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Design of thrust bearing for hydrogen liquefaction turbine

Zhenyang Zhang[1, 2, *], Bingyan Yu[1, 2], Junzhe Wu[1], Zhijiu Sun[3], Jianli Gan[3]

[1]Beijing Institute of Aerospace Test Technology, Beijing, China

[2]Aerospace Hydrogen Energy Technology Co., Ltd, Beijing, China

[3]Zhejiang Provincial Energy Group Beihai Hydropower Co., Ltd, Lishui, China

* zhangzy_buaa@163.com

Abstract. With rapid development of hydrogen energy industry, liquid hydrogen, as a carrier with the highest energy storage density, is an effective way to ensure the large-scale application of hydrogen energy. Thrust bearing, which is the core component of hydrogen liquefaction turbine expander, has always been difficult to design. In this paper, the stress characteristics of the thrust bearing for hydrogen liquefaction are analyzed, and the analysis for designed prototype is performed. The numerical simulation results are compared with the design method. Results show the optimum clearance of thrust bearing and clearance of closed bearing has obvious influence on the peak value and position of stiffness caused by eccentricity. Numerical simulation results reflect the internal flow structures in bearing clearance, while the maximum stiffness is relatively conservative compared to analysis of radius ratio.

Keywords: Hydrogen liquefaction, turbine, thrust bearing, design method.

1. Introduction

With the implementation of carbon peaking and carbon neutralization in China, hydrogen liquefaction is an effective way to realize the efficient utilization of hydrogen energy. The core cryogenic equipment in hydrogen liquefaction system is the turbine expander. As the most important component to ensure the safety and efficiency of expander, thrust bearing is the difficulty and emphasis in system developing. Thrust bearing with small friction resistance, almost zero wear rate, low noise, small vibration keeps a stable, smooth and precise operation.

Thrust bearing with orifice restrictors is convenient to process for the simple structure, and stable overall performance, which has become the priority research object of many scholars. Nakamura and Yoshimoto[1] studied the influence of various loads on the thrust bearing with composite restrictor in both theoretical and practical methods. Results show that thrust bearing has better performance and advantages. Renn and Hsiao[2] explored the mass flow characteristics of aerostatic thrust bearing through numerical and experimental study, and concluded the mass flow coefficient of simple orifice which is relatively consistent with throttling theory. Masaaki and Yoshimoto[3] conducted a comparative study on the static characteristics of thrust bearings with different annular orifice diameters using finite element simulation analysis. Bhat et al.[4] performed static and dynamic research on aerostatic thrust bearing with single orifice row. Gherca et al.[5] mainly explored the characteristics of unidirectional thrust bearing with different stripes under steady-state and transient states through finite element method. Domestic research institutions such as Xi'an Jiaotong University[6,7] studied the effects of different parameters, such as the distributions of thrust bearing orifices, structure and pressure, on the static and dynamic characteristics of the bearing.

Researchers currently focus on the rotating machinery in normal temperature or near liquid nitrogen temperature, while there is less investigation on the thrust bearing used in the turbine machinery at liquid hydrogen boiling point, especially the whole process from basic performance design to three-dimensional numerical simulation. This paper conducts the theoretical stress and kinetic analysis and designs the thrust bearing for hydrogen liquefaction turbomachinery, which meets the characteristic calculation and finite element simulation results. The exploration shows

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effective analysis methods of thrust bearing and the influence of parameters on bearing performance.

2. Thrust bearing stress analysis

The support of thrust bearing is formed by the force on the bearing surface and thrust disc on the axis due to the gas film. The high pressure gas after dried and purified ejects into bearing clearance through orifices on bearing surface, and fills the gap to establish gas film with high pressure and high viscosity, which floats the thrust disc and bears the load for smooth rotating in the axial direction. The orifices could adjust pressure of film with variation of bearing clearance.

The clearance between bearing surface with orifice and thrust disc is the thrust bearing gas film thickness h. The compressed gas with pressure p_0 enters the bearing clearance through throttle holes from inlet, and the gas pressure at outlet of the orifice is p_d. The out flux is divided into radial outflow part $\dot{m}_1$ and radial inflow $\dot{m}_2$. The two parts flow into the inner and outer exit structures with pressure p_a. A certain pressure distribution is formed in the gas film, as shown in Figure 1. The thrust bearing capacity W can be obtained by integrating the differential pressure along the thrust disc surface.

The flow resistance decreases and flow rate increases with gas film thickness increasing at constant orifice diameter d, which leads to a larger pressure drop through orifice and results in the drop of p_d and thrust. The gas film thickness could be adjusted adaptively according to the load within the range of bearing capacity, so as to achieve the balance between bearing capacity and load.

