Effect of Pad-Preload on Fluid Film Coefficients in Tilting Pad Journal Bearing

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***Abstract*—Analytical model has been developed to analyze the effect of pad-preload on bearing parameters. A four pad centrally pivoted tilting pad journal bearing has considered for analysis. The infinite short bearing assumption has taken for determination of fluid film pressure and fluid film forces. Fluid film thickness, pressure along pads, hydrodynamic forces, stiffness and damping coefficients has presented in the form of graphs concerning journal eccentricity ratios and pad-preload coefficients. The separate Matlab™ code has made for investigation purpose. Investigation showed that pad preload is vital for the stability of tilting pad journal yet higher value could cause bearing failure.**

***Index Terms*-Tilting Pad Journal Bearing, Preload Coefficient, Fluid Film Coefficients, Hydrodynamic Forces.**

I. INTRODUCTION & LITERATURE SURVEY

Tilting pad journal bearings contain the number of arcuate pads which may be typically four or five. Each pad of the bearing is free to rotate about a pivot and cannot support a moment. As a result, the destabilizing forces greatly reduce or eliminated, and the bearing is no longer a potential source of rotor dynamic instability. This property has made tilting pad journal bearings the standard fluid-film bearing for most high-speed applications [2]. Preload is a dimensionless number, describing the relationship between the shaft and bearing pad curvature, which combines to form bearing oil wedge. As oil wedge clearance changes, bearing stiffness and damping are influenced, to provide a common method to describe these variations, the concept of preload has applied.

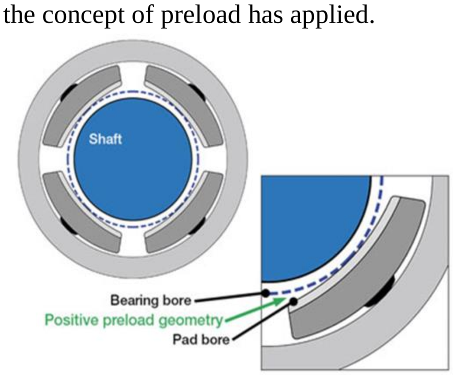


Fig.1.TPJB with positive pad-preload geometry [11]

Preload usually use to adjust bearing coefficients to obtain specific rotor response characteristics. The preload has defined as [11]:

$$m=1-(C_b/C_p) \qquad (1)$$

Here, C_b is Bearing Diametrical Assembling Clearance, and C_p is Diametrical Pad Clearance. Industrial machines usually have preload in between 0.1 to 0.5. These values may change by adjusting bearing assembly clearance (C_b). Decreasing the assembly clearance will increase preload. Lund [1] presented design curves considered no preload condition and assumed that pivot has located on the inner pad surface. Dimond et al. [2] investigated later that low preload leads to more stable systems. Orcutt & Arwas [3] analyzed that preloading the tilting-pad bearing results in greater stiffness and damping, thus improving dynamic characteristics. Ettlels [4] investigated that high bearing stiffness in any direction can obtain by radial preloading of the pads also the pad-preload can be increased by thermal expansion of the shaft and pads, particularly during start-up when the film can contract to a dangerously low thickness. White and Chan [5] investigated pad journal bearings with small preload and operating at sub-synchronous frequencies. They investigated lower damping and higher stiffness compared to synchronous excitation. Nicholas [6] investigated fractional frequency whirl in the un-preloaded tilting-pad bearing at high speeds and very light loads. They determined that Preloading can stop instability completely. Decreasing pad preload decreases bearing stiffness but increases effective damping thereby providing improved stability performance and an overdamped non-responsive second critical speed. As preload decreases, the instability threshold speed increases. By increasing both bearing stiffness and damping by increasing pad, preload causes less motion at the bearings. Nicholas [6] investigated the effects of pad preload for an axial compressor. It has observed that as preload decreases, the synchronously reduced stiffness remains about the same but damping increases. Mahfouz and Adams [7] used geometrical preloading is to achieve high fluid film stiffness and tracking. They concluded that stabilization against pad flutter could improve by geometric preloading. Okabe and Cavalca [8] observed that stiffness increase due to an increase in pad preload. They also mentioned that locus has gone toward bearing center as preload has increased, while orbit size has decreased. They observed an increase of pad preload produced a sharp rise of the locus associated with a stiffness decrease. Cha and Glavatskih [9] mentioned that Preload could set in the range from zero to one. Lightly loaded bearings are purposely preloaded to stiffen the rotor-bearing system. The preload can change by altering the pad radial clearance. As the pad preload increases, the size of the journal orbit decreases. Results for different preload factors show that the compliant bearings need to be preloaded more to produce journal orbits and eccentricities comparable to the white metal bearings. It has concluded that without preload, the vibration level and bearing metal temperature were very high. Various Authors adopted numerical methods to solve the Reynolds equation at each time step to update the stiffness and damping coefficients. As a result, simulation time increased, and parametric studies become rigorous. So in this article, the simplified method is adapted to perform the effect of pad preload on TPJB's fluid film properties. The single code has been developed in MATLAB™ to solve nonlinear equations and to generate design plot, which has used for further analysis.

