Research Article

Study on Static Characteristics of Annular Array Microporous Aerostatic Bearing

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In this paper, an aerostatic bearing model in the form of microorifice annular array is designed to explore the static performance of aerostatic bearing when the orifice diameter and working air film thickness are of the same order of magnitude. Firstly, the simulation model of the new bearing is established, the pressure distribution of the bearing gas film under the influence of the orifice diameter is analyzed by the CFD method, and the performance of the aerostatic bearing is compared with the traditional toroidal throttling method with the orifice diameter of 0.1 mm; at the same time, the effects of air supply pressure, orifice diameter, and orifice number on the static performance of the bearing under different air film thickness are analyzed; finally, a new bearing is developed, and the simulation results are verified by the experimental test. The results show that when the orifice diameter and the air film thickness are of the same order of magnitude, the maximum stiffness of the aerostatic bearing is significantly improved and the microporous throttled aerostatic bearing is more suitable for the field of high stiffness. The greater the air supply pressure, the greater the bearing capacity and stiffness of the bearing. The number of orifices has an effect on improving the bearing capacity of the bearing, but when the number of orifices is large, the effect will become less obvious. This study can provide a new theoretical reference for the design of aerostatic bearing.

1. Introduction

Aerostatic bearing has the advantages of high precision, high temperature resistance, simple structure, and sanitation. It has been widely used in the fields of ultra-precision manufacturing, aerospace experiment, and precision measurement [1–5]. As an important pressure compensation unit of aerostatic bearing, the performance of restrictor affects the application of aerostatic bearing. However, due to the compressibility of air, the problems of low bearing capacity and poor stiffness of aerostatic bearing have become a great obstacle to its further application.

In order to improve the performance of aerostatic bearing, researchers have done a lot of research from various throttling methods. Gao used CFD method to study the influence of six different restrictor structures on the working characteristics of aerostatic bearing [6]. Jeng and others analyzed and compared the influence of single row and double row orifice arrangement on bearing stiffness according to different factors such as air supply pressure, air chamber size, orifice geometry, and external load [7]. Ye carried out experimental research on four kinds of aerostatic bearings with different throttling cavity shapes: spherical, rectangular, grooveless, and diamond [8]. G. Belfort studied the flow coefficient at the outlet of single supply hole and annular hole through experiments and approximately established the approximate expressions of supply pressure, film thickness, and flow coefficient based on Reynolds number and orifice geometry [9]. Li simulated and analyzed the small hole throttling aerostatic bearing with different geometric parameters, analyzed the influence of bearing geometric parameters on its bearing capacity, stiffness, and mass flow, and optimized the size of this kind of bearing [10]. Aoyama changed the right angle connection at the junction of orifice and air chamber to fillet connection, which reduced the strength of cyclone and reduced the microvibration of
bearing [11]. Li used the large eddy simulation method to analyze the time-varying flow field of the toroidal throttled aerostatic bearing. It was found that the vortex shedding would cause the pressure fluctuation in the gas film. At the same time, it was pointed out that the design methods of reducing the orifice diameter, increasing the bearing diameter, reducing the throttling cavity depth, and reducing the air supply pressure can reduce the microvibration of the aerostatic bearing under load [12]. There are also some researchers using the aerostatic bearing with pressure equalizing groove throttling. Yoshimura studied the gas film fluctuation mechanism of thrust bearing with T-groove. Through simulation and experimental verification, it is known that the gas film fluctuation of thrust bearing with T-groove is caused by the change of Reynolds number caused by airflow and air velocity at the bearing edge connected to the external atmospheric environment [13]. Du systematically studied the influence of structural parameters such as length, depth, number, and position of pressure equalizing groove on bearing capacity and stiffness. The results show that the depth and layout position of pressure equalizing groove have a great influence on load performance [14]. In order to improve the stability of grooved hydrostatic gas thrust bearing, Ma set a damping hole array in the high-pressure area of the bearing [15]. With the in-depth research, especially the development of computer simulation technology, researchers began to focus on the impact of bearing geometry on bearing performance, especially for the small orifice diameter. Chen Xuedong conducted a systematic theoretical and experimental study on the orifice flow pattern with a minimum orifice diameter of 0.1 mm through the CFD simulation analysis of toroidal throttling and orifice throttling aerostatic bearing and analyzed the intensity of cyclones in gas chambers with different shapes [16]. Yoshimoto and others analyzed the intensity of cyclones in gas chambers with different areas [17]. The grid quality is good, which can meet the calculation requirements.