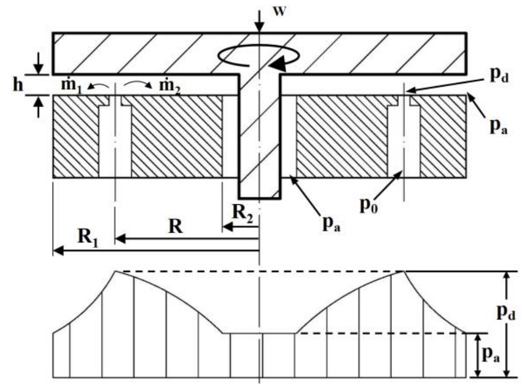

Figure 1. Thrust bearing structure and pressure distribution on thrust surface.

In this paper, thrust bearing with single orifice row and pressure equalizing ring groove is analyzed, designed and calculated. Out flux from orifice is evenly distributed in the ring groove to be pressure equalized, and flows radially inward and outward without circumferential and axial movement. The preliminary design structural parameters are listed in Table 1.

Table 1. Structural parameters of thrust bearing surface.

Parameters	Value
Outer diameter of thrust surface $2R_1$/mm	80
Inner diameter of thrust surface $2R_2$/mm	27
Diameter of orifice distribution $2R = 2\sqrt{R_1R_2}$/mm	46.5
Number of orifices *n*	12
Ring groove width *b*/mm	0.6
Ring groove depth h_g/mm	0.1
Orifice diameter *d*/mm	0.3

According to the radial uniform flow above, the gas equation of motion in cylindrical coordinates is obtained as follows:

$$\frac{\partial p}{\partial r} = \eta \frac{\partial^2 v_r}{\partial z^2} \tag{1}$$

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$$\frac{\partial p}{\partial z} = 0 \tag{2}$$

$$\frac{\partial p}{\partial \theta} = 0 \tag{3}$$

It could be concluded from equation (2) and (3) that pressure p is independent from z and θ, and the equation of motion in the thrust bearing clearance is:

$$\frac{dp}{dr} = \eta \frac{\partial^2 v_r}{\partial z^2} \tag{4}$$

Assuming the bearing film thickness in the clearance is uniform, when inlet pressure p_0 is constant, the outlet pressure of each orifice is equal to be p_d. The total mass flow rate $\dot{M}$ in the gap is:

$$\dot{M} = n\dot{m} = n\phi A p_0 \sqrt{2\rho_a/p_a}\varphi \tag{5}$$

Where $\dot{m}$ is the mass flow rate from a single orifice, and A is the area of ring groove. ϕ is the flow coefficient and φ is the flow function. The outward flow from orifice $\dot{m}_1$ and radial inward $\dot{m}_2$ are:

$$\dot{m}_1 = 2\pi r \int_0^h \rho v_r dz \ (\mathrm{R} < \mathrm{r} \le R_1) \tag{6}$$

$$\dot{m}_2 =- 2\pi r \int_0^h \rho v_r dz \ (R_2 \le \mathrm{r} < \mathrm{R}) \tag{7}$$

$$\dot{M} = \dot{m}_1 + \dot{m}_2 \tag{8}$$

According to the equations above, the gas film pressure could meet:

$$p^2 = p_d^2 - (p_d^2 - p_a^2)\ln(r/R)/\ln(R_1/R) \ (\mathrm{R} < \mathrm{r} \le R_1) \tag{9}$$

$$p^2 = p_d^2 + (p_d^2 - p_a^2)\ln(r/R)/\ln(R/R_2) \ (R_2 \le \mathrm{r} < \mathrm{R}) \tag{10}$$

The bearing capacity W is the pressure difference between gas film and ambient on the other side of the thrust disc:

$$W = 2\pi \int_{R_2}^{R} prdr + 2\pi \int_{R}^{R_1} prdr - \pi(R_1^2 - R_2^2)p_a \tag{11}$$

And bearing stiffness according to eccentricity e is:

$$K_W = dW/de \tag{12}$$

3. Analysis of thrust bearing characteristic

3.1 Analysis of optimum radius ratio R_1/R_2

The thrust bearing radius ratio R_1/R_2 affects the bearing capacity and stiffness at the same time. Distributions are calculated under different R_1/R_2 using 8bar helium as gas source with constant orifice diameter, shown in Figure 2.

The maximum bearing capacity reaches $W_{max} = 1868.3\mathrm{N}$ at $R_1/R_2 = 4$, and maximum bearing stiffness is $K_{W,max} = 65.7\mathrm{N}/\mu\mathrm{m}$ at $R_1/R_2 = 2.25$. Due to limitation of the size and structure of the expander, the initial structural parameter radius ratio $R_1/R_2 = 80/27 = 2.96$ lies in the optimum range of (2.25, 4) according to curves in Figure 2, which holds enough capacity and stiffness for supporting.

