Numerical analysis of the static performance of an annular aerostatic gas thrust bearing applied in the cryogenic turbo-expander of the EAST subsystem

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Abstract
The EAST superconducting tokamak, an advanced steady-state plasma physics experimental device, has been built at the Institute of Plasma Physics, Chinese Academy of Sciences. All the toroidal field magnets and poloidal field magnets, made of NbTi/Cu cable-in-conduit conductor, are cooled with forced flow supercritical helium at 3.8 K. The cryogenic system of EAST consists of a 2 kW/4 K helium refrigerator and a helium distribution system for the cooling of coils, structures, thermal shields, bus-lines, etc. The high-speed turbo-expander is an important refrigerating component of the EAST cryogenic system. In the turbo-expander, the axial supporting technology is critical for the smooth operation of the rotor bearing system. In this paper, hydrostatic thrust bearings are designed based on the axial load of the turbo-expander. Thereafter, a computational fluid dynamics-based numerical model of the aerostatic thrust bearing is set up to evaluate the bearing performance. Tilting effect on the pressure distribution and bearing load is analyzed for the thrust bearing. Bearing load and stiffness are compared with different static supply pressures. The net force from the thrust bearings can be calculated for different combinations of bearing clearance and supply pressure.

Keywords: EAST, helium cryogenic system, aerostatic thrust bearing, inclination, bearing load

(Some figures may appear in colour only in the online journal)

1. Introduction
The EAST, a fully superconducting tokamak, was built at the Institute of Plasma Physics, Chinese Academy of Sciences (CASIPP) at the end of 2004 and was first cooled down in 2006 [1]. The cold components such as superconducting magnets, bus-lines and current leads are cooled by a helium refrigerator. Different operating modes, such as cool down, warm up, standby and normal, are required for the cryogenic system being used today. In the refrigeration cycle, turbo-expanders are used to generate cooling capacity. As important items of refrigeration equipment, they can be used in cryogenic systems such as hydrogen and helium refrigeration [1–5].

In a turbo-expander, the bearing technology is critical for its smooth operation at high speed. As an optional candidate, an aerostatic thrust bearing with small feeding holes possesses low friction and larger stiffness and damping coefficients [6]. The performance of an aerostatic bearing is affected by geometric parameters, such as orifice diameter and film thickness, etc. A smaller film thickness is conducive to better bearing performance [7]. The bearing load of an externally pressurized air bearing decreases but mass flow rate increases with greater film thickness [8].
When the gas film thickness increases to a certain extent, pressure depression appears at the outer circumference of the supplying holes. Gao et al used the computational fluid dynamics (CFD) method to analyze the pressure depression, gas vortices and turbulence intensity [9]. Eleshaky attributed flow relaminarization to the transition shock region from supersonic to subsonic [10]. Yohimoto et al investigated the flow structure in the bearing clearance with a single inlet by solving Navier–Stokes equations directly. Pressure recovery was caused by the flow separation at the upper wall and decrease in the air velocity after the minimum pressure; air flow in the bearing clearance changed from laminar to turbulent in the region where rapid pressure recovery occurred [11]. Teo and Spakovszky incorporated both compressibility and viscosity into the resistance model to predict the axial stiffness and mass flow rates [12]. The existing researches focused on the performance of the bearing component. But the axial force balance in the high-speed rotor system is rarely covered.

In a cryogenic turbo-expander (figure 1), the rotor is usually composed of an expansion wheel, a brake wheel, a shaft, journal bearings and thrust bearings. In field operation, a higher rotor speed was anticipated to have a larger cooling capacity and compactness. Therefore, a stable bearing system was critical for the smooth operation of the rotor bearing system. In addition, an axial force originates from the expansion wheel and the braking wheel might change direction in the whole speed range. The turbo-expander should operate smoothly in both on design and off design modes [4]. These uncertainties show the importance of comprehensive bearing design and assembly techniques.

The design parameters of a helium turbo-expander are shown in table 1. For the upgrade of the cryogenic system, an axial load on the rotor bearing system was predicted and a hydrostatic thrust bearing was designed in this paper. The numerical model was validated with the experimental results. The pressure distribution and bearing load of a single-rowed aerostatic thrust bearing were analyzed numerically. The tilting effect on the bearing performance was also evaluated. Bearing load and stiffness were compared with different static supply pressures. The results may provide some insights for bearing design and further research.

