EFFECT OF LOADED-PAD THICKNESS ON THE STATIC BEHAVIORS OF FIVE-PAD HYDRODYNAMIC JOURNAL BEARINGS

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Abstract - The use of tilting pad journal bearings (TPJBs) has increased in the recent past due to their stabilizing effects on the rotor bearing system. Most of the studies about steady state and dynamic behaviors of TPJBs are usually evaluated by means of thermohydrodynamic models assuming nominal dimensions for the bearing. However, machining errors could lead to actual bearing geometry and dimensions different from the nominal ones. In particular, for TPJB the asymmetry of the bearing geometry leads to unexpected behavior of the bearing. In this paper, effects of non-uniform clearance on TPJBs' performances such as clearance profile, shaft locus, power loss, pad tilt angle, and pressure distribution are evaluated using an analytical model. The results show that effects of non-uniform clearance on static characteristics of TPJBs are considerable when evaluating the efficiency of the bearing in particular and system in general.

Key words - Pad thickness; five-pad; journal bearing; static behavior

1. Introduction

Since tilting pad journal bearings (TPJBs) are more stable and efficient than conventional bearings, they have been frequently applied to many rotor bearing systems. The essential feature of tilting-pads is that they modify their configuration to adapt to every operating condition, creating several convergent-divergent gaps around the circumference and thus making the system highly stable. A lof of numerical methods for calculating dynamic coefficients for tilting-pad journal bearings, extensive theoretical and experimental studies on dynamic and stability analysis have been conducted. For the development of journal bearings, many effective methods have been applied, such as Newton-Raphson method, pad assembly technique, finite elements method, and genetic algorithm to calculate static as well as dynamic characteristics of the TPJBs.

The effects of bearing clearance and pad clearance on the steady-state and dynamic behavior in TPJBs were theoretically studied in [1]. It can be concluded that the pad clearance has little effect on the K_{yy} term at small values of bearing clearance ratio and the bearing clearance has more effect than pad clearance. The effects of preload factor, clearance ratio, load directions, pad thickness, working temperature on performance of TPJBs were also investigated by theoretical and/or experimental activities in some literatures [2] - [9].

A mathematical model was developed to study the nonlinear dynamics of rotor supported on a TPJB [10]. The results refer to nonsynchronous and chaotic vibrations associated with phenomena other than the classical oil whirl. In particular, it has been shown that two suggested loading mechanisms successfully lead to the appearance of rich nonlinear vibrations for the rotor and its supporting pads. The first one is represented by through-pivot and on-the-pad loading case. The second one is the case of a concentric rotor with no static biasing load. For the pivot loaded pads, large static loads lead to large rotor displacements in the direction orthogonal to the applied load. It was demonstrated that stabilization against pad flutter could be improved by geometric preloading.

The dynamic characteristics of TPJBs with asymmetric pad support using Reynolds equation and an adiabatic model for the oil film was studied in [11]. It was applied to a five-pad bearing, and then the pad relative clearance has influenced the dynamic stiffness and damping coefficients, while the length-to-diameter ratio has just affected the direct dynamic stiffness coefficients.

The importance of pad and pivot flexibility in predicting impedance coefficients for the TPJBs, presented measured changes in bearing clearance with operating temperature, and summarize the differences between measured and predicted frequency dependence of dynamic impedance coefficients was taken-into-account in [12]. The bearing clearance with pentagonal shape was obtained as a function of operating temperature. Hot-bearing clearances and cold-bearing clearances were plotted and measured hot-clearances are approximately 30% smaller than measured cold ones and were inversely proportional to pad surface temperature. The effect of employing a full bearing model versus a reduced bearing model (where only journal degrees of freedom are retained) in a stability calculation for a realistic rotor-bearing system is assessed.

A misaligned journal bearing system using mixed lubrication model coupling for the asperity contact effect, elastic deformation, effect of viscosity on oil temperature and pressure was investigated in [13]. The performance of model was calculated using finite difference method, and a test-rig presented in this paper was used to experimental evaluate the friction and temperature characteristics of the system. It can be concluded that the pressure distribution, film thickness, and elastic deformation of the journal bearing under misalignment conditions should be considered.

The main effects of turbulent flow in TPJBs are presented in [14]. Turbulence plays an important role in the design of hydrodynamic bearings in order to increasing power density in turbomachines and consequently the operating speeds of the fluid-film bearings.

The static and dynamic characteristics of a large cylindrical journal bearing operating under severe conditions namely at very low speed and high specific pressure were described in [15]. The bearing was assumed to be uniform geometry with diameter of 160mm and the length of 145mm.

