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Design of AMBs and Rotordynamics for a 30 kW, 60 000 rpm Permanent-magnet Machine

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Abstract

Due to the long lifetime requirement and high rotating speed, an active magnetic bearing is designed for a 30 kW, 60 000 rpm permanent magnet machine application. The electromagnetic distribution characteristics of the active magnetic bearing are analyzed under the static levitation and the maximum bearing capacity using 2D finite element method. According to the large error problems of the calculation of the critical speed of the rotor assembly, this paper presents a model updating method to establish the finite element model of the rotor assembly. The calculated results of the first two bending critical speeds are verified by using the modal testing. The errors of the first two bending critical speed are 3.2 % and 1.8, respectively.

Keywords : High-speed permanent-magnet machine, Active Magnetic Bearings, Air gap flux density, Critical speed, Model updating.

1. Introduction

A common advantage of high speed machines is the reduction of system weight for a given magnitude of power conversion. Another commonly benefit is the improvement in reliability as a result of direct drives which is the elimination of intermediate gearing. Permanent magnet (PM) machines are thought to be the obvious choice, since they have high efficiency, high power density, small size, and low acoustic noise[1~3]. Due to the long lifetime requirement and high rotating speed, the magnetic bearing technology comes into consideration and application in machine systems.

The active magnetic bearings (AMBs) have the advantage of the long-life, lubrication-free, adjustable stiffness and damping. B. Liu et al. [4] carried out an optimization-based AMB controller design method. They proposed that the main barriers today for broader industrial AMB application one was the cost. However, simple structure, convenient fixing, low loss, high reliability, all of them are required for equip application.

The rotor of the high speed PM machine is an assembly including the clearance fit and interference fit. The most common methods used in critical speed can be subdivided into two wide classes: transfer matrix method and finite element method (FEM) [5]. The transfer matrix method has an obvious advantage in terms of the calculation of the rotor assembly's critical speed. It can be flexible and convenient to adjust the contributions of the stiffness and the mass from the components. However, the rotor assembly has to be divided into a number of massless beams and rigid bodies, which will carry out the great manual computations [6]. These limitations are now making its yield to the FEM. The FEM has the advantage of the modeling of the complex construction and accurate dynamics calculation to a single component. but it has a large errors to deal with the contact element each other due to their strong-nonlinearity. In order to realize the accurate calculation of the critical speed to the rotor assembly, the adjustments of the mass and the stiffness will be implemented to balance the contribution of components to rotor assembly.

In this paper, a 30kW, 60 kr/min high speed PM magnetically levitated motor with surface mounted SmCo PM and Inconel718 sleeve has been designed. The AMB from electromagnetic performance point of view is designed and optimized. The length and the outer diameter of rotor from mechanical point of view are designed and optimized.

2. Constructions of machine and magnetic bearing

The cross section view of the machine is shown in Figure 1. The rotor is supported by two radial AMBs and two thrust AMBs on each side and there are no additional bearings on the rotor. Based on the theoretical analysis and experience, a major improvement has been made to acquire a better electromagnetic performance and mechanical property. The machine has 24-slots stator with a distributed winding, and two-pole radial magnetized solid PM. The stator core is composed of 0.2mm laminations of low-loss steel. It is cooling by the water on the outside of jacket. The PM needs shrink-fitting a metallic sleeve to withstand strong mechanical stress from high-speed centrifugal force. The rotor is cooling by the pressurized cooling air through the rotor surface.

