial direction as well as about its leading edge. Externally adjustable pad bearing will perform as a conventional partial arc bearing when the adjustments are set to zero. This paper deals with the effect of misalignment on steady state characteristics of a centrally loaded 120° single pad externally adjustable fluid film bearing. The bearing has an aspect ratio of one and operates over a wide range of eccentricity ratios, adjustments, adiabatic parameters and degrees of misalignment. Static performance characteristics calculated are presented in terms of load carrying capacity, attitude angle, friction variable and total end leakage. Reynolds equation incorporated with simplified adiabatic model of Pinkus and Bupara is solved using finite difference method. A comparative study predicts that, static performance of the bearing is superior with negative radial and tilt adjustments.

Keywords: *Static characteristics, externally adjustable, tilt and radial adjustments, single pad, misalignment, simplified adiabatic solution.*

1. INTRODUCTION

Journal misalignment is an important factor that can induce a disruption of the bearing performance. The phenomenon of misalignment, in various forms of journal bearings like, plain, grooved and profiled, has been increasingly studied over the last sixty years. However, these forms of journal bearings are reactive to the operating loads and conditions and can offer only a little improvement on the performance. Another limitation is that, in all these bearings the bearing surface once manufactured and put in place cannot be altered. Hence, adjustable bearing is one option. Adjustable fluid film bearing has been devised whereby the hydrodynamic conditions can be changed as required in a controlled manner.

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Martin [1], in 1999, did the first study entirely devoted to adjustable pad bearing. He outlined the principles of thermo-hydrodynamic lubrication and viscosity variations, together with an expanded version of the governing pressure field equation as related to the novel adjustable hydrodynamic fluid film bearing. In 2002, Martin and Parkins [2] demonstrated that continuously adjustable pad bearing with inverse orientations, i.e., a rotor on a stationary shaft; has excellent attributes like high speed, high stability, low power absorption, low cost and ease of maintenance and replacement and proved that it has superior static characteristics. In 2002, Martin [3] expanded the governing Reynolds equation to take into account the non-uniform variations in fluid-film thickness and also predicted operating characteristics of one form of continuously adjustable hydrodynamic bearing. In 2004, Martin [4] designed a multi-body dynamics rig for testing a novel rotor fluid-film bearing system and gave a description of the technique to derive displacement coefficients. In 2008, Shenoy and Pai [5] studied the static performance of a single pad externally adjustable fluid film bearing having conventional orientation and concluded that a reduction in the film thickness improves the performance of such bearings. All the above studies consider a perfectly aligned adjustable bearing.

This paper investigates the effect of misalignment on the steady state performance characteristics of a single pad externally adjustable fluid film bearing for a variety of simulated operating conditions. Figure 1 shows the schematic diagram of single pad externally adjustable fluid film bearing having conventional orientation, i.e., rotating shaft.

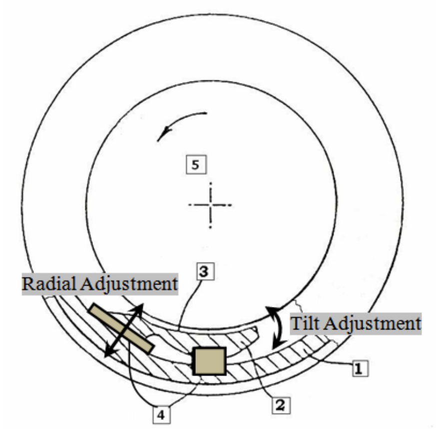


Figure 1: Schematic diagram of a single pad externally adjustable fluid film bearing.

2. THEORY

Pressure distribution in the clearance space between journal and bearing is governed by Reynolds equation.

$$\frac{\partial}{\partial x}\left[h^3\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[h^3\frac{\partial p}{\partial z}\right]=6U\eta\frac{\partial h}{\partial x}+12\eta\frac{\partial h}{\partial t} \tag{1}$$

