Static Performances Study on a Three-pad Air Journal Bearing Based on Near Field Acoustic Levitation

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Introduction

Air lubricated bearing has the advantages of high speed, low cost and less heat generation owing to the inherently low viscosity of gas [1, 2]. These advantages enable their successful applications in high-speed spindles, inertial gyroscopes and air compressors [3, 4]. Since the aerodynamic effect is not evident in lower rotation speed, the friction and wear will happen for the bearing and rotor during start-up and shutdown stages [5]. The squeeze film bearing which combining the near field acoustic levitation (NFAL) technology and the gas film bearing was introduced to meet the special low wear requirements, even at lower rotor rotation speeds.

Up to now, several studies have indicated that the feasibility of the squeeze film bearing. Ha et al. [5] proposed a squeeze film bearing with elastic hinges and three piezoelectric plates. However, this bearing has a relatively low squeeze-film load capacity. In the research area of the acoustic levitation, it is common sense that the load capacity increases with the increase of the structure vibration amplitude. In order to acquire higher load capacity, Zhao at al. [6] proposed a novel active non-contact journal bearing which consists of three Langevin transducers as the bearing structure. Based on this model, this paper studies static performances of the three-pad squeeze-film bearing. At first, the nonlinear compressible Reynolds equation is solved by finite difference methods to obtain the pressure distribution. And then, the pressure distribution of three-pad bearing is studied. In the end, the effects of vibration amplitude of pad, eccentricity of the rotor and bearing nominal clearance on static performances are discussed.

Numerical Analysis

The geometry of the three-pad squeeze film bearing is shown in Fig. 1. The *y*-axis locates in the vertical position and points to the direction of the rotor's weight. The *x*-axis is perpendicular to the *y*-axis and points to the right direction. The center point, inner radius and length of the bearing are defined as O, R and L, respectively. Similarly, the center point and outer radius of the rotor are defined as O_r and r, respectively. The nominal clearance of the bearing equals to the difference of R and r. The distance of O and O_r is named by eccentricity displacement e. The ratio of eccentricity displacement to nominal clearance is called eccentricity.

The differential equation governing the pressure distribution in fluid film lubrication is widely used in the field of numerical analysis of gas film bearing, which is called Reynolds equation. This equation is based on the Navier-Stokes equations and the continuity equation [7]. The expression of Reynolds equation which both considering the aerodynamic and squeeze effects is as follows

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{ph^3}{12\mu}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{ph^3}{12\mu}\frac{\partial p}{\partial z}\right)=\frac{\omega}{2}\frac{\partial(ph)}{\partial\theta}+\frac{\partial(ph)}{\partial t}, \quad (1)$$

where μ is the viscosity coefficient of ambient air, ω is the rotor rotation speed. The film thickness and pressure are named by h and p, respectively. The first term in the right of equation represents the self-acting aerodynamic effect. The second term represents the self-acting squeeze effect.

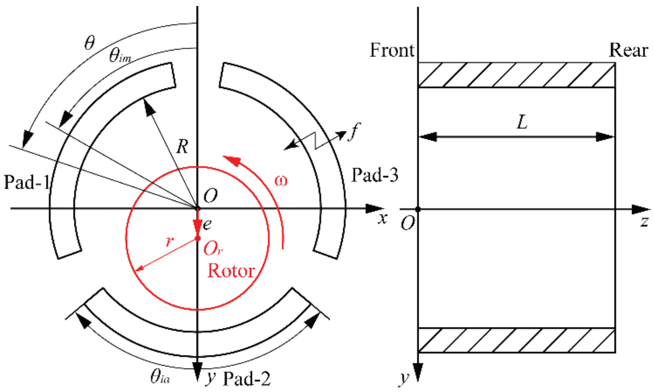


Figure 1: Coordinate system and geometric parameters of the three-pad squeeze bearing

The deformation of every pad is treated as uniform in this study. The squeeze film thickness between the rotor and the bearing is expressed by

$$h=C+e\cdot\cos(\theta)+\xi_i\cdot\sin(t), \quad (2)$$

where ξ_i is maximum vibration amplitude for number i pad. Hereafter, the pressure distribution can be acquired by using the Finite Difference Methods (FDM).

Similar to the oil journal bearing, the bearing forces are acquired by integrating the pressure in horizontal and vertical directions [8]. The bearing forces of squeeze-film air bearing are expressed by

$$F_x=\int_0^{2\pi}\int_0^L(p-p_a)\cdot\sin(\theta)Rd\theta dz$$
$$F_y=\int_0^{2\pi}\int_0^L(p-p_a)\cdot\cos(\theta)Rd\theta dz, \quad (3)$$
$$F=\sqrt{F_x^{\,2}+F_y^{\,2}}$$

where F_x and F_y are the bearing force components in *x*- and *y*-directions, respectively. The total bearing force is named by F.