To sum up, most scholars have done a lot of research on the aerostatic bearing with an orifice diameter of more than 0.1 mm. Restricted by the processing capacity, there is a lack of theoretical and experimental analysis on the aerostatic bearing with an orifice diameter of less than 0.1 mm (when the orifice diameter and the working air film thickness are of the same order of magnitude). Taking the aerostatic bearing with orifice diameter of 0.05 mm as the research object, this paper proposes a model of annular array microporous throttling aerostatic bearing, simulates its hydrostatic characteristics by CFD method, and compares the simulation results with the performance of traditional annular throttling aerostatic bearing; at the same time, the effects of orifice diameter, orifice number, and air supply pressure on the static characteristics of the new bearing are analyzed; finally, the orifice has a diameter of 0.05 mm bearing, and experimental analysis is carried out.

2. Physical Model and Calculation Method

2.1. Physical Model. Figure 1 shows the physical model of annular array microporous throttling aerostatic bearing. The bearing radius is R, the orifice diameter is d, the orifice depth is h, the film thickness is h, the orifice distribution radius is r, and the number of orifices is N. The air compressor sends pressurized gas into the air chamber of the bearing, then flows between the working surface and the base through the orifice in the air chamber, and finally reaches the external air.

The bearing model has a ring array of orifices on the bearing surface. Through the cooperation of these orifices, a high-pressure area can be established in the bearing to enhance its bearing capacity; at the same time, the orifice diameter in the model is small, and its scale has the same order of magnitude as the working air film thickness of the bearing. Through the coordinated change of gas resistance of the same order of magnitude, the throttling efficiency is increased and the bearing performance is improved.

2.2. Meshing. Due to the axial center symmetry of the gas film physical model of the new bearing, in order to simplify the difficulty of mesh generation and improve the calculation efficiency, the minimum sector area of a throttling area is used for modeling, and the whole model is divided into n small sector areas along the center. At the same time, due to the great difference in the size of the gas film structure in the axial and radial directions, the structured grid is used to divide the model. After the preliminary division, the grid at the connection between the orifice outlet and the air film thickness is densified to improve the grid quality and accuracy. Figure 2 shows a structured grid divided by the minimum sector area of an orifice area.

The grid quality in this paper adopts $3 \times 3 \times 3$ determinant. The range is 0–1, and the grid quality is greater than 0.65, which is mainly concentrated in the range of 0.8–1. The grid at the gas film is 10 layers, and the grid at the orifice is divided into 100 layers. The grid quality is good, which can meet the calculation requirements.

2.3. Governing Equations and Environmental Variables. At normal temperature, without considering the influence of temperature thermal coupling, the gas is an ideal gas. Since the residence time of the gas in the gas film is very short, it is assumed that the flow process is adiabatic, there is no heat exchange, and the influence of wall roughness is ignored. The gas control equation is as follows.

\[ \frac{\partial p}{\partial t} + \frac{\partial (pu)}{\partial x} + \frac{\partial (pv)}{\partial y} + \frac{\partial (pw)}{\partial z} = 0. \] (1)

In formula (1), $u$ is the velocity in the $x$ direction; $v$ is the speed in the $y$ direction; $w$ is the velocity in the $z$ direction; $\rho$ is density; $t$ is the time.

Momentum equation:
In formula (2), $\mu$ is the dynamic viscosity coefficient of the gas; $f$ is the volumetric force.

$f$ belongs to external force, and the influence of volume force is not considered in this study, which has been ignored.

SIMPLE algorithm is the most widely used flow field calculation method in engineering at present. It belongs to the pressure correction method. The pressure field obtained by SIMPLE algorithm in each iteration step cannot fully meet the momentum equation, so it needs to iterate repeatedly until it converges. Its core is the process of “guess correction.” This paper selects K- $\varepsilon$. The turbulence model is simulated by simple algorithm. Air is used as the bearing medium. The whole simulation calculation is carried out under normal temperature and standard atmospheric pressure. The environmental variable parameters are shown in Table 1.

