The Study on Dynamic Characteristics of Twin-Screw Compressor Rotor

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ABSTRACT

In the working process of twin-screw compressor, the rotors are subjected to multiple physical effects of the gas temperature, pressure and force, and presents a periodic change. In this paper, three-dimensional Computational Fluid Dynamics (CFD) simulation of screw compressor is carried out, and the characteristics of temperature distribution, pressure distribution and gas force distribution on the rotors' surface are studied. Firstly, the rotor domain, suction and exhaust face grids are generated by TwinMesh, and then they are imported into the CFX calculation model. Through the fluid-structure interaction analysis and calculation, the strain shape of the screw rotor under the multiple alternating physical action of gas temperature, pressure and force is analyzed, and the deformation of the rotor structure caused by the pressure and temperature in the working process is obtained, which plays a good guiding role in the design of the screw rotor and the improvement of the performance of the compressor.

1. INTRODUCTION

Screw compressors can be divided into two types: single-screw compressors and twin-screw compressors. Twin-screw compressors are composed of a pair of female and male rotors that mesh with each other in parallel. Screw compressors are widely used in industrial production and chemical industries due to their compact structure, stable operation, convenient maintenance, and long life. In recent years, screw compressors have developed rapidly: on the one hand, screw profiles and structural designs have Made great progress. On the other hand, the introduction of a screw rotor special milling machine, especially a grinding machine, has improved the processing accuracy and efficiency of the key parts, and effectively improved the performance of the screw compressor.

In recent years, many scholars at home and abroad have done a lot of research on screw compressor. N. Stosic et al. presented some results of mathematical modelling and experimental investigation of the influence of oil injection upon the screw compressor working process. Xing analyzed the screw rotor meshing conditions and gave a general method for the design of the female and male rotor profile. N. Stosic et al. introduced a suitable procedure for optimization of the screw compressor shape, size, dimension and operating parameters, which results in the most appropriate design for a given compressor application and fluid. Cai et al. provided a simplified method for calculating the contact line shape and length of screw compressor. D. Zaytsev et al. deduced a forming method of rotor profile of twin-screw compressor. LIU et al. simulated the lubrication problem between the rotor and the bearing. An studied the influence of rotor speed on the selection of screw compressor. Li et al. verified the speed transmission ratio of screw rotor. Zhao et al. made a dynamic simulation study on the working condition of a certain type of screw compressor by using virtual prototyping technology. Seshaish. N. et al. studied the influence of internal pressure and leakage of oil injected compressor. Wu et al. presented a design method for the rotor profile of twin-screw compressor from an arbitrary sealing line. Ahmed El Shorbagy designed the rotor tooth profile of twin-screw compressor, and analyzed the three-dimensional flow field of the tooth profile by using CFX and TwinMesh. Feng et al. proposed the mean pressure model and sector pressure model to calculate the axial force on the rotor of the twin-screw refrigeration compressor. Rane et al. studied a twin-screw compressor and found that the refinement of the grid along the circumferential direction of the rotor profile directly affected the prediction of mass flow. Li et al. studied the rotor structural characteristics of twin-screw compressor under fluid-solid coupling.

In summary, at present, there are many studies on the rotor profile and speed of screw compressor, but there are few studies on the dynamic characteristics of the rotor. In this paper, through the computer simulation of the twin-screw refrigeration compressor, the force and deformation of the screw rotor are studied and analyzed.

2. METHODS

In this paper, the twin-screw refrigeration compressor is simulated by CFD, and the heat flux module of CFX in ANSYS Workbench is used for calculation. The rotor segment is meshed by TwinMesh software, and the inlet and outlet part is modeled by SolidWorks and imported into ANSYS Mesh (ICEM) for meshing. The simulation solution is solved by 12-core parallel operation, which reduces the time needed to solve the problem.

Based on the actual working conditions of twin-screw refrigeration compressors, a fluid-solid thermal coupling model is established in this paper. CFX is used to calculate the flow field distribution of the twin-screw compressor during operation, and the temperature distribution of the female and male rotors during operation are calculated by fluid-solid thermal coupling. Then the pressure field distribution calculated in the CFX flow field is imported as a result file into the ANSYS statics module to calculate the force and deformation characteristics of the female and male rotors in the working process.

Table 1 shows the detailed structural parameters of the twin screw refrigeration compressor. The ratio of the rotor length to the outside diameter of the male rotor (L/D) was 1.09742. The theoretical volume flow rate of the compressor was 3.797 $m^3 \cdot min^{-1}$.