The pivot flexibility in combination with the fluid film plays an important role for improving agreement between theory and experiment on static and dynamic behaviors of TPJBs. In [16], the model used hinges on a single-pad with two degree-of-freedom has been developed to investigate the effect of various geometries and operating conditions to bearings' performances. It can be concluded that both stiffness and damping coefficients depend on the applied force frequency and the flexibility of the pivot contact region that provides support for the pad.

In [17] authors presented a method to evaluate the thermal behavior of a five-pad TPJB using cooling pads. The influences of oil supply grooves at the trailing edge of the two loaded pads on the maximum pad temperature of a large TPJB in non-flooded design were carefully considered. Experimental and numerical results were compared on a tested bearing. The experimental results show a significant decrease of the maximum pad temperatures. An overall temperature reduction is obtained for the rear pad in circumferential direction.

However, almost papers studied static characteristics of TPJB by means of thermohydrodynamic models using nominal dimensions for the bearing. In practical applications, the error machining on pads usually occurs leading to thickness of pads is not identical to each other. It means that the dimensions of all pads are identical or the bearing is uniform configuration.

In this paper, effects of loaded-pad thickness, i.e., pad #1, on the performances of TPJBs such as clearance profile, shaft locus, power loss, pad tilt angle, and pressure distribution are evaluated using an analytical model. A five-pad TPJB with different thickness in one pad is analytically modelled. Its steady state behaviors are compared with a symmetric bearing.

2. Description of the bearing

The TPJBs consists of a set of pads which are individually pivoted, as shown in Figure 1. The bearing has five pads, load-on-pad configuration. The pivots are threaded into an outer retaining ring (or bearing shell) for positioning and preload adjustment.

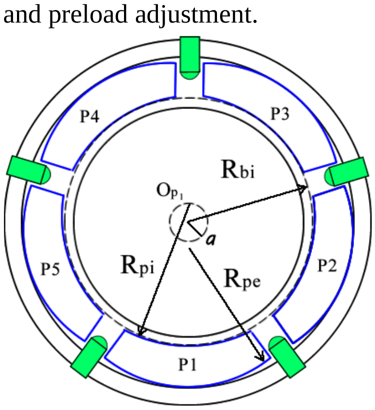


Figure 1. *Geometry of five-pad TPJB*

Much of the advantage of the TPJBs depends on the pads being able to track the motions of the rotor axis. Tracking depends on the pad inertia and the stiffness and damping of the fluid film. During operation, each pad will assume an inclination such that the resultant of the fluid film forces passes through the pivot point. Thus the pivot location influences the pad inclination and the magnitude of the hydrodynamic pressures that are generated.

In this paper, two bearing models, namely uniform bearing and non-uniform bearing, are used to evaluate effects of non-uniform clearance on performances of TPJBs. For the numerical prediction, a finite-difference code was developed to solve for the steady-state and dynamic characteristics of a TPJB using the Reynolds equation. Pad and rotor position are determined using a Newton–Raphson algorithm that employs the analytically perturbed fluid-film stiffness and damping matrices.

The MATLAB code includes a thermal model to determine bearing fluid temperatures and viscosities. The temperature of inlet oil used for numerical model is kept at 40 °C and is similar with the practical operating condition of TPJBs. The structural and operating parameters of two bearings are given in Table1.

Table1. *Two bearing geometric characteristics and operating conditions*

Item	***Uniform bearing***	***Non-uniform bearing***
Bearing diameter (*mm*)	100	100
Machined clearance (*mm*)	0.125	0.125
Assembled clearance (*mm*)	0.07	0.07
Bearing length (*mm*)	70	70
Angular amplitude of pads (°)	60	60
Upper pads	3,4	3,4
Lower pads	1,2,5	1,2,5
Lubricant	ISO VG46	ISO VG46
Speed range (*rpm*)	1000 – 3000	1000 – 3000
Thickness – Pad 1 (*mm*)	**16**	**15.99**
Thickness – Pad 2-5 (*mm*)	16	16

3. Results and discussion

In this paper, comparisons between two kinds of bearing are made in terms of dynamic coefficients, pressure distributions on each pad, shaft locus, power loss, and pad tilt angle using the complete dynamic model. The rotor is run with rotational speeds from 1000 rpm to 3000 rpm and is applied by a static force of 5 kN in vertical direction. During operating, the temperature of oil is assumed to keep at 40°C. Moreover, effect of non-uniform clearance is also considered in terms of load direction. In this case, the rotational speed of rotor is fixed at 1200 rpm (20 Hz) and direction of static load is changed from -90° to 270°, in steps of 18°.

