Analysis of the dynamic characteristics of downsized aerostatic thrust bearings with different pressure equalizing grooves at high or ultra-high speed

Ruzhong Yan¹,² and Haojie Zhang¹,²

Abstract
This study adopts the DMT (dynamic mesh technology) and UDF (user defined functions) co-simulation method to study the dynamic characteristics of aerostatic thrust bearings with equalizing grooves and compare with the bearing without equalizing groove under high speed or ultra high speed for the first time. The effects of air film thickness, supply pressure, rotation speed, perturbation amplitude, perturbation frequency, and cross section of the groove on performance characteristics of aerostatic thrust bearing are thoroughly investigated. The results show that the dynamic stiffness and damping coefficient of the bearing with triangular or trapezoidal groove have obvious advantages by comparing with that of the bearing without groove or with rectangular groove for the most range of air film thickness, supply pressure, rotation speed, perturbation amplitude, especially in the case of high frequency, which may be due to the superposition of secondary throttling effect and air compressible effect. While the growth range of dynamic stiffness decreases in the case of high or ultra-high rotation speed, which may be because the Bernoulli effect started to appear. The perturbation amplitude only has little influence on the dynamic characteristic when it is small, but with the increase of perturbation amplitude, the influence becomes more obvious and complex, especially for downsized aerostatic bearing.

Keywords
Aerostatic thrust bearing, dynamic characteristics, dynamic mesh technology, high speed, pressure equalizing groove

Date received: 16 November 2020; accepted: 27 April 2021

Handling Editor: James Baldwin

Introduction
Aerostatic bearings are one of the most important parts of precise and ultra-precise machinery, which are widely used in aerospace, biomedicine, high-precision machine tools and measuring instruments. The aerostatic bearings have the advantages of high precision, low friction and no pollution, but at the same time, due to the compressibility of the air film, they have low stiffness, and low bearing capacity in steady operation, moreover, the low stability under micro-perturbation when subject to dynamic perturbation forces. The traditional evaluation of aerostatic bearing is mostly based on the static characteristics or the dynamic characteristics of stationary state and without pressure equalizing groove. However, with the rapid development of ultra-precision and ultra-high-speed machines, the research on the dynamic characteristics of aerostatic bearing under high speed or...
ultra high speed becomes more and more significant in the evaluation of its working performance.

Accordingly, some researchers have studied the dynamic characteristics. The dynamic stiffness is much more sensitive to the perturbation frequency rather than the nanoscale perturbation amplitude by simulations and experiments. The static and dynamic characteristics of aerostatic bearings with numerous small feed holes less than 0.05 mm diameter by using the finite difference method and reported that a higher maximum stiffness could be obtained by reducing the diameter of feed holes and the damping coefficient increased remarkably with decreasing bearing clearance but was insensitive to the number of feed holes. Chen et al. theoretically analyzed the nonlinear frequency dependence of these dynamic characteristics which is due to compressibility of the air. The difference of dynamic coefficients between coupled and uncoupled perturbation Reynolds equation is more prominent when the perturbation frequency ratio and rotating speed increase by solving the coupled perturbation Reynolds equation. Deb reported that stiffness and damping characteristics provide the limitations of stability for the bearing which lead to the instability and critical frequency which are obtained along with inertia number. A comparison between a bearing with orifices and bearings either with a groove or made from porous material is carried out using frequency response measurements. The results show that the bearing with orifices is better damped than the other types for the highest load capacity. Pneumatic hammer instability tends to occur at low perturbation frequencies at small orifice diameters (<0.25 mm), large gap heights (≥20 μm) and large supply pressures. Nishio et al. confirmed that aerostatic thrust bearings with small feed-holes could have larger stiffness and damping coefficient than bearings with compound restrictors. In order to suppress the micro-perturbation, the air supply pressure should be kept as small as possible, while the small air pocket diameter and orifice diameter are also needed. The impulse response fluctuation amplitudes of the X-shaped grooved aerostatic bearing decreases with groove widths and depths. Cui et al. presented that the stiffness and stability of aerostatic circular pad bearings are influenced significantly by geometrical and material parameters. The depths of the circumferential PEGs (pressure-equalizing grooves) have great impacts on the load performance. Compared with opening the circumferential PEGs, opening the axial PEGs is more helpful to improve the load capacity, even opening only one or two axial PEGs and the dynamic stiffness increases with the increase of frequency, and the equivalent damping coefficient decreases with the increase of frequency. Belforte et al. reported that the load carrying capacity and the stiffness of the aerostatic thrust bearing with groove increases significantly, especially at low gas film thickness, where the air flow only has a slight increase. Maamari et al. proposed an orifice based aerostatic bearing with a compliant back plate to improve performance, the results show that an increase by a factor of three in stiffness compared to conventional bearing while ensuring a positive damping. Qiao et al. concluded that the trapezoidal section has the maximum static bearing capacity and static stiffness, while air consumption rate of unit load is the lowest. The adoption of the pressure-equalizing grooves can substantially improve the load capacity and static stiffness of the bearing and make the bearing maintain a uniform stress, which enhances operating accuracy and life of the bearing and the effects of porous bearing and the form of restrictor on the change law of static stiffness and damping coefficient. Chen et al. revealed from the simulations that a better static performance can be obtained based on the proper value of axial groove length. The method combining traditional engineering simplification algorithm and CFD can analyze the static and dynamic performances of the complex structure bearing with high accuracy and low time cost.

