Design and analysis of 5 kW helium turbine for EAST cryoplant

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Abstract: The helium turbine is an important core component in the refrigerator, which provides support and guarantee for the stable operation of the cryogenic system. This paper refers to the parameters of one of the four helium turbines in EAST, designs a 5 kW helium turbine. The overall structure of the turbine has been described, and the design and analysis of the impeller and bearing have been carried out. Through CFD analysis, the internal pressure and temperature change law of the flow is analyzed. The variable-condition operating characteristics of the turbine are predicted, and it is found that the turbine can maintain efficient operation within a wide range of mass flow changes. The bearing performance is calculated by solving the Reynolds equation. After determining the optimal bearing clearance 28 μm based on the principle of maximum stiffness, bearing performance parameters of load capacity and stiffness can be further calculated.

1. Introduction

Nowadays, many countries have built large-scale scientific devices to provide support for mankind to explore solutions to energy problems and expand basic physical theories [1-2]. The cryogenic system is an indispensable part of many large scientific devices, and large helium refrigerators have a wide range of applications in the field of international thermonuclear fusion research [3-4]. During the fusion experiment, a pulsed thermal load is generated during plasma discharge, causing sudden changes in the operating conditions of the cryogenic system, and also impacting the operation of the helium turbine [5].

The EAST device built in China was successfully discharged for the first time in 2009 and occupies an important position in the field of international thermonuclear fusion research [6]. In the EAST experiment, the load of the cryogenic system is not constant. During the cooling process of the magnet, the low-temperature system has a heavy load. When the cooling of the magnet is completed, the low-temperature system is switched to maintain operation. The helium turbine in the refrigerator is the main cold-producing component. Therefore, its adaptability to variable working conditions plays an important role in the stable and efficient operation of the cryogenic system.

This paper refers to the parameters of one of the four helium turbines in EAST, improves the mass flow, and designs a 5 kW helium turbine with good adaptability to variable working conditions.
2. Structural design

2.1 Turbine machine structure
This article adopts a vertical structure design, from bottom to top, flow part (nozzle, impeller), impeller end support structure, lower bearing and eddy current brake device, upper bearing and end positioning cover plate. The upper and lower bearing seats are fastened to the cooling cavity structure of the eddy current brake device by long bolts, and the bearing seats are provided with air supply holes and are machined with an annular cavity. An annular O-ring is installed at both ends of the bearing, which realizes an air supply structure with the annular cavity after being placed in the bearing seat. The O-ring bearing structure can also play a role in sealing the bearing gas to prevent shaft end leakage, and at the same time realize the flexible detachable function of the bearing and the rotor.

The designed turbine uses an eddy current brake device. The eddy current braking is a non-contact braking method. It uses the principle of electromagnetic induction to generate induced eddy currents on the eddy current ring, thereby applying a magnetic torque opposite to the direction of movement to the rotor, thereby maintaining the stable operation of the rotor. From the point of view of energy balance, the gas does work on the impeller through the flow, which is converted into Joule heat on the vortex ring through the eddy current brake device. And this part of Joule heat is exported through the circulating cooling water, so as to realize the energy conservation in the turbine.

2.2 Impeller design
The design of the three-dimensional (3D) impeller is the focus of the design of the flow part and the starting point of the entire design work. In the initial stage of the design, it can be assumed to be a one-dimensional steady-state adiabatic flow. Select initial parameters (such as expansion ratio, inlet temperature, pressure, etc.) based on design requirements, and then perform trial calculations on impeller geometric parameters. The initial parameters and trial calculation parameters are shown in table 1. Then, a two-dimensional design was carried out, and the meridian flow channel was adjusted by adjusting the hub and the streamline of the back of the wheel. The 3D design of the impeller is carried out by adjusting the space bending and torsion modeling of the blade, and setting the thickness of the leading edge and the trailing edge of the blade.

Since the input parameters of the nozzle are related to the height of the impeller, it is necessary to design the nozzle after the impeller parameters are determined. The nozzle is a guiding device that accelerates the gas flowing in the radial direction and enters the impeller at a designed angle. For the gas flowing at subsonic speed, the diverging flow channel is adopted, and for the gas flowing at supersonic speed, the zooming type is adopted.

The final designed impeller and nozzle parameters are shown in table 2, and the 3D geometric
structure is shown in Figure 2.