II. Mathematical Model

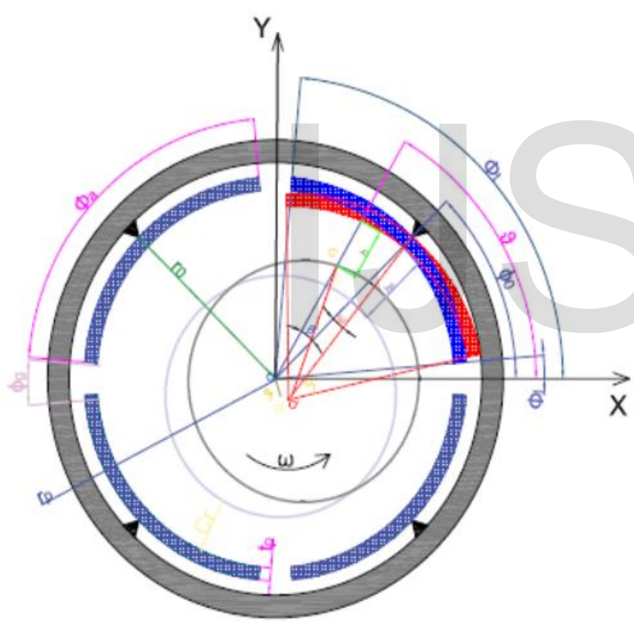


Fig.2. Geometrical Parameters of TPJB

The first step of modeling hydrodynamic bearings is to develop an equation for the bearing fluid film thickness H along each pad, which has to consider the displacement of the journal center about bearing center, pad preload, and pad pivoting position. The modified expression of fluid film thickness derived from Santos et al. [10], and it has expressed as:

$$H = f(X, Y, \dot{X}, \dot{Y}, m, \emptyset_0, \psi, R_0) \qquad (2)$$

The hydrodynamic pressure P determined by solving the Reynolds equation by considering bearing is infinitely short. The pressure distribution has evaluated from Reynolds equation (3), was depends on the fluid film thickness (H).

$$\frac{1}{{R_b}^2}\frac{\partial}{\partial\vartheta_b}\left(\frac{H^3}{\mu}.\frac{\partial P}{\partial\vartheta_b}\right)+\frac{\partial}{\partial Z}\left(\frac{H^3}{\mu}.\frac{\partial P}{\partial Z}\right)=6\left(\omega.\frac{\partial H}{\partial\vartheta_b}+2\frac{dH}{dt}\right) \qquad (3)$$

TABLE I
VALUES OF BEARING GEOMETRIC PARAMETERS

Sr. No	Symbol	Name	Values (Unit)
1	C_r	Bearing Radial Clearance	125 (Microns)
2	N	Journal Revolution	6000 (RPM)
3	D_j	Journal Diameter	50 (mm)
4	R_0	Pivot Distance from Bearing Centre	25 (mm)
5	T_p	Pad Thickness	0 (Meters)
6	L_B	Bearing Length	25 (mm)
7	m	Pad Preload	0-0.9
8	ϕ_{01}	Pad-1 Pivoting Angle	45 (Degrees)
9	μ	Fluid Dynamic Viscosity	8.95e-4 (Pa*sec)
10	Φ_l	Pad-1 Leading Edge Angle	5 (Degrees)
11	Φ_t	Pad-2 Trailing Edge Angle	85 (Degrees)
12	N_p	Number of Pads	4

