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Comparison between the effects of single-pad and double-pad aerostatic bearings with pocketed orifices on bearing stiffness

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This study compares the stiffness of the single-pad and the double-pad aerostatic bearings with pocketed orifices. The stiffness comparison is based on the given loads, which provide useful information for bearing design. Bearing design parameters include supply pressure, pocket size, orifice design, and applied load. Therefore, the analysis of double-pad aerostatic bearings is complex because of the independent design of upper and lower bearings. Results show that a double-pad aerostatic bearing with the same upper and lower bearing is not the best design and that double-pad aerostatic bearings have higher stiffness than the single-pad aerostatic bearings.

1. Introduction

Aerostatic bearings are commonly used in ultra-precision machines because of generating almost zero friction and low heat. The aim of the bearing design is to improve the bearing performance. Improving bearing stiffness is one method of to increase system stability. Pocketed orifice bearing analysis revealed that several design parameters such as bearing size, pocket size, orifice design, supply pressure, and bearing load affected bearing performance.

In aerostatic bearing analysis, the discharge coefficient of orifice is usually assumed to be a constant and this value is obtained from experimental results. There are many factors to affect the discharge coefficient value. Belforte et al. [1] conducted an experimental study to determine the discharge coefficient of orifice-type restrictors. The results showed that the annular orifices and the shallow pocket orifices had one discharge coefficient and deep pocket orifices had two discharge coefficients. They also presented approximation functions for discharge coefficients based on the Reynolds number and feeding system geometry and examined the effect of the pocket depth (d) on pressure distribution in the pocket. For a given film thickness (h), uniform pressure distribution was achieved in the pocket if $d \geq h$.

Chen and He [2] studied the effect of the recess shape on the bearing performance. They found that the rectangular recessed bearings had higher load capacity than the spherical recessed and non-recessed bearings. The mass flow rate was largest in the rectangular recessed bearing and smallest in the non-recessed bearing. Moreover, they examined the effect of the orifice diameter on the bearing performance. At a certain supply pressure and film thickness, load capacity increased with the increase of the orifice diameter. Li and Ding [3] studied the influence of the geometrical parameters of the aerostatic thrust bearings with pocketed orifice type restrictors on bearing performance. They indicated that bearings performed well if orifice diameter and film thickness were small and air chamber diameter was large. Ignoring the influence of orifice length on bearing performance may result in large errors if orifice diameter was sufficiently small.

Schenk et al. [4] showed that the load capacity increased almost linearly and bearing stiffness increased rapidly at the operating point when supply pressure increased. Chen et al. [5] studied the effects of the operational conditions and geometric parameters on the stiffness of the aerostatic journal bearings. They developed a reliable theoretical model to calculate the gas-bearing stiffness and this model was validated by experimental results. The results showed that for a given film thickness, stiffness was improved when supply pressure increased. They also found that the stiffness of the pocketed orifice was higher than that of the inherent orifice.

Pneumatic instability is an important issue in the analysis of aerostatic bearing. Ye et al. [6] studied the effect of the recess shape on pneumatic hammering and found that the non-recessed aerostatic bearings were causing less pneumatic hammering than the recessed aerostatic bearings. They also studied the relationship between the supply pressure and the pneumatic instability. For a given load, reducing supply pressure may increase pneumatic stability. They also examined the effect of recess volume on pneumatic instability and found that a larger recess volume caused more self-excited vibrations. Talukder and Stowell [7]

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Nomenclature

B	Bearing width (m)
b	Pocket width (m)
c	Bearing clearance (m)
C_d	Orifice discharge coefficient
d_o	Orifice diameter (mm)
h	Film thickness (m)
k	Specific heat ratio of air
L	Bearing length (m)
l	Pocket length (m)
$\dot{m}_b$	Mass flow rate from bearing (kg/s)
$\dot{m}_o$	Mass flow rate through orifice (kg/s)
p	Film pressure (N/m^2)
P_a	Ambient pressure (N/m^2)
P_r	Pocket pressure (N/m^2)
P_s	Supply pressure (kg/cm^2)
U	Bearing speed (m/s)
W	Bearing load (N)
W_{upper}	Load capacity of upper bearing (N)
W_{lower}	Load capacity of lower bearing (N)
μ	Viscosity of air (N-s/m^2)
ρ	Density of air (kg/m^3)
ρ_s	Density of supply air (kg/m^3)

studied pneumatic hammering in an externally pressurized orifice-compensated air journal bearing. They indicated that pneumatic hammering was related to recess volume and orifice diameter and was easily avoided in journal bearings. Their experimental results showed that pneumatic hammering can be prevented using a low supply pressure, small orifice diameter, high load operation, and external damping. Bhat et al. [8] studied the performance of inherently compensated flat pad aerostatic bearings subjected to dynamic perturbation forces. They concluded that pneumatic hammer instability tended to occur at low perturbation frequencies, small orifice diameters, large gap heights, and large supply pressures. The results of [6–8] imply that the aerostatic bearings operating at small film thicknesses caused less pneumatic hammering.

