Effect of environmental pressure on dynamic characteristics of aerodynamic gas bearing of boost turbo expander

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Abstract. Improvement of the environmental pressure of gas bearing can enhance the load capacity of dynamic gas bearing. This study analyzes the effect of environmental pressure on the dynamic characteristics of a bump foil gas aerodynamic bearing. Results show that an increase in the environmental pressure enhances the direct dynamic stiffness $K_{xx}$ and $K_{yy}$ and the direct dynamic damping $D_{xx}$ and $D_{yy}$ of gas film. The direct stiffness $K_{xx}$ and $K_{yy}$ and direct damping $D_{xx}$ and $D_{yy}$ of gas film increase with the increase of the environmental pressure of gas bearing. The load capacity of bearing is related to the direct stiffness of gas film. Therefore, the direct stiffness $K_{yy}$ can be increased by increasing the environmental pressure of gas bearing, consequently improving the load capacity of the bearing.

1. Introduction

A turbo expander is an equipment used to obtain low temperature and recycle energy. In a cryogenic turbo expander, the temperature of the inlet medium is low, the expansion ratio is high, and the specific enthalpy drop of the expansion medium is large. Generally, the rotor speed of the turbo expander is very high that it exceeds $10^4$ r/min [1-3], e.g., the rotation speed of the turbo expander reaches 196 kr/min in the literature [4]; a high-speed turbo expander with gas bearing has also been studied in literature [5], and the designed rotation speed is 300 kr/min with maximum rotation speed of 342 kr/min. The operation requirements of the equipment are difficult to achieve with oil-lubricated bearing under such high-speed motion. The advantages of gas bearing include light weight, high speed, absence of friction, non-pollution, and steady running even at high- and ultra-speed motions [6]. Hence, this bearing is widely utilized. Gas bearing also presents low gas-film stiffness, small bearing capacity, and limited applicability (suitable for small turbo expanders only) [7-9]. The literature [10] recommends replacing the fan brake of a turbo expander with a booster and supplying pressurized gas to the gas bearing to enhance the environmental pressure at both ends of the bearing. The results show that improving the environmental pressure of gas bearing can improve the load capacity of dynamic gas bearing. Because the rotation speed of hydrodynamic gas bearing is quite high, the highest speed can reach hundreds of thousands of revolutions per minute, the dynamic characteristics display a complicated nonlinear stochastic process. The barycenter of a rotor can move randomly with the change of load when the bearing is affected by external factors. Dynamic characteristic coefficients of gas film directly affect the dynamic behavior and stability of a rotor-bearing system [11, 12]. Jia and Peng et al. [13] studied the influences of different speeds and eccentricity ratios on the dynamic stability of spherical spiral groove hybrid gas bearing rotor system, and their results show that the influences of speeds and eccentricity ratios on the dynamic characteristics of gas film are significant. Yang and Li et al. [14] studied the effects of bearing number, perturbation frequency of the journal and pads, eccentricity ratios, preload, and length-to-diameter ratio of the bearing on the dynamic coefficients of...
aerodynamic tilting-pad journal bearing. Numerical results indicate that the dynamic stiffness and damping coefficients of tilting-pad gas bearing are closely related with these factors. Chen and Wang [15] analyzed the chaotic behaviors and vibration phenomenon of a flexible rotor with different mass and bearing number supported by ultra short aero-lubricated bearing (USAB) system. The results show that system exist chaotic motions over the specific ranges, and the maximum lyapunov exponent is positive as chaos occurred.

The dynamic stiffness and damping coefficients of gas film can be used to characterize dynamic characteristics of gas film. For the rotor supported by gas bearings, these coefficients can be used to calculate the critical speed and analyze the stability of the rotor-bearing system [13]. The present study uses the small perturbation method to solve the dynamic Reynolds equation and analyze the effects of environmental pressure on the dynamic properties of hydrodynamic gas bearing. The Newmark-β method is also adopted to analyze the influence of environmental pressure on the stability of the rotor system.

2. Effect of environmental pressure on dynamic properties of aerodynamic gas bearing

2.1. Mathematical model

Figure 1 shows the schematic of a bump foil gas aerodynamic bearing, in which φ represents the attitude angle of the bearing.