In conclusion, there is almost no research work that has been done on the dynamic characteristics aerostatic thrust bearing with pressure equalizing groove at high-speed and ultra high speed state. Therefore, to address the gap in the literature, the steady-state Reynolds equation and perturbation Reynolds equation are obtained, based on linear perturbation method for aerostatic thrust bearing, the perturbation Reynolds equations are solved to obtain the dynamic stiffness and damping coefficient of the aerostatic thrust bearing. CFD simulation models are built up to investigate the dynamic characteristics of the aerostatic thrust bearing with pressure equalizing groove at high speed or ultra-high speed state. The effects of air film thickness, supply pressure, perturbation frequency, perturbation amplitude, rotation speed, and cross section of the groove on the dynamic characteristics of the aerostatic thrust bearings with pressure equalizing groove are analyzed in this research.

### Aerostatic thrust bearing with/without annular pressure equalizing groove

The illustration of aerostatic thrust bearing with pressure equalizing groove is shown in Figure 1(a). There are four orifice restrictors distributing uniformly in circumference of the bearing, cylindrical, air chamber exists at the outlet of each restrictor. The aerostatic bearing without pressure equalizing groove has exactly the same dimensions as the bearing with pressure equalizing groove. The cross section shape of the thrust bearing are shown in Figure 1(b), B and C represent enlarged views of the restrictor and equalizing groove respectively. The three different sections of annular pressure equalizing grooves
are illustrated in Figure 1 C, C, and C, which are all 0.05 mm in width and 0.1 mm in depth. The principal dimensions of aerostatic thrust bearings studied in this paper are shown in Table 1.

### Table 1. Principal dimensions of aerostatic thrust bearings.

|                        | With groove | Without groove |
|------------------------|-------------|----------------|
| Inner radius: $r_0$    | 8.9 mm      | 8.9 mm         |
| Outer radius: $r_1$    | 24.7 mm     | 24.7 mm        |
| Restrictor position diameter: $r_2$ | 16.5 mm | 16.5 mm |
| Orifice diameter: $d_o$ | 0.1 mm      | 0.1 mm         |
| Orifice number: $n_o$  | 4           | 4              |
| Chamber diameter: $h_o$ | 1 mm        | 1 mm           |
| Chamber depth: $d_i$   | 0.05 mm     | 0.05 mm        |
| Groove cross-section   | Triangle    | Trapezoid      | Rectangle |
| Groove depth: $h_g$    | 0.05 mm     | 0.05 mm        | 0.05 mm   |
| Groove width: $d_g$    | 0.1 mm      | 0.1 mm         | 0.1 mm    |
| Trapezoid angle: $\alpha$ | $60^\circ$ | -              |

Numerical model

**Governing equations**

The Reynolds equations for aerostatic thrust bearing are obtained based on the following assumptions:

(a) Since the gas film thickness is much smaller than the length and width of the gas film, the gas pressure along the gas film thickness direction is a constant.