3.1. Shaft locus

For a bearing used in practice, the static loads (gravity load, external static load, etc.) are predetermined while the static equilibrium position is unknown. So, to obtain the dynamic coefficients, we have to find the static equilibrium position first. Suppose that the static load applied on rotor is fixed, the rotational speed of rotor is increased gradually from 1000 rpm to 3000 rpm. Figure 2 shows the shaft centers of two bearings at twenty positions corresponding to twenty rotational speed values of rotor.

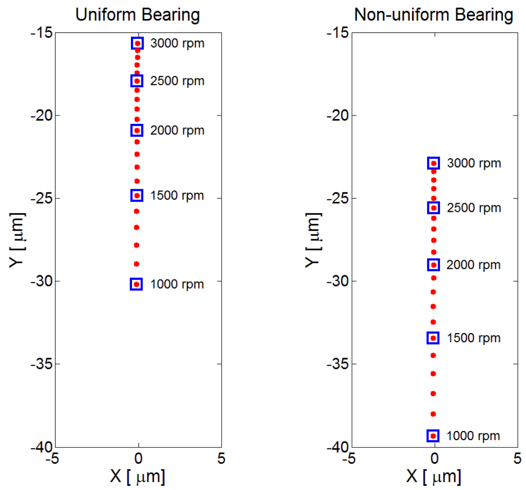


Figure 2. *Shaft locus of bearing vs. rotational speed*

It is clearly seen that when rotational speed increases, the center of shaft tend to climb to the equilibrium position of bearing. It should be noted that the displacements of shaft in the non-uniform bearing are always larger than those of in the uniform bearing, nearly about 10 μm at each position corresponding to the speed of rotor.

It can be explained that because the thickness of pad #1 in the non-uniform bearing is less than 10 μm compared to pad #1 in the uniform bearing, the center of shaft in the non-uniform bearing will be lower than 10 μm in the uniform bearing. At 1000 rpm, the center of shaft in the uniform bearing is more than 15 μm while this value for the non-uniform bearing is nearly 23 μm.

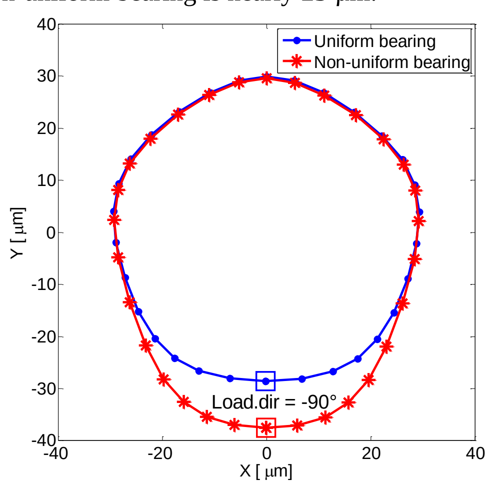


Figure 3. *Shaft locus of bearing vs. load direction*

In terms of load direction, Figure 3 shows the effect of non-uniform clearance on bearings when varying static load directions. For the uniform bearing, the shaft locus has a pentagon shape (look like a circular shape with a radius approximately 30μm). This is because for a TPJB, the clearance profile generates a polygon, having number of sides corresponding to the number of pads in the bearing [18], [19] and [20]. On contrary, an eclipse shape is presented for shaft locus of the non-uniform bearing. It is clearly seen that the displacement of shaft when pad #1 is loaded in the non-uniform bearing is always greater than that of the uniform one, especially in the vertical direction (-90°) with the difference of 10μm.

3.2. Minimum oil-film thickness

Figure 4 shows the effect of non-uniform clearance on minimum oil-film thickness under varying rotational speed and static load direction. Oil-film thickness increases with increasing of speed on both bearings and this thickness of the non-uniform bearing is always higher than in the uniform bearing, especially at high speed. It can be explained that when the rotor speed increases, the center of shaft tend to climb to the equilibrium position of bearing, that means the shaft is moved up to the bearing center.