(1) Dynamic Reynolds equation

The dimensionless form of compressible gas dynamic Reynolds equation under isothermal condition is expressed as follows:

$$\frac{\partial}{\partial \theta} \left( P H^3 \frac{\partial P}{\partial \theta} \right) + \frac{\partial}{\partial \xi} \left( P H^3 \frac{\partial P}{\partial \xi} \right) = \Lambda \frac{\partial (P H)}{\partial \theta} + 2 \gamma A \frac{\partial (P H)}{\partial T}$$

(1)

The dimensionless parameters are

$$\theta = \lambda / R; \xi = z / R; P = p / p_a; H = h / c; \Lambda = \left( 6 \omega \mu R^2 \right) / (p_a c); T = \omega t; \gamma = \omega / \omega_s$$

(2)

where λ is the circumferential coordinate (m), z is the axial coordinate (m), and R is the bearing radius (m). In this equation, p is the gas-film pressure (Pa), and $P_a$ is the environmental pressure (Pa); h is the gas-film thickness (m), $\omega$ is the angular velocity of the journal (rad/s), $\omega_s$ is whirl angular velocity of the journal, $\gamma$ is the dimensionless whirling frequency, $t$ is the time (s), $P$ is the dimensionless gas-film pressure, $H$ is the dimensionless gas-film thickness, and $c$ is the bearing radial gap (m), which is calculated as $c = R_1 - R_2$; $T$ is the dimensionless time, $\xi$ is the dimensionless axial coordinate, $\theta$ is the dimensionless circumferential coordinate, and $A$ is the bearing number.

(2) Dimensionless gas-film thickness equation
Considering the specific conditions of bump foil and top foil, the following assumptions are made:

1) by using a single-bump foil, the stiffness of the bearing surface is well distributed with a fixed value;
2) the top foil with the whole displacement of bump foil is only considered between the two adjacent wave crests; and 3) the deformation caused by load only depends on the point load.

Define that the foil stiffness is $k_b$, the damping is $c_b$, the relationship between pressure of gas-film and foil deformation is:

$$p = k_b u + c_b \frac{\partial u}{\partial t}$$

where $k_b = \frac{E_b t_b L}{2i s^2 (1 - v_s^2)}$, $c_b = \eta k_s / \omega$, $E_b$ is the bump foil elasticity modulus (N/m$^2$), $t_b$ is the bump foil thickness (m), $i$ is the bump foil radius (m), $v_s$ is the poisson’s coefficient of bump foil, $u$ is the radial deformation displacement of bump foil (m).

The dimensionless form is: $P = K_b U + C_b \gamma \frac{\partial U}{\partial T}$

The dimensionless parameters are

$$K_b = \frac{k_b C}{P_a}, C_b = \frac{C_b \omega c}{P_a} = \eta K_b, U = \frac{u}{c}$$

The corresponding expression of dimensionless gas-film thickness is

$$H = 1 + \varepsilon \cos \theta + U$$

where $\varepsilon$ is the eccentricity ratio, which is calculated as $\varepsilon = e / c$; $e$ is the eccentric distance (m), $c$ is the bearing radial gap (m), $K_b$ is the dimensionless stiffness of bump foil, $C_b$ is the dimensionless damping of bump foil, $U$ is the dimensionless radial deformation displacement of bump foil.

3) Dynamic stiffness and damping coefficient of gas film

If the eccentricity ratio is provided, the distribution of static pressure of gas bearing can be figured out, and the force acted on the journal can be calculated using an integral. On the basis of the coordinates shown in Figure 1, the corresponding force can be expressed as

$$\left\{ F_x, F_y \right\} = \int_{-\pi}^{\pi} \int_0^{2\pi} p \left\{ \sin \theta, \cos \theta \right\} r d\theta dz$$

when the journal is disturbed by small displacement and small velocity, the force acted on the journal can be expressed by the Taylor series expansion:

$$\vec{F} = \vec{F}_0 + \frac{\partial \vec{F}}{\partial x} \left|_{x_0} \right. \Delta x + \frac{\partial \vec{F}}{\partial y} \left|_{y_0} \right. \Delta y + \frac{\partial \vec{F}}{\partial \dot{x}} \left|_{\dot{x}_0} \right. \Delta \dot{x} + \frac{\partial \vec{F}}{\partial \dot{y}} \left|_{\dot{y}_0} \right. \Delta \dot{y} + O\left( \Delta x^2, \Delta y^2, \Delta \dot{x}^2, \Delta \dot{y}^2 \right)$$

