Design and Research of New Type High-speed Hydrostatic Spindle Test Rig

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Abstract. Aiming at the air dynamic pressure foil bearing performance test rig, design and research a new high-speed static pressure spindle test rig. Carry out aerodynamic design of the driving turbine and compressor, obtain the three-dimensional impeller shape and carry out the strength calculation, design a new high-speed static pressure spindle test rig; use the finite element method to model the test bed rotor system, and use the transfer matrix method to analyze the shaft system structure critical speed and mode, so that the test rig can effectively avoid the critical speed and run stably. Analyze the unbalanced response and stability of the rotor system, and reasonably arrange the test rig device to achieve the purpose of measuring the performance of the dynamic pressure foil bearing, and ensure that the turbine static pressure spindle test rig can provide for the measurement of the air dynamic pressure foil bearing performance stable high speed.

1. Introduction

The Air foil bearing is a kind of self acting dynamic pressure gas bearing. It uses surrounding air as lubricant[1], elastic foil as supporting element, and relies on the dynamic pressure effect between the flexible foil surface and the shaft to support the rotor system[2]. With the characteristics of high speed, low friction, no pollution and long service life[3], air foil bearing is widely used in rotating machinery such as air dynamic pressure suspension turbine cooler, oil-free lubricating turbine expander, centrifugal blowers with foil bearings, etc[4].

Since the 1990s, some universities and research institutions in China have only begun theoretical and experimental research on air foil bearings. The team of Professor Hou Yu[5] from the Institute of Refrigeration and Cryogenics of Xi’an Jiaotong University has carried out theoretical and experimental research on air foil bearings in turboexpanders. Yang Lihua et al. [6] built a radial bearing experimental platform, and used the friction moment method and radial displacement response spectrum method to test the bearing lift off speed. The Professor Feng Kai [7] of Hunan University has recently research on foil bearing and rotordynamic. Meanwhile, the member of his research team, Lv Peng and Liu Yuman[8,9] carried out the theoretical and experimental research on flexibility aerodynamic pressured bearing, and experimentally studied the static characteristics of air foil bearings. Xie Shilong, Nanjing University of Aeronautics and Astronautics[10], discussed the influence of the fixing method on the structural rigidity of the wave foil gas bearing foil, and an experimental bench was set up. The experimental bench consisted of the experimental bench body, the lubrication cooling system, and the measurement system.
In 2011, Tae Ho Kim and Luis San Andres de et al.[11] Proposed a new experimental method to identify the structural stiffness and damping of bearings. However, the second generation wave foil bearing has not been tested. In 2015, Franck Balducchi et al.[12] built a pneumatic rotor test rig. The nonlinear response of a rigid rotor supported by gas foil bearing to different unbalance is studied. However, due to the heating problem of the motor, it is impossible to test the performance of the hydrodynamic foil bearing at high speed for a long time, which leads to the inaccuracy of the relevant theoretical model. Therefore, a new high-speed hydrostatic spindle test-bed using aerostatic bearing to carry the spindle is proposed, which can maintain high-speed and stable operation for a long time, which is important for the performance test of aerodynamic foil bearing research significance.

2. Aerodynamic design and strength calculation of turbine and compressor

2.1 Aerodynamic design

Referring to the parameters of the air turbine expander and the existing domestic and foreign turbine expanders, and according to the specific characteristics and requirements, the basic thermal parameters and structural parameters of the driving turbine and compressor are selected to optimize the reaction degree, wheel diameter ratio and other parameters. In addition to its thermal performance, the stability of mechanical performance is also an important factor, which determines the feasibility and operation reliability of mechanical design[8]. Therefore, it is necessary to carry out multi-objective optimization design for its thermal performance and mechanical performance. The constraints are considered comprehensively, and other thermal and structural parameters are selected from the database, and the experience of experts is added to improve the search efficiency. Main design results: The operating speed is 70000 rpm, the air mass flow is 0.276 kg/s, the compressor diameter is D = 92 mm, the driving turbine D = 85 mm, ρ = 0.53, U1 = 0.624, and the adiabatic efficiency is greater than 0.65. Three-dimensional modeling of the final turbine and compressor are shown in Figures 1(a) and (b).

![Figure 1](a)Three-dimensional modeling of turbine; (b) Three-dimensional modeling of compressor.

The calculation and simulation results of the turbine and compressor are as follows: the turbine inlet flow is 0.27 kg/s, the outlet flow is 0.22 kg/s, the overall efficiency is 92.5%, the inlet/outlet pressure ratio is 1.5, the compressor inlet flow is 0.27 kg/s, the outlet flow is 0.22 kg/s, the overall efficiency is 74.1%, the inlet / outlet pressure ratio is 2.04, and the error of the simulation results is within the allowable range.