### 2. Axial load and thrust bearing design

#### 2.1. Axial load prediction

The axial load of the turbo-expander for helium refrigeration is predicted based on the pressure distribution on the expansion wheel and braking wheel. The forces on the expansion wheel and braking wheel are shown in figures 2(a) and (b), respectively. The resultant axial force \( F_{\text{ea}} \) is computed using equation (1)

\[
F_{\text{ea}} = F_{e1} - F_{e2} - F_{e3} - F_{e4} - F_{e2},  \tag{1}
\]

where \( F_{e1} \) is the acting force from the back of the expansion wheel, \( F_{e2} \) is the acting force from the upper blade of the expansion wheel, \( F_{e3} \) is the acting force from the exit of expansion wheel and \( F_{e4} \) is the acting force at the hub of radius \( R_{c2}'' \). The acting force can be derived from pressure integration on the acted surface by equation (2)

\[
F_{ei} = \int 2\pi r p_{ei}(r) dr \quad i = 1, \ldots, 4,  \tag{2}
\]

where \( p_{ei}(r) \) is the pressure distribution at radial locations for \( F_{ei} \). For \( F_{e1}, F_{e2}, F_{e3} \) and \( F_{e4} \), the integration limits of the variable \( r \) are from \( R_b \) to \( R_{c1} \), \( R_{c2}' \) to \( R_{c1} \), \( R_{c3}'' \) to \( R_{c2}' \), 0 to \( R_{c3}'' \), respectively. It is usually assumed to be linear with radial location as shown in equation (3)

\[
p_{ei}(r) = a_i + b_i r \quad i = 1 \ldots 4,  \tag{3}
\]

where, \( a_i \) and \( b_i \) are coefficients that derive from the pressure distribution on the surface of the expansion wheel. For \( F_{e1} \), it is assumed that the gauge pressure at \( R_b \) is the ambient pressure, 0Pa and the gauge pressure at \( R_{c1} \) is the supply pressure of the expander, 1.25 MPa. \( a_1 \) and \( b_1 \) can be derived from the above known conditions. For other \( p_{ei} \), the coefficients \( a_i \) and \( b_i \) can be obtained with the same method. \( F_{e1} \) and \( F_{e2} \) can be computed from equations (4) and (5),

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate in expansion wheel (g s(^{-1}))</td>
<td>34.5</td>
</tr>
<tr>
<td>Flow rate in braking wheel (g s(^{-1}))</td>
<td>19.7</td>
</tr>
<tr>
<td>Inlet pressure of expansion wheel (MPa)</td>
<td>1.25</td>
</tr>
<tr>
<td>Outlet pressure of expansion wheel (MPa)</td>
<td>0.60</td>
</tr>
<tr>
<td>Inlet pressure of braking wheel (MPa)</td>
<td>0.59</td>
</tr>
<tr>
<td>Outlet pressure of braking wheel (MPa)</td>
<td>0.61</td>
</tr>
<tr>
<td>Inlet temperature of expansion wheel (K)</td>
<td>34</td>
</tr>
<tr>
<td>Outlet temperature of expansion wheel (K)</td>
<td>29</td>
</tr>
<tr>
<td>Capacity (kW)</td>
<td>( \sim 1.0 )</td>
</tr>
<tr>
<td>Isentropic efficiency</td>
<td>( \geq 60% )</td>
</tr>
</tbody>
</table>

![Figure 1. Cryogenic turbo-expander.](image-url)
respectively:

\[ F_{e3} = 2\pi \left( a_1 \frac{R_{e1}^2 - R_{eb}^2}{2} + b_1 \frac{R_{el}^3 - R_{eb}^3}{3} \right) \]  
(4)

\[ F_{e4} = 2\pi \left( a_2 \frac{R_{e2}^2 - R_{eb}^2}{2} + b_2 \frac{R_{e2}^3 - R_{eb}^3}{3} \right) . \]  
(5)

It is assumed that the pressure at the exit of the flow and shroud is continuous, and therefore \( F_{e3} \) and \( F_{e4} \) are computed using equation (6):

\[ F_{e3} + F_{e4} = 2\pi \left( a_3 \frac{R_{e2}^{1.2}}{2} + b_3 \frac{R_{e2}^{1.3}}{3} \right). \]  
(6)

If the average pressure \( p_{e2m} \) at the flow exit is used,

\[ F_{e3} + F_{e4} = \pi R_{e2}^{1.2} p_{e2m}. \]  
(7)

The reaction force from the exiting mass flux is computed by equation (8)

\[ F_{e2c} = q_{em} c_{e2mr}, \]  
(8)

where, \( q_{em} \) is the mean mass flux and \( c_{e2mr} \) is the mean velocity at the exit of expansion wheel.

