Thermo-mechanical-dynamic coupling analysis on cold compressor rotor

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Abstract. Cold compressor rotor works in complex thermal filed, which endures huge temperature differences from room temperature to low-temperature. To study the effect of thermal field on deformation and dynamics of cold compressor rotor, a thermo-mechanical-dynamic coupling analysis is carried out by finite element software ANSYS. Firstly, a thermal analysis on the entire compressor is carried out by CFX and the temperature distribution is obtained. Secondly, a static structural analysis considering thermal loads and contact effect is performed. The thermal stress and deformation are obtained. Finally, a pre-stress modal analysis in working thermal field is carried out. The results show that: The axial distance between blade tip and thrust disc reduces by 117.41 µm. The first two natural frequencies of the cold compressor rotor have a slight improvement by less than 2.16% and higher order natural frequency is more sensitive to thermal effect.

1. Introduction

The cold compressor is one of the core components of large-scale helium refrigeration system and its rotor works in the complex thermal field [1-2]. The temperature field can cause thermal stress and deformation. In addition, the thermal field could change the elasticity modulus of rotor. Both of two factors above may contribute to change of dynamics. Many studies in thermal effect on rotor dynamics have been done by simplified rotor models [3-5]. Some studies usually are focused on high temperature rotating machinery, for example steam turbine [6-7] and combustion gas turbine [8]. Thermal stress and deformation arise inevitably since steam turbines and combustion gas turbines work in high temperature environments. Elasticity modulus decreases with the increasing of temperature. These factors contribute to deteriorating of dynamic characteristics in high temperature rotating machinery. But the situation is more complicated for cold compressors. The parts of rotor including motor and the magnetic bearings work at above room temperature, which is slightly more than reference temperature 293 K. The part including impeller works at greatly less than reference temperature. Therefore, it is necessary to investigate the deformation and dynamic characteristics of the cold compressor rotor at the working temperature field.

In this paper, thermal deformation and dynamics of the cold compressor in the complex temperature field are studied by thermo-mechanical-dynamic multi-physics coupling analysis. Firstly, the temperature field distribution of the rotor system is figured. Then, a static structural analysis is
performed. Lastly, the pre-stress modal analysis is carried out and natural frequencies considering thermal field and contact effect are obtained.

2. Steady-state thermal field of cold compressor rotor
The schematic diagram of a cold compressor is shown in Figure 1. The cold compressor rotor is suspended by active magnetic bearing (AMB) and driven by high-speed induction motor. The water jacket is designed to cool motor and magnetic bearings. Thrust magnetic bearings are used to adjust the axial position of rotor. Heat leakage in cold compressor from room temperature to low temperature is an urgent problem [2]. In order to reduce axial heat leakage, unique structure is adopted for cold compressor. Firstly, a portion of the rotor is designed as vacuum hollow structure. Secondly, the thermal anchor maintains at a certain temperature and thus could take a lot of heat. Thirdly, Thermal insulation material is adopted outside the casing. All of three factors mentioned above promote the maintenance of huge temperature difference along the axial direction. According to the temperature level, it can be divided into three temperature zones including the room temperature zone, low temperature zone and transition temperature zone. There is an axial temperature gradient more than 300 K. In detail, the motor and magnetic bearings are major heat production units, whose temperature is slightly higher than the room temperature. The impeller compresses the cryogenic helium gas, lowest temperature of which is less than 5 K.

![Figure 1. Schematic diagram of cold compressor.](image)

The whole compressor is taken as the computational domain. The number of elements in domain is 9.3 million and the grids have been verified by independence. The heat transfer coefficients between shell and ambient are set to 10 W/(m²·K). The mass flow rate in cooling water jacket is 20 g/s and inlet temperature is 283 K. The heating sources include magnetic bearings and motor. The heat sources for each part are shown as table 1. The rotor term for heating units include rotor of magnetic bearings and motor. The helium gas is 20 kPa in gap between rotor and stator. The viscous work terms are including.

| Parts                      | Motor Stator | Upper AMB Stator | Lower AMB Stator | Thrust AMB Stator | Rotor |
|----------------------------|--------------|------------------|------------------|-------------------|-------|
| Heating source (W)         | 279.0        | 27.2             | 27.2             | 7.6               | 139.0 |
The whole cold compressor temperature is calculated by CFX and the temperature contour of cold compressor rotor is extracted and showed in figure 2. The temperature distribution along axial direction is extracted and shown in figure 3. The origin of coordinates is located at center of rotor left end. The extracted temperature is along centre line in solid section and wall in vacuum wall section. The maximum temperature at 316.22 K appears in the middle of the motor rotor and the lowest temperature at 5 K appears on the impeller. There is a large axial temperature gradient mainly in the vacuum hollow segment, while the radial temperature gradient is not large. The axial temperature gradient could lead to changed elastic modulus in the axial direction and further change the rotor dynamics.