Under steady state condition, time dependent term in Eqn. 1 is neglected. The governing equation in non-dimensional form will then be,

$$\frac{1}{\bar{\eta}}\left[\frac{\partial}{\partial\theta}\left(\left(\bar{h}\right)^3\frac{\partial\bar{p}}{\partial\theta}\right)\right]+\frac{1}{\bar{\eta}}\left(\frac{R^2}{L^2}\right)\left[\frac{\partial}{\partial\bar{z}}\left(\left(\bar{h}\right)^3\frac{\partial\bar{p}}{\partial\bar{z}}\right)\right]=6\frac{\partial\bar{h}}{\partial\theta} \tag{2}$$

Film thickness equation given by Guha [6], for a misaligned journal, is modified to by superimposing the effect of adjustments [7] provided to the pad.

$$\bar{h}=1+\varepsilon\cos\theta'+\varepsilon'\left(\bar{z}-0.5\right)\cos\left(\theta'-\alpha\right)+R_{adj}+T_{adj} \tag{3.a}$$

$$\varepsilon'=2DM\left\{\left(\left(1+R_{adj}+T_{adj}\right)^2-\left(\varepsilon\sin\alpha\right)^2\right)^{1/2}-\varepsilon\left|\cos\alpha\right|\right\} \tag{3.b}$$

For simplicity, only misalignment in one plane is considered in this work, that is $\alpha=0$. It is noted that, when

$\alpha=0$ and $R_{adj}=T_{adj}=0$;

$\varepsilon'=2DM\left(1-\varepsilon\right)$.

A simplified adiabatic model proposed by Pinkus and Bupara [8], is employed to study the effect of temperature on the bearing performance. This adiabatic model [8] assumes an exponential relationship between viscosity and temperature and further relates the temperature within the film at any angular position to the film thickness in the circumferential direction. The required parameters are taken from Martin [1].

$$\bar{\eta}=e^{-\beta(T-T_i)} \tag{4.a}$$

$$T-T_i=\frac{1}{\beta}\ln\left(1+E\int_{\theta_1}^{\theta_2}\frac{d\theta}{\left(\bar{h}\right)^2}\right) \tag{4.b}$$

2.1 Pressure Boundary Condition

Pressure along the bearing edges were set to zero. Mathematically it is given as,

$$\bar{p}=\bar{p_a}=0;\ at\ \bar{z}=0\ and\ \bar{z}=1.0\ \text{and}$$
$$\bar{p}=\bar{p_a}=0;\ at\ \theta=\theta_1\ and\ \theta=\theta_3\ . \tag{5}$$

Cavitation is allowed to occur at ambient pressure by setting all calculated negative pressure equal to zero throughout the iterative solution scheme. This implies that, the lubricant film ruptures and reforms when,

$$\bar{p}=\frac{\partial\bar{p}}{\partial\theta}=0 \tag{6}$$

2.2 Steady State Performance Characteristics

The load carrying capacity along the line of centers and in its perpendicular directions is evaluated in the non dimensional form.

$$\bar{W}_r=-\int_0^1\int_{\theta_1}^{\theta_2}\bar{p}\cos\theta'\,d\theta\,d\bar{z} \tag{7.a}$$

$$\bar{W}_t=\int_0^1\int_{\theta_1}^{\theta_2}\bar{p}\sin\theta'\,d\theta\,d\bar{z} \tag{7.b}$$

Resultant force acting on the journal

$$\bar{W}=\sqrt{\left(\bar{W}_r\right)^2+\left(\bar{W}_t\right)^2} \tag{8}$$

$$\phi_0=\tan^{-1}\left(\frac{\bar{W}_t}{\bar{W}_r}\right) \tag{9}$$

After each calculation, the attitude angle calculated from Eqn. 9 is compared with the assumed value of the attitude angle $\left(\psi\right)$. The value of ψ is modified with a small increment of ψ and the Reynolds equation Eqn. 2 is solved using this modified value until ψ is equal to ϕ_0.