In equation (3), Z is axial coordinate, R_b is the bearing radius, ϑ_b is the circumferential coordinate, μ is the viscosity of the fluid, H is the dimensional fluid film thickness, P is fluid film pressure, and t is the time. Pressure distribution has found by applying infinite short bearing assumption. The reaction forces of the bearing can be obtained from the integration of pressure distribution. These forces are functions of the displacements and the instantaneous journal centre velocities along X-direction and Y-direction. Initially, the equilibrium position (X, Y) of the shaft inside the bearing is found depending on the rotational speed. The differential contact area ($dA = R_b.L.d\vartheta.dZ$) is considered in the integration.

$$F_X = \int_{\vartheta_1}^{\vartheta_2}\int_{-\frac{L}{2}}^{\frac{L}{2}} P.R_b \cos\vartheta_b \; d\vartheta_b.dZ \qquad (4)$$

$$F_Y = \int_{\vartheta_1}^{\vartheta_2}\int_{-\frac{L}{2}}^{\frac{L}{2}} P.R_b \sin\vartheta_b \; d\vartheta_b.dZ \qquad (5)$$

The fluid film moments have calculated as a cross product of force component and the distance between force applications to pad pivot. There were some assumptions taken as fluid was considered incompressible and the viscosity was assumed to be constant across the fluid. Half Somerfield solution was applied, and the cavitation effect has neglected. The fluid film forces of the whole bearing can be obtained by vector summation of forces from each pad to journal along bearing axis as:

$$F_{bX} = \sum_{i=1}^{4} F_X^i \; , F_{bY} = \sum_{i=1}^{4} F_Y^i \qquad (6)$$

The stiffness and damping coefficients of bearing determined as:

$$S_{bxx} = \frac{\partial F_{bX}}{\partial X}, S_{bxy} = \frac{\partial F_{bX}}{\partial Y}, S_{byx} = \frac{\partial F_{bY}}{\partial X}, \quad S_{byy} = \frac{\partial F_{bY}}{\partial Y} \qquad (7)$$

$$D_{bxx} = \frac{\partial F_{bX}}{\partial \dot{X}}, D_{bxy} = \frac{\partial F_{bX}}{\partial \dot{Y}}, D_{byx} = \frac{\partial F_{bY}}{\partial \dot{X}}, \quad D_{byy} = \frac{\partial F_{bY}}{\partial \dot{Y}} \qquad (8)$$

Figure (3) represents a flow chart included steps involved in the analysis.

