Influence of cooling temperature on load-carrying performance of a radial HTS magnetic bearing

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Abstract. High temperature superconducting (HTS) magnetic bearing can operate in the harsh environment of low temperature and negative pressure with low loss and self-stability. Therefore, it has been gradually applied in flywheel energy storage, cryogenic liquid pump, cold compressor and some other cryogenic rotating machinery. The load-carrying performance of HTS magnetic bearing is affected by the cooling temperature of HTS directly. In this paper, the linear model is used to modify the temperature dependence to evaluate the influence of cooling temperature on load-carrying performance quantitatively, and the results are analyzed and displayed from the aspects of maximum levitation force, levitation stiffness and hysteresis effect. The research results of this paper have practical significance for the design and safe operation of HTS levitation system.

1. Introduction

High temperature superconducting (HTS) magnetic bearing can operate in the harsh environment of low temperature and negative pressure with low loss and self-stability. Therefore, it has been gradually applied in flywheel energy storage[1], cryogenic liquid pump[2], cold compressor[3] and some other cryogenic rotating machinery. At the same time, the HTS levitation system is a complex multi-field coupling system[4]. The cooling temperature of HTS will have a certain impact on its electromagnetic characteristics, and then it will affect the bearing capacity of the system. It is very important for the design and safe operation of HTS magnetic bearing to evaluate the influence of cooling temperature on load-carrying performance quantitatively.

Generally, the modeling and calculation of HTS levitation system is complicated, especially for the radial bearing with complex structure. Its complexity lies in solving the Maxwell equations including nonlinear electromagnetic constitutive relation of HTS and describing the changing external field of HTS under complex structure, which is equivalent to describing the relative motion of permanent magnet and HTS. Our team has proposed a simple and convenient method model for the calculation of axial force and radial force of radial HTS magnetic bearing[5, 6]. On the basis of these studies, this paper will further consider the temperature correction to evaluate the influence of cooling temperature on load-carrying performance of a radial bearing quantitatively.
2. Model establishment

2.1. Physical model

Our team has designed a radial HTS magnetic bearing[3]. The bearing adopts the form of internal permanent magnet rotor and external HTS stator. The permanent magnet rotor consists of alternating permanent magnetic rings and silicon steel sheets. The HTS stator is made of superconducting ring material produced by ATZ in Germany. Figure 1 shows the picture of permanent magnet rotor and HTS stator. The size of HTS stator is ID 50 mm × OD 80 mm × 60 mm. The size of the rotor is ID30 mm × OD48 mm × 70 mm, which consists of two permanent magnetic rings with the size of ID30 mm × OD48 mm × 2.8mm at both ends, nine permanent magnetic rings with the size of ID30 mm × OD48 mm × 5.6 mm in the middle, and several silicon steel sheets with the size of ID 30 mm × OD48 mm × 1.4 mm.

![Figure 1. Picture of permanent magnet rotor and HTS stator.](image)

2.2. Mathematical model

According to the theory of electromagnetic field, the governing equation of the solution domain is as follows when the displacement current can be ignored,

\[ \nabla \times (\rho \nabla \times H) + \mu_\epsilon \mu_0 \frac{\partial H}{\partial t} = 0 \]  

Where, \( \rho \) and \( \mu_\epsilon \) are the electrical conductivity and relative permeability of the materials in the corresponding domain. For the HTS domain, the relative permeability is 1. The conductivity is the virtual conductivity calculated according to the \( E - J \) relationship. In this paper, the power exponent model is used to describe the nonlinear electromagnetic constitutive relation.

\[ \rho_v = \frac{E}{J_c} \left( \frac{J}{J_c} \right)^n \]  

Generally, the critical current density is only considered as a function of the magnetic flux density \( B \). In this paper, we need to introduce temperature correction to evaluate the effect of cooling temperature. In the temperature correction model, the most commonly used is the linear model[4, 7-10], as shown in formula (3). It is worth noting that in the calculation of this paper, the heat exchange and diffusion of the HTS itself are not considered, and the temperature of the HTS is assumed to be uniformly distributed at each calculation temperature.
\[ J_c(B,T) = J_{c0} \frac{B_0}{B_0 + |B|} \frac{T_c - T}{T_c - T_0} \quad (3) \]

Where, \( E_c \) is the critical electric field strength; \( T \) is the temperature; \( T_c \) is the critical temperature of HTS; \( T_0 \) is the temperature of liquid nitrogen; \( J_c(B,T) \) is the critical current density; \( E_c, n, J_{c0} \), \( B_0 \) are the characteristic parameters of HTS. Finally, we can get the electromagnetic force by integrating the HTS domain.

\[ F = \iiint (J \times B) dV \quad (4) \]

2.3. Solution method

In this paper, the Maxwell equations above (with the magnetic field intensity \( H \) as the independent variable) are solved with the help of COMSOL Multiphysics® software. It is also a common method for many scholars to solve the problem of HTS in recent years. The key of the solution process is to apply a variable magnetic field boundary to the HTS domain, which is equivalent to describing the relative motion of permanent magnet and HTS. The traditional method is to describe the external field of permanent magnet changing with time and space based on analytical formula[11, 12], or calculate the steady-state magnetic field distribution first by finite element method and then couple it with the magnetic field of HTS in two directions[13]. These methods have some limitations or disadvantages. In reference[14], the moving mesh in COMSOL Multiphysics® is proposed to solve this problem, which facilitates the modeling and solving of this kind of problem greatly. Generally speaking, this paper also adopts the method of moving mesh and solves the equations above based on \( H \) method.