Numerical results

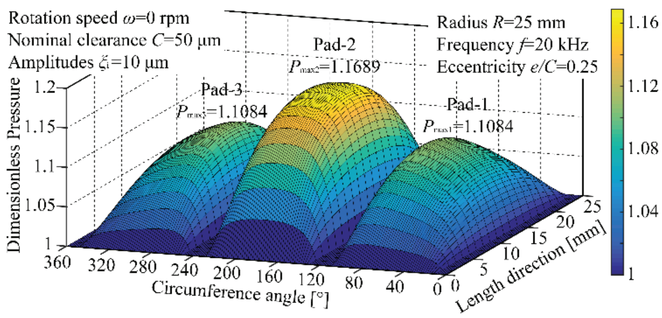
The static erformances of the three-pad bearing are predicted using the parameters shown in Table 1.

Table 1: Basic parameters of three-pad bearings

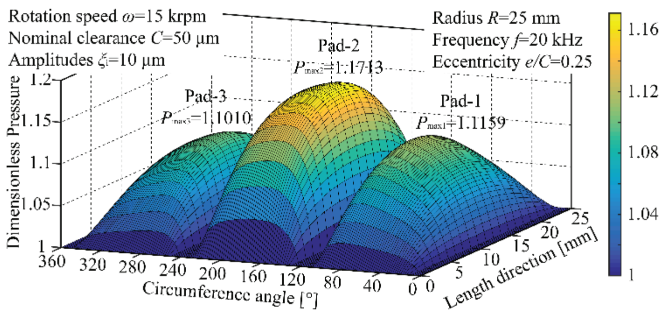
Parameters	Three-pad	Parameters	Three-pad
Rotor radius (r)	24.95 mm	Amplitude (ξ_i)	10 μm
Bearing radius (R)	25 mm	Eccentricity (e/C)	0.25
Clearance (C)	50 μm	Rotation speed (ω)	15 krpm
Bearing length (L)	25 mm	Pad number (i)	3
Pad arc length (θ_{ia})	100°	Pad mid length (θ_{im})	60°, 180°, 240°

Pressure distribution

Substitute the parameters in Table 1 into the above-mentioned equations, the pressure distributions can be acquired. Figure 2 compares the pressure distribution at the rotor rotation with no speed and with speed ω=15 krpm. It is clearly showing that the pressure distribution is symmetrical about the xOz plane at no speed condition. However, the pressure distribution is not symmetrical at the rotor with speed condition due to the aerodynamic effect.



(a)



(b)

Figure 2: Pressure distributions of at the rotor rotation with no speed (a) and with speed (b)

The bearing forces can be acquired by using Eq. (3). The changes in bearing force of two different rotation speed conditions within one cycle at two directions are shown in Fig. 3. In x-direction, owing to the symmetric pressure distribution, the bearing force equals to zero at no speed condition. At the rotor with speed condition, the bearing force value is always a positive value. In y-direction, the bearing force value alternates between negative and positive. Whereas, the bearing force value difference is very small at two speeds. Owing to the aerodynamic effect, the total bearing force at speed condition is higher than that at no speed condition.

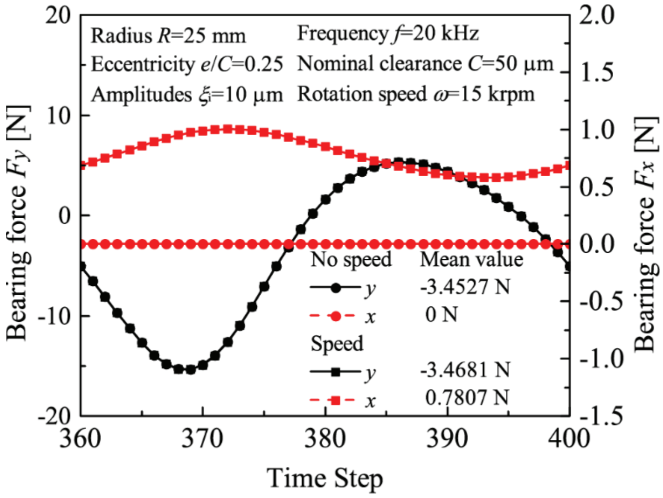


Figure 3: Bearing forces of two different rotation speeds in two directions

Parametric analysis

Eccentricity

The dependence of bearing force on the eccentricity is shown in Fig. 4. It can be seen from the figure that the bearing force increases as the eccentricity increases. However, the relationship between the bearing force and eccentricity is no-linear. The growth gradient of the bearing force increases with the increment in the eccentricity. This result may be explained by the fact that the squeeze effect will be more evident since increases the eccentricity is equivalent to reduce the squeeze film thickness. Comparing with no speed condition, the bearing force is higher with rotor speed. There are very small value differences between no speed and speed. A possible explanation for this might be that the influence of the aerodynamic effect on the bearing force is small in this condition.