Table 1: Structural parameters of the screw compressor

Parameters	Male Rotor	Female Rotor
Number of Teeth	5	6
Diameter of Rotor D/mm	138.507	109.76
Length of Rotor L/mm	152	152
Wrap Angle of Male Rotor	300 °	250 °
Screw Lead of Male Rotor T/mm	181.761	218.113

3. SIMULATION SETTINGS

3.1 Boundary Condition Settings

The turbulence model in this paper was the SST k-ω model, which combines the advantages of the k-ω model in the near-wall region calculation and the advantages of the k- ε model in the far-field calculation. The refrigerant was R134a in twin-screw compressor. The inlet and outlet temperatures and pressures is shown in Table 2. To simplify the calculation, the heat transfer of the oil was not considered in the CFD simulation. The rotor speed was 3000 rpm.

Table 2: Boundary conditions of computational fluid dynamics (CFD) simulation

Parameters	Value
Inlet Pressure	0.285 MPa
Inlet Temperature	274 K
Discharge Pressure	1.505 MPa
Discharge Temperature	355 K
Volume flow	3.254 $m^3 \cdot min^{-1}$

In the simulation, it is also necessary to set the pressure of the initial flow field in the twin-screw compressor. In this paper, the pressure of the initial flow field is set as the inlet pressure of the compressor, and the reference pressure is 0 bar. All wall surfaces of screw compressors are non-slip wall surfaces.

3.2 Meshing

As the structure of the female and male rotor is relatively complicated, this paper focuses on the deformation and force of the rotor under the condition of heat and force. Therefore, the model of the rotor is simplified. The grid of the rotor segment needs to establish a structural grid with the same number of grid points, so TwinMesh is used to mesh the rotor segment. The rotor grid is divided as shown in Figure 1. The total number of the female and male rotor grids is 1344401.

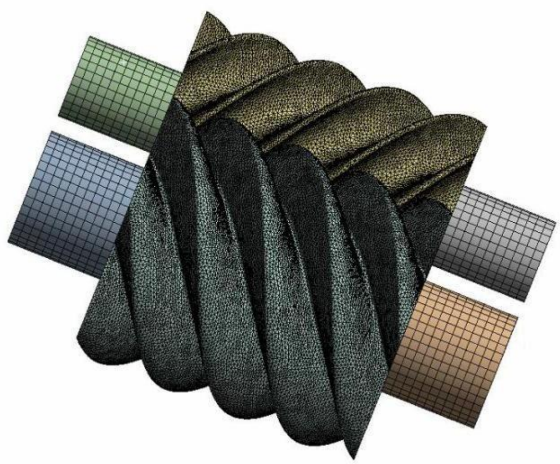

Figure 1: The rotor meshing

4. RESULTS

The effect of the pressure and temperature distribution inside the compressor on the efficiency and performance of the compressor is a topic for engineers in the compressor field. The distribution of pressure and temperature in the compressor was calculated by the CFD software. Figure 2 shows the distribution of pressure and temperature with the male rotor and female rotor. In the process of working, the pressure on the rotor surface is greatest near the exhaust orifice.

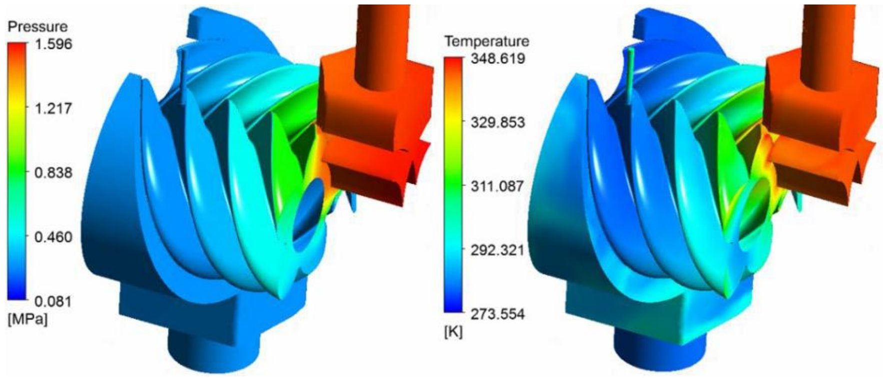

Figure 2: The distribution of pressure and temperature inside the compressor

During the working process of the screw compressor, the rotor is deformed due to the change of temperature and rotor force. The deformation of the twin-screw compressor rotor under three different conditions: pressure field, temperature field and pressure temperature field were analyzed in this paper.