The static performance of aerostatic bearing mainly includes bearing capacity and stiffness. The bearing capacity is the integral of the pressure distribution on the bearing surface, and the expression is...
3. Results and Discussion

3.1. Simulation Results of Microporous Throttling. In this section, a specific calculation example is given to illustrate the influence of microporous restrictor on the performance of aerostatic bearing. The structural dimensions of the bearing are shown in Table 2. Calculate the pressure distribution on the bearing surface under different air film thickness $h$.

Figure 4 is a cloud diagram of pressure distribution when the air film thickness is equal to 3 mm, 5 mm, 7 mm, 9 mm, 11 mm, and 13 mm. The pressure at the orifice is the largest, a circular isobaric surface is formed in the annular area surrounded by the orifice, and the pressure accumulation effect is obvious. In the area of orifice and bearing edge, the bearing pressure decreases rapidly. Going farther away from the orifice, the pressure decreases gradually and finally tends to the set outlet atmospheric pressure. Comparing the pressure under different air film thickness, it can be seen that, with the increasing air film thickness, the pressure on the circular isobaric surface decreases gradually. In the area of orifice and bearing edge, the pressure is also decreasing, but the decreasing speed is slow; when the air film thickness is greater than 11 μm, the pressure in this area almost drops to the ambient pressure.

In order to further study the influence of microporous throttling on aerostatic bearing, the static characteristics of aerostatic bearing with traditional toroidal throttling are compared and analyzed. Although the two models have some differences in physical structure, from the gas model, the main difference is that the orifice diameter of aerostatic bearing with toroidal throttling is relatively large. In this paper, the diameter of annular orifice selected for comparative calculation is 0.1 mm, and the calculation parameters are consistent with other environmental variables. According to formulas (3) and (4), the comparison results are shown in Figure 5.

Figure 5(a) is a comparison diagram of the bearing capacity of the two bearings with the air film thickness. It can be seen from Figure 5(a) that the bearing capacity of toroidal throttling is greater than that of microporous throttling, and the difference between the two curves increases first and then decreases under the same air film thickness. When the air film thickness $h$ is less than 5 mm, the two bearing capacity curves are basically consistent. This is because when the air film thickness is small, the gas resistance of orifice and air film is high, the bearing has strong ability to maintain pressure, and the gas resistance of orifice and gas film are maintained at a high level. With the increase of air film thickness, the bearing capacity curve of microporous throttling bearing decreases faster than that of toroidal throttling bearing, and the pressure difference between the two curves increases. When the clearance $h$ increases to 15 mm, the bearing capacity difference between the two

### Table 1: Model simulation environment variable parameters.

| Parameter       | Outlet pressure, MPa | Temperature K | Density kg·m$^{-2}$ | Viscosity Pa·s |
|-----------------|----------------------|---------------|---------------------|----------------|
| Value           | 0.1                  | 300           | 1.225               | $1.7894 \times 10^{-5}$ |

$W = \int_0^x \int_0^y [p(x, y) - Pa] dx\, dy$. 

In formula (3), $Pa$ is the standard atmospheric pressure; $p(x, y)$ is the film pressure. The stiffness of aerostatic bearing is the variation of bearing capacity under small displacement, and its expression is

$$K = \frac{dW}{dh}$$ (4)

In formula (4), $dh$ is the small change of gas film and $dW$ is the variation of bearing capacity.

### 2.4. Boundary Condition. The orifice inlet is connected with the air chamber inside the bearing, so the pressure in the air chamber is equal to the pressure at the inlet and also the air supply pressure provided by the air compressor.

$$p = p_i.$$ (5)

The outlet of the gas film is connected to the external environment, so it is set to 1 standard atmospheric pressure.

$$p = p_a.$$ (6)

Because only a part of the whole bearing is calculated, the split section of the model is set as the symmetrical boundary, its normal velocity is zero, and the normal gradient of physical quantity on the symmetrical plane is zero. Set other walls as walls without slip boundary conditions. The $y+$ is a nondimensional distance to the wall; $y+$ should be between 0–5.

### 2.5. Independence Detection of the Model Partition Method. In order to verify whether the model division method in this paper has an impact on the calculation results, the sector area of one orifice, the sector area of two orifices, and the sector area of three orifices are used as different models for grid division, as shown in Figure 3.