2.3 Oil Flow

Total volume flow rate $\overline{Q}_z$ is evaluated as,

$$\overline{Q}_z = \overline{Q}_1 + \overline{Q}_2 = \int_{\theta_1}^{\theta_2} \frac{\overline{h}^3}{12\overline{\eta}} \frac{\partial \overline{p}}{\partial \overline{z}}\bigg|_c d\theta \qquad (10)$$

Slope of pressure curves at the sides of the bearing is given as, $\left.\frac{\partial \overline{p}}{\partial \overline{z}}\right|_c$

$$\overline{Q}_1 = -\int_{\theta_1}^{\theta_2} \frac{\left(\overline{h}\right)^3}{12\overline{\eta}} \frac{\partial \overline{p}}{\partial \overline{z}}\bigg|_{\overline{z}=0} d\theta \qquad (11.a)$$

$$\overline{Q}_2 = -\int_{\theta_1}^{\theta_2} \frac{\left(\overline{h}\right)^3}{12\overline{\eta}} \frac{\partial \overline{p}}{\partial \overline{z}}\bigg|_{\overline{z}=1} d\theta \qquad (11.b)$$

2.4 Friction Parameter

Friction Force as per Pinkus and Sternlicht, [9], is given as,

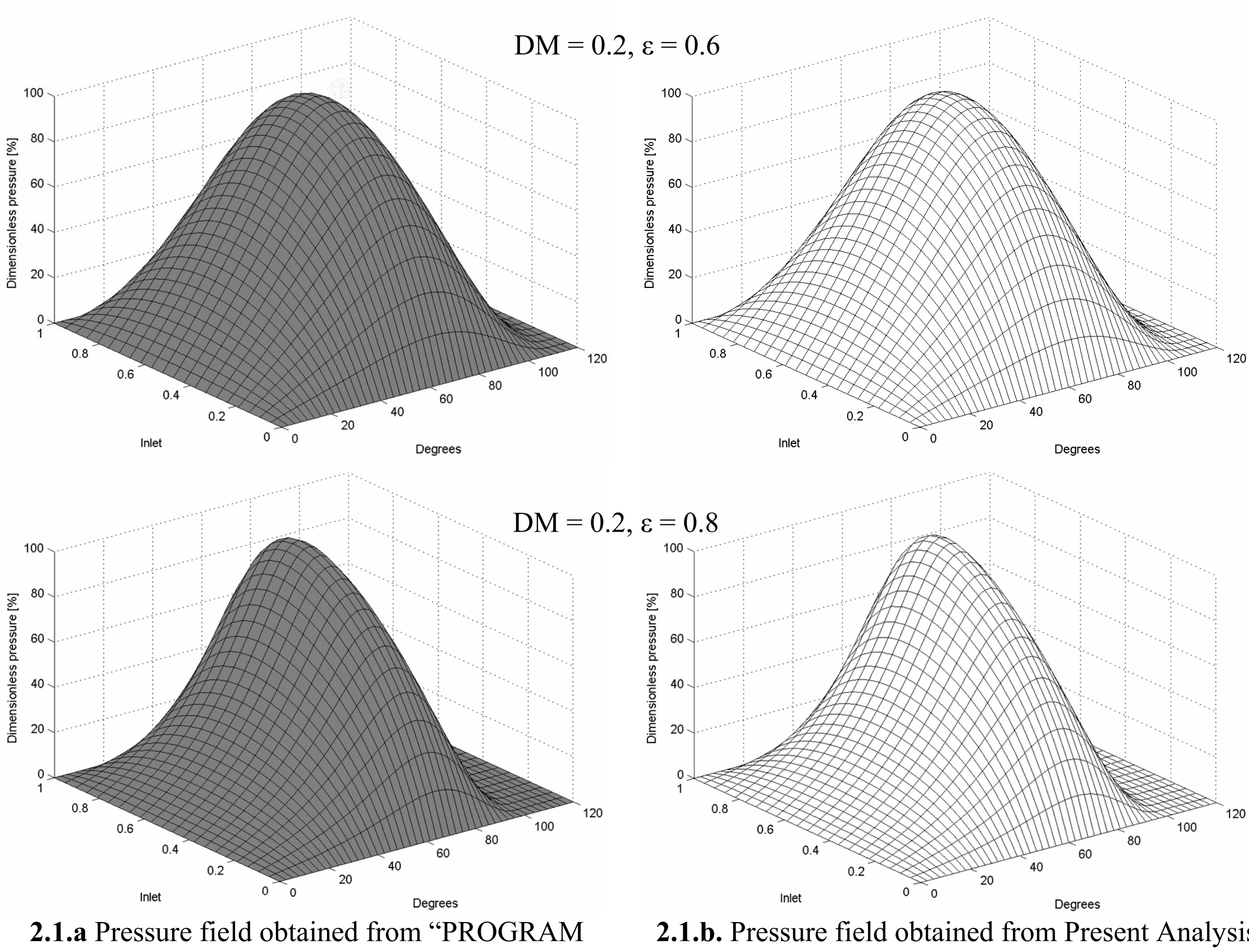
$$F = \iint \tau dA$$

$$\overline{F} = \int_0^1 \int_{\theta_1}^{\theta_2} \frac{\overline{h}}{2} \frac{\partial \overline{p}}{\partial \theta} d\theta \, d\overline{z} + \int_0^1 \int_{\theta_1}^{\theta_2} \frac{\overline{\eta}}{\overline{h}} d\theta \, d\overline{z} \qquad (12)$$

$$\frac{\overline{F}}{\overline{W}} = f\left(\frac{R}{C}\right) \qquad (13)$$

3. VALIDATION

To validate the present numerical scheme, a result obtained from the present analysis, with zero tilt and zero adjustment configuration, is compared with the results obtained by executing the MATLAB program 'PARTIAL' given by Stachowiak and Batchelor [10]. Table 1 lists the performance parameters for comparison. Similarly, Fig. 2 demonstrates the comparison of the pressure fields for misaligned bearings obtained from program PARTIAL [10] and the present analysis. This assessment validates the present analysis.