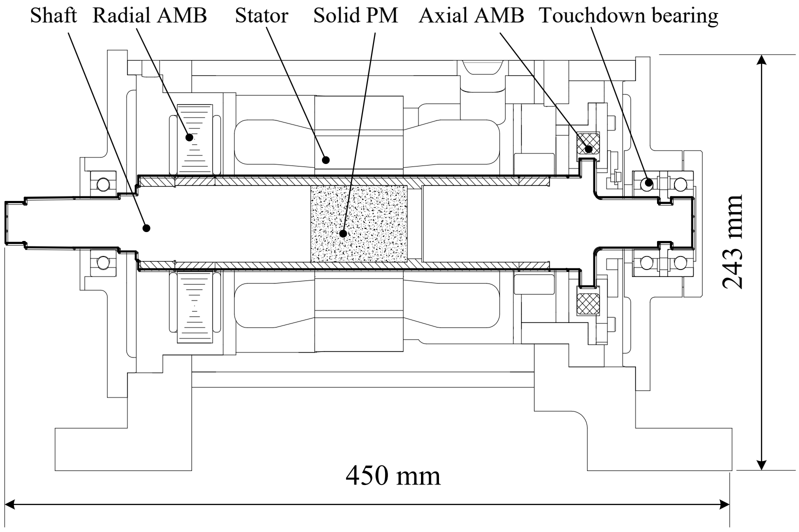


Fig.1 Cross-sectional view of high-speed PM machine

There are two fundamental assumptions in the design stage. The air-gap field of the AMB uniformly distributed and magnetic flux *NI* invariability under the small displacement conditions. According to the formulas of Maxwell electromagnetic force, the electromagnetic attraction can be derived as

$$f = \frac{B^2 A}{\mu_0} \tag{1}$$

where μ_0 is the permeability of vacuum, $\mu_0 = 4\pi \times 10^7 Vs / Am$. B is the air-gap flux density, $B = \mu_0 \mu_r H$. H is the magnetic field intensity of the air-gap. μ_r is the relative permeability. A is the projected area of the magnetic pole surface.

For the iron core lamination of the AMB stator composed by silicon steel sheet, $\mu_r \gg 1$， that is

$$B = \frac{\mu_0 NI}{2s} \tag{2}$$

where s is the air-gap length, NI is the magnetomotive force, N is the number of turns of winding, I is the current. For the eight poles magnetic bearing, the included angle between the acting surface and the resultant force is 22.5. The electromagnetic force of the AMB can be expressed as

$$f = \frac{\mu_0 N^2 A i^2}{4s^2} \cos\alpha \tag{3}$$

where i is the control current.

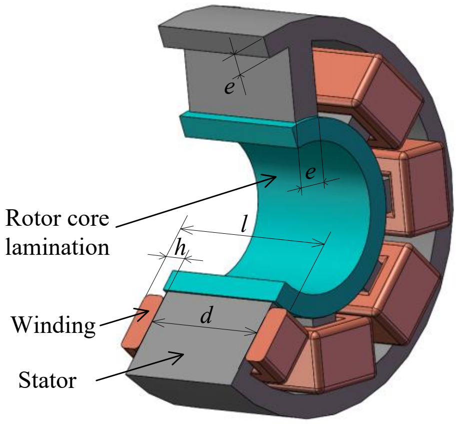
While the AMB under the maximum bearing capacity, the magnetic induction intensity in the upper air-gap pole will reach maximum. The value is often taken as the saturation flux density of the material. The maximum bearing capacity can be written as

$$F_{\max} = \frac{B_{Max}^2 S_a}{2\mu_0} \tag{4}$$

where S_a is the area of the magnetic pole, $B_{\max}$ is the saturation flux density.

Due to the adjustable bias current to the AMB, the bearing capacity and the stiffness can be control according to the actual working conditions. The sequence of the polarity of stator poles in a given rotational plane is

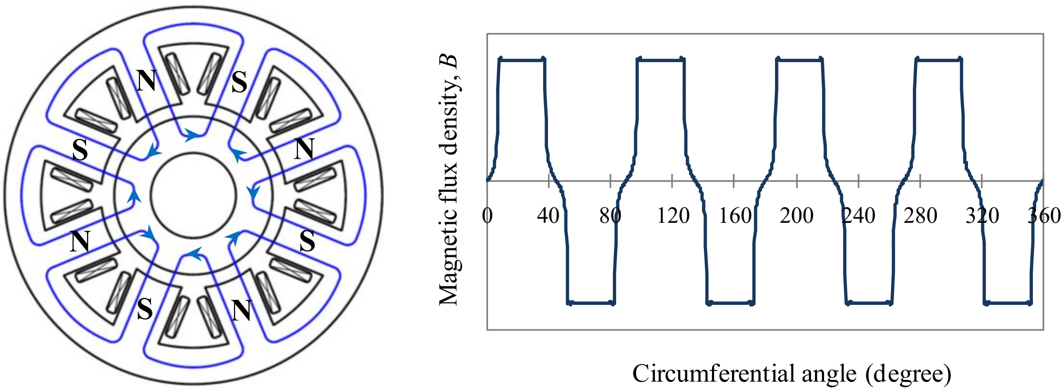
N-S-N-S-N-S-N-S. In order to obtain the maximum admissible magnetomotive force and reduce the size and weight, the optimizing of leg width and air gap on maximum carrying force is executed. The optimized geometry of the radial AMB is shown in Figure 2. The polarity configuration of bearing is selected the heteropolar.