(b) The airflow is isothermal in the aerostatic thrust bearings.

(c) The air viscosity in the bearing gas film is sticky degree of constant.

(d) Based on the above assumptions, the Reynolds equation can be deduced as:

$$\frac{\partial}{\partial x} \left( h^3 \rho \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \rho \frac{\partial P}{\partial z} \right) = 12 \mu \frac{\partial (Ph)}{\partial t} + 6 \mu \frac{\partial}{\partial x} (Phu),$$

where $P$ is the pressure distribution, $h$ is the air film thickness, $\mu$ is the viscosity of air, $x$ and $y$ are the rectangular coordinates, $t$ represents the flow time, $u$ is the rotation speed.
In the case of small perturbation of the aerostatic bearing, the pressure perturbation equation and the film thickness perturbation equation are respectively expressed as equation (2) and equation (3):

\[ P = P_0 + P_1, \]
\[ h = h_0 + h_1, \quad (3) \]

where 0 and 1 represent the steady state and perturbation state respectively, \( P_0 \) is the pressure in steady state, \( P_1 \) is the dynamic pressure in perturbation at a specified time, \( h_0 \) is the film thickness in steady state, \( h_1 \) is the displacement values in perturbation at a specified time.

Combining the above-mentioned equations, steady state Reynolds equation is obtained as:

\[ \frac{\partial}{\partial x} \left( \frac{P_0 h_0^3}{\mu} \frac{\partial P_0}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{P_0 h_0^3}{\mu} \frac{\partial P_0}{\partial y} \right) = 6\mu \frac{\partial}{\partial x} (Ph_d), \quad (4) \]

and perturbation Reynolds equation is given by:

\[ (h_0 + h_1)^2 \left[ (3P_0 + 3P_1 + h_0 + h_1) \left( \frac{\partial P_1}{\partial x} \frac{\partial (P_1 + h_1)}{\partial x} + \frac{\partial P_1}{\partial y} \frac{\partial (P_1 + h_1)}{\partial y} \right) 
\quad + (h_0 + h_1)(P_0 + P_1) \left( \frac{\partial^2 P_1}{\partial y^2} + \frac{\partial^2 P_1}{\partial x^2} \right) \right] 
\quad = 6\mu \left( \frac{2}{\partial t} \frac{\partial P_1}{\partial x} + u \frac{\partial P_1}{\partial x} \right) (h_0 + h_1) + \left( \frac{2}{\partial t} \frac{\partial h_1}{\partial x} + u \frac{\partial h_1}{\partial x} \right) (P_0 + P_1). \quad (5) \]

The film thickness \( h_1 \) and the pressure \( P_1 \) are calculated by:

\[ h_1 = h_d e^{j\omega t}, \quad (6) \]
\[ P_1 = P_d e^{j\omega t}, \quad (7) \]

where \( \omega \) is the perturbation angular speed and denotes as \( \omega = 2\pi f \), \( h_d \) is the perturbation amplitude of displacement, \( P_d \) is the perturbation amplitude of pressure.

From the equations (6) and (7), the following formulas are derived:

\[ \frac{\partial h_1}{\partial t} = h_d \omega e^{j\omega t}, \quad (8) \]
\[ \frac{\partial P_1}{\partial t} = P_d \omega e^{j\omega t}, \quad (9) \]

Substituting the equations (6)–(9) into the perturbation Reynolds equation and combining the steady state Reynolds equation to calculate \( P_1, P_0, \) and \( P_d \), then the change of bearing capacity load \( W \) are defined as:

\[ W = \int \int P_1 dx dy, \quad (10) \]

\[ W = -K_d h_1 - D \frac{\partial h_1}{\partial t} = -K_d h_d e^{j\omega t} - D h_d \omega e^{j\omega t}, \quad (11) \]

Hence, the dynamic stiffness \( K_d \) and dynamic damping \( D \) can be calculated respectively as:

\[ K_d = -Re(W), \quad (13) \]
\[ D = -Im(W). \quad (14) \]