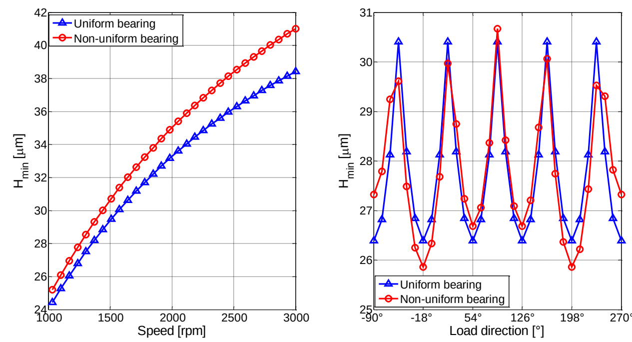


Figure 4. *Minimum oil-film thickness of two bearings*

At speed of 3000 rpm, the minimum oil-film thickness of uniform bearing and non-uniform bearing is nearly 41 μm and 38 μm, respectively.

In terms of load directions, the non-uniform clearance has a little effect on minimum oil-film thickness of bearings. It is clearly seen from Figure 4 that the minimum oil-film thickness in the uniform bearing changes periodically with load directions, from 26.5 μm to 30.5 μm, while this periodicity variation cannot be seen in the non-uniform bearing due to asymmetric geometry.

3.3. Pressure distributions

As known that, pressure distribution of oil film is the most fundamental characteristic of bearing. The pressure distributions of oil-film on each pad are obtained by integrating Reynolds equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right)=\frac{\partial}{\partial x}\left(\rho\frac{h}{2}(U_1+U_2)\right)$$
$$+\frac{\partial}{\partial z}\left(\rho\frac{h}{2}(W_1+W_2)\right)-\rho U_2\frac{\partial h}{\partial x}-\rho W_2\frac{\partial h}{\partial z}+\rho(V_2-V_1) \quad (1)$$

where h is the oil-film thickness, p is the pressure in the fluid-film, μ is the lubricant dynamic viscosity, z is the axial direction, x is the tangential direction, ρ is the density of oil. The velocity vector component of the shaft is described by U1, V1, W1 and that of the pads by U2, V2, W2. In this paper, the oil-film thickness is evaluated considering a fixed-pivot point. The oil film thickness is given by:

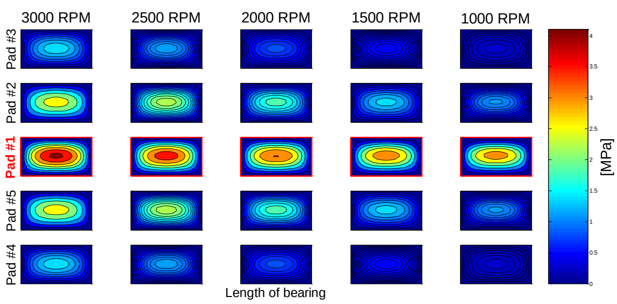
$$h=C_p-\left(C_p-C_b\right)\cos\left(\theta-\alpha\right)-\left(R+t\right)\delta\sin\left(\theta-\alpha\right)$$
$$+X\cos\left(\theta\right)+Y\sin\left(\theta\right)+\delta_r \quad (2)$$

This expression is applicable to pads with both the line type and ball–socket pivots. Radial displacement term δ_r represents pad displacements due to both inclination and deflection.

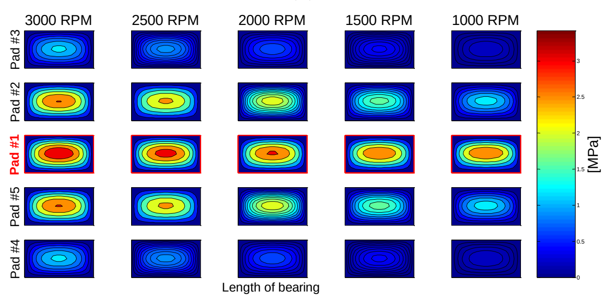
To illustrate the effect of non-uniform clearance on oil-film pressure distribution, Figure 5 shows the pressure distributions on each pad of two bearings in load-on-pad configuration at five different rotational speeds of rotor.

Pressure on each pad increases proportional to increasing of rotational speed of rotor. It is clearly seen that the pressure distributions on two bearings are quite similar. At the pad 1 of the uniform bearing, the maximum pressure at 3000 rpm is more than 4.0 MPa; this value for the non-uniform bearing is approximately 3.5 MPa.

It can be explained that in the non-uniform bearing, thickness of pad #1 is smaller than compared to that on the uniform bearing. It makes the oil-film thickness at pad #1 in the non-uniform bearing larger, so the oil-film pressure is smaller than that on the uniform one.