e—Leg width; ***d***—Bearing width (magnetically active); ***h***—Winding head height; ***l***—Bearing length

Fig. 2 Geometry of the radial active magnetic bearing

In order to reduce the eddy current loss of the iron core lamination, it is made up of the 0.2 mm silicon steel sheet. The air-gap length between the inner diameter of the stator core and outer diameter of the rotor core is a very important design variable. It will decide the maximum magnetomotive force of the magnetic circuit and the processing and assembling precision. In general, the air-gap length is taken as 0.2 mm~1 mm. In the specified number of the ampere-turns, increasing the current and reducing the number of turns can improve the dynamic response speed and also reduce the axial length of the AMB. Correspondingly, the bending critical speed of the rotor will be improvement. The magnetic circuit and magnetic flux density distribution of the configuration form with NSNSNSNS are shown in Figure 3, and the structure parameters are listed in Table 1.



(a) (b)

Fig. 3 Configuration form of AMB with NSNSNSNS. (a) Schematic diagram of the magnetic circuit; (b) Magnetic flux density distribution of the rotor.

Tab. 1 Structure size and electromagnetic parameter of the radial AMB

Items	Value	Unit
Stator outer diameter	140	mm
Stator inner diameter	55.8	mm
Rotor core outer diameter	55	mm
Rotor core inner diameter	44	mm
Air gap length	0.4	mm
Bias current	0.95	A
Maximum bearing capacity	154	N
Maximum current	1.9	A

Displacement stiffness	0.4	N/μm
Current stiffness	180	N/A

3. Analysis of the electromagnetic characteristic

According to the design parameter of the AMB, the 2D finite element model is established. The electromagnetic distribution characteristic of the AMB is analyzed under the static levitation and the maximum bearing capacity. If the rotor can be steadily levitation in the air-gap, it is the premise to realize high speed rotation and can validate the reasonability of the structural design and the control system of the AMB. The flux density and flux line distribution at maximum carrying force are obtained by using the finite element commercial software Ansoft. After the 0.95 A bias current input the electromagnetic coil, the magnetic line of force and the air gap flux density waveform are shown in Figure 4 and 5.