**Boundary conditions and mesh model**

The dynamic characteristics of the aerostatic thrust bearing is numerically studied by adopting the CFD software ANSYS-Fluent which adopts the combination of modified fluid mechanics formulas and experimental models, the whole flow process of complex airflow field can be simulated accurately. It is vital to solve the pressure distribution of aerostatic bearing with groove structure under high speed and ultra high speed. Due to the periodic and boundary conditions of the 3D air flow model, one fourth of the air flow field is considered in the computations to reduce the cost. The boundary conditions of the 3D air flow model with pressure equalizing groove are depicted in Figure 2, pressure inlet and pressure outlet are specified at the orifice supply, the inner and outer peripheries of the bearing respectively. Periodic and adiabatic boundary conditions are prescribed at the periodic walls, rotation wall is the boundary condition of rotation speed. The rest other walls of the 3D air flow model are assumed to be adiabatic, stationary and rigid walls. For the analysis of dynamic characteristics, a sinusoidal boundary condition is applied to the rotation and perturbation wall by user-defined-functions, and the dynamic mesh modeling approach will be employed in this study. The 3D air flow model with pressure equalizing groove, illustrated in Figure 2, takes the same boundary conditions as the bearing without pressure-equalizing groove. The grid model divided by ICEM CFD is shown in Figure 3.

**Verification of model**

The accuracy of the DMT and UDF co-simulation method needs to be verified. Therefore, the dynamic
stiffness and dynamic damping of the aerostatic thrust bearing with orifice restrictor are analyzed to prove the accuracy of the DMT and UDF co-simulation method. The basic dimensions and operating parameters of the verification bearing are shown in Table 2. Based on the principal parameters, the comparison between the co-simulation method and reported work are carried out which are shown in Figure 4, it can be observed that both dynamic stiffness and dynamic damping of the simulation are very close to the reported results, which verify the accuracy of the DMT and UDF co-simulation method.

**Numerical simulation**

Figure 5 shows the total pressure disturbance vary with rotation time of the bearing with triangular pressure equalizing groove at film thickness 12 μm, perturbation amplitude 0.4 μm, supply pressure 0.6 MPa, and rotation speed 150,000 r/min and perturbation frequency 5000 Hz. One-fourth of the pressure graphs are restored to the global pressure graphs by using the boundary condition of rotation period for the sake of conversion. It can be obtained from Figure 5 that the total pressure disturbance of bearing with triangular groove has periodic variation characteristics. Figure 6 shows that the curves of total pressure change over

| Table 2. Principal parameters of verification bearing and operation. |
|----------------|-----------------|-----------------|----------------|----------------|----------------|----------------|
| Description    | Inner radius    | Outer radius    | Mean radius     | Orifice radius | Film thickness | Perturbation amplitude | Supply pressure |
| Number          | 25 mm           | 50 mm           | 37.5 mm         | 0.2 mm         | 25 μm          | 0.25 μm             | 0.4 MPa        |

**Figure 2.** The computational model with pressure equalizing groove and boundary conditions.

**Figure 3.** The grid model with pressure equalizing groove.

**Figure 4.** Verification of model: (a) comparative dynamic stiffness between the reported work and simulation and (b) comparative dynamic damping between the reported work and simulation.
rotation time which approximate to the harmonic function and consistent with the formulas derived above. Moreover, the amplitude and phase of the total pressure are remarkably affected by the frequency, and the total pressure increases with the frequency increase, which may be due to a combination of compressible effects of air, Bernoulli effects, and secondary throttling effects of the equalizing groove.

**Figure 5.** The pressure distribution of the bearing with the triangular groove in a period time: (a) $t = 0$ or $T/2$, (b) $t = T/4$, and (c) $t = 3T/4$.

**Figure 6.** The total pressure at different perturbation frequency under 150,000 r/min.
Figure 7 shows the total pressure disturbance of the bearing with different types of grooves at film thickness 12 μm, perturbation amplitude 0.4 μm, supply pressure 0.8 MPa, rotation speed 20,000 r/min and perturbation frequency 100 Hz. The results show that the pressure of the bearing with triangular section groove is the highest, followed by trapezoidal section groove, and rectangular section groove is the smallest. Figure 8 shows the total pressure of the bearing with different types of grooves at time period. Although the pressure at the equilibrium position of the triangular groove bearing is greater than that of the trapezoidal groove, the pressure amplitude of the trapezoidal section groove is larger and its phase is smaller, so it can be inferred that the dynamic stiffness of the trapezoidal section groove is greater than that of the triangle section groove which is verified by Figure 8(b).