2.1.a Pressure field obtained from "PROGRAM PARTIAL," [10]

2.1.b. Pressure field obtained from Present Analysis

Figure 2. Comparison of pressure distribution in the bearing having DM = 0.2 and ε = 0.6 and 0.8

Table 1: Comparison of present analysis with PROGRAM PARTIAL [10] for DM = 0.2.

ε	Attitude Angle		Load Capacity		Friction variable		Maximum Pressure	
	Present Analysis	PROGRAM PARTIAL[10]	Present Analysis	PROGRAM PARTIAL[10]	Present Analysis	PROGRAM PARTIAL[10]	Present Analysis	PROGRAM PARTIAL[10]
0.1	71.29	72.39	0.0249	0.0249	87.31	87.32	0.0297	0.0291
0.2	58.92	59.18	0.0530	0.0528	46.12	44.58	0.0635	0.0630
0.4	44.11	44.08	0.1385	0.1374	23.71	21.77	0.1756	0.1743
0.6	35.63	35.71	0.3302	0.3282	14.72	12.92	0.4689	0.4659
0.8	27.12	27.32	1.0236	1.0152	8.62	7.41	1.8700	1.8396

4. RESULTS AND DISCUSSIONS

Figures 3 to Fig.10, illustrates the result of present analysis for operating conditions listed in Table 2. Static performance characteristics obtained are in terms of load carrying capacity, attitude angle, oil flow rate and friction variable.

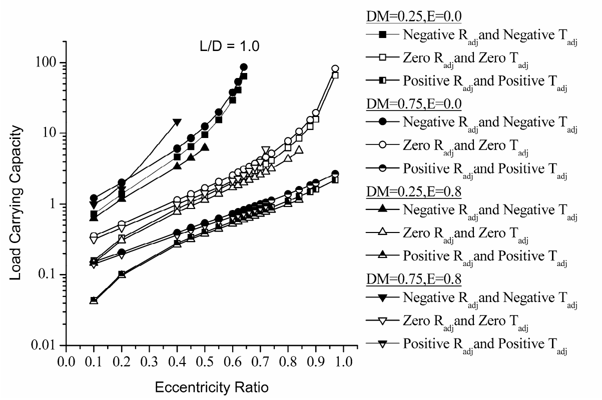


Figure 3. Load carrying capacity Vs Eccentricity Ratio for various operating conditions.

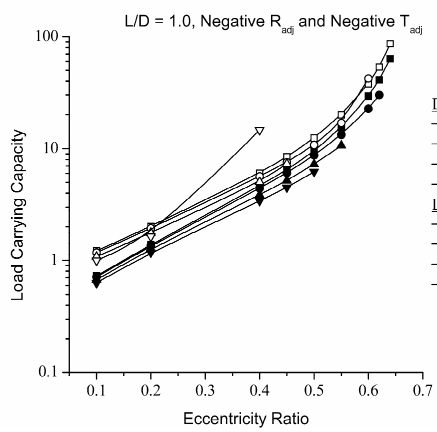


Figure 4. Load carrying capacity Vs Eccentricity ratio with negative adjustment configuration for various operating conditions.