4.1 Pressure Field

In the working process of twin-screw compressor, the pressure distribution in different tooth grooves is constantly changing, so the forces on the male and female rotors of the compressor are constantly changing.

The pressure field distribution calculated in the CFX flow field is imported into the ANSYS statics module as a result file. Figure 3 is the force diagram of the male rotor. In the working process, the axial force on the rotor, the radial force on the suction end and the radial force on the discharge end all show periodic changes. Figure 4 is the force diagram of the female rotor. The force change of female rotor is similar to that of male rotor, but the overall force is smaller than that of male rotor.

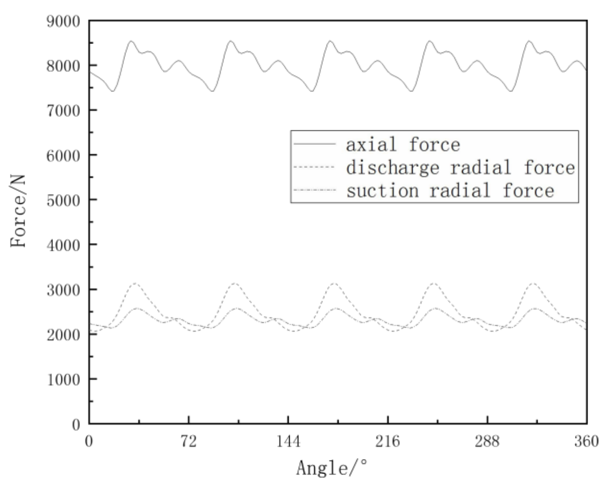

Figure 3: Male rotor force

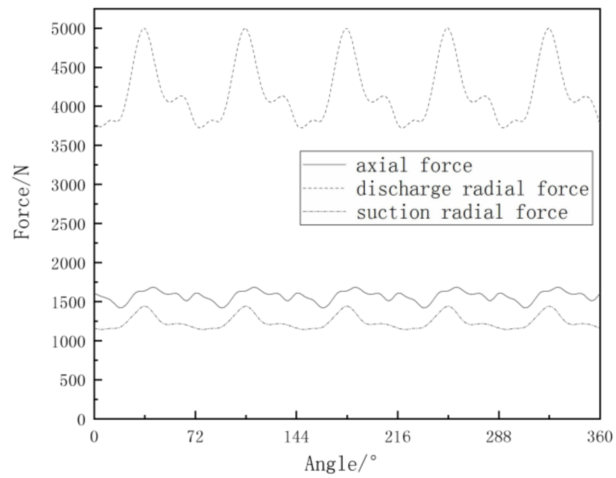

Figure 4: Female rotor force

Figure 5 shows that the internal torque of the rotor varies with the angle of the male rotor. And the power change of the rotor is shown in Figure 6.