The designed rotor speed is 150 000 rpm. With the design parameters in table 2, the axial load on the expansion wheel is computed to be 161.4 N.

For the braking wheel, similar assumptions to those used in the expansion wheel are adopted for predicting the axial force. The main parameters of the braking wheel are listed in table 3. The axial load on the braking wheel is 242.6 N.

Therefore, under the designed condition, the net force on the rotor by combining the expansion wheel and the braking wheel is 81.2 N. Due to the back to back configuration of the expansion wheel and braking wheel at the two ends of the rotor, the net force points towards the expansion wheel. The net force is balanced by two thrust bearings at the thrust disc.

### 2.2. Hydrostatic thrust bearing design

The viscosity and density of air and helium with pressure at 278 K are shown in figure 3. From 0.1 MPa to 1.4 MPa, density keeps a linear relation with pressure for the two kinds of gas. The density of air is greater than that of helium. Air density varies from around 1.3 to 16.4 kg m\(^{-3}\). The density of helium varies from around 0.17 to 2.2 kg m\(^{-3}\). The viscosities of the two fluids are quite independent of pressure. The viscosity of air and helium are around 17.6 \(\times\) \(10^{-6}\) Pa s and 18.9 \(\times\) \(10^{-6}\) Pa s, respectively. Therefore, the design of the helium hydrostatic thrust bearing can be referenced from that of an air-lubricated thrust bearing.

The designed aerostatic thrust bearing with one row of static supplying holes is shown in figure 4. The inner and outer diameters of the thrust disc are 12.5 and 30 mm, respectively. There are eight supplying holes of 0.35 mm diameter uniformly distributed in a circumferential direction. The high-pressure gas is supplied into the thrust disc through the holes. The radial location of the single row of supplying holes in the thrust bearing is determined by equation (9) to ensure the balance of flow resistance on the two sides of the supplying hole. The radial location of the thrust holes is 10 mm after rounding off. The cross-section at \(X = 0\) of the fluid control volume and boundary conditions in the bearing is shown for further boundary setting in the following

<table>
<thead>
<tr>
<th>Table 2. Design parameters of the expansion wheel.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>( R_{e1} ) (mm)</td>
</tr>
<tr>
<td>( R_{e2}' ) (mm)</td>
</tr>
<tr>
<td>( R_{e2}'' ) (mm)</td>
</tr>
<tr>
<td>( R_{eb} ) (mm)</td>
</tr>
<tr>
<td>( c_{e2mr} ) (m s(^{-1}))</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 3. Design parameters of the braking wheel.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>( R_{k1} ) (mm)</td>
</tr>
<tr>
<td>( R_{k2} ) (mm)</td>
</tr>
<tr>
<td>( R_{kb} ) (mm)</td>
</tr>
<tr>
<td>( c_{k2mr} ) (m s(^{-1}))</td>
</tr>
</tbody>
</table>
numerical model. The fluid film thickness is 10 μm, which is small compared with the diameter of the supplying hole. We find

\[ R_e = \exp\left(\frac{\ln(R_i) + \ln(R_o)}{2}\right) = (R_i R_o)^{1/2}, \]

where, \( R_i \) and \( R_o \) are the inner and outer radii, respectively, of the thrust bearing.

3. Numerical model

3.1. Governing equations

In this analysis, the fluid flow path in the thrust bearing is chosen as the control volume. The governing equations for the fluid in the control volume are given below, in which the time-dependent element can be ignored for steady-state flow.
Continuity:
\[ \frac{\partial \rho}{\partial t} + \nabla (\rho \mathbf{u}) = 0. \]  

Momentum in the \(x\), \(y\) and \(z\) directions:
\[
\begin{align*}
\frac{\partial (\rho \mathbf{u}_x)}{\partial t} + \nabla (\rho \mathbf{u}_x \mathbf{u}_x) &= -\frac{\partial p}{\partial x} + \Delta (\mu \nabla (\mathbf{u})) + S_{Mx}, \\
\frac{\partial (\rho \mathbf{u}_y)}{\partial t} + \nabla (\rho \mathbf{u}_y \mathbf{u}_y) &= -\frac{\partial p}{\partial y} + \Delta (\mu \nabla (\mathbf{u})) + S_{My}, \\
\frac{\partial (\rho \mathbf{u}_z)}{\partial t} + \nabla (\rho \mathbf{u}_z \mathbf{u}_z) &= -\frac{\partial p}{\partial z} + \Delta (\mu \nabla (\mathbf{u})) + S_{Mz},
\end{align*}
\]

where \(x\), \(y\) and \(z\) represent the Cartesian coordinates, \(\rho\) is gas density, \(\mathbf{u}\) is the velocity vector for \((u, v, w)\), \(p\) is pressure, \(\mu\) is the dynamic viscosity and \(S_{Mx}, S_{My}, S_{Mz}\) are the momentum sources.