**Table 1.** The initial parameters and trial calculation parameters.

| Parameters       | Values | Parameters       | Values |
|------------------|--------|------------------|--------|
| Inlet temperature| 22 K   | Helium mass flow | 110 g/s|
| Inlet pressure   | 6.0 bar| Speed ratio      | 0.62   |
| Outlet pressure  | 1.2 bar| Wheel diameter ratio | 0.45 |
| Outlet pressure  | 1.2 bar| Enthalpy drop ratio | 0.51 |

**Table 2.** Geometric parameters of impeller and nozzle.

| Parameters                  | Values    | Parameters                  | Values    |
|-----------------------------|-----------|-----------------------------|-----------|
| Impeller inlet diameter     | 38.2 mm   | Impeller outlet diameter    | 23.5 mm   |
| Impeller inlet blade angle  | 90°       | Impeller outlet blade angle | 35.6°     |
| Impeller inlet blade height | 1.4 mm    | Impeller outlet blade height| 6.5mm     |
| Nozzle inlet blade angle    | 90°       | Nozzle outlet blade angle   | 17.8°     |
| Number of impeller blades   | 12        | Number of nozzle blades     | 21        |
| Impeller outlet temperature | 12.9 K    | Design impeller power       | 5.1 kW    |
| Impeller speed              | 108000 rpm| Isentropic efficiency       | 85.7%     |

**Figure 2.** The 3D geometric structure diagram of impeller and nozzle.

2.3 Bearing structure
The bearing is one of the core components of a helium turbine. It is used to support the rotor structure and is an important guarantee for the stable operation of the turbine. The working temperature zone of the helium turbine impeller is generally lower, and the rotor speed is higher. Taking into account the purity of the helium system and the demand for high-speed lubrication, helium turbines are generally lubricated with helium. The eddy current brake helium turbine designed in this paper is intended to use helium static pressure to lubricate the bearings. The bearing structure is shown in Figure 3, and the
bearing parameters are listed in Table 3.

![Figure 3. Structure of the bearing.](image)

**Table 3. Helium lubricated bearing parameters.**

| Parameters                              | Values                              |
|-----------------------------------------|-------------------------------------|
| Bearing diameter ($D$)                  | 22 mm                               |
| Bearing length ($L$)                    | 6.0 bar                             |
| Supply orifice diameter ($d_s$)         | 0.25 mm                             |
| Number of the supply orifices (row×column) | 2×8                               |
| Atmospheric pressure ($p_a$)            | 1 bar                               |
| Supply pressure ($p_s$)                 | 6 bar                               |
| Helium viscosity ($\mu$)                | $1.9793 \times 10^{-5}$ Pa·s        |
| Helium density ($\rho$)                 | 0.16412 kg/m³                      |
| Isentropic exponent of helium ($\kappa$) | 1.67                               |
| Temperature (T)                         | 293 K                               |
| Rotational speed ($\omega$)             | 108000 rpm                          |

3. Analysis and discussion

3.1 Impeller flow characteristics

Based on ANSYS CFX, the designed impeller and nozzle are simulated and analyzed. It can be seen from the 3D structure that the impeller and nozzle have the characteristics of periodic flow passages. Therefore, single-passage model can be used for Computational Fluid Dynamics (CFD) simulation, and the mesh of the impeller is shown in Figure 4. For more accurate results, a mesh boundary layer should be set near the wall of the blade, hub and shroud. The model equations involved in the simulation can be found in ANSYS solver theory, which are not listed here.

The pressure distribution cloud diagram of the impeller blade pressure surface, suction surface and flow channel meridian surface is shown in Figure 5. It can be seen that along the flow direction, the pressure at the inlet of the impeller is higher, and the pressure in the flow channel gradually decreases. Except for a low pressure area near the inlet of the suction surface affected by the turbulence, on the
whole, the pressure change in the flow channel is relatively uniform, and there is no obvious pressure sudden change area. It shows that the air flow in the flow channel is relatively smooth, without blockage and obvious shock wave loss. At the same position of the streamline, the pressure surface has the highest pressure, the meridian surface is the second, and the impeller suction surface has the lowest pressure.

Figure 4. Schematic diagram of meshing in the channel.

Figure 5. Distribution of pressure clouds in the impeller flow channel.

Figure 6. Variation of pressure at different streamwise locations.

Figure 7. Variation of temperature at different streamwise locations.

Figure 8. Impeller power changes with mass flow.

Figure 9. Isentropic efficiency changes with mass flow.
Figure 6 and 7 respectively illustrate that the pressure and temperature vary with the streamwise location, and the values are processed by cross-section weighted average. Streamwise 0–1 means nozzle inlet to outlet, streamwise 1–2 means impeller inlet to outlet. It can be seen that the pressure drop in the nozzle is relatively large, and the pressure change in the impeller is also mainly concentrated in the first half. This shows that the gas accelerates in the nozzle, converts pressure potential energy into kinetic energy, and enters the impeller to push the impeller to do work. The temperature distribution of the flow part is similar to the pressure distribution. The simulation shows that the outlet temperature of the impeller is 13.16 K, and the isentropic efficiency is 84.5%. Although it is slightly different from the design result, it is still within the acceptable range.