By setting the same boundary conditions and calculation parameters, the surface pressure distributions of the three models are calculated. Further processing shows that the sum of the sector area pressure of one orifice is 12.4 N, the sum of the sector area pressure of two orifices is 24.9 N, and the sum of the sector area pressure of three orifices is 37.4 N. Through comparison, it can be seen that the sector area of the number of orifices is directly proportional to the sum of the pressure in this area. Therefore, the whole model is divided into $n$ minimum sector areas along the center (each sector area has only one orifice) for solution and calculation, which will not affect the final calculation results.
bearings is the largest, and the difference reaches 347 N. This is because the increase of air film thickness rapidly reduces the gas resistance at the air film thickness, resulting in the rapid release of pressure in the restrictor, while the gas flow of toroidal throttling is greater than that of microporous throttling, resulting in the increase of pressure drop of microporous throttling. With the further increase of the air film thickness, the difference between the two begins to decrease gradually. This is because the air film thickness has exceeded the optimal working clearance, the gas resistance decreases sharply, and the flow difference caused by the orifice diameter decreases.

Figure 5(b) is a comparison curve of the stiffness of the two bearings with the air film thickness. It can be seen from Figure 5(b) that their stiffness increases first and then decreases. The stiffness of microporous throttling is greater than that of toroidal throttling, and there is an optimal stiffness value. The optimum stiffness value of microporous throttling corresponds to the air film thickness at 13 mm, and the maximum stiffness value is 50.5 N·mm⁻¹, while the optimum stiffness value of toroidal throttling corresponds to the air film thickness at 23 mm, and the maximum stiffness value is 31.3 N·mm⁻¹. When the air film thickness is greater than 20 mm, the annular throttled aerostatic bearing has greater stiffness than the microporous throttled aerostatic bearing. This is because according to the calculation formula (4) of stiffness, stiffness is the ratio of the change of bearing capacity to the change of air film thickness. When the air film thickness is small, the change of bearing capacity is small, so the ratio is small. When the air film thickness increases beyond the optimal value, the stiffness decreases.
increases to the optimal stiffness range, the bearing capacity changes sharply and its ratio increases, so the stiffness increases first. When the air film thickness is far from the optimal stiffness range, the change of bearing capacity decreases, so the stiffness begins to decrease again.

Figure 5(c) is a comparison diagram of air consumption of two bearings with air film thickness. It can be seen from Figure 5(c) that when the air film thickness $h$ is less than 12 mm, the gas consumption of the bearings with two different throttling methods is basically the same. When the air film thickness $h$ is greater than 12 mm, the gas consumption of toroidal throttling increases significantly. Therefore, the growth rate of gas consumption in toroidal throttling mode is greater than that in microporous throttling after the air film thickness is greater than 12.5 mm, and the difference between gas consumption in two different throttling modes is larger and larger.

Therefore, it can be concluded that, under the condition of small air film thickness, although the bearing capacity of microporous throttling aerostatic bearing is slightly reduced, its stiffness is significantly improved, and the air consumption is also small. The performance of microporous throttling aerostatic bearing is obviously better than that of traditional toroidal throttling aerostatic bearing.
3.2. Effect of the Orifice Diameter on Static Performance of Microporous Throttled Aerostatic Bearing. This section mainly analyzes the influence of orifice diameter on the static performance of microporous throttled aerostatic bearing. The bearing diameter \( D \) is 60 mm, the air supply pressure \( p \) is 0.5 MPa, the distribution radius \( r \) of bearing orifice is 20 mm, the number of holes \( N \) is 80, and the micropore diameter \( d \) is 0.03 mm, 0.04 mm, and 0.05 mm, respectively. The calculation results are as follows.

Figure 6 shows the film pressure distribution of aerostatic bearing under different orifice diameters when the air film thickness is equal to 5 mm. It can be seen from Figure 6 that the high-pressure area is concentrated in the center of the bearing surrounded by the orifice, the pressure in the orifice and the edge area of the bearing is low, and the pressure difference is large. With the increase of the orifice diameter, the color of the bearing center surrounded by the orifice becomes darker and the corresponding gas film pressure becomes larger and larger. The pressure changes in the orifice and bearing edge area are not obvious. Overall, with the increase of orifice diameter, the pressure on the whole bearing surface of the bearing increases.