The subscript “0” shows the equilibrium position, and the vector force $F$ is

$$\bar{F} = \left\{ F_x, F_y \right\}$$

Given that $\Delta x$, $\Delta y$, $\Delta \dot{x}$, and $\Delta \dot{y}$ are all close to zero, the corresponding second-order approximation reminder $O\left( \Delta x^2, \Delta y^2, \Delta \dot{x}^2, \Delta \dot{y}^2 \right)$ can be ignored; thus, Formula (6) is linear. By combining Formulas (6) and (7), the stiffness and damping coefficients of foil gas bearing can be expressed as

$$\left\{ K_{xx}, K_{yy} \right\} = \frac{\partial \vec{F}}{\partial x} \left|_{x_0} \right., \left. \left\{ K_{xy} \right\} = \frac{\partial \vec{F}}{\partial y} \right|_{y_0},$$

$$\left\{ D_{xx}, D_{yy} \right\} = \frac{\partial \dot{\vec{F}}}{\partial x} \left|_{x_0} \right., \left. \left\{ D_{xy} \right\} = \frac{\partial \dot{\vec{F}}}{\partial y} \right|_{y_0}$$

To explain the two formulas, the stiffness and damping coefficients can be expressed as
(4) Boundary conditions

In the region of gas-film, mesh is divided with difference step $\Delta \theta$ and $\Delta \xi$ when using finite difference method to solve the pressure distribution of dynamic gas bearing. By dividing the $m$ grid in the direction of $\theta$ ($0 \sim 2\pi$), the step length of each grid is $\Delta \theta=2\pi/m$; by dividing the $n$ grid in the direction of $\xi$ (axial direction), the step length of each grid is $\Delta \xi=1/n$. Figure 2 shows the mesh generation that define the gas-film area of dynamic gas bearing.

Figure 2. Mesh generation diagram of gas-film area of dynamic gas bearing

The environmental boundary conditions are

$$
P_x(i, j) = P_x(m, j) = P_x(i,1) = 0 \quad (i = 1: m, j = 1: n)$$

$$
P_y(i, j) = P_x(m, j) = P_y(i,1) = 0 \quad (i = 1: m, j = 1: n)$$

$$
P_z(i, j) = P_z(m, j) = P_z(i,1) = 0 \quad (i = 1: m, j = 1: n)$$

$$
P_v(i, j) = P_v(m, j) = P_v(i,1) = 0 \quad (i = 1: m, j = 1: n)$$

The symmetrical boundary conditions are
\[
\begin{align*}
P_i (i, n-1) &= P_i (i, n+1) \quad (i = 1 : m) \\
P_i (i, n-1) &= P_i (i, n+1) \quad (i = 1 : m) \\
P_i (i, n-1) &= P_i (i, n+1) \quad (i = 1 : m) \\
P_i (i, n-1) &= P_i (i, n+1) \quad (i = 1 : m) \\
\frac{\partial P_i (i, j)}{\partial \xi} &= \frac{\partial P_i (i, j)}{\partial \xi} = 0 \quad (i = 1 : m, j = 1 : n) \\
\frac{\partial P_i (i, j)}{\partial \xi} &= \frac{\partial P_i (i, j)}{\partial \xi} = 0 \quad (i = 1 : m, j = 1 : n)
\end{align*}
\]

2.2. Model parameters

The bearing structure and relevant parameters of lubricating gas are summarized in Table 1.

| Name                                      | Value            |
|-------------------------------------------|------------------|
| Bearing radius \( R \) (m)               | \( 50 \times 10^{-3} \) |
| Bearing width \( L \) (m)                | \( 75 \times 10^{-3} \) |
| Radius gas \( c \) (m)                   | \( 100 \times 10^{-6} \) |
| Top foil thickness \( t_p \) (m)         | \( 101.6 \times 10^{-6} \) |
| Bump foil thickness \( t_b \) (m)        | \( 76.2 \times 10^{-6} \) |
| Bump foil cycle length \( s \) (m)       | \( 4.064 \times 10^{-3} \) |
| Bump foil diameter \( 2l \) (m)          | \( 3.434 \times 10^{-3} \) |
| Bump foil elasticity modulus \( E_b \) (N/m2) | \( 207 \times 10^9 \) |
| Poisson’s coefficient of bump foil \( \nu_b \) | 0.3 |
| Dynamic viscosity of lubricating gas \( \mu \) (Pa·s) | \( 1.932 \times 10^{-5} \) |
| Friction coefficient of foil bearings \( \eta \) | 0.25 |

2.3. Results and discussion

(1) The effects of rotation speed on the dynamic stiffness and dynamic damping of bearing, when the slenderness ratio \( L/D \) is 0.75, the eccentricity ratio is 0.6, and the whirling frequency is 0.5, is explained below.