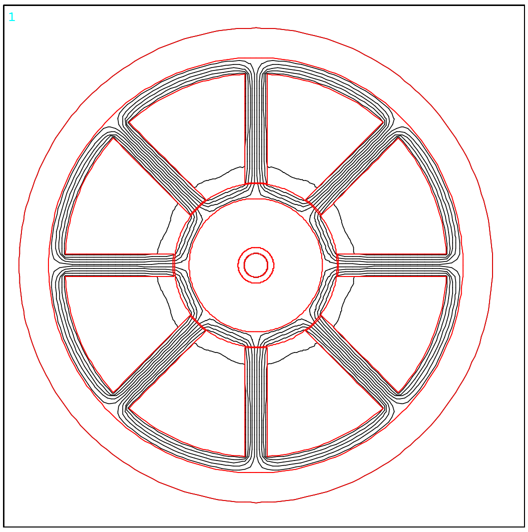


Fig. 4 Magnetic line of force of the radial AMB

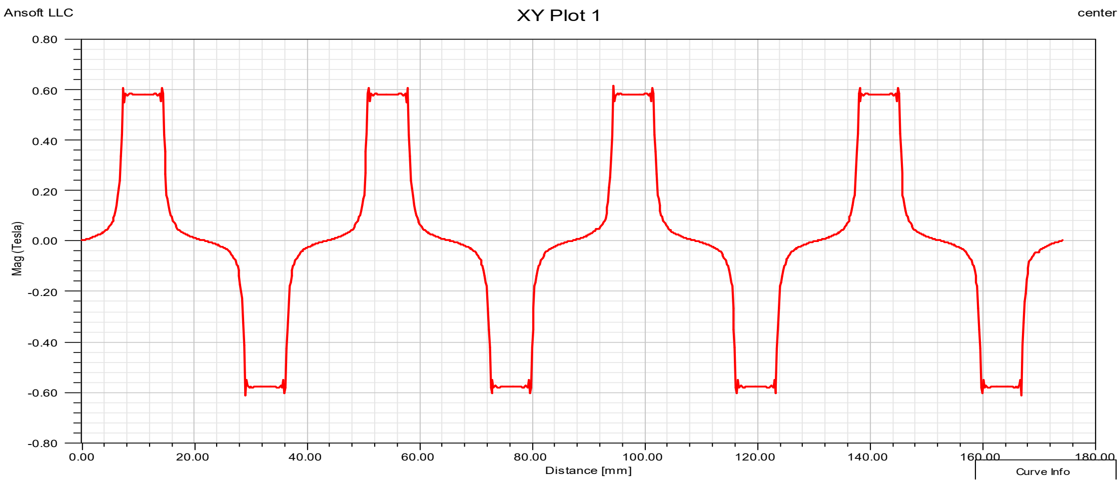


Fig. 5 Air gap flux density waveform of the radial AMB

As can be seen from Figure 4, the magnetic field distribution in iron core and air gap is homogeneous distribution while the AMB is at the static state of levitation. The bias magnetic flux density 0.58 T is half of the saturation magnetic induction. After the control current in the Y direction input the electromagnetic coil, the magnetic line of force and the air gap flux density waveform are shown in Figure 6 and 7.