Figure 7(a) is the influence curve of orifice diameter on bearing capacity. As can be seen from Figure 7(a), the orifice diameter has no obvious effect on the maximum bearing capacity of the bearing. No matter what orifice diameter is, the maximum bearing capacity of the bearing is constant. When the air film thickness \( h \) is 3 mm, the maximum variation range of bearing capacity under three orifice diameters is only 30 N. With the increase of air film thickness, the larger the orifice diameter is, the greater the bearing capacity of aerostatic bearing is. Under the same air film thickness, the difference of bearing capacity of different orifice diameters first increases and then decreases. The smaller the orifice diameter is, the most obvious the change of bearing capacity is. When the air film thickness \( h \) increases to about 10 mm, the orifice diameter has the most significant effect on the bearing capacity.

Figure 7(b) is the influence curve of orifice diameter on bearing stiffness. As shown in Figure 7(b), the aerostatic bearings with different orifice diameters have an optimum stiffness value. The smaller the orifice diameter is, the greater the optimum stiffness value is. When the orifice diameter is 0.03 mm, the maximum stiffness is 76.5 N-mm\(^{-1}\), the orifice diameter increases to 0.05 mm, and the stiffness value is 50.5 N-mm\(^{-1}\). At the same time, the smaller the orifice diameter is, the smaller the air film thickness corresponding to the maximum stiffness is. When the orifice diameter \( d \) is 0.03 mm, 0.04 mm, and 0.05 mm, respectively, the air film thickness \( h \) of the optimal stiffness of the bearing is 9 mm, 11 mm, and 13 mm, respectively. When the thickness of the air film is greater than 13 mm, stiffness decreases. The smaller the orifice diameter, the faster the decline rate. Here, the reason why the stiffness first increases and then decreases is the same as that in Figure 5(b), which is determined by the change of bearing capacity in the change range of air film thickness.

Figure 7(c) shows the influence curve of orifice diameter on bearing air consumption. As can be seen from Figure 7(c), under the same air film thickness, the larger the orifice diameter, the greater the air consumption of the bearing. The gas consumption increases with the increase of air film thickness. When the air film thickness is small, the influence of orifice diameter on gas consumption is little due to the limitation of the whole flow field. With the increase of air film thickness, the gas consumption of bearing with large orifice diameter increases greatly, and the difference is also larger and larger. When the air film thickness \( h \) is greater than 15 mm, the corresponding gas consumption tends to be stable when the orifice diameter \( d \) is 0.03 mm and 0.04 mm, while when the orifice diameter is 0.05 mm, the gas consumption continues to increase with the increase of air film thickness.

3.3. Effect of the Orifice Number on Static Performance of Microporous Throttled Aerostatic Bearing. This section analyzes the effect of the number of orifices on the static performance of microporous throttled aerostatic bearing under different air film thickness. The bearing diameter \( D \) is 60 mm, the air supply pressure \( p \) is 0.5 MPa, the bearing distribution radius \( r \) is 20 mm, and the hole diameter \( d \) is 0.05 mm. The number of orifices \( N \) is 60, 70, and 80, respectively.

Figure 8 shows the pressure distribution of local air film of aerostatic bearing with different number of orifices when the air film thickness \( h \) is equal to 7 mm. The more the number of orifices, the closer the distance between adjacent orifices, the stronger the pressure accumulation at the bearing center surrounded by orifices, and the smaller the pressure difference between two adjacent orifices. It further shows that the number of orifices has an important influence on improving the bearing capacity of aerostatic bearing.
Figure 9(a) shows the influence curve of the number of orifices on the bearing capacity. It can be seen from Figure 9(a) that when the air film thickness is small, the number of orifices has little effect on the maximum bearing capacity of the bearing. When the air film thickness is equal to 5 μm, the bearing capacity under the number of three orifices is equal to 989 N, 976 N, and 961 N, respectively, of which the maximum difference is 28 N. With the increase of air film thickness, the more the number of orifices under the same air film thickness, the greater the bearing capacity.

Figure 9(b) shows the influence curve of the number of orifices on the bearing stiffness. As can be seen from Figure 9(b), the stiffness increases first and then decreases. Under different number of orifices, the bearing has an optimal stiffness value. The maximum stiffness values of bearings with the number of orifices equal to 60 and 70 are basically the same, but the corresponding air film thickness values are different, which are 11 μm and 13 μm, respectively. The maximum stiffness of the bearing with the number of orifices equal to 80 is slightly lower. This is because the more the number of orifices, the smaller the variation range of bearing capacity with the increase of air film thickness, so the stiffness value is too small.