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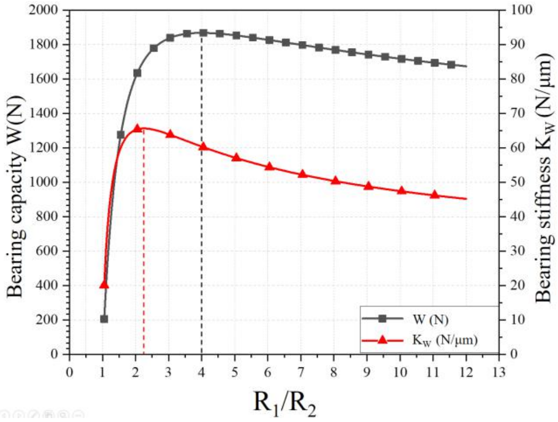

Figure 2. Variations of bearing capacity and stiffness with radius ratio variation.

3.2 Sided bearing optimal clearance

Relationship between bearing thrust and clearance shows the characteristics on one single side. The influence of inlet pressure p_0 variation from 5 to 10bar on bearing capacity and stiffness is obtained and shown in Figure 3, with outlet pressure $p_a = 1\text{bar}$. The bearing capacity decreases with increasing of gas film thickness, which equals to bearing clearance, and the stiffness has a peak value.

The maximum stiffness supported by the bearing is at the maximum slope of the bearing capacity curve. Both capacity and stiffness are positive related to p_0. The preset 8bar leads to thrust large enough at maximum stiffness with gas film thickness h=24.4μm. Considering the cost and level of compressor, which needs too much energy consumption to keep high pressure supply, and processing of bearing, the preset condition could ensure the operation safety and 24.4μm is optimal.

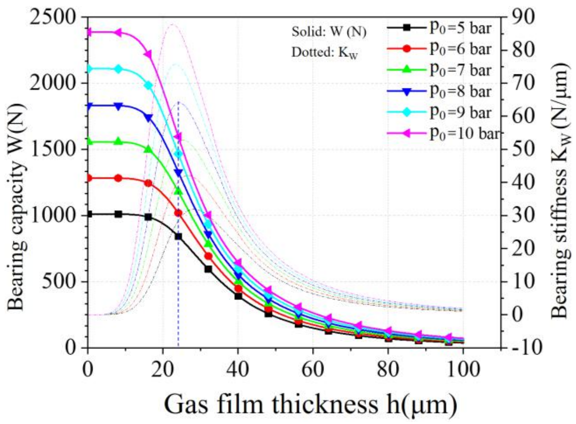

Figure 3. Variations of bearing capacity and stiffness with p_0.

3.3 Bearing characteristic analysis

The closed thrust bearing with gas supported on both sides of the thrust disc is used in actual turbomachinery. The closed bearing capacity and stiffness are the resultant on both sides of thrust disc. On the basis of single-sided bearing above, the relationship between resultant capacity and eccentricity e can be obtained by fitting the bearing characteristic curve.

After rounding the optimal clearance above to $2h_0 = 50\mu m$ for analysis, the closed bearing characteristics are shown in Figure 3(a), which reveals the maximum resultant stiffness $K_W = 127.7 N/\mu m$ in the middle position without eccentricity. If the clearance is reduced for large stiffness, it is difficult to realize processing, but if it is increased, it will lead to the decrease of stiffness. Especially when h_0 deviates a lot from the optimal clearance, the closing stiffness will be significantly reduced such as $2h_0 = 100\mu m$ in Figure 3(b). The stiffness curve is bimodal, and the maximum is also significantly reduced to $K_W = 64.1 N/\mu m$ located at both sides of the equilibrium

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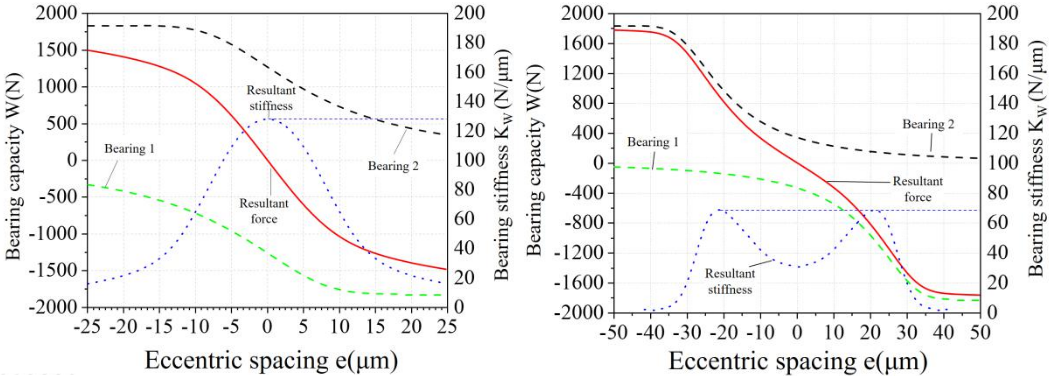
position. Therefore, bearing clearance should be designed close to the optimal clearance as far as possible.