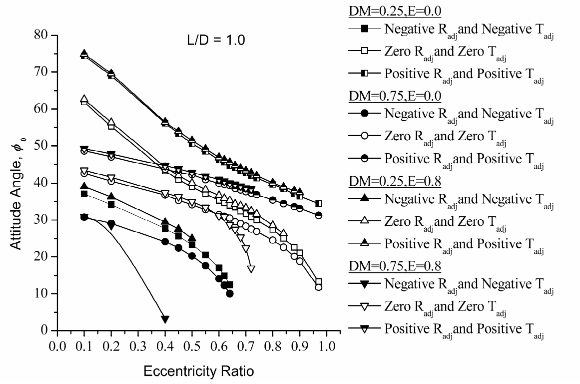


Figure 5. Attitude Angle Vs Eccentricity Ratio Vs Eccentricity Ratio for various operating conditions.

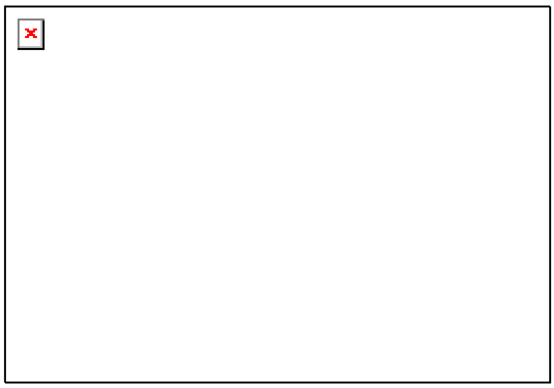


Figure 6. Attitude Angle Vs Eccentricity Ratio with negative adjustment configuration for various operating conditions.

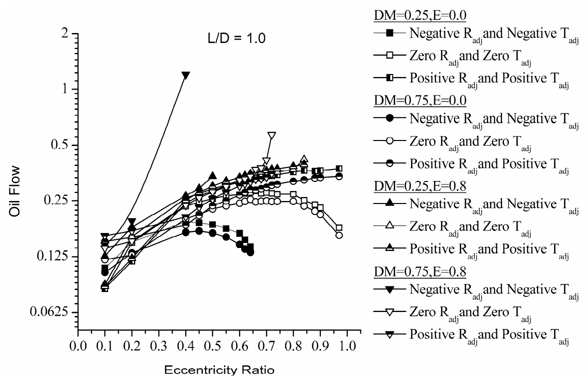


Figure 7. Oil Flow Vs Eccentricity Ratio for various operating conditions.

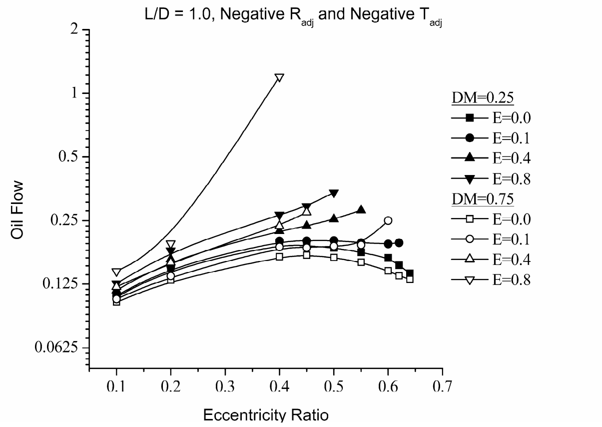


Figure 8. Oil Flow Vs Eccentricity Ratio with negative adjustment configuration for various operating conditions.