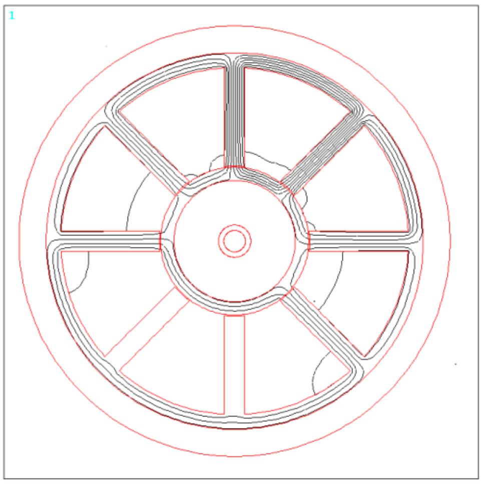


Fig. 6 Magnetic line of force at maximum bearing capacity

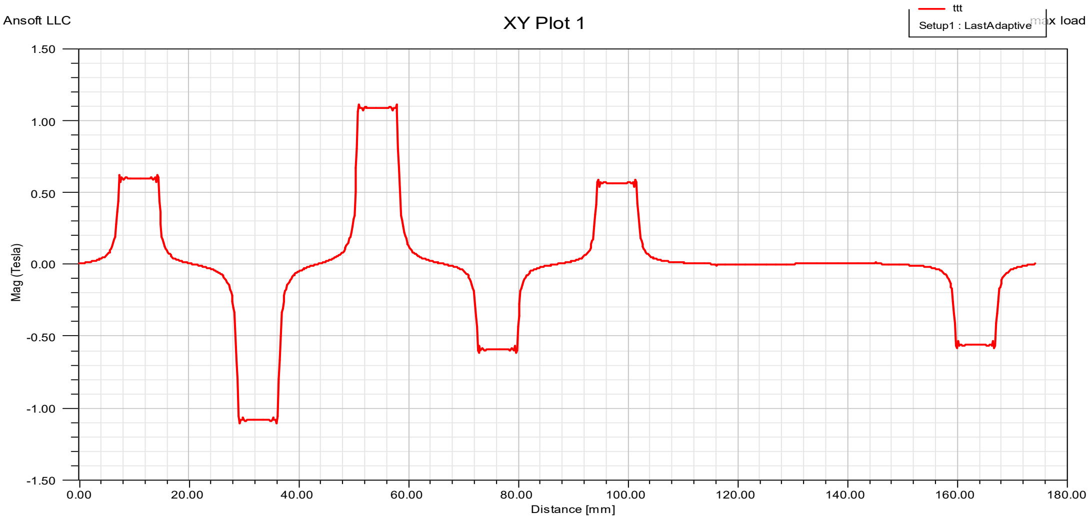


Fig. 7 Air gap flux density waveform at maximum bearing capacity

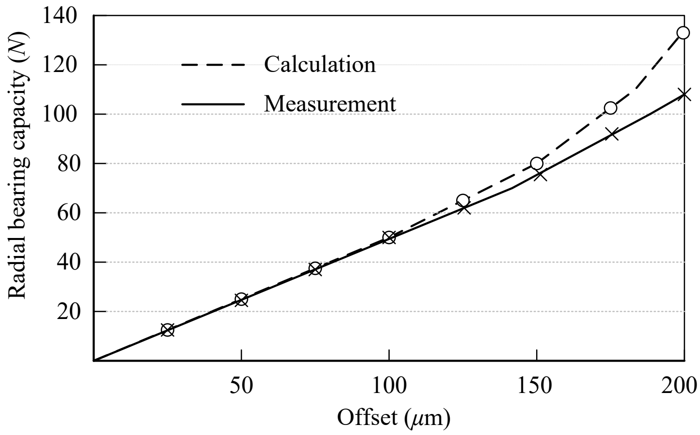


Fig. 8 Relationship of the radial bearing capacity and the displacement

Compared with the static, the distribution of the magnetic line of force is no longer uniform. The maximum magnetic field intensity appear on the Y direction, and the distribution of the magnetic line of force is dense. While the AMB is at the maximum bearing capacity, the maximum magnetic flux density in the iron core is 1.1 T, which is two times to the bias magnetic flux density. It indicate that the electromagnetic performance of the ferromagnetic material is fully used. The relationship of the radial bearing capacity and the displacement is shown in Figure 8. It is observed that there is a good linear relation between the radial bearing capacity and the displacement near the equilibrium position.

When the rotor is off center, the linearity will get worse. The radial AMB meets the requirements of the radial bearing capacity. It has a well characteristic of the linear magnetic force. The calculation is in good agreement with the measurement. The design method of the AMB used in this paper is reasonable.

5. Rotor dynamics analysis

The rotor under the centrifugal force often appears as forced vibration. While the natural frequency and the excitation frequency from high rotational speed are identical, the resonance will occur. This speed is called critical speed. In order to ensure the stable operation of the machine, the design of the rotor's bending critical speed must avoid the rated speed. Therefore, the accurate calculation of the bending critical speed is one of the most important design parameters in the machine design stage.

The general FEM can adjust the elasticity modulus and the density to the rotor components. Model updating is employed to correct the FEM modeling through adjustments of the modulus of elasticity in [7]. However, a too small value of the elasticity modulus will result in the variation of the material performance. This paper take the 30 kW,60000 rpm high speed PM machine as an example as shown in Figure 9. The analysis of the critical speed is carried out through the adjustment of the material density of the rotor components. Because the modal of the rotor assembly is mainly provide by the sleeve, the others components will be the design variables. The material densities of the nut, AMB, PM, front shaft, rear shaft are arranged as ρ_1, ρ_2, ρ_3, ρ_4, ρ_5, respectively. The first two bending modes of the rotor assembly can be obtained by FEM based on the density updating as shown in Figure 10.