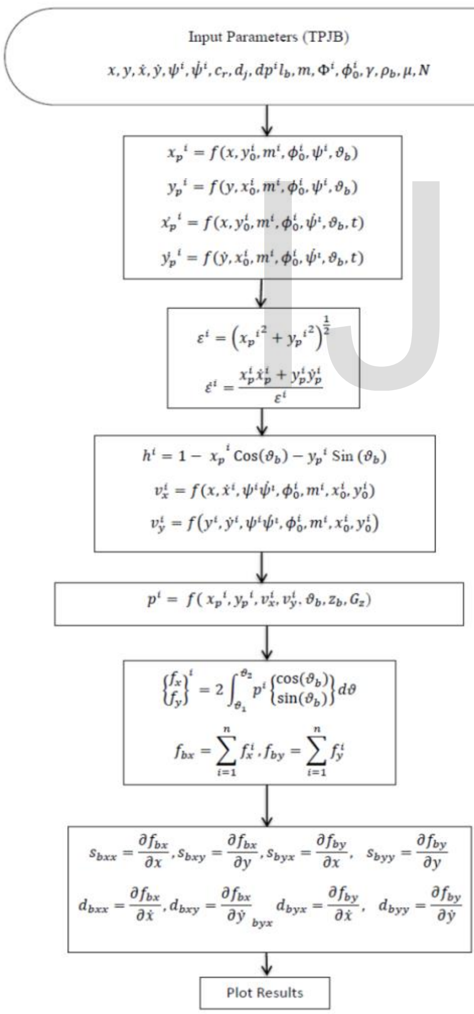


Fig.3. Flow Chart for Determination of Fluid Film Coefficients

III. RESULTS & DISCUSSION

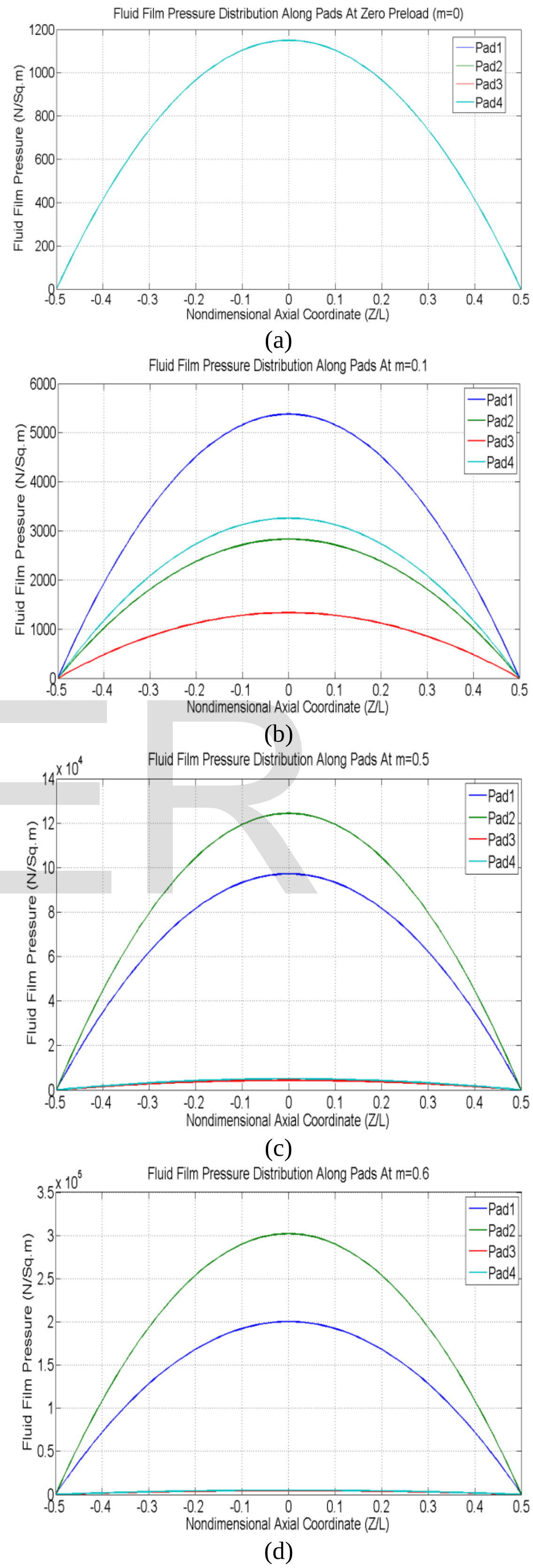


Fig.4. Axial Pressure Distribution, (a) m=0, (b) m=0.1, (c) m= 0.5, (d) m=0.6