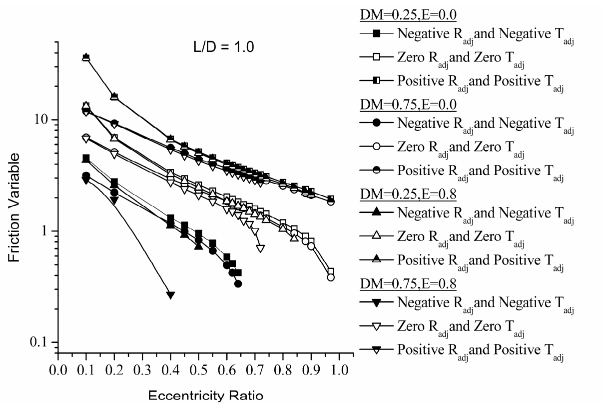


Figure 9. Friction Variable Vs Eccentricity Ratio for various operating conditions

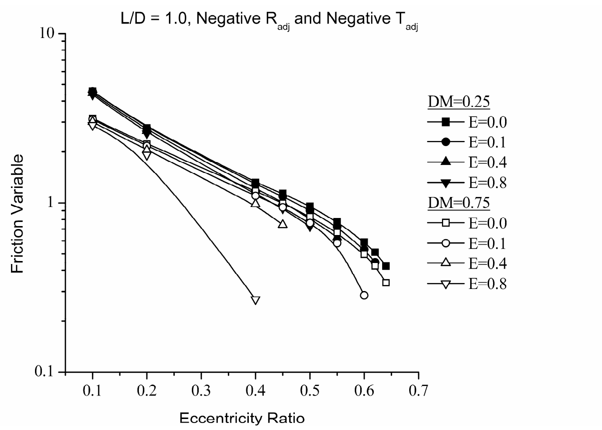


Figure 10. Friction Variable Vs Eccentricity Ratio with negative adjustment configuration for various operating conditions.

Table 2: Various operating parameters considered for the present analysis.

Parameters	
R_{adj}	$\pm$ 0.25C
δ	$\pm$ 0.0061°
DM	0.25 and 0.75
E	0.0, 0.1,0.4,0.8

Figure 3 depicts that, the load carrying capacity is higher with negative adjustment (i.e., negative R_{adj} and negative T_{adj}) configuration. Load carrying capacity reduces with a decrease in viscosity except for a bearing having negative adjustment configuration and operating at higher values of eccentricity ratio and DM. Misalignment has a positive influence on load carrying capacity. From Fig.4, it is seen that for a bearing with negative adjustment (i.e., negative R_{adj} and negative T_{adj}) configuration, DM has more influence on load carrying capacity than the adiabatic parameter E. Effect of various operating conditions on attitude angle is shown in Fig. 5. Pad configuration with negative R_{adj} and negative T_{adj} results in lesser attitude angle, which indicates that the dynamic stability of the system may improve. In Fig.5, it is also seen that, for a given DM, an increase in E marginally increases the attitude angle, except for a bearing with negative R_{adj} and negative T_{adj}, wherein E's influence is significant. At lower eccentricity ratio, DM has considerable influence on attitude angle than E. For eccentricity ratio lesser than 0.4 and for all values of E, it is observed that positive adjustment configuration with $DM = 0.75$ results in lesser attitude angle than zero adjustment configurations with $DM = 0.25$. Similarly, for eccentricity ratio greater than 0.6, zero adjustment configuration with $DM = 0.75$ and $E = 0.8$, results in lesser attitude angle than that of a bearing having same adjustment configuration with $DM = 0.75$ and $E = 0.0$. From Fig.6, it is seen that for a bearing with negative adjustment configuration, attitude angle reduces with an increase in DM. For $DM = 0.25$, and increase in E increases the attitude angle. However, for $DM = 0.75$, at higher values of eccentricity ratio, an increase in E reduces the attitude angle. Figure 7 depicts that at lower eccentricity ratio and for all value of E, positive adjustment configuration with $DM = 0.25$, results in minimized end leakage. However, for the same operating condition, $DM = 0.75$ results in higher end leakage. On the other hand, at higher eccentricity ratio, oil flow rate is less with negative adjustment configuration. For $E = 0.0$ or 0.1, increase in DM reduces flow however, for a given DM increase in E increases flow. From Fig. 8 it is observed that, with negative adjustment configuration, for a value of E below 0.4, an increase in DM reduces flow. However, for E greater than or equal to 0.4, an increase in DM increases the flow. E has higher influence on side flow than DM. Figure 9 illustrates that, negative R_{adj} and negative T_{adj} configuration results in lesser friction. However, for a particular adjustment configuration, friction variable reduces with an increase in E and DM. This trend in friction variable is obvious because the shear resistance to fluid flow reduces with the increase in E and reduced film thickness. It is seen that at lower eccentricity ratio, DM has significant influence on friction than E. For eccentricity ratio greater than 0.6, zero adjustment configuration with $E = 0.0$ and $DM = 0.75$ has higher friction than for the same operating condition with $DM = 0.25$, $E = 0.8$. Similarly, for eccentricity ratio greater than 0.4, negative adjustment configuration with $E = 0.8$ and $DM = 0.25$ has lesser friction than for the adjustment configuration with $DM = 0.75$ and $E = 0.0$. From Fig. 10 it is seen that at higher eccentricity ratio and DM, it is observed that, the friction variable reduces as parameter E increases.