Simulation results and discussions

The influence of film thickness

The simulation results about the influence of the air film thickness on the dynamic stiffness and damping coefficient for the aerostatic thrust bearing with or without groove at air supply pressure 0.6 MPa, rotation speed 20,000 r/min, and perturbation amplitude 0.4 μm are shown in Figures 9 and 10. In Figure 9(a), aerostatic bearings with and without pressure equalizing groove of different cross section show similar trends. The dynamic stiffness first remains almost constant at the frequency between 0 Hz and 1000 Hz, then decreases until 3000 Hz, and finally increases with the increase of frequency. There is a rapid ascent in the dynamic stiffness of the bearings type C₁ (the bearing with the triangular cross section groove) and type C₂ (the bearing with the trapezoidal cross section groove) between 3000 Hz and 10,000 Hz. The value of dynamic stiffness in this range about the bearings type B (the bearing without groove) and type C₃ (the bearing with the rectangular cross section groove) increase slowly. The main reason for this phenomenon may be the secondary throttling effect. The bearing type C₁ remains constant at low frequencies and continues to grow at high frequencies in Figure 9(b) to (d), while the other types of bearings first increase and then slightly decrease in the high frequency range in Figure 9(b) and (c). The
Figure 8. The total pressure of the bearing with different types of grooves at time period.

Figure 9. Dynamic stiffness under different gas film thickness: (a) dynamic stiffness at $h = 8\,\mu m$, (b) dynamic stiffness at $h = 12\,\mu m$, (c) dynamic stiffness at $h = 16\,\mu m$, and (d) dynamic stiffness at $h = 20\,\mu m$. 
The dynamic stiffness of bearing type C has an obvious advantage in both high and low frequency bands in Figure 9(c) and (d). Meanwhile, by comparing the four figures in Figure 9, the dynamic stiffness of the aerostatic thrust bearing of the same type increases with the decrease of air film thickness when the perturbation frequency does not change. Figure 10 shows the damping coefficient under different gas film thickness. It can be seen that the damping coefficient of different types of aerostatic bearing decreases with the increase of perturbation frequency in the same film thickness, and that of the same type of aerostatic bearing of fixed frequency increases as the film thickness decrease. The damping coefficient drops most rapidly in the range of 0 Hz–50 Hz, flattens out between 50 Hz and 500 Hz and then gradually approaches zero, and that of the bearing type C of different thickness is always the highest of the four types of bearings.

**The influence of supply pressure**

The influence of the supply pressure on the dynamic stiffness and damping coefficient for the aerostatic thrust bearing with or without groove are studied at the gas film thickness 12 μm, rotation speed 20,000 r/min and perturbation amplitude 0.4 μm. Figure 11(a) to (c) show the variation curve of dynamic stiffness with perturbation frequency at supply pressure of 0.4, 0.8, and 1 MPa, respectively. The figure about supply pressure 0.6 MPa is exactly the same as Figure 9(b), similarly, the damping coefficient at supply pressure 0.6 MPa is identical to Figure 10(b). The dynamic stiffnesses of four different types of aerostatic bearing change in the same variation trend in Figures 11(a) and 9(b), which increases with the increase of frequency at first and decreases slightly with the increase of frequency after 5000 Hz. The dynamic stiffnesses of bearings of type C1, C2, and C3 all decrease slowly with the increase of perturbation frequency, and then increase with the increase of frequency in Figure 11(b) and (c). For type B bearing, with the increase of frequency, that is basically unchanged in the first, then increases, and then begins to decrease after 5000 Hz in Figure 11(b) and (c). By comparing the Figures 11(a) to (c) and 9 (b), it can be found that the dynamic stiffness the bearing of type C1 is always highest under 0.4 MPa, however, with the increase of supply pressure to 0.6, 0.8, and 1 MPa, the dynamic stiffness of type C2 gradually is the highest in all frequency band. Moreover, the dynamic stiffnesses of type C1 and type C2 are always greater than those of type C3 and type B in the case of high supply pressure and high perturbation frequency, because of the superposition of secondary throttling effect and air compressible effect. For the same type of bearing and when the frequency does not change, the dynamic stiffness increases with the increase of supply pressure. As can be seen from Figure 12, the damping coefficient of type C1 bearing and type C2 bearing is only slightly different, and both greater than that of type C3 bearing and type B bearing. The damping coefficient of the same type bearings increases with the increase of supply pressure when the frequency is constant, and that of the different types of bearings decreases with the increase of perturbation frequency for the same supply pressure. It can be concluded that type C1 bearing and type C2 bearing have better stability under high supply pressure and high perturbation frequency.