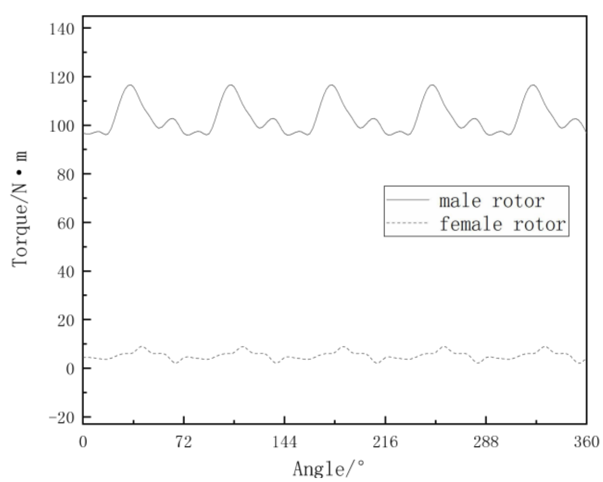

Figure 5: Rotor internal torque

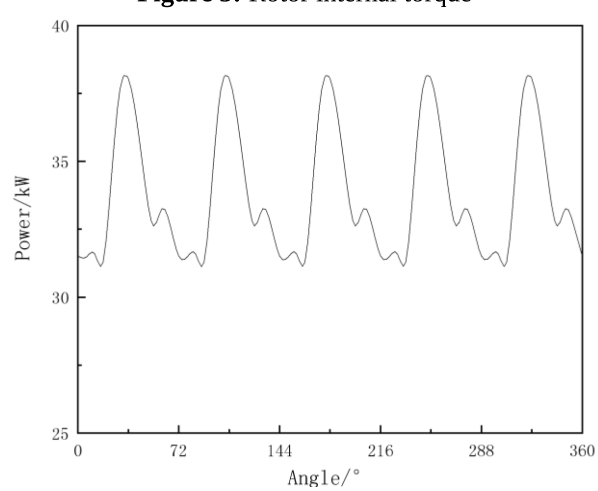

Figure 6: Power change

The forces on the female and male rotor have similar changes. With the increase of the rotation angle of the male rotor, the forces of female and male rotor change little. Among the axial force of female and male rotor and the radial force of suction and discharge end, the axial force of male rotor is the largest, and its maximum value can reach 8548N.For the internal torque of the male rotor, with the increase of the angle of rotation of the male rotor, its value gradually increases and then decreases. For the internal torque of the female rotor, its value basically does not change with the increase of the rotation angle of the male rotor. In addition, the internal gas torques of the female and male rotors are both positive value, which indicates that both torques are gas resistance moments. The power also increases at first and then decreases, and the maximum value reaches 38.16kW.
The following uses the rotor at four different angles of 12°, 24, 36°and 60°as examples to calculate the deformation of the rotor as shown in Figure 7.

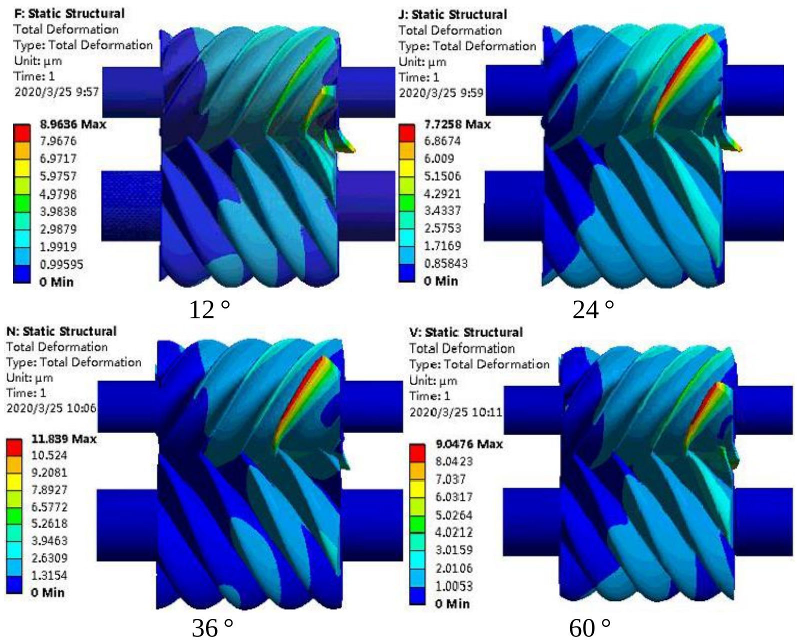

Figure 7: Deformation of the rotor at different angles

It can be seen from Figure 7 that the deformation of the female rotor is greater than that of the male rotor at different angles. This is because the teeth of the male rotor are thicker than the teeth of the female rotor, and the tooth stiffness of the male rotor is better. Moreover, the deformation of the rotor near the discharge end is larger than that of the suction end. This is because the compressor tooth groove has a higher pressure near the discharge end.
Through fluid-solid coupling calculation, the maximum deformation of the rotor under the pressure field varies with the angle of the male rotor as shown in Figure 8.

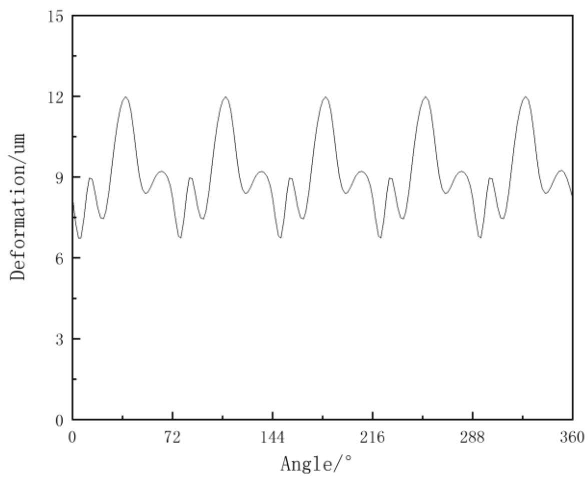

Figure 8: Deformation of the rotor in pressure field

It can be seen from Figure 8 that the maximum deformation of the rotor changes continuously with the change of the angle of the male rotor. When the angle of the male rotor reaches 36°, the deformation of the rotor reaches the maximum of 11.84μm.