3. Stress analysis of cold compressor rotor in thermal field

3.1 Contact analysis between components and shaft

Contact is a kind of interference behaviour between parts. Contact analysis often consumes a lot of computer resources, and the appropriate model is the key to efficiently solving the problem [9-11]. The contact setting has an important effect on the stress solution. The motor rotor and the magnetic bearing rotor are interference fitted on shaft. The torque is transmitted through the friction between rotors and shaft. When friction is involved, some friction laws and models need to be considered, and all of them are non-linear. The friction response may be chaotic and solution is difficult to converge. The methods applied in this paper consider the interference friction contact effect between the sleeve and shaft. In contact analysis, a 3D surface-to-surface model is built. The contact pair includes target face and contact face. The outer surface of the shaft is taken as contact face. Motor and magnetic bearings rotors are taken as target face. Some of the contact settings are as follows:

a. The augmented Lagrange method.
b. Contact detection is located at Gauss integration point.
c. Initial penetrations including geometry penetration and offset.
d. Normal penalty stiffness factor 1.
e. Penetration tolerance factor 0.1.

The amount of interference is a key factor to determine contact stress. Material and temperature also have an important influence on stress. According to section 2, thrust disc, silicon steel sheets and sleeve are in room temperature zone. Impeller is in cryogenic temperature zone. In order to determine appropriate amount of interference, the stress analysis under different amount of interference are performed and shown in Figure 4. The max equivalent stress increases with the increasing of amount of interference. In order to avoid structure damage, the amount of interference should be controlled within a certain range. At the same time, the parts and the shaft should also ensure a certain amount of interference so that there can transmit a certain friction torque. To protect the max equivalent stress less than 200 MPa, the amount of interference for thrust disc, silicon steel sheets, sleeve and impeller can be 0.01 mm, 0.016 mm, 0.016 mm, 0.0025 mm respectively. The static structural analysis is carried out considering contact behaviour and working thermal field.
3.2 Thermal stress and deformation of rotor in thermal field

The temperature distribution of rotor is imported to static structure analysis. The stress solution of the cold compressor rotor is shown in Figure 5. The maximum equivalent (Von-Mises) stress is 193.36 MPa and happened at interference assembly and it is in a safe stress range. In transition segment with big temperature difference, maximum equivalent stress is about 94.78 MPa and it is complete thermal stress with same material due to non-uniform thermal field. Axial active magnetic bearings adjust the thrust disc to adjust the axial position of the rotor. Thermal deformation due to large axial temperature difference between thrust disc and blade tip reduces by 117.41 µm.

![Figure 5. Equivalent stress contour map of combined rotor in Non-uniform thermal field.](image)

4. The pre-stress modal analysis

The low temperature resistant steel alloy is utilized as shaft material. The elastic modulus of shaft material at 5 K increases 8.2% than at 320 K. A pre-stress modal analysis in thermal field and a pre-stress modal analysis in reference temperature are respectively carried out. The thermal effect on dynamic characteristics could be obtained by comparison of two methods. The calculation results are shown in table 2.

|                  | Room temperature | Thermal field | Difference (%) |
|------------------|------------------|---------------|----------------|
| 1<sup>st</sup> mode (Hz) | 1327.1           | 1342.1        | +1.13%         |
| 2<sup>nd</sup> mode (Hz)   | 2126.1           | 2172.0        | +2.16%         |
Considering the thermal effect of cold compressor rotor, the natural frequencies increase by less than 2.16%. Besides, higher order natural frequency is more sensitive to thermal field in cold compressor rotor. This has an opposite trend with steam turbines and combustion gas turbines, whose dynamic characteristics deteriorated in working thermal field. The first two order modal results considering thermal field and interference friction contact are shown in Figure 6.

![First order bending modal shape (1342.1Hz).](image1)

![Second order bending modal shape (2172.0Hz).](image2)

**Figure 6.** First two order modal shapes considering thermal field and interference fit.

5. **Conclusions**

A thermo-mechanical-dynamic multi-physics filed coupling analysis is carried out to obtain thermal effect on dynamic characteristics of cold compressor rotor. Following conclusions can be made: thermal deformation between thrust disc and blade tip due to large axial temperature difference reduces by 117.41 µm. The tip clearance becomes larger without considering the deformation of the volute. The dynamics of the cold compressor rotor have a slight improvement by less than 2.16%. Besides, higher order natural frequency is more sensitive to thermal effect.

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