**The influence of rotation speed**

The influence of the rotation speed on the dynamic stiffness and damping coefficient for the aerostatic thrust bearings with or without groove are investigated at the gas film thickness 12 μm, supply pressure 0.6 MPa and perturbation amplitude 0.4 μm, the rotation speeds of the rotational wall boundary condition are 100,000, 150,000, and 200,000 r/min respectively, which are shown in Figure 13(a) to (c). As for the dynamic stiffness and damping coefficient under the boundary condition when the rotational wall speed is 20,000 r/min, which are shown in Figures 9(b) and 10(b), respectively. It can be seen that the dynamic stiffness of type C1 bearing is always lowest in the low frequency range at both high and low speeds in Figures 13(a) and (c) and 6(b), except at 150,000 r/min rotation speed. The dynamic stiffness of different types of bearings increase with the increase of frequency when the rotation speed is below 200,000 r/min, however these dynamic stiffness increase as the frequency increase at the initial stage and then decrease when rotation speed is 200,000 r/min and the frequency exceeds 8000 Hz in Figure 13(c). The bearing of type C1 has the maximum dynamic damping and the best stability at low speeds, however, with the increase of speed, the bearing of type C2 replaces the type C1 as the bearing with the best damping coefficient by comparing Figures 14(a) to (c) and 10(b). In conclusion, by comparing bearings of the same type with different rotation speed, the dynamic stiffness and damping coefficient only increase slightly with the increase of rotation speed. For bearings of the different types with same speed, the dynamic stiffness generally increase with the increase of perturbation frequency, while the dynamic stiffness will decline when the rotation speed is 20,000 r/min and the frequency exceeds 8000 Hz, the amount of growth at low speed is significantly higher than that at high speed, those which mainly because the Bernoulli effect cancels out a greater amount of the effect of high speed gas flow and gas film compression effect. In this case, the damping coefficient of different type bearings are decrease with the frequency increase when the rotation speed is constant, that of the same type bearing increases first and
then decreases as the speed increases under the frequency is fixed. The damping coefficient of type C2 bearing is higher and more stable by comparing with other types of bearings.

The influence of perturbation amplitude

The influence of the perturbation amplitude on the dynamic stiffness and damping coefficient are shown in Figures 15 and 16, four different amplitudes of 0.2, 0.3, 0.4, and 0.5 µm are adopted respectively. The supply pressure is 0.6 MPa, the rotation speed is 20000 r/min and the film thickness is 12 µm. The curves of dynamic stiffness and damping coefficient at the amplitude of 0.4 µm are the same as those in Figures 9(b) and 10(b). It can be found that the dynamic stiffness of bearings type C1 and Type C2 with secondary throttling effect first increases with the increase of frequency, then decreases, and finally presents an upward trend in high frequency band and the dynamic stiffness of type C1 is higher than that of type C2 in the high frequency stage by comparing Figure 15(a) and (b). In the case of high perturbation amplitude, Figures 15(c) and 9(b) show similar variation rules, the dynamic stiffness of bearing with pressure equalizing groove increases with the increase of frequency, while the dynamic stiffness of bearing without micro-groove structure decreases slightly with the increase of frequency when the frequency exceeds 8000 Hz. Comparing the dynamic characteristics at a smaller perturbation amplitude with that at a larger perturbation amplitude, the influence of smaller perturbation amplitude on the dynamic characteristics is little, but as the increase of perturbation amplitude, the influence becomes more obvious and complex. It can be concluded that the damping coefficient of different types of bearing decreases with the increase of frequency under same perturbation amplitude by comparing Figures 16 and 10(b), the change

Figure 10. Damping coefficient under different gas film thickness: (a) damping coefficient at \(h = 8\) µm, (b) damping coefficient at \(h = 12\) µm, (c) damping coefficient at \(h = 16\) µm, and (d) damping coefficient at \(h = 20\) µm.
Figure 11. Dynamic stiffness under different supply pressure: (a) dynamic stiffness at $P = 0.4 \text{ MPa}$, (b) dynamic stiffness at $P = 0.8 \text{ MPa}$, and (c) dynamic stiffness at $P = 1 \text{ MPa}$.