(a)



(b)

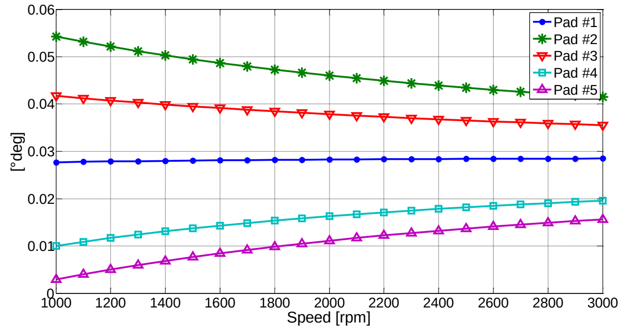
Figure 5. *Pressure distribution of (a) uniform bearing and (b) non-uniform bearing*

3.4. Pad tilt angle

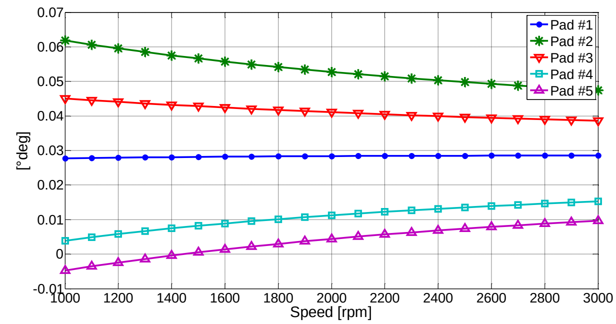
It can be seen that from Figure 6 the loaded pad, namely pad #1, in both tested bearings has a very small movement, nearly 0.03°. Besides, this angle value is quite stable when rotor speed increases.

On contrary, pad #2 has larger movements but this displacement decreases with increasing of rotational speed of shaft. In this kind of bearing, the tilt angle of pad #5 is the smallest and it increases proportional to rotational speed. Note that the shaft is assumed to be rotated in anticlock-wise direction, so the rotor tends to locate on the left hand side when the shaft is rotated.

In general, the non-uniform clearance has a little impact on the pad tilt angle of TPJBs. And the tilt pad angle of five pads on two kinds of bearings will be small at high speed.



(a)



(b)

Figure 6. *Pad tilt angle of (a) uniform bearing and (b) non-uniform bearing*

3.5. Power loss

The estimated bearing power losses as a function of the rotational speed is given in Figure 7. The power losses in both bearings increase linearity with increasing of rotational speed of rotor. When speed of rotor increases from 1000 rpm to 3000 rpm, the power loss of two bearings increases dramatically, from 0.4 kW to 2.3 kW (about 6 times). It can be seen that, power loss in the non-uniform bearing is always smaller than the uniform one, but the difference is insignificant. However, this evidence can be useful for design bearing to reduce the power loss. In particularly, the power losses of two bearings at low speed (1000 rpm – 1500 rpm) are quite similar.

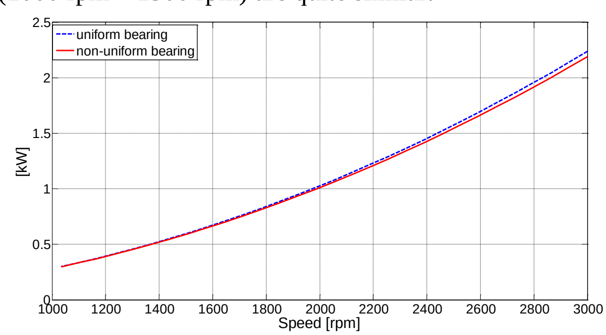


Figure 7. *Power loss vs. rotational speed*

4. Conclusions

The paper presents effects of non-uniform clearance on steady-state by comparison two models of five-pad TPJBs. Because machining errors could lead to actual bearing geometry and dimensions different from the nominal ones. In particular, for TPJB the asymmetry of the bearing geometry leads to unexpected behavior of the bearing. The analysis model shows that it is necessary to consider effects of the non-uniform clearance on the static characteristics of TPJBs such as shaft locus, minimum oil-film thickness, pressure distribution and power loss.

Some conclusions can be drawn from the paper:

1. The pad thickness has the larger effect to the shaft locus and clearance profile. The clearance profile of the five-pad TPJB has a pentagon shape.

2. The minimum oi-film thickness increases with increasing of rotational speed. The bearing with loaded-pad error has larger film thickness compared to the uniform bearing.

3. The pad tilt angle of pad#1 of both bearings nearly stable when working. However, this value has a different trend for the pad #2 and pad #5.

4. The power loss for the both bearings increases with rotor speed. But the difference between two bearings is very small and can be ignored.

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