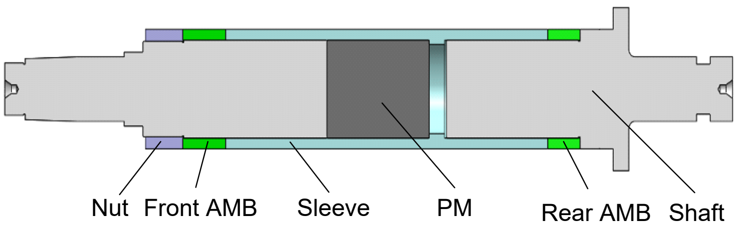
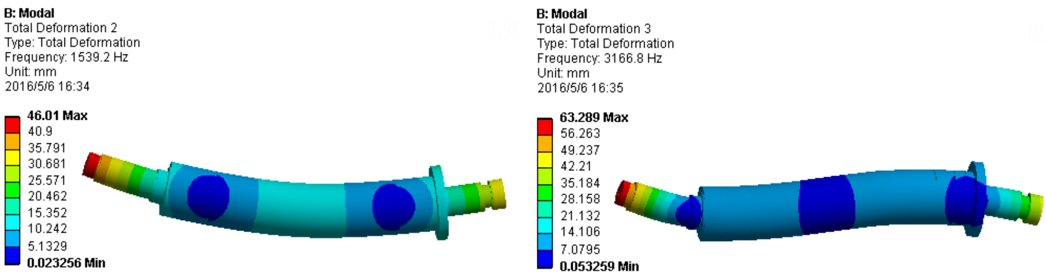


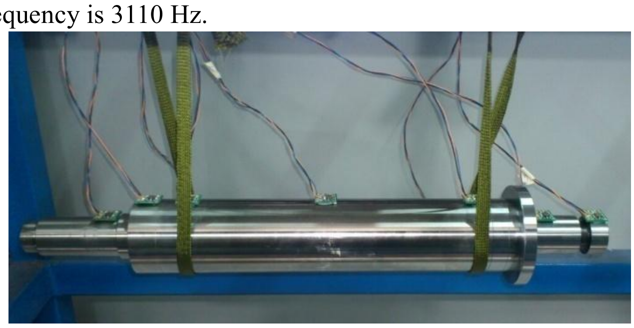
Fig. 9 Cross-sectional view of rotor assembly



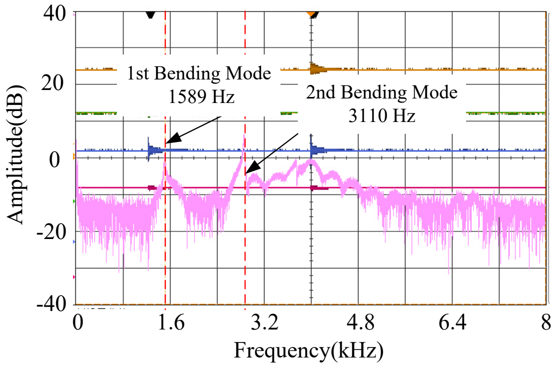
(a) (b)

Fig.10 First two bending mode of rotor assembly using model updaing FEM. (a) first bending natural frequency 1539 Hz; (b) second bending natural frequency 3167 Hz.

The calculated accuracy should be validated by modal testing to the prototype. The measurement is implemented to the 30 kW, 60 kr/min rotor assembly of the high speed PM electrical machine. A free-free natural frequency of rotor assembly is measured using an acceleration sensor as shown in Figure 13. The frequency domain of rotor-shaft assembly by the fast Fourier transformation (FFT) spectrum of the sensors output signals is shown in Figure 14. The spectrum has clear peaks for the first two bending natural frequencies. The 1^{st} bending natural frequency is 1589 Hz and the 2^{nd} bending natural frequency is 3110 Hz.



(a)



(b)

Fig.11 Modal test bench and the frequency spectrum of the rotor assembly. (a) Modal testing of rotor assembly with free-free condition; (b) Frequency spectrum of the rotor assembly from acceleration sensor.

As can be seen from Figure 11, the spectrum has clear peaks for the first two bending natural frequencies. The 1st bending natural frequency is 1589 Hz and the 2nd bending natural frequency is 3110 Hz. The updated finite element modeling can accurately calculate the bending critical speed of the rotor assembly. The errors of the first two bending natural frequency are 3.2 % and 1.8 %, respectively.

6. Conclusion

This paper is analyzed the AMB and the rotordynamics for the 30 kW, 60000rpm high speed PM machine. The radial AMB meets the requirements of the radial bearing capacity. It has a well characteristic of the linear magnetic force. The accurate critical speed can be obtained by using model updating method to the rotor assembly. The errors of the first two bending natural frequency are 3.2 % and 1.8 %, respectively. The design of the AMB can provide the enough bearing force for the high speed rotational rotor, and the rotor dynamics can avoid the resonance since the first bending critical is greater than the rated speed.

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