4.2 Temperature Field

The change of temperature will lead to thermal stress in the working process of rotor. The female and male rotor will have thermal deformation under the temperature field.

When only affected by the temperature field, the deformation of the rotor of the twin-screw compressor at different angles is shown in Figure 9. And the maximum deformation of the rotor is obtained by fluid-solid coupling calculation as shown in Figure 10.

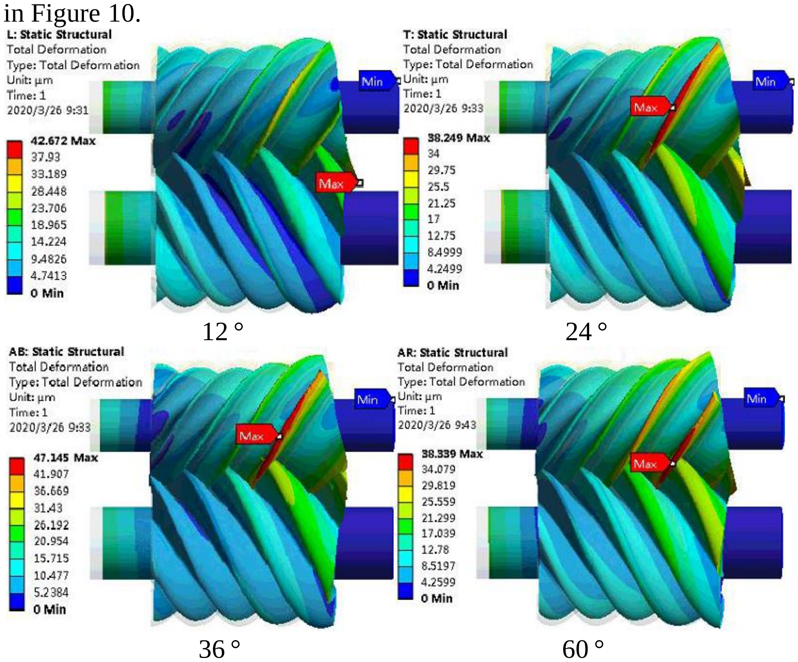

Figure 9: Deformation of the rotor at different angles

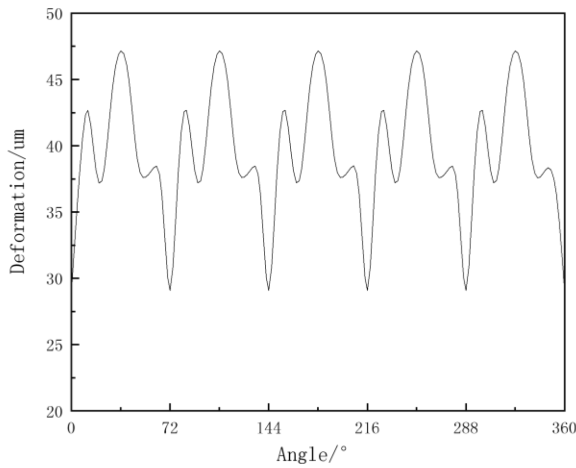

Figure 10: Deformation of the rotor in the temperature field

As a whole, under the temperature field, the thermal expansion and deformation of the male rotor near the discharge end is smaller than that of the female rotor, and the shrinkage deformation of the male rotor near the suction end is larger than that of the female rotor. It can be seen from Figure 10 that the maximum deformation of the rotor changes periodically. When the angle of the male rotor reaches 36 °,the deformation of the rotor reaches the maximum of 47.15μm.

4.3 Pressure and Temperature Field

In the working process of the twin-screw compressor, the force and deformation of the female and male rotor are affected by the temperature field and pressure field. The research on the rotor under the joint action of the two physical fields is closer to the actual situation of the rotor in the working process.

The deformation of the rotor of twin-screw compressor at different angles under the combined action of temperature and pressure fields is shown in Figure 11. As a whole, when the temperature and pressure fields are combined, the deformation of the female rotor is still larger than that of the male rotor. The maximum deformation position of the rotor at different angles is about 36°. This is due to the end of the compressor's external compression process at this time, the discharge temperature and discharge pressure have reached the maximum, resulting in the largest deformation there. The maximum deformation of the rotor is located on the female rotor, and the maximum deformation position is around the discharge orifice, so the deformation of the rotor is mainly affected by the discharge pressure and discharge temperature.