3.2. The CFD method and boundary conditions

The single-row thrust bearing is analyzed based on commercial CFD software. The governing equation and the boundary condition of the fluid are solved by the SIMPLE scheme, which is based on the finite-volume method. Iteration is performed until a converged solution is obtained. Due to the turbulence flow in the static thrust bearing, the RNG \(k-c\) turbulence model is adopted. Taking into account the variation of density with pressure, helium is chosen as the ideal gas for the bearing clearance. A residue of \(10^{-5}\) is set for convergence criteria.

In this model, the inlet of the thrust bearing is set as the pressure inlet boundary of 0.61 MPa in figure 4. The circular surfaces at the radial ends are set as the pressure outlet boundary of 0 MPa. The thrust disc is set as a moving wall with a rotational speed of 150 000 rpm.

3.3. Geometry and mesh

In the axial direction along the gas film, the gas film is 5–25 \(\mu m\) thick. This gas film thickness is much less than the bearing dimension. In this model, bearing load is set as an evaluation indicator. Figure 5 shows the calculated bearing load with a bearing clearance of 20 \(\mu m\) based on different mesh grid elements. When there are more than 1.5 million grid elements, the bearing load no longer changes much.

4. Model validation

For model validation, figure 6 shows the numerical results with air compared with the experimental [13] and MTI
The inner and outer diameters of the bearing are 20 and 38 mm, respectively. The numerical results agree with the results from MTI and the experiment quite well. However, when the bearing clearances are larger than 15 μm, the deviation between the experiment and CFD method is quite small. When the bearing clearance decreases, the deviations between CFD and experiment become larger, which might be attributed to the manufacture and assembly of the thrust bearing and thrust disc in the experiment. Due to the similar trends of density and viscosity with pressure, the same model can be used for both helium and air.

5. Numerical results and analysis

5.1. Pressure distribution

The pressure distribution is important for the bearing load. The pressure profile of the thrust bearing for a 10 μm film thickness is shown in figure 7. In figure 7(a), the pressure spikes correspond to the supplying holes. Due to the symmetrical geometry of the bearing, the pressure distribution at the A–A cross-section (X = 0, Y > 0) is shown in figure 7(b). The maximum pressure in the bearing is located around the supply holes. From the supply holes to the outlet of the thrust bearing, the pressure drops almost exponentially to the surrounding ambient pressure.

The gas film thickness of the static thrust bearing is important for pressure distribution and smooth operation of the rotor bearing system. The pressure distribution at the A–A cross-section (X = 0) with different gas film thicknesses is shown in figure 8. The maximum pressure for the three gas films is almost the same, around 0.61 MPa at the supply holes. For a 5 μm film thickness, the pressure attenuated to ambient pressure from a supply hole to the inner and outer radius of the bearing. There is no pressure depression for gas film thicknesses of 0 and 15 μm. When the bearing clearance...
increases to 25 μm, pressure depression appears near the static supply hole. Local negative pressure appears after leaving the supply holes. The peak pressure in the bearing clearance is around 0.1 MPa. It follows that a smaller gas film thickness is conducive to a higher pressure in the bearing.

The bearing load and mass flux are important parameters, as shown in Figure 9. The bearing load increases from 67.7 to 138.8 N when the bearing clearance decreases from 25 to 5 μm. The gas consumption in the bearing is smaller for the bearing with a thinner gas film. The mass flux increases from 2.9 × 10⁻³ to 104 × 10⁻³ g s⁻¹ when the bearing clearance increases from 5 to 25 μm.

In an aerostatic thrust bearing, the effects of supply pressure on the bearing load and stiffness are shown in Figures 10(a) and (b), respectively. Four supply pressures, 0.2, 0.4, 0.6 and 0.8 atm (atmospheric pressure) were chosen for the thrust bearing. With a higher supply pressure, the bearing load increases. With smaller bearing clearances, the increment in bearing load is larger with the same supply pressure increment of 0.2 atm. Bearing stiffness can be derived directly from the relation of load and displacement. Under the same supply gas pressure, the bearing stiffness maintains an inverse relation with bearing clearance. However, when the supply gas pressure is smaller, such as 0.4 and 0.2 atm, the stiffness stays relatively stable with larger gas film thicknesses. This can be attributed to the characteristics of the pressure distribution with gas film thickness.