Figure 9(c) shows the influence curve of the number of orifices on the air consumption of the bearing. As can be seen from Figure 9(c), the gas consumption increases...
Figure 8: Air film pressure distribution of bearings with different number of orifices ($p = 0.05$ MPa, $d = 0.05$ mm, $h = 7$ mm).

Figure 9: Continued.
gradually with the increase of air film thickness. Under the condition of small air film thickness, the number of orifices has little effect on the air consumption of bearing; when the air film thickness is less than 7 μm, the air consumption of the 3 bearings is basically equal to 1100; With the increase of air film thickness, the more the number of orifices, the greater the air consumption of the bearing, and the faster the growth trend.

3.4. Effect of Air Supply Pressure on Static Performance of Microporous Throttling Aerostatic Bearing. This section analyzes the impact of different air supply pressures on the static performance of bearings. The bearing diameter D is equal to 60 mm, the orifice diameter d is equal to 0.05 mm, the number of orifices N is equal to 80, and the air supply pressure p is equal to 0.4 MPa, 0.5 MPa, and 0.6 MPa, respectively. The simulation results are shown in the figure below.

Figure 10(a) shows the influence curve of air supply pressure on bearing capacity. From Figure 10(a), it can be seen that the greater the air supply pressure, the greater the bearing capacity. Compared with other parameters, the air supply pressure has the greatest bearing on the bearing capacity. The smaller the air film thickness, the greater the influence of air supply pressure on bearing capacity. When the air film thickness is 3 mm, the difference reaches the maximum value of 180 N; with the increase of air film thickness, the influence of air supply pressure on bearing capacity will gradually decrease.

Figure 10(b) shows the influence curve of air supply pressure on bearing stiffness. As can be seen from Figure 10(b), the overall bearing stiffness increases first and then decreases, and the position of gas film thickness corresponding to the stiffness peak is basically the same, which is equal to 13 mm. The bearing stiffness is directly proportional to the air supply pressure. The greater the air supply pressure, the greater the increase of stiffness.

Figure 10(c) shows the influence curve of air supply pressure on bearing air consumption. As can be seen from Figure 10(c), the air supply pressure has a severe impact on the air consumption of the bearing. The greater the air supply pressure, the greater the air consumption of the bearing.

4. Experiment

4.1. Test Bearing and Test Bench. In this paper, a new type of bearing with orifice diameter of 0.05 mm is developed by laser machining. Limited by the technical conditions of laser processing, the ratio of the length and diameter of the orifice is no more than 1:10. Therefore, the orifice position of the experimental bearing is a stainless steel plate with a thickness of 0.5 mm. The new aerostatic bearing with orifice diameter of 0.05 mm developed in this paper is shown in Figure 11.

Figure 12 shows the static performance test bench of aerostatic bearing. The test bench is composed of pedestal, cylinder, loading rod, tested bearing, microdisplacement sensor, pressure sensor, and display equipment. The cylinder is installed on the pedestal. One end of the cylinder is connected with a loading rod. The loading rod presses the tested bearing to provide preload for the bearing. A pressure sensor is installed on the lower surface of the bearing and a microdisplacement sensor is installed on the upper surface of the bearing. All mechanical structures of the test bench are installed in the granite frame, and the force loading presents a closed loop, reducing the interference of external factors and the influence of overall frame deformation.

The basic flow of static experimental test: firstly, the cylinder preloads the tested bearing and adjusts the pressure
Figure 10: Effect of air supply pressure on bearing air consumption.

Figure 11: Physical drawing of aerostatic bearing of the annular array microporous restrictor.
sensor and microdisplacement sensor to make it within the working range. Then increase the air supply of the cylinder and increase the load. Record the load data through the pressure sensor and the displacement change of the microdisplacement sensor at the same time. In the test process, the tested bearing is inflated and deflated for many times at the same air film thickness position to eliminate the experimental system error.