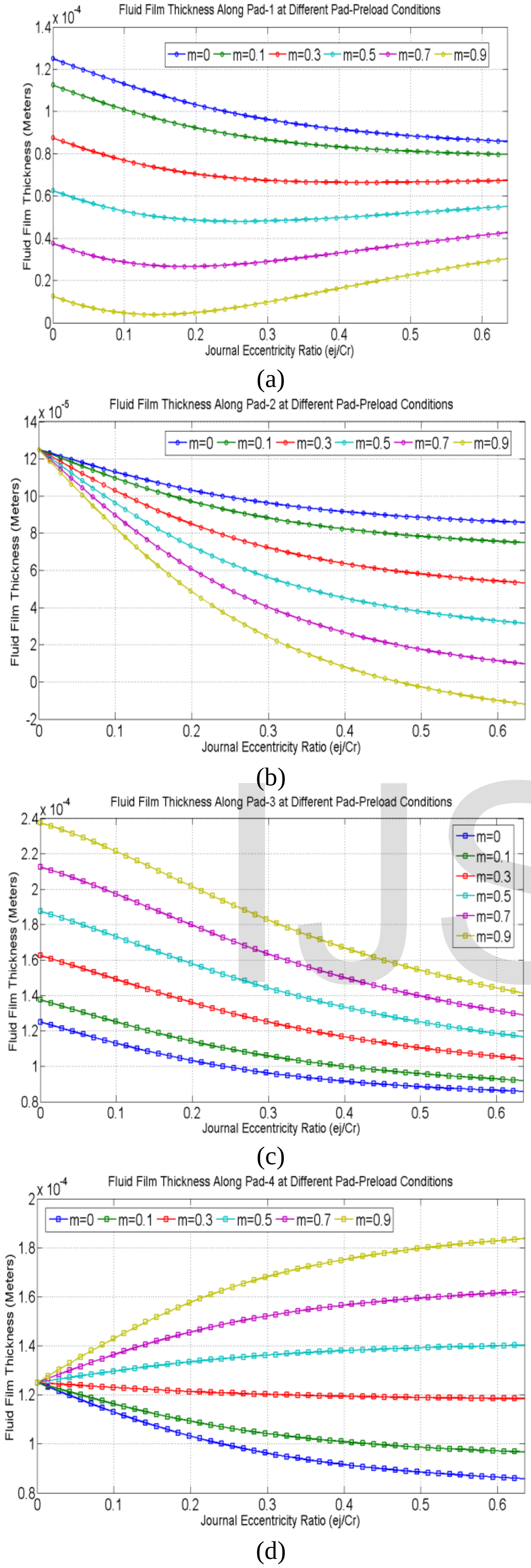


Fig.5. Fluid Film Thickness with Journal Eccentricity Ratio along Different Pads (a) Pad-1, (b) Pad-2, (c) Pad-3, (d) Pad-4

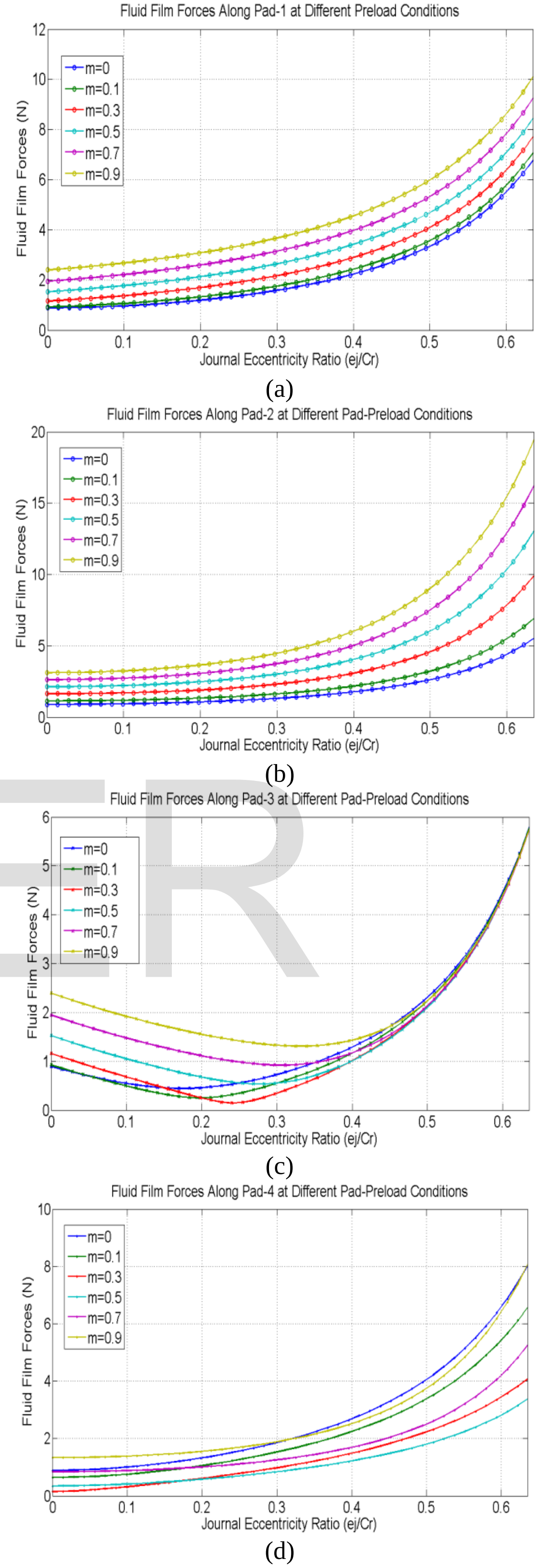
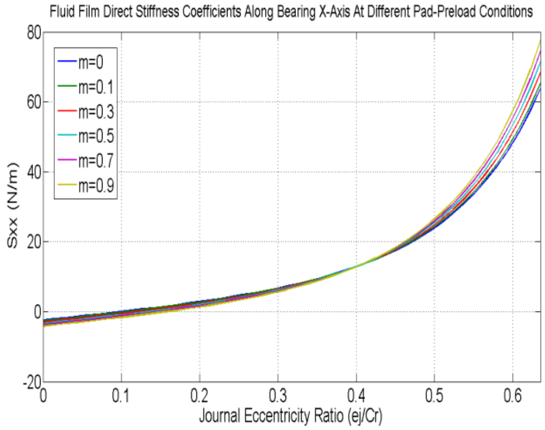
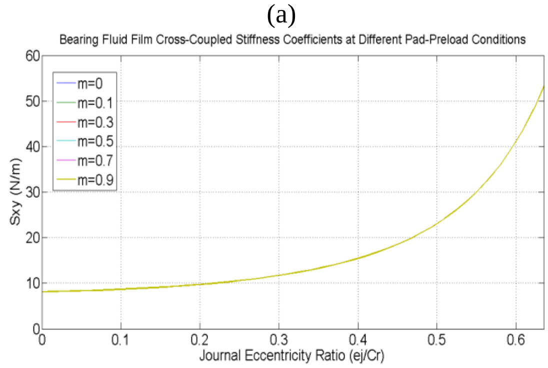