Figure 3 displays the relationship curve of dynamic stiffness and dynamic damping of bump foil gas bearing with rotation speed when the slenderness ratio \( L/D \) is 0.75, the eccentricity ratio is 0.6, and the whirling frequency is 0.5. As shown in the figure, the direct stiffness \( K_{xx} \) and \( K_{yy} \) of gas film
increase with the increase of rotation speed. With the increase of rotation speed, the pressure of gas film increase and the direct stiffness of gas film also increase. However, the cross-coupling stiffness $K_{xy}$ and $K_{yx}$ change relatively slow. The direct damping $D_{xx}$ and $D_{yy}$ decrease with the increase of rotation speed, whereas the cross-coupling damping $D_{xy}$ and $D_{yx}$ are close to 0 with the increase of rotation speed.

(2) The effects of whirling frequency on the dynamic stiffness and dynamic damping of bearing, when the slenderness ratio $L/D$ is 0.75, the eccentricity ratio is 0.6, and the rotation speed is 100kr/min, is explained below.

Figure 4 displays the relationship curve of dynamic stiffness and dynamic damping of bump foil gas bearing with whirling frequency when the slenderness ratio $L/D$ is 0.75, the eccentricity ratio is 0.6, and the rotation speed is 100kr/min. As shown in the figure, the whirling frequency is closely related to the dynamic stiffness and dynamic damping of bump foil gas bearing. The direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase with the increase of whirling frequency, the cross-coupling stiffness $K_{xy}$ and $K_{yx}$ change relatively slow. The direct damping $D_{xx}$ and $D_{yy}$ decrease with the increase of whirling frequency, whereas the cross-coupling damping $D_{xy}$ and $D_{yx}$ are close to 0 with the increase of whirling frequency.

(3) The effect of environmental pressure on the dynamic stiffness and dynamic damping of bearing at different rotation speeds, when the slenderness ratio $L/D$ is 0.75, the eccentricity ratio is 0.6, and the whirling frequency is 0.5, is described as follows:

In this paper, we kept the bearing parameters unchanged to analyze the bearing performance under different speeds. The load capacity of bearing was different for the fixed structures under different eccentricity ratios and speeds. Bearing load wasn’t taken into account in this paper. The bearing structures need to be re-designed under different speeds if the load keeps unchanged.
Figure 5 illustrates the effect of environmental pressure on direct dynamic stiffness and direct dynamic damping at different rotation speeds. It is observed that the direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase with the increase of environmental pressure. When the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the rotation speed is 60 kr/min, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $7.48 \times 10^6$ N/m to $9.13 \times 10^6$ N/m and from $8.3 \times 10^6$ N/m to $1.44 \times 10^7$ N/m, respectively, indicating 22.1% and 49.4% increases, correspondingly. When the rotation speed is 100 kr/min, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $8.7 \times 10^6$ N/m to $1.09 \times 10^7$ N/m and from $9.13 \times 10^6$ N/m to $1.44 \times 10^7$ N/m, respectively, showing 25.29% and 57.7% increases, correspondingly. When the rotation speed is 200 kr/min, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $9.36 \times 10^6$ N/m to $1.3 \times 10^7$ N/m and from $9.8 \times 10^6$ N/m to $1.69 \times 10^7$ N/m, respectively, which recorded 38.89% and 72.44% increases, correspondingly. The cross-coupling stiffness $K_{xy}$ and $K_{yx}$ slightly decrease and increase, respectively, with the increase of environmental pressure. The direct damping $D_{xx}$ and $D_{yy}$ increase with the increase of environmental pressure. When the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the rotation speed is 60 kr/min, the direct damping $D_{xx}$ and $D_{yy}$ increase from $1.3 \times 10^3$ N·s/m to $2.56 \times 10^3$ N·s/m and from $8.8 \times 10^2$ N·s/m to $2.27 \times 10^3$ N·s/m, respectively, which showed increases 96.92% and 157.95%, correspondingly. When the rotation speed is 100 kr/min, the direct damping $D_{xx}$ and $D_{yy}$ increase from $7.73 \times 10^2$ N·s/m to $1.8 \times 10^3$ N·s/m and from $5.08 \times 10^2$ N·s/m to $1.41 \times 10^3$ N·s/m, respectively, recording increases of 132.86% and 177.56%, correspondingly. When the rotation speed is 200 kr/min, the direct damping $D_{xx}$ and $D_{yy}$ increase from $4.05 \times 10^2$ N·s/m to $9.67 \times 10^2$ N/m and from $2.36 \times 10^2$ N·s/m to $7.48 \times 10^2$ N·s/m, respectively, indicating increases of 138.77% and 216.95%, correspondingly. The cross-coupling damping $D_{xy}$ and $D_{yx}$ slightly increase and decrease, respectively, with the increase of environmental pressure. The higher the rotation speed, the greater effect of environmental pressure on the direct stiffness and direct damping is. Meanwhile, because the bearing has only load in the vertical direction, the increase of $K_{yy}$ tend to be more obvious.