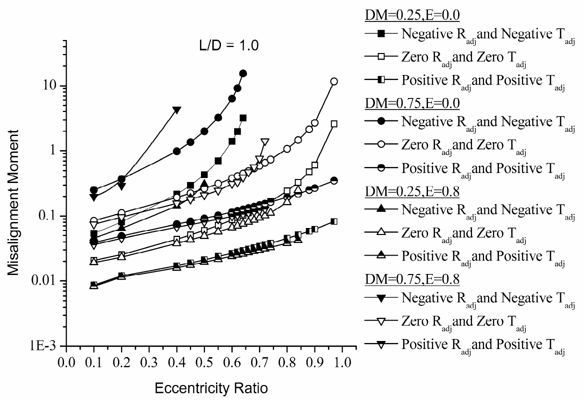


Figure 11. Misalignment Moment Vs Eccentricity Ratio for various operating conditions

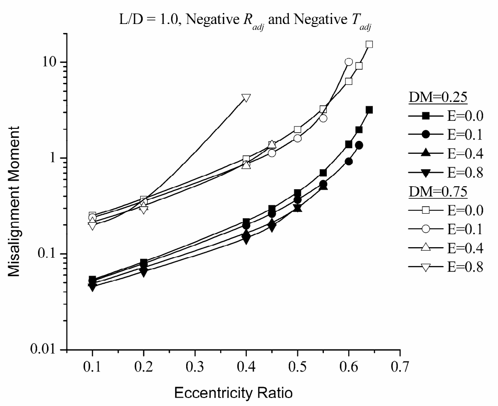


Figure 12. Misalignment Moment Vs Eccentricity Ratio with negative adjustment configuration for various operating conditions.

Figure 11 illustrates that, negative R_{adj} and negative T_{adj} configuration results in higher misalignment moment. However, for a particular adjustment configuration and DM, misalignment moment reduces with an increase in E. It is seen that at lower eccentricity ratio, DM has significant influence on misalignment moment than E. For eccentricity ratio lesser than 0.4 and a particular value of E, bearing having negative adjustment configuration that is operating with $DM = 0.25$, has less misalignment moment than a one with zero adjustment configuration and $DM = 0.75$. Similarly, for eccentricity ratio lesser than 0.8 and a particular value of E, zero adjustment configuration with $DM = 0.25$ has lesser misalignment moment than for positive adjustment configuration with $DM = 0.75$. From Fig. 12 it is seen that at higher eccentricity ratio and DM, it is observed that, the misalignment moment increases as parameter E increases. However, an inverse trend is observed for lower values of DM.

5. CONCLUSION

- The load capacity decreases with an increase in the adiabatic parameter, whereas it increases with the degree of misalignment for all values of the eccentricity ratio. However, at higher eccentricity ratio, the increase in load carrying capacity with degree of misalignment is not considerable. An externally adjustable pad bearing with negative adjustment configuration results in better load capacity.
- Attitude angle reduces with an increase in the degree of misalignment. At lower values of degree of misalignment, an increase in adiabatic parameter decreases the attitude angle. Such an effect is not observed at the higher values of degree of misalignment. Bearing with negative adjustment configuration results in reduced shaft attitude angle.
- The end leakage flow increases with the adiabatic parameter. At lower values of adiabatic parameter, an increase in degree of misalignment reduces the flow. However, at higher value of the adiabatic parameter, an inverse effect is observed with an increase in the degree of misalignment. Flow is less for an externally adjustable pad bearing with negative adjustment configuration, which is operating with higher eccentricity ratio.
- At lower values of eccentricity ratio and a particular value of adiabatic parameter, frictional coefficient reduces with an increase in the degree of misalignment. At higher values of the eccentricity ratio, the frictional coefficient always decreases with an increase in adiabatic parameter. The effect of pad adjustments on the frictional coefficient is significant at lower values of the eccentricity ratio. Friction coefficient is less for an externally adjustable

pad bearing with negative adjustment configuration.