2.2 Strength calculation
Figure 2. Stress contour

The computer-solved Mises stress value is obtained. The resulting stress distribution of the compressor impeller is shown in Figure 2. According to the cloud diagram display, the maximum stress value of the impeller occurs at the root of the leading edge of the blade is 126 MPa, and the yield strength of the material is 300 MPa, which is less prone to yield failure than the allowable stress of the material, and is in a safer range. After the aerodynamic strength calculation, the results are almost the same.

3. Rotor dynamics calculation of hydrostatic spindle system

Using the transfer matrix method, the state vector of the ith section of the rotor is \( Z_i \), which is composed of the radial displacement \( X_i \) of the section, the deflection angle \( \alpha_i \), the bending moment \( M_i \) and the amplitude of the shear \( Q_i \),

\[
Z_i = \begin{bmatrix} X_i, A_i, M_i, Q_i \end{bmatrix}^T
\]

(1)

It has a certain relationship with the state vector \( Z_{i+1} \) of section \( i+1 \), that is

\[
Z_{i+1} = T_i Z_i
\]

(2)

Where \( T_i \) is called the transfer matrix of a component between two sections. When there are \( r \) elements in the state vector, \( T_i \) is a square matrix of order \( r \times r \) whose elements can be obtained by analyzing the force and deformation relations on the stressed components.

\[
T_i = B_i D_i = \begin{bmatrix}
1 + \frac{l^2 (1 + \nu) (m\omega^2 - K_{ij})}{6EJ} & 1 + \frac{l^2 (I_p - I_d) \omega^2}{2EI} & \frac{l^2}{2EI} & \frac{(1 + \nu)}{6EJ} \\
\frac{6EI}{l^2 (m\omega^2 - K_{ij})} & \frac{6EI}{l^2} & \frac{l}{2EI} & \frac{1}{6EJ} \\
\frac{6EI}{l} & \frac{6EI}{l} & \frac{1}{2EI} & \frac{l}{2EI} \\
\frac{l (m\omega^2 - K_{ij})}{(m\omega^2 - K_{ij})} & \frac{l (I_p - I_d) \omega^2}{2EI} & \frac{l}{2EI} & \frac{1}{6EJ}
\end{bmatrix}
\]

(3)

Establish the rotor system model, use the small disturbance method to calculate the stiffness and damping value of hydrostatic bearing, select 25th node and 35th node to apply the bearing position, the model is shown in the figure 3.
Figure 3. Dynamic model of rotor system

3.1 Calculation of critical speed under rigid support

The natural frequency of the rotor is determined by the stiffness and mass of the rotor and their distribution, independent of the initial conditions. The former three steps natural frequency of the rotor is the first three critical speed under the rigid support. It reflects whether the design of the rotor itself is reasonable.

Figure 4. The first order modes and modes of the rotor

Figure 5. Second order modes and modes of vibration
3.2 Unbalance response calculation of the rotor system of the experimental platform

The unbalance response is used to study the sensitivity of the rotor to the unbalance on the position. The unbalance response of the rotor is analyzed by the transfer matrix method, which is similar to the calculation of the critical speed. The recurrence of the state vector starts from the left most section of the rotor. The state vector of section $i (i=2,3,\ldots,N,N+1)$ is

$$
\begin{align*}
X_A & = \begin{bmatrix} a_{i1} & a_{i2} & a_{i3} & a_{i4} \\ a_{i3} & a_{i2} & a_{i3} & a_{i4} \\ a_{i3} & a_{i2} & a_{i3} & a_{i4} \\ a_{i4} & a_{i2} & a_{i3} & a_{i4} \end{bmatrix} \begin{bmatrix} X_N \\ A \\ M \\ Q \end{bmatrix} = \begin{bmatrix} a_{i1} & a_{i2} \\ a_{i1} & a_{i2} \\ a_{i1} & a_{i2} \\ a_{i1} & a_{i2} \end{bmatrix} \begin{bmatrix} X_N \\ A \\ M \\ Q \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ b_3 \end{bmatrix} 
\end{align*}
$$

(4)

For the right most $N+1$ section, and considering its boundary conditions are

$$
\begin{align*}
\begin{bmatrix} M \\ Q \end{bmatrix}_{N+1} & = \begin{bmatrix} a_{i1} & a_{i2} \\ a_{i1} & a_{i2} \end{bmatrix} \begin{bmatrix} X_N \\ A \\ M \\ Q \end{bmatrix} + \begin{bmatrix} b_3 \\ b_4 \end{bmatrix}_{N+1} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}
\end{align*}
$$

(5)

For the elastic undamped critical speed in the speed range of $0\% - 125\%$ of the trip speed, a separate damping unbalance response analysis is carried out. The unbalance shall be applied according to the specific shape of undamped vibration mode. The applied unbalance is calculated based on the total mass of the cantilever end and applied at the maximum displacement of the shafting.