4.2. Experimental Test Results. Compare and test the bearing capacity of annular throttled aerostatic bearing with orifice diameter equal to 0.1 mm and microporous aerostatic bearing with orifice diameter equal to 0.05 mm, as shown in Figure 13. The experimental results show that when the air film thickness is equal to 5 μm, the maximum bearing capacity of microporous throttling bearing is 879.4 N and that of toroidal throttling bearing is 896.2 N, which is basically consistent with the theoretical analysis results. With the increase of air film thickness, the bearing capacity of microporous throttling bearing changes sharply with the increase of air film thickness; in particular when the air film thickness is equal to 12 μm, the descending gradient is the largest. It can also be explained that the stiffness of the bearing is the largest under this film thickness. The experimental results are basically consistent with the theoretical simulation results. It is further verified that reducing the orifice diameter can effectively improve the stiffness of the bearing.

This paper tests the bearing capacity of the new bearing under five different conditions when the orifice diameter is equal to 0.05 mm. The comparison results of experimental test and simulation are shown in Figure 14. From the orifice diameter, the bearing capacity of the 3 kinds of orifices is 857.4 N, 869 N, and 879.3 N, respectively. With the increase of air film thickness, the more the number of orifices under the same air film thickness, the greater the bearing capacity. At the same time, it can be found that when the air film thickness is 11 μm, the variation of bearing capacity is the largest, which is basically consistent with the results of theoretical analysis. From the air supply pressure, the bearing capacity under the 3 air supply pressures is 724.5 N, 879.3 N, and 1041.2 N, respectively; The greater the air supply pressure, the greater the bearing capacity. It can also be seen that when the air film thickness is equal to 12 μm, the variation of bearing capacity is the largest, indicating that the bearing can obtain the maximum stiffness under this air film thickness.

The experimental results show that the trend of the experimental results is basically consistent with the theoretical simulation. Under the condition of small air film thickness, the bearing capacity of the experimental bearing is slightly smaller than the simulation results as a whole; the error between the theoretical simulation results and the maximum bearing capacity of the experimental test is 12%~15%. However, with the increase of air film thickness, the difference decreases significantly; take the aerostatic bearing with the air supply pressure equal to 0.5 MPa and the number of orifices equal to 70 as an example; when the film
gap is 5 μm, 7 μm, 9 μm, 11 μm, and 13 μm, respectively, the difference between theoretical simulation value and experimental test value is 160.588 N, 156.927 N, 153.76 N, 133.362 N, and 108.766 N, respectively. This is caused by the roughness and machining error of the bearing surface. With the increase of air film thickness, the influence of roughness decreases obviously. Therefore, the experimental results are closer to the theoretical results.

5. Conclusion

In this paper, the simulation and experimental research on the aerostatic bearing with the same order of magnitude of orifice diameter and air film thickness are carried out for the first time. An annular array aerostatic bearing with a diameter equal to 0.05 mm is successfully developed. The simulation and experimental comparative analysis are carried out for different influencing factors, and the following conclusions are obtained:

1. Reducing the orifice diameter has a positive effect on improving the bearing stiffness. Microporous throttling is more suitable for the case of small working air film thickness.

2. The number of orifices has an obvious effect on improving the bearing capacity of the bearing but has little effect on improving the bearing stiffness.

3. Increasing the air supply pressure is an effective way to improve the bearing capacity and stiffness. This study provides a new theoretical reference for the design of aerostatic bearing.

Data Availability

All models and codes generated or used during the study are included in the article. The raw/processed data required to reproduce these findings cannot be shared at this time as the data also form part of an ongoing study.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Authors’ Contributions

All authors contributed to the study conception and design. Material preparation, data collection, and analysis were performed by Zhao Xiao-Long. The first draft of the manuscript was written by Zhao Xiao-Long, and all authors commented on previous versions of the manuscript. All authors read and approved the final manuscript.

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References

[1] D. Chen, C. Huo, X. Cui, R. Pan, J. Fan, and C. An, “Investigation the gas film in micro scale induced error on the performance of the aerostatic spindle in ultra-precision machining,” *Mechanical Systems and Signal Processing*, vol. 105, pp. 488–501, 2018.

[2] Z.-W. Lu, J.-A. Zhang, and B. Liu, “Research and analysis of the static characteristics of aerostatic bearings with a multi-hole integrated restrictor,” *Shock and Vibration*, vol. 2020, no. 1, pp. 1–11, Article ID 7426928, 2020.