(4) The effect of environmental pressure on the dynamic stiffness and dynamic damping of bearing at different eccentricity ratios, when the slenderness ratio $L/D$ is 0.75, the rotation speed is 100 kr/min, and the whirling frequency is 0.5, is explained below.
Figure 6. The relation curve of dynamic stiffness and dynamic damping of gas film with rotation speed eccentricity ratio

Figure 7. The relation curve of dynamic stiffness and dynamic damping with environmental pressure at different eccentricity ratio

Figure 6 shows the relationship curve of dynamic stiffness and damping of bump foil gas bearing with eccentricity ratio when the slenderness ratio L/D is 0.75, the rotation speed is 100 kr/min, and the whirling frequency is 0.5. Figure 7 illustrates the effect of environmental pressure on direct dynamic stiffness and direct dynamic damping at different eccentricity ratios. The direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase with the increase of eccentricity ratio. No obvious change is observed in the cross-coupling stiffness $K_{xy}$ with the increase of eccentricity ratio. The cross-coupling stiffness $K_{yx}$ decrease slightly with the increase of eccentricity ratio. The direct damping $D_{xx}$ and $D_{yy}$ of gas film increase with the increase of eccentricity ratio. No obvious change in the cross-coupling damping $D_{xx}$ and $D_{yy}$ with the increase of eccentricity ratio is observed. When the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the eccentricity ratio is 0.4, the direct stiffness $K_{xx}$ and $K_{yy}$ increase
from $7.46 \times 10^6$ N/m to $9.73 \times 10^6$ N/m and from $8.32 \times 10^6$ N/m to $1.28 \times 10^7$ N/m, respectively, which indicated increases of 30.43% and 53.85%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ increase from $6.06 \times 10^2$ N·s/m to $1.66 \times 10^3$ N·s/m and from $4.61 \times 10^2$ N·s/m to $1.46 \times 10^3$ N·s/m, respectively, increasing by 173.93% and 216.7%, correspondingly. When the eccentricity ratio is 0.6, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $8.7 \times 10^6$ N/m to $1.09 \times 10^7$ N/m and from $9.13 \times 10^6$ N/m to $1.44 \times 10^7$ N/m, respectively, showing increases of 25.29% and 57.72%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ increase from $7.73 \times 10^2$ N·s/m to $1.8 \times 10^3$ N·s/m and from $5.08 \times 10^2$ N·s/m to $1.41 \times 10^3$ N·s/m, respectively, which indicated increases of 132.86% and 177.56%, correspondingly. When the eccentricity ratio is 0.8, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $9.52 \times 10^6$ N/m to $1.08 \times 10^7$ N/m and from $9.23 \times 10^6$ N/m to $1.51 \times 10^7$ N/m, respectively, revealing increases of 13.45% and 63.6%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ increase from $1.25 \times 10^3$ N·s/m to $2.11 \times 10^3$ N·s/m and from $7.21 \times 10^2$ N·s/m to $1.5 \times 10^3$ N·s/m, expressing increases of 68.8% and 108.04%, correspondingly. As the eccentricity ratio increases, the influence of environmental pressure on the direct stiffness $K_{xx}$ and $K_{yy}$ increase and decrease, respectively. Furthermore, the influence of environmental pressure on the direct damping of gas film decrease with the increase of eccentricity ratio.