For eddy current brake helium turbines, a constant speed adjustment method is generally adopted. Figures 8 and 9 reflect the variable operating performance of turbine power and efficiency under different mass flow. When the mass flow is lower than 44 g/s, the impeller power is less than 1 kW. In general, during actual operation, the impeller power will not be lower than 1 kW. On the whole, the power of the impeller increases with the increase of the flow rate. And when the mass flow is greater than 44 g/s, the linear relationship is more obvious. From the change characteristics of the isentropic efficiency, the turbine can be operated at a higher efficiency when the flow rate is greater than 44 g/s. It shows that the designed impeller has good adaptability to variable working conditions.

3.2 Bearing performance

Since the clearance between the rotor and the bearing is extremely small, its flow can be regarded as isothermal and laminar movement, and its pressure distribution can be determined by the following dimensionless Reynolds equation.

\[
\frac{\partial}{\partial \varphi} \left( PH^3 \frac{\partial P}{\partial \varphi} \right) + \frac{\partial}{\partial \lambda} \left( PH^3 \frac{\partial P}{\partial \lambda} \right) + Q \delta_j = \lambda \frac{\partial (PH)}{\partial \varphi}
\]  

(1)

On the node without an orifice, \( \delta = 0 \). On the supply orifice, \( \delta = 1 \).

The dimensionless parameter process is as follows:

\[
P = \frac{P}{P_a}, \quad H = \frac{h}{c}, \quad \varphi = \frac{x}{R}, \quad \lambda = \frac{z}{R}, \quad \Lambda = \frac{6 \mu \omega R^2}{R p_c^2}, \quad Q = \frac{12 \mu R^2}{R p_c^2 R_a^2 \rho \bar{v}}
\]

Where \( P \) is the helium film pressure, \( H \) is film thickness, \( \Lambda \) is the bearing number, \( \varphi \) is circumferential angle, \( \lambda \) is dimensionless bearing length, \( c \) is the radius clearance, \( \bar{v} \) is velocity entering the orifice, and the other terms have been defined in Table 3.

By the Galerkin finite-element method, the Reynolds equation can be solved. Obtained pressure field distribution, the characteristic parameters can be further calculated. The dimensionless load component can be acquired by the following integral formula.

\[
\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \int_0^{\lambda_R} \int_0^{2\pi} \left( P - 1 \right) \begin{bmatrix} \cos \varphi \\ \sin \varphi \end{bmatrix} d\varphi d\lambda
\]

(2)

Where \( F_x \) and \( F_y \) respectively represent the component forces of the bearing.

The load capacity is described by

\[
W = R^2 p_a \sqrt{F_x^2 + F_y^2}
\]

(3)

The stiffness is calculated by

\[
K = \frac{dW}{d \varepsilon}
\]

(4)

Where \( \varepsilon \) is the eccentricity ratio.

The rotor-bearing clearance is an important parameter that affects the performance of the bearing. Under the eccentricity ratio is 0, the stiffness under different clearances is calculated. Select the maximum rigidity divided by the corresponding bearing clearance as the optimal clearance of the turbine bearing. It can be seen from Figure 10 that the optimal clearance of the bearing is about 28 \( \mu \text{m} \). After
determining the optimal bearing clearance, the load characteristics and stiffness characteristics of the bearing at rated speed are further predicted.

Figure 11 shows the variation of load with eccentricity ratio. It can be seen that as the eccentricity ratio increases, the load increases rapidly. Especially when the eccentricity ratio is high, the load growth rate is greater due to the appearance of the dynamic pressure effect. The stiffness characteristics of the bearing show the law of first decreasing and then increasing, as shown in Figure 12. This shows that when the eccentricity is less than 0.5, the static effect dominates, causing the stiffness to decrease with the increase of the eccentricity ratio. When the eccentricity ratio is greater than 0.5, the dynamic pressure effect becomes more and more obvious, so that the stiffness increases with the increase of the eccentricity ratio.

4. Conclusion
This paper designs an eddy current brake helium turbine for large refrigerator. The overall structure of the turbine has been described, and the design and analysis of the impeller and bearing have been carried out.

The turbine can reduce the pressure of the helium from 6.0 bar to 1.2 bar, and the temperature from 22 K to 12.9 K. The designed turbine can provide about 5 kw of cooling capacity for refrigerator. Furthermore, the variable-condition operating characteristics of the turbine are predicted through CFD simulation, and it is found that the turbine can maintain high-efficiency operation when the mass flow rate is greater than 44g/s.

The bearing performance is calculated by solving the Reynolds equation. After determining the
optimal bearing clearance 28 μm based on the principle of maximum stiffness, bearing performance parameters of load capacity and stiffness can be further calculated.

5. References

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