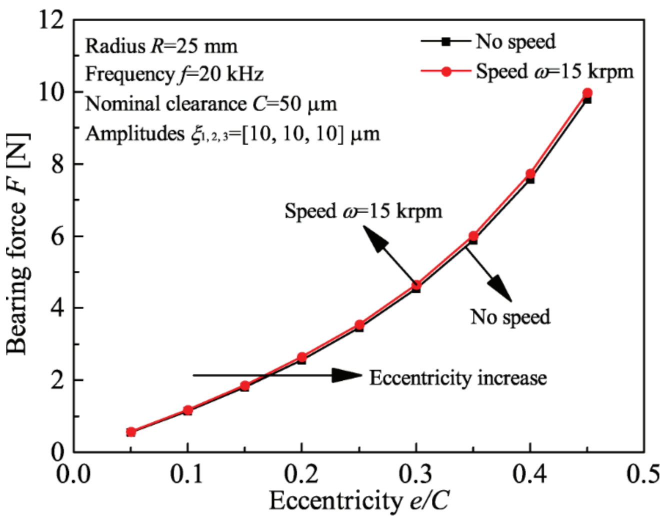


Figure 4: Bearing force versus eccentricity at two rotation speeds

Amplitude

Figure 5 shows the variation of the bearing force with the increment in the amplitude at two speeds. The bearing force increases as the vibration amplitude increases. The exponential relationship between the bearing force and amplitude is striking. Prior studies have found similar results [9]. Interestingly, the difference of bearing force value between no speed and with speed is decreasing with the vibration amplitude increases. Previous studies revealing the aerodynamic effect plays a dominant role in the bearing force in lower vibration amplitude [10]. Correspondingly, the squeeze effect is very important in the higher vibration amplitude.

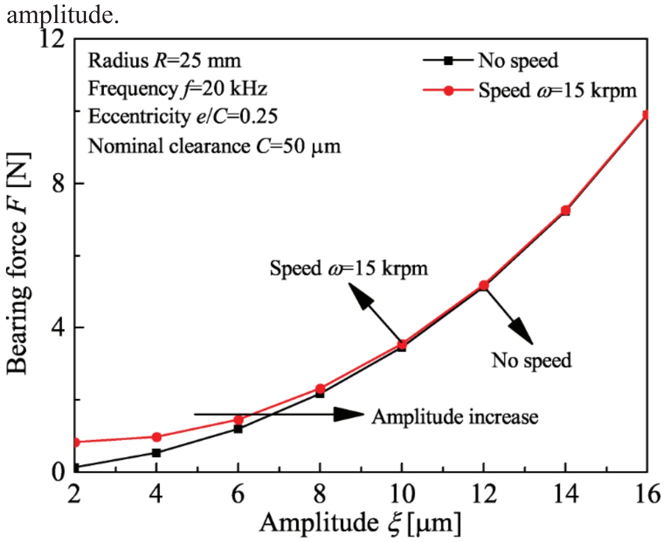


Figure 5: Bearing force versus amplitude at two rotation speeds

Nominal clearance

Figure 6 presents the effect of the nominal clearance on the bearing force. It is observed that the bearing force decreases with the clearance increases. The results seem to indicate that the nominal clearance has no effect on the aerodynamic effect. Therefore, to acquire a higher load-carrying capacity, the nominal clearance between the rotor and the bearing should be designed as small as possible.

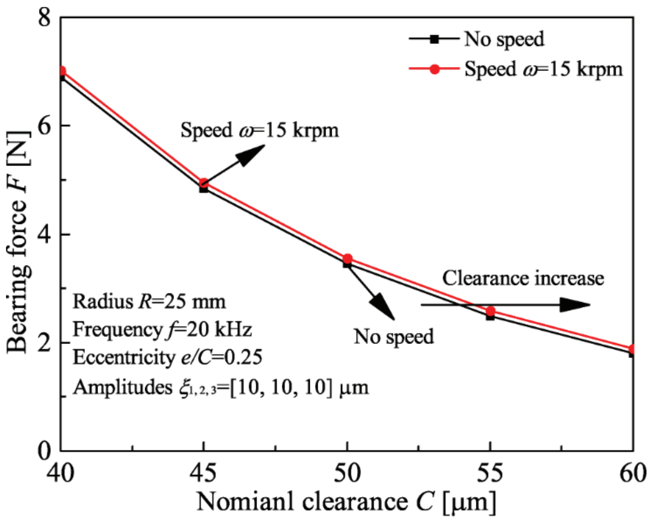


Figure 6: Bearing force versus nominal clearance at two rotation speeds

Conclusions

This paper presented the static performances of the three-pad bearing. The Reynolds equation was solved by FDM to obtain the pressure distribution of the bearing. By comparing the bearing forces of two different rotation speed conditions at two directions, it was found that the bearing force value difference between them is very small. This outcome further confirmed by the bearing force results in different eccentricities. However, in lower amplitude, the bearing force value difference between them is relatively higher. In the end, the higher load-carrying capacity can be acquired by decreases the nominal clearance.

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