Fig.6. Fluid Film Forces with Journal Eccentricity Ratio along Different Pads (a) Pad-1, (b) Pad-2, (c) Pad-3, (d) Pad-4



(a)



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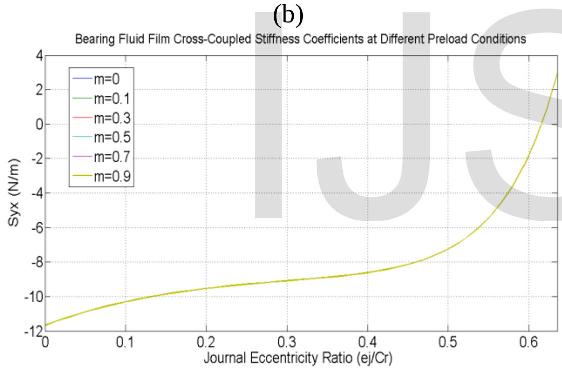

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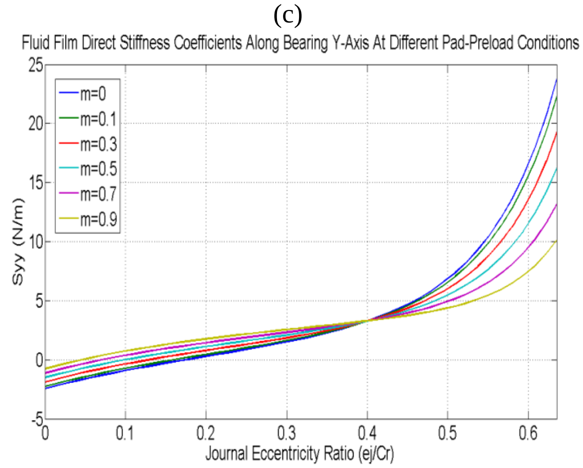
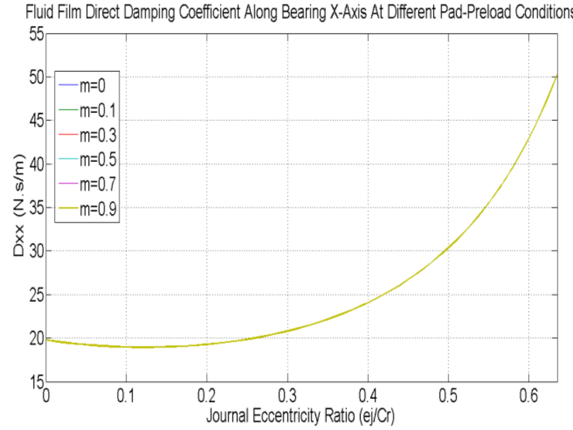
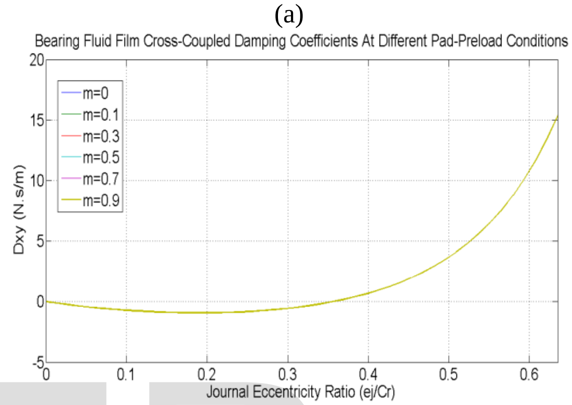