**Figure 6.** Third mode and mode shape of rotor

**Figure 7.** Response at the left bearing of the first-order unbalance response analysis of the shaft system
Figure 8. Response at the right bearing of the first-order unbalance response analysis of the shaft system

When calculating the unbalanced response, the AF value corresponding to the peak value of the left and right bearings is 2.84 and 3.87 respectively, and the calculation of the isolation margin is as follows:

14400rpm:

$$SM_{left} = 17 \left( 1 - \frac{1}{AF - 1.5} \right) = 17 \left( 1 - \frac{1}{2.84 - 1.5} \right) = 4.31$$

(6)

15000rpm:

$$SM_{left} = 17 \left( 1 - \frac{1}{AF - 1.5} \right) = 17 \left( 1 - \frac{1}{3.87 - 1.5} \right) = 9.82$$

(7)

The vibration peak value of the left bearing is large at 14400rpm, and its isolation margin is 4.31%. The working speed of 70000rpm falls outside the isolation margin, and the unbalanced response of the left bearing is qualified.

The isolation margin SM of the right bearing is the minimum value between the calculated value and 16, so it is 9.82.

The vibration peak value of the right bearing is large at 150000 rpm, its isolation margin is 9.82%, and the working speed of 70000 RPM falls outside the isolation margin. The unbalanced response at the right bearing is qualified.

According to rule 2.1.3 of API617[13], the unbalanced response at the left and right bearings is qualified.

3.3 Stability analysis

During the operation of the rotor, large-scale vibration may be caused by some reasons. These reasons are caused by the oil film force in the bearing, the fluid force in the seal, the friction between the rotor and the stator, and the mass and stiffness of the rotor are not equal in each radial direction. And many factors that have yet to be recognized. Rotation of the rotor will cause the rotor to lose stability, referred to as instability. Instability is extremely dangerous for the safe operation of the rotor. The stability analysis is shown in Figure 11.
For the elastic undamped critical speed in the speed range of 0% - 125% of the trip speed, the damping unbalance response should be analyzed. The unbalance shall be applied according to the specific shape of undamped vibration mode. The applied unbalance is calculated based on the total mass of the cantilever end and applied at the maximum displacement of the shafting.

4. Turbine static pressure spindle experimental bench construction

According to the design and calculation of the experiment table, the turbine static pressure spindle experiment table was constructed. The layout of the experiment table is shown in figure 9. Divided into drive module, loading module, measurement module and visualization system. The driving module uses an air compressor to input the high-pressure air source. The rotor system has been dynamically detected in the previous work, and the speed can reach the working speed. The measurement module consists of a series of sensors. The visualization system is output and processed by Labview software written by the team.
Figure 11. Site map of aerodynamic spindle experiment table

4.1 Sensor arrangement
After the experiment table is set up, various sensors need to be reasonably arranged, mainly including speed sensors, displacement sensors, temperature sensors, and force sensors, to measure the speed, displacement, temperature, and friction torque of foil dynamic pressure air bearings. The performance test of the bearing capacity, start-stop times, take-off speed, and stability of the foil dynamic pressure air bearing is realized.

Figure 12. Sensor layout of aerodynamic spindle experiment table

5. Air foil bearing lift off speed and bearing capacity test
Lift off speed test of aerodynamic bearing is carried out by using the built experimental platform. Figure 11 is the actual picture of foil bearing, and the specific parameters of bearing are shown in the table.

Figure 13. Experimental bearing
Table 1. Geometrical parameters of foil bearing

| Dimension parameters | Value       |
|----------------------|-------------|
| Foil thickness       | 0.07[mm]    |
| Foil height          | 0.22[mm]    |
| Bump pitch           | 3.3[mm]     |
| Diameter             | 28[mm]      |
| Foil width           | 25[mm]      |

Figure 14. Friction torque diagram

At the beginning of the experiment, the friction torque of the foil bearing is transmitted to the tension sensor through the torque rod, and then it is obtained through the data analysis platform. When the friction torque reaches the maximum value, it indicates that the hydrodynamic foil bearing is in the lift off stage, when the friction torque reaches a stable value, the rotational speed is the lift off speed of the bearing, the lift off speed is 4000rpm. when test bench runs for about 28 s, the friction torque decreases slightly and then increases rapidly when the driving device is turned off. The lift off speed measured in the experiment is 4000rpm.

6. Conclusion
(1) At a safe working speed, the aerodynamic design and strength check of the drive mechanism of the static pressure spindle experiment table were performed. The Mises stress was displayed as MPa, which met the requirements of the experiment table working speed.

(2) Based on the finite element method, the rotor system of the experiment table was modeled. The first three critical speeds and modes of the rotor system were studied, and the working speed of the experiment table was effectively avoided. Calculation, according to API617 Rule 2.1.3, the unbalance response verification is qualified; the stability analysis also meets the requirements.

(3) Based on the previous design and theoretical calculation of the experimental bench, a turbine static pressure spindle experimental bench was set up to rationally arrange the various sensors. Prepare for subsequent experiments with air foil bearings.

(4) The lift off speed of the five foil aerodynamic bearing was measured on the experimental platform, and the lift off speed of the bearing was 4000 rpm.
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