Nakamura and Yoshimoto [9,10] studied the static tilt characteristics of the aerostatic rectangular double-pad thrust bearings with compound restrictors. They studied the effects of applied load types on tilt moment and also compared the compound restrictor and the feed-hole restrictor tilt moments [9]. The results showed that the aerostatic thrust bearings with compound restrictors had larger tilt moments. They also indicated that the double-pad thrust bearings had higher stiffness than the single-pad thrust bearings. In a subsequent study, they compared the tilt stiffness of the single row and double row admission thrust bearings [10]. The results showed that the double row admission thrust bearings can improve tilt stiffness in pitch and roll directions.

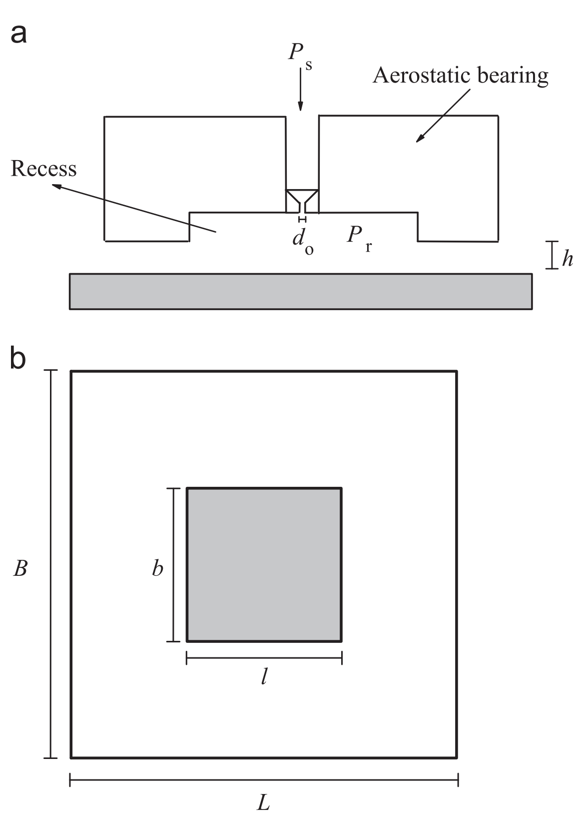

Fig. 1. (a) Diagram of single-pad aerostatic bearing with orifice-type restrictor. (b) Design of bearing and pocket size.

In aerostatic bearings, bearing stiffness affects the stability and precision of the system. Therefore, this study compares the stiffness of the single-pad and the double-pad aerostatic bearings. Double-pad aerostatic bearing analysis is complex because of the independent design of upper and lower bearings. To reduce the complexity of bearing analysis, upper bearing design was fixed and only lower bearing design was investigated.

2. Aerostatic bearing modeling

2.1. Single-pad aerostatic bearings

Fig. 1 shows a single-pad aerostatic bearing with an orifice-type restrictor. Externally pressurized air flows into the rectangular pocket via an orifice restrictor. Air pressure is decreased from P_s to P_r. It is assumed that the pocket depth is large compared to the film thickness and that P_r is uniform in the pocket [1]. The orifice discharge coefficient is assumed to be a constant and the mass flow rate of air through the orifice is calculated using Eq. (1)

$$\dot{m}_o = \frac{\pi d_o^2}{4} C_d \sqrt{2\rho_s P_s \left(\frac{k}{k-1}\right)\left[(P_r/P_s)^{2/k} - (P_r/P_s)^{(k+1)/k}\right]} \quad \text{if } \frac{P_r}{P_s} \geq \left(\frac{2}{k+1}\right)^{k/(k-1)}$$

$$\dot{m}_o = \frac{\pi d_o^2}{4} C_d \sqrt{\rho_s P_s k \left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}} \quad \text{if } \frac{P_r}{P_s} < \left(\frac{2}{k+1}\right)^{k/(k-1)} \tag{1}$$

where P_s is supply pressure, ρ_s is supply air density, P_r is pocket pressure, d_o is orifice diameter, C_d is discharge coefficient, and k is specific heat ratio of air.

In aerostatic bearings, an air film separates the bearing surface and reaction plate. The supply air forms this film and it can provide the bearing load capacity, stiffness, and damping. The pressure distribution of the film can be obtained by solving Reynolds equation. When solving Reynolds equation, uniform pressure in the pocket is used as the boundary condition. The general form of Reynolds equation is shown in Eq. (2)

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial y}\right) = \frac{U}{2}\frac{\partial(\rho h)}{\partial x}, \tag{2}$$

where p is film pressure, h is film thickness, ρ is air density, μ is air viscosity, and U is speed. Assume that the air is an ideal gas ($\rho = p/RT$)