The maximum deformation of the rotor is obtained by fluid-solid coupling calculation as shown in Figure 12. When the angle of the male rotor reaches 36°, the deformation of the rotor reaches the maximum of 56.69μm.

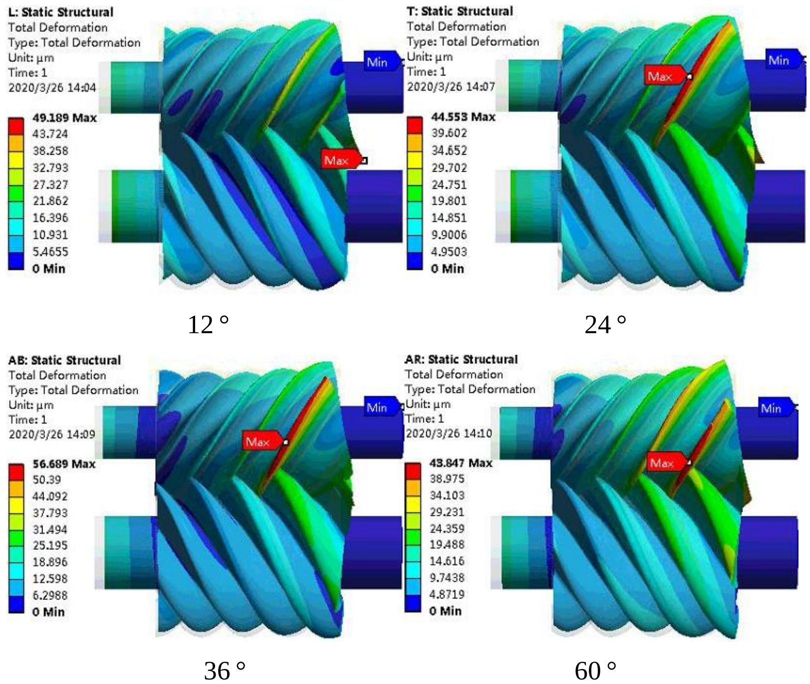

Figure 11: Deformation of the rotor at different angles

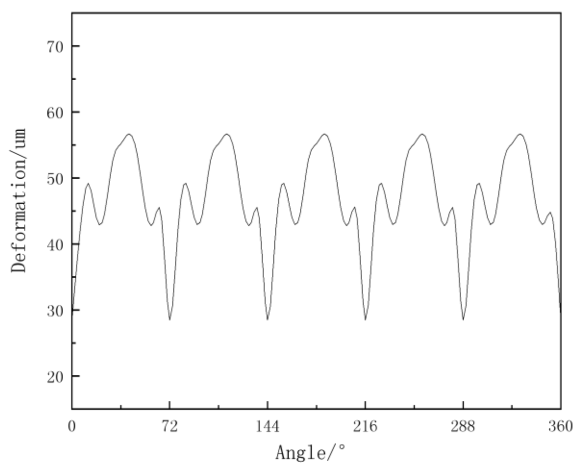

Figure 12: Deformation of the rotor in the pressure and temperature field

5. CONCLUSIONS

In this paper, the strain shape of the screw rotor under the multiple alternating physical action of gas temperature and pressure is analyzed, and the deformation of the rotor structure caused by the pressure and temperature in the working process is obtained. Our results can be concluded as follows.

Under the action of gas pressure, the force of the female and male rotor shaft of the twin-screw compressor is obtained by calculation. Under this condition, the maximum axial force on the male rotor is about 8500N, the maximum radial force on the discharge end is about 3100N, and the maximum radial force on the suction end is about 2500N. The radial force on the discharge end of the female rotor is the largest, and the maximum value is about 5000N. The maximum axial force is about 1700N and the maximum radial force at the suction end is about 1400N.

Through the analysis of the force and deformation of the rotor of the compressor under the combined action of pressure field and temperature field, it can be seen that the deformation of the rotor is mainly affected by the internal temperature distribution of the compressor and tends to expand at the discharge end and shrink at the suction end. Under the combined action of the two physical fields, the maximum deformation of the rotor is 56.69 μm. In this paper, the research on the force and deformation of the rotor provides a theoretical basis for the bearing selection and minimum clearance design of the twin-screw compressor, and the study has important guiding significance for improving the performance and structural safety of the twin-screw compressor.

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