The specific modeling methods we adopted are as follows. When solving the axial force of radial bearing, we establish a two-dimensional axisymmetric model. When solving the radial force of radial bearing, we establish a three-dimensional model. The displacement function is defined to describe the motion process of the permanent magnet rotor and control the deformation of the mesh. The specific setting method is shown in reference[5, 6]. The model is shown in figure 2.

![Figure 2. Modeling diagram.](image)

3. Method validation

3.1. Verification of the levitation force calculation of radial HTS magnetic bearing

Our team has carried out some calculation and experimental research on the axial and radial force of radial HTS magnetic bearing. The specific parameters of the calculation model are introduced in reference[6]. The calculation and experimental results of radial force are also given in reference[6]. The comparison between the calculated and experimental results of axial force is shown in figure 3. In the actual working state, the displacement of the rotor of bearing is usually allowed to be in the range
of tens of microns, so the calculation in this paper is limited to the range of small displacement. Within the acceptable engineering error range, we think that the calculated value of the levitation force is basically consistent with the experimental value.

![Figure 3](image.jpg)

**Figure 3.** The comparison between the calculated and experimental results of axial force.

### 3.2. Verification of temperature correction model

In reference[15], the levitation force of a simple axial levitation system at different temperatures is measured experimentally. The relevant performance and experimental parameters are listed in table 1. According to the numerical method above, we simulated the experimental process. Figure 4 shows the comparison between the calculated and experimental values of the levitation force at the maximum displacement at different temperatures. In the calculation process, although some of the electromagnetic parameters are selected according to the empirical values, the calculated results are consistent with the experimental results generally. The results show that the linear temperature correction model can reflect the influence of temperature on the levitation force well within the acceptable range of engineering error.

| Parameter                          | Symbol | Value          |
|------------------------------------|--------|----------------|
| Remanent magnetism of PM           | $B_r$  | 1.05 T         |
| Critical current criterion         | $E_c$  | $1 \times 10^4$ V/m |
| Zero-field critical current density| $J_{c0}$ | $2.4 \times 10^8$ A/m² |
| Power law exponent                 | $n$    | 21             |
| Critical magnetic flux density     | $B_0$  | 0.37 T         |
| Size of HTS                        |        | $\varnothing 30 \times 14$ mm |
| Size of PM                         |        | $\varnothing 30 \times 15$ mm |
| The displacement range of the experiment |       | 150 mm-5 mm     |
4. Results and discussion

On the basis of the method above, we can study the load-carrying performance of our radial HTS bearing at different temperatures quantitatively. Figure 5 and 6 show the variation curves of calculated axial force and calculated radial force under different cooling temperatures at a given stroke, which show similar laws. The value of levitation force reaches its maximum at the maximum displacement. The lower the cooling temperature is, the greater the maximum levitation force is. Taking the levitation force at 75K as the standard, we calculated the relative proportion of levitation force at different temperatures, as shown in figure 7. It can be seen that the levitation force changes sharply with temperature between 70-80K, but it changes slowly and tends to saturation below 60K. The saturation levitation force below 60K is about 1.2-1.5 times of that at 75K. In the temperature range above 80K, the levitation force decreases rapidly. This is also confirmed by the experiment in reference[16], although they are levitation systems with different sizes and structures.

Figure 4. The comparison between the calculated and experimental values of the levitation force at the maximum displacement at different temperatures.

Figure 5. The variation curves of the calculated axial force of radial HTS bearing under different cooling temperatures.
Figure 6. The variation curves of the calculated radial force of radial HTS bearing under different cooling temperatures.

In addition to the maximum levitation force, cooling temperature also causes the stiffness difference of levitation force. But the difference is not obvious in the small displacement range. For example, in the range of radial displacement within 0.1 mm and axial displacement within 0.05 mm, the difference of levitation force is not obvious. With the increase of displacement range, the nonlinearity of levitation force increases obviously, and the stiffness difference caused by temperature is more obvious.

Finally, it should be noted that the cooling temperature also affects the hysteresis effect of the levitation force. We calculated the hysteresis area of the reciprocating levitation force, and the results show that the higher the cooling temperature is, the larger the hysteresis area is, as shown in figure 8. It is concluded that the higher the cooling temperature is, the more obvious the hysteresis effect of the levitation force is, and the larger the corresponding loss is.

Figure 7. The calculated ratio of maximum levitation force at different temperatures.
5. Conclusion
The influence of cooling temperature on the levitation force can be well described by using the linear model to modify the critical current density. It also shows that the influence of cooling temperature on the levitation force is mainly due to the influence of temperature on the critical current density.

The cooling temperature of HTS will affect the radial and axial load-carrying performance of radial bearing, which can be reflected in the maximum levitation force, levitation stiffness, hysteresis effect and so on. When the cooling temperature is lower than 60K, the bearing capacity tends to be saturated, which is about 1.2-1.5 times of that at 75K. When the cooling temperature is higher than 80K, the levitation force decreases rapidly. The lower the cooling temperature, the greater the levitation force when the same displacement occurs and the levitation stiffness will be. However, the stiffness difference caused by the temperature is not obvious in the small displacement range, but it becomes obvious beyond a certain displacement range. At the same time, the hysteresis area of reciprocating levitation force increases with the increase of cooling temperature.

6. References
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