(d)

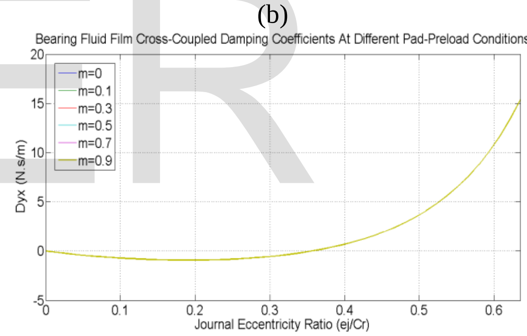
Fig.7. Bearing Fluid Film Stiffness Coefficients with Journal Eccentricity Ratio (a) Sxx, (b) Sxy, (c) Syx, (d) Syy



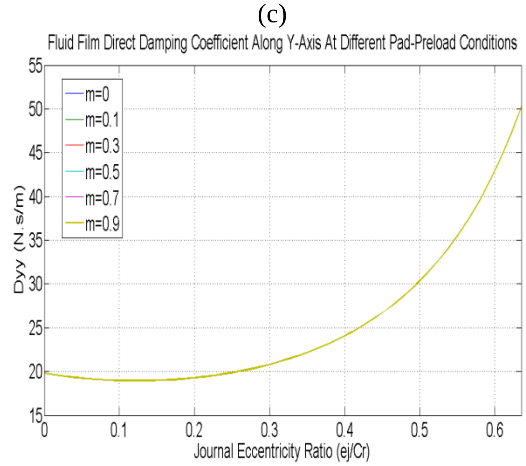
(a)



(b)



(c)



(d)

Fig.8. Bearing Fluid Film Damping Coefficients with Journal Eccentricity Ratio (a) Dxx, (b) Dxy, (c) Dyx, (d) Dyy

Figure 4 represents axial pressure distribution along pads at different pad preload conditions. Figure 4(a) indicates that without pad preload condition the pressure distribution along the axial direction is equal. Figure 4(b), 4(c) and 4(d) represent axial pressure distribution at different positive preload conditions in increasing order. The results show that fluid film pressure increases with positive preload along pads; also there were different values of fluid film pressure observed along different pads. The pressure difference also increased between loaded and unloaded pads as the value of pad-preload increases.

Figure 5 represents fluid film thickness along all pads at different preload conditions. It has been observed that for same journal eccentricity ratio fluid film thickness would decrease when the preload coefficient increases while for unloaded pads this phenomenon would be reversed.

Figure 6 represents fluid film forces along pads with journal eccentricity ratio at different preload conditions. It has observed in figure 6(a), and 6 (b) loaded pads that the values of fluid film forces increase with journal eccentricity ratio when pad preload increase while as figure 6 (d) indicates that at pad-4, unloaded pad, fluid film forces found at zero preload coefficient. Figure6 (c) indicates that when pad preload coefficient was above 0.35 the values of fluid film forces along all pads increase rapidly because higher values of pad preload cause higher convergence of oil film, i.e., higher fluid film forces.

Figure7 (a) represents the direct stiffness coefficient along bearing X-direction increases with pad preload. 7(d) indicates that at higher preload values (above 0.4) the space between journal and pad would reduce, and that may cause complete unloading of the pad. Figure 7(b) and 7 (c) indicate there is no effect or marginal effect of pad preload on cross-coupled stiffness coefficients, so preloading is provided to stiffen the bearing, but higher values of preload may cause catastrophic failure of bearing or completely unloading of pads.

Figure 8 represents that there is not any substantial effect of pad preload on direct as well as cross-coupled damping coefficients at steady state conditions, but in the dynamic condition, it affects the bearing response.

IV. Conclusion

Following results has been concluded by analyzing results:

1. Pad-preloading influences the pressure and forces distribution along pads; hence it provides better convergence of fluid film along pads.
2. Higher values of pad preload may cause unloading of pads or pad- journal interaction, which may cause catastrophic failure of bearing.
3. Pad preloading provides marginal stiffening of bearing hence optimum value of pad preload increases bearing performance.
4. Effect of pad preload on bearing damping coefficients is negligible at steady state condition.

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