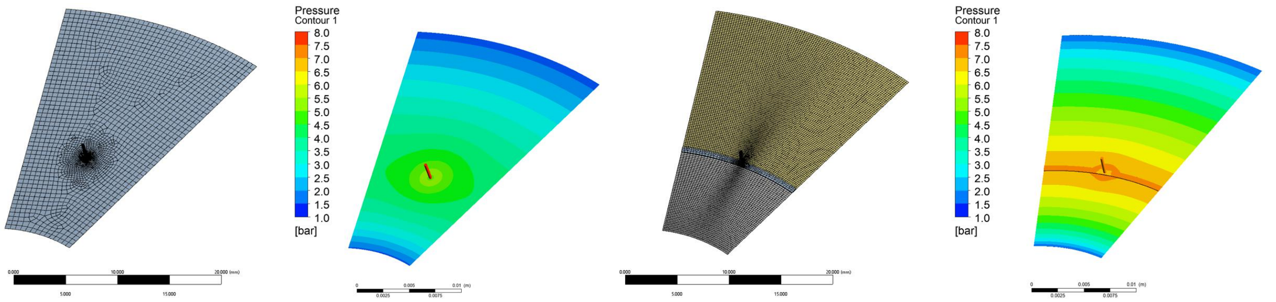


(a) $2h_0 = 50\mu m$ (b) $2h_0 = 100\mu m$

Figure 4. Characteristic of closed thrust bearing.

4. Results and analysis

The numerical simulation of thrust bearing is performed 1/12 of the whole gas film due to the rotationally symmetrical structure and the periodic boundary conditions. The thrust bearings with or without pressure equalizing ring groove structure are simulated and analyzed, shown in the Figure 5.



(a) meshing without groove (b)result without groove (c)meshing with groove (d) result with groove

Figure 5. Numerical simulation of bearing gas film.

Sixteen pressure distribution cases are calculated for clearance from 5 to 80μm with 5μm interval at $p_0 = 8bar$ and $p_a = 1bar$. The h=20μm case as an example is shown in Figure 5(b) and (d). Pressure distribution with ring groove in Figure 5(d) reveals the gas film assumption in analysis of optimum radius ratio above, where outflow from orifice is evenly distributed in the groove and flows inward and outward with pressure reduced gradually. For structure without ring groove in Figure 5(b), outflow diffuses circumferentially and decreases sharply to result in a weak thrust effect.

The variation of bearing capacity with gas film thickness is shown in Figure 6 consisting of maximum values from 16 simulation cases above fitting by the quintic polynomial, and the optimal clearance and maximum stiffness are calculated according to the fitting curve and shown in Table 2. The optimal clearance with ring groove is close to the optimum radius ratio analysis result with large stiffness.

5. Conclusion

Thrust bearing with orifice employs the optimal bearing clearance, which is large enough to be easy to realize in processing, to balance the bearing capacity and stiffness simultaneously. The clearance of closed bearing affects the stiffness caused by eccentricity obviously, which impacts on

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the peak value and peak position significantly, however, it has little influence on the bearing capacity.

Numerical simulation provides better understandings of the internal flow structure of gas film and verifies the assumptions in analysis of radius ratio. Results show that thrust bearing with pressure equalizing ring groove has larger optimal clearance, which is close to radius ratio analysis result, and leads to relatively conservative maximum stiffness.

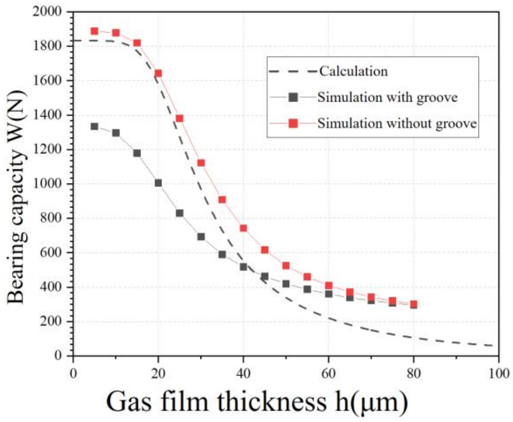

Figure 6. Variation of bearing capacity with gas film thickness.

Table 2. Comparison of optimal clearance and maximum stiffness calculation.

	Optimum clearance(μm)	Maximum stiffness(N/μm)
Analysis of radius ratio	24.4	64.1
With ring groove	24.7	48.5
Without ring groove	20.1	33.5

Acknowledgments

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