- The misalignment moment decreases with an increase in the adiabatic parameter for all values of the eccentricity ratio. Such an effect is not observed for the cases of degree of misalignment.

Adiabatic solution of a misaligned single pad externally adjustable fluid film bearing, operating in with negative T_{adj} and negative R_{adj} configuration results in better load carrying capacity. This adjustment configuration, in conjugation with increasing adiabatic parameter, substantially increases the lubricant end leakage, friction variable and shaft attitude angle. However, for this adjustment, the bearing performance is superior when compared to other adjustment configurations.

NOMENCLATURE

C	Radial clearance (m)
E	Dimensionless number, $\dfrac{2\omega\beta\eta_i}{c_p\rho g}\left(\dfrac{R}{C}\right)^2$
F	Friction force (N), $\overline{F}=\dfrac{FC}{\eta_i\omega R^2 L}$
L	Length of the bearing (m)
R	Radius of the journal (m)
T	Temperature (°C)
T_i	Inlet Temperature (°C)
U	Peripheral velocity of the journal (m/s) $U=\omega R$
W	Load carrying capacity (N), $\overline{W}=\dfrac{WC^2}{\eta_i UR^2 L}$
e	Steady state eccentricity (m), $\varepsilon=\dfrac{e}{C}$
f	Coefficient of friction
h	Film thickness (m), $\overline{h}=\dfrac{h}{C}$
t	Time (s)
x	Coordinate axis in circumferential direction (m), $x=R\theta$
z	Coordinate axis in the axial direction (m), $\overline{z}=\dfrac{z}{L}$
Q_1	Lubricant leakage flow rate from one end of the bearing (m3/s), $\overline{Q_1}=\dfrac{Q_1 L}{UCR^2}$
Q_2	Lubricant leakage flow rate from other end of the bearing (m3/s), $\overline{Q_2}=\dfrac{Q_2 L}{UCR^2}$
Q_z	Total Lubricant leakage flow rate (m3/s), $\overline{Q_z}=\dfrac{Q_z L}{UCR^2}$
R_{adj}	Normalized radial adjustment, $\dfrac{Radial\ Adjustment}{C}$
R_p	Radius of the pad (m)
T_{adj}	Normalized tilt adjustment, $\dfrac{Tilt\ Adjustment}{C}=\dfrac{R_p\chi\tan(\delta)}{C}$
W_r	Radial component of load carrying capacity (N), $\overline{W_r}=\dfrac{W_r C^2}{\eta_i UR^2 L}$
W_t	Tangential component of load carrying capacity (N), $\overline{W_t}=\dfrac{W_t C^2}{\eta_i UR^2 L}$
p	Film pressure (N/m2), $\overline{p}=\dfrac{pC^2}{\eta_i\omega R^2}$
ϕ_0	Angle of the line of centers with the vertical direction (rad)
ψ	Assumed attitude angle (rad)
η	Lubricant viscosity (Ns/m2), $\overline{\eta}=\eta/\eta_i$
η_i	Lubricant viscosity at inlet (Ns/m2)
η_L	Local viscosity of lubricant (Ns/m2)
ω	Angular velocity of the journal (rad/s)
θ,θ^*	Angular coordinate in the bearing (rad)
θ_1,θ_2	Angles of start and end of hydrodynamic film (rad)
τ	Shear due to viscous effects (N/m2)
δ	Tilt angle (rad)
χ	Span of the pad (rad)
DM	Degree of misalignment
α	Angle between the central eccentricity vector and projected journal axis (rad)
ε'	Non-dimensional projected distance of the journal, $\varepsilon'=e'/C$

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