Figure 12. Damping coefficient under different supply pressure: (a) damping coefficient at $P = 0.4 \text{ MPa}$, (b) damping coefficient at $P = 0.8 \text{ MPa}$, and (c) damping coefficient at $P = 1 \text{ MPa}$.
Figure 13. Dynamic stiffness under different rotation speed: (a) dynamic stiffness $u = 100,000$ r/min, (b) dynamic stiffness at $u = 150,000$ r/min, and (c) dynamic stiffness at $u = 200,000$ r/min.

Figure 14. Damping coefficient under different rotation speed: (a) damping coefficient at $u = 100,000$ r/min, (b) damping coefficient at $u = 150,000$ r/min, and (c) damping coefficient at $u = 200,000$ r/min.
Figure 15. Dynamic stiffness under different perturbation amplitude: (a) dynamic stiffness at \( h_d = 0.2 \, \text{\textmu}m \), (b) dynamic stiffness at \( h_d = 0.3 \, \text{\textmu}m \), and (c) dynamic stiffness at \( h_d = 0.5 \, \text{\textmu}m \).

Figure 16. Damping coefficient under different perturbation amplitude: (a) damping coefficient at \( h_d = 0.2 \, \text{\textmu}m \), (b) damping coefficient at \( h_d = 0.3 \, \text{\textmu}m \), and (c) damping coefficient at \( h_d = 0.5 \, \text{\textmu}m \).
curve of damping coefficient of the same type of bearing under different amplitudes can be divided into two situations, one is that the damping coefficient of bearing with groove increases first and then decreases with the increase of perturbation amplitude; the other one is that the damping coefficient of bearing without micro-groove structure decreases with the increase of perturbation amplitude.

**Conclusions**

In this paper, the dynamic characteristics of aerostatic bearing with different cross-section pressure equalizing groove and the bearing without pressure equalizing groove at high speed and ultra-high speed are studied for the first time by means of DMT of CFD and UDF. The effects of the section shape of the pressure equalizing groove, perturbation amplitude, film thickness, perturbation frequency, supply pressure, and rotation speed on the dynamic characteristics of aerostatic thrust bearing are studied. Several conclusions can be drawn from this study which are listed below:

1. The DMT and UDF can be combined to solve the transient fluid dynamics equations and to study the dynamic characteristics of aerostatic bearing with micro-groove structure at high speed or ultra-high speed, which is extremely difficult to study under experimental conditions.

2. The variation law of dynamic stiffness is complex. In this study, the dynamic stiffness of the aerostatic bearing increases with the increase of frequency when only one of the conditions is changed, such as air film thickness, supply pressure, perturbation amplitude, groove section shape, perturbation frequency and rotation speed, except when the gas film thickness is changed only and the gas film thickness is 8 μm, and the air supply pressure is changed only and the air supply pressure is 0.8 MPa or 1 MPa.

3. The damping coefficient of the aerostatic bearing decreases with the increase of frequency when only one of the conditions, such as air film thickness, supply pressure, rotation speed and groove cross-section shape, is changed except the change of perturbation amplitude. When the frequency is fixed, the damping coefficient of bearings with pressure equalizing groove increases with the decrease of air film thickness and perturbation amplitude, or the increase of supply pressure and rotation speed.

4. For the most range of air film thickness, supply pressure, rotation speed, perturbation amplitude, the dynamic stiffness and damping coefficient of aerostatic thrust bearing with triangular equal-pressure groove or trapezoidal groove have obvious advantages by comparing with that of the bearing without groove or with rectangular groove, especially in high frequency band.