(5) The effect of environmental pressure on the dynamic stiffness and dynamic damping of bearing at different slenderness ratios (L/D), when the eccentricity ratio is 0.6, the rotation speed is 100 kr/min, and the whirling frequency is 0.5, is explained below.

![Figure 8. The relation curve of dynamic stiffness and dynamic damping of gas film with slenderness ratio](image)
Figure 8 shows the relationship curve of dynamic stiffness and dynamic damping of foil gas bearing with slenderness ratio when the eccentricity ratio is 0.6, the rotation speed is 100 kr/min, and the whirling frequency is 0.5. Figure 9 presents the effect of environmental pressure on the direct dynamic stiffness and direct dynamic damping of gas film at different slenderness ratios. The direct stiffness $K_{xx}$ and $K_{yy}$ increase with the increase of environmental pressure. No obvious change in the cross-coupling stiffness $K_{xy}$ and $K_{yx}$ with the increase of slenderness ratio has been observed. The direct damping $D_{xx}$ and $D_{yy}$ increase with the increase of slenderness ratio. The cross-coupling damping $D_{xy}$ and $D_{yx}$ slightly increase and decrease, respectively, with the increase of slenderness ratio. When the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the slenderness ratio is 0.65, the direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase from $7.36 \times 10^6$ N/m to $8.52 \times 10^6$ N/m and from $7.51 \times 10^6$ N/m to $1.18 \times 10^7$ N/m, respectively, showing increases of 15.76% and 57.1%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ of gas film increase from $6.4 \times 10^6$ N·s/m to $1.5 \times 10^7$ N·s/m and from $4.33 \times 10^6$ N·s/m to $1.15 \times 10^7$ N·s/m, respectively, which increased by 134.37% and 165.13%, correspondingly. When the slenderness ratio is 0.75, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $8.7 \times 10^6$ N/m to $1.09 \times 10^7$ N/m and from $9.13 \times 10^6$ N/m to $1.44 \times 10^7$ N/m, respectively, indicating increases of 25.29% and 57.72%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ of gas film increase from $7.73 \times 10^6$ N·s/m to $1.8 \times 10^7$ N·s/m and from $5.08 \times 10^6$ N·s/m to $1.41 \times 10^7$ N·s/m, respectively, increasing by 132.86% and 177.56%, correspondingly. When the slenderness ratio is 0.85, the direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase from $1.01 \times 10^7$ N/m to $1.28 \times 10^7$ N/m and from $1.08 \times 10^7$ N/m to $1.71 \times 10^7$ N/m, respectively, indicating increases of 26.73% and 58.33%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ of gas film increase from $9.22 \times 10^6$ N·s/m to $2.1 \times 10^7$ N·s/m and from $5.76 \times 10^6$ N·s/m to $1.68 \times 10^7$ N·s/m, respectively, increasing by 127.77% and 191.67%, correspondingly. When the slenderness ratio is 1, the direct stiffness $K_{xx}$ and $K_{yy}$ increase from $1.22 \times 10^7$ N/m to $1.55 \times 10^7$ N/m and from $1.32 \times 10^7$ N/m to $2.14 \times 10^7$ N/m, respectively, expressing increases of 27.05% and 62.12%, correspondingly. The direct damping $D_{xx}$ and $D_{yy}$ increase from $1.17 \times 10^7$ N·s/m to $2.58 \times 10^7$ N·s/m and from $6.74 \times 10^6$ N·s/m to $2.08 \times 10^7$ N·s/m, respectively, reflecting increases of 120.51% and 208.6%, correspondingly. As the slenderness ratio increases, no obvious change in the direct stiffness and direct damping $D_{xx}$ is observed, whereas, the influence of environmental pressure on the direct damping $D_{yy}$ becomes significant.