In this high-speed rotor bearing system there are two thrust bearings at the thrust disc. The axial force of the rotor is balanced by the two bearings of face-to-face configuration. The net force from the two thrust bearings can be calculated for different combinations of bearing clearance and supply pressure. With the same supply pressure of 0.8 atm, the bearing net force is 86.6 N with two bearing clearances of 10 and 30 μm; and 98.3 N for 5 and 25 μm. Both kinds of bearing clearance partition can meet the demand of the cryogenic turbo-expander.

5.2. Effect of misalignment on the bearing performance

In the rotor bearing system, misalignment is inevitable due to the manufacturing and assembly processes. A schematic of misalignment of the thrust bearing is shown in Figure 11. The gas film thickness is uniformly h₀ with no inclinations. When the symmetrical axis of the thrust disc inclines from the z-axis with an angle of ψ from the direction (0, 0, 1), the gas film thickness h between the thrust disc and the thrust bearing in the thrust bearing varies on the surface. The inclination at the outer radius of the thrust disc is Δh.

In real operation, the direction vector can be directed in the plane XOY with altitude angle ψ from the direction (1, 0, 0). In this analysis, two altitude angles are chosen, 0° and 22.5°. Three inclination angles, 0.008°, 0.015°, 0.023° are analyzed with a bearing clearance of 10 μm (corresponding to inclination distances Δh of 2, 4 and 6 μm at the outer radius of the thrust disc, respectively). The pressure profiles at cross-section B–B with different inclination angles are shown in Figure 12. The pressure spikes of the bearings are around 0.61 MPa, which corresponds to the supply gas pressure. If there is no inclination, then the pressure is symmetrical at the two sides of ψ = 0°. With increasing inclination, the bearing clearance increases at X > 0 and decreases at X < 0. The pressure becomes smaller on the side with a thicker gas film. For the 10 μm bearing clearance, there is no obvious pressure depression with inclination.

The bearing load and mass flux with inclination are shown in Figures 13(a) and (b). With a bearing clearance of 10 μm, the bearing loads are around 120 N. At an altitude angle of 0°, the bearing load increases by about 7 N when the inclination angle reaches 0.024°. At an altitude angle of 22.5°, the bearing load increases by about 13 N. For both altitude angles of 0° and 22.5°, the mass flux increases with inclination angle from 17.2 × 10⁻³ to 19.6 × 10⁻³ g s⁻¹. The mass flux of ψ = 22.5° is a little bit higher than ψ = 0° when the inclination is larger than 0.016°.

6. Conclusions

In this paper a hydrostatic thrust bearing for the EAST cryogenic turbo-expander was designed based on the axial force from an expansion wheel and braking wheel. The tilting effect on the annular static thrust bearing was investigated numerically. The CFD method was used to determine the pressure distribution and flow in the micrometer-thick gas film. Bearing parameters such as bearing load and gas consumption were compared under inclinations. The following conclusions might be drawn:
(1) Pressure depression appeared in the aerostatic thrust bearing with increase in the thrust bearing clearance. Negative (relative to ambient) pressure appeared when the bearing clearance was larger than a certain value. Mass flux increased almost linearly with bearing clearance.

(2) With increasing inclination, both bearing load and gas consumption increased a little. For the designed bearing, bearing load could meet the demand of the axial load of the rotor bearing system. In other words, the annular hydrostatic thrust bearing could endure a certain amount of bearing inclination.

(3) The bearing load increases with higher supply pressure. With smaller bearing clearances, the bearing load increment is larger with the same pressure increment of 0.2 atm. Under the same supply gas pressure, the bearing stiffness remains in inverse proportion to bearing clearance. However, when the supply gas pressure is smaller, such as 0.4 and 0.2 atm, the stiffness stays relatively stable with larger gas film thicknesses.

(4) In this high-speed rotor bearing system there are two thrust bearings at the thrust disc. The axial force of the rotor is balanced by the two bearings in face-to-face configuration. The net force from the two thrust bearings can be calculated with different combinations of bearing clearance and supply pressure. Although a smaller bearing clearance is beneficial for a higher bearing load, the vibration amplitude of the rotor cannot be ignored. For the stability of the system, a greater bearing clearance margin is recommended as long as the bearing load meets the requirement.

(5) The density variation with pressure was considered in this numerical analysis. The numerical results with an ideal gas and turbulence model could predict the bearing performance quite well, which provided an accurate model for further analysis.

Acknowledgments

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