**Declaration of conflicting interests**

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

**Funding**

The author(s) received no financial support for the research, authorship, and/or publication of this article.

**ORCID iD**

Haojie Zhang https://orcid.org/0000-0003-4512-8721

**References**

1. Yu P, Chen X, Wang X, et al. Frequency-dependent nonlinear dynamic stiffness of aerostatic bearings subjected to external perturbations. Tribol Int 2015; 16: 1771–1777.

2. Miyatake M and Yoshimoto S. Numerical investigation of static and dynamic characteristics of aerostatic thrust bearings with small feed holes. Tribol Int 2010; 43: 1353–1359.

3. Chen X, Zhu J and Chen H. Dynamic characteristics of ultra-precision aerostatic bearings. Adv Manuf 2013; 1: 82–86.

4. Shi J, Cao H and Jin X. Investigation on the static and dynamic characteristics of 3-DOF aerostatic thrust bearings with orifice restrictor. Tribol Int 2019; 138: 435–449.

5. Deb RK and Khan IA. Numerical simulation of aerostatic bearing stiffness, damping and critical frequency properties using linear stability analysis. J Inst Eng (India) Series C 2020; 101: 571–578.

6. Charki A, Diop K, Champmartin S, et al. Numerical simulation and experimental study of thrust air bearings with multiple orifices. Int J Mech Sci 2013; 72: 28–38.

7. Bhat N, Kumar S, Tan W, et al. Performance of inherently compensated flat pad aerostatic bearings subject to dynamic perturbation forces. Precis Eng 2012; 36: 399–407.

8. Nishio U, Somaya K and Yoshimoto S. Numerical calculation and experimental verification of static and dynamic characteristics of aerostatic thrust bearings with small feedholes. Tribol Int 2011; 44: 1790–1795.

9. Li YF, Yin YH, Yang H, et al. Micro-vibration analysis and optimization of aerostatic bearing with pocketed orifice-type restrictor. J Appl Fluid Mech 2018; 4: 1115–1124.

10. Chen MF and Lin YT. Static behavior and dynamic stability analysis of grooved rectangular aerostatic thrust
bearings by modified resistance network method. *Tribol Int* 2002; 35: 329–338.

11. Cui H, Wang Y, Wang B, et al. Numerical simulation and experimental verification of the stiffness and stability of thrust pad aerostatic bearings. *Chin J Mech Eng* 2018; 31: 23.

12. Du J, Zhang G, Liu T, et al. Improvement on load performance of externally pressurized gas journal bearings by opening pressure-equalizing grooves. *Tribol Int* 2014; 73: 156–166.

13. Zhao X, Dong H, Fang Z, et al. Study on dynamic characteristics of aerostatic bearing with elastic equalizing pressure groove. *Shock Perturbation* 2018; 2018: 1–12.

14. Belforte G, Colombo F, Raparelli T, et al. Comparison between grooved and plane aerostatic thrust bearings: static performance. *Meccanica* 2011; 46: 547–555.

15. Maamari N, Krebs A, Weikert S, et al. Stability and dynamics of an orifice based aerostatic bearing with a compliant back plate. *Tribol Int* 2019; 138: 279–296.

16. Qiao Y, Luo R and Shi K. Analysis on the influence for the sectional shape of compound pressure-equalizing groove to the supporting and bearing characteristics of precision aerostatic bearing. *Appl Mech and Mater* 2018; 494–495: 598–601.

17. Yan R, Wang L and Wang S. Investigating the influences of pressure-equalizing grooves on characteristics of aerostatic bearings based on CFD. *Ind Lubr Tribol* 2019; 71: 853–860.

18. Cui H, Wang Y, Yue X, et al. Numerical analysis of the dynamic performance of aerostatic thrust bearings with different restrictors. *Proc Inst Mech Eng J J Eng Tribol* 2019; 233: 406–42318.

19. Chen X, Mills JK and Bao G. Static performance of the aerostatic journal bearing with grooves. *Proc Instit Mech Eng J J Eng Tribol* 2020; 234: 1114–1130.

20. Li Y, Yin Y and Cui H. An ESA-CFD combined method for dynamic analysis of the aerostatic journal bearing. *Lubr Sci* 2020; 32: 387–403.