The direct stiffness $K_{xx}$ and $K_{yy}$ and direct damping $D_{xx}$ and $D_{yy}$ of gas film increase with the increase of the environmental pressure of gas bearing. The load capacity of bearing is related to the direct stiffness of gas film. Therefore, the direct stiffness $K_{yy}$ can be increased by increasing the environmental pressure of gas bearing, consequently improving the load capacity of the bearing.
3. Conclusion
In this work, the rotor system model of foil dynamic gas bearing is built, and the effects of environmental pressure on the dynamic characteristics of aerodynamic gas bearing are analyzed. The results show that the direct stiffness $K_{xx}$ and $K_{yy}$ of gas film increase with the increase of environmental pressure (when the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the rotation speed is 100 kr/min, at which $K_{xx}$ increases by 25.29%, whereas $K_{yy}$ increases by 57.72%). The cross-coupling stiffness $K_{xy}$ and $K_{yx}$ slightly decrease and increase, respectively, with the increase of environmental pressure. The direct damping $D_{xx}$ and $D_{yy}$ of gas film increase with the increase of environmental pressure (when the environmental pressure increases from 0.1 MPa to 0.5 MPa, and the rotation speed is 100 kr/min, at which $D_{xx}$ increases by 132.86%, whereas $K_{yy}$ increases by 177.56%). The direct stiffness $K_{xx}$ and $K_{yy}$ and direct damping $D_{xx}$ and $D_{yy}$ of gas film increase with the increase of the environmental pressure of gas bearing. The load capacity of bearing is related to the direct stiffness of gas film. Therefore, the direct stiffness $K_{yy}$ can be increased by increasing the environmental pressure of gas bearing, consequently improving the load capacity of the bearing.

References
[1] Niu L, Hou Y, Sun W, et al. The measurement of thermodynamic performance in cryogenic two phase turbo-expander[J]. Cryogenics 2015; 70:76-84.
[2] Li M, Liu X, Zhu R, et al. Rotor Dynamics Behavior of Tilting Pad Bearing Supported Turbo Expander Considering Temperature Gradient[J]. Journal of Computational and Nonlinear Dynamics 2016; 11(2): 021004.
[3] Jiang C B. Turbo-expander and its development[J]. Cryogenic Technology 2001; 5 : 1-9.
[4] Yang S J, Chen S T, Chen X Y, et al. Study on the coupling performance of a turboexpander compressor applied in cryogenic reverse Brayton air refrigerator[J]. Journal of Energy Conversion and Management 2016; 112:386-399.
[5] Chen R G, Hou Y, Chen C Z. Study of miniature high speed turboexpander with gas lubricated bearing[J]. Journal of Xi’an Jiaotong University 2010; 1(44):61-65.
[6] R. Dupont. On an isotropic and centrifugal force invariant layout of a conically shaped gas lubricated high-speed spiral-groove bearing[J]. Precision Engineering 2003; 27: 346-361.
[7] Hou Y, Yang S J, Chen X Y. Application of foil gas bearing in cryogenic turbo expander[J]. Journal of Harbin Engineering University 2015; 36(4):489-493.
[8] Kim D, Creary A, Chang S S, et al. Mesoscale Foil Gas Bearings for Palm-Sized urbomachinery: Design, Manufacturing, and Modeling[J]. Journal of Engineering for Gas Turbines and Power 2009; 131(4): 042502.
[9] Hou Y, Zhu Z H, Chen C Z. Comparative test on two kinds of new compliant foil bearing for small cryogenic turbo-expander[J]. Cryogenics 2004; 44(1):69-72.
[10] Li Y Y, Lei G, Sun Y, Wang L. Effect of environmental pressure enhanced by a booster on the load capacity of the aerodynamic gas bearing of a turbo expander[J]. Tribology International 2017; 105:77-84.
[11] Jia C H, Yang W, Qiu M. Numerical Simulated of Dynamic Stability of Aerodynamic Bearing Rotor System[J]. Journal of System Simulation 2014; 26(8):1763-1768.
[12] Jerry C.T. Su , K.N. Lie. Rotor dynamic instability analysis on hybrid air journal bearings[J]. Tribology International 2006; 39:238-248.
[13] Jia C H, Pang H J, Ma W S, et al. Dynamic stability prediction of spherical spiral groove hybrid gas bearings rotor system[J]. Journal of Tribology 2017; 139:021701-1-021701-12.
[14] Yang L H, Li H G, Yu L. Dynamic stiffness and damping coefficients of aerodynamic tilting pad journal bearings. Tribology International 2007; 40(9):1399-1410.
[15] Chen J H, Wang C C. Chaotic and dynamic analysis of a flexible rotor supported by ultra short aero-lubricated bearing system[J]. Journal of Applied Research and Technology 2015; 13:328-341.