[3] P. Zhang, “Accuracy Prediction Model of an Orifice-Compensated Aerostatic Bearing,” *Precision Engineering*, vol. 72, 2021.

[4] Z. Xiao-Long, Z. Jun-An, D. Hao, F. Zhou, and L. Jun-ning, “Numerical simulation and experimental study on the gas-solid coupling of the aerostatic thrust bearing with elastic equalizing pressure groove,” *Shock and Vibration*, vol. 2017, pp. 1–11, 2017.

[5] Q. Gao, L. Qi, S. Gao, L. Lu, L. Song, and F. Zhang, “A FEM based modeling method for analyzing the static performance of aerostatic thrust bearings considering the fluid-structure interaction,” *Tribology International*, vol. 156, no. 3–4, Article ID 106849, 2021.

[6] S. Gao, K. Cheng, S. Chen, H. Ding, and H. Fu, “CFD based investigation on influence of orifice chamber shapes for the design of aerostatic thrust bearings at ultra-high speed spindles,” *Tribology International*, vol. 92, pp. 211–221, 2015.

[7] Y. R. Jeng and S. H. Chang, “Comparison between the effects of single-pad and double-pad aerostatic bearings with pocketed orifices on bearing stiffness,” *Tribology International*, vol. 66, no. 7, pp. 12–18, 2013.
[8] Y. X. Ye, X. D. Chen, Y. T. Hu, and X. Luo, “Effects of recess shapes on pneumatic hammering in aerostatic bearings,” Proceedings of the Institution of Mechanical Engineers - Part J: Journal of Engineering Tribology, vol. 224, no. 3, pp. 231–237, 2010.

[9] G. Belforte, T. Raparelli, V. Viktorov, and A. Trivella, “Discharge coefficients of orifice-type restrictor for aerostatic bearings,” Tribology International, vol. 40, no. 3, pp. 512–521, 2007.

[10] Y. Li and H. Ding, “Influences of the geometrical parameters of aerostatic thrust bearing with pocketed orifice -type restrictor on its performance,” Tribology International, vol. 40, no. 7, pp. 1120–1126, 2007.

[11] T. Aoyama, K. Koizumi, Y. Kakinuma, and Y. Kobayashi, “Numerical and experimental analysis of transient state micro-bounce of aerostatic guideways caused by small pores,” CIRP Annals, vol. 58, no. 1, pp. 367–370, 2009.

[12] Y. Li, J. Zhao, H. Zhu, and Y. Lin, “Numerical analysis and experimental study on the microvibration of an aerostatic thrust bearing with a pocketed orifice-type restrictor,” Proceedings of the Institution of Mechanical Engineers - Part J: Journal of Engineering Tribology, vol. 229, no. 5, pp. 609–623, 2014.

[13] T. Yoshimura, T. Hanafusa, T. Kitagawa, T. Hirayama, T. Matsuoka, and H. Yabe, “Clarifications of the mechanism of nano-fluctuation of aerostatic thrust bearing with surface restriction,” Tribology International, vol. 48, pp. 29–34, 2012.

[14] J. Du, G. Zhang, T. Liu, and S. To, “Improvement on load performance of externally pressurized gas journal bearings by opening pressure-equalizing grooves,” Tribology International, vol. 73, no. 5, pp. 156–166, 2014.

[15] W. Ma, J. Cui, Y. Liu, and J. Tan, “Improving the pneumatic hammer stability of aerostatic thrust bearing with recess using damping orifices,” Tribology International, vol. 103, pp. 281–288, 2016.

[16] X. Chen, H. Chen, X. Luo, Y. Ye, Y. Hu, and J. Xu, “Air vortices and nano-vibration of aerostatic bearings,” Tribology Letters, vol. 42, no. 2, pp. 179–183, 2011.

[17] S. Yoshimoto, M. Yamamoto, and K. Toda, “Numerical calculations of pressure distribution in the bearing clearance of circular aerostatic thrust bearings with a single air supply inlet,” Journal of Tribology, vol. 129, no. 2, pp. 384–390, 2007.

[18] H. Yu, G. Wang, and Z. Li, “Study on static characteristics of aerostatic thrust bearing with a single continuous slot-restrictor,” Lubrication Engineering, vol. 44, no. 001, pp. 70–75, 2019.