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The design of a centrifugal blower rotor with magnetic bearings based on rotor dynamics

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Abstract

A maglev centrifugal blower rotor with magnetic bearings is designed and developed by rotor dynamics. The shaft diameter is initially set according to the magnetic bearings and engineering experience, the finite element model calculation shows the first bending critical speed of the initially designed rotor without silicon steel sheets is closed to the rated speed. To avoid the resonance, the shaft diameter of magnetic bearings section increases and the length of left and right enameled wire section are reduced, thus the first bending critical speed increases, which and the second rigid critical speed have more than 15% margin to the rated speed within support stiffness of 10^5 N/m~10^7 N/m, the first bending critical speed of the revised rotor with silicon steel sheets increases a little compared to that without silicon steel sheets. At the condition of dynamic balance grade of G2.5 and the damping ratio of 0.707, the vibration amplitudes of the revised rotor are far less than the protective gap, the gyroscopic effect influences more and more with the increase of angular speed, so that the vibration amplitudes with different support stiffness tend to be the same, then the lower support stiffness such as 5×10^5 N/m can be chosen to facilitate the control system design.

Keywords : maglev centrifugal blower rotor, FEM model, critical speed, support stiffness, vibration amplitude

1. Introduction

The centrifugal blower with magnetic bearings has the advantages of high rotational speed, lower noise and compact arrangement compared with the conventional blowers. It increases the rotor design difficulty to consider the characteristics of magnetic bearings, rotor dynamics is the most important aspect of the optimization design of the rotor with magnetic bearings[1-6], the first bending critical speed and the vibration amplitude of rotor at the location of magnetic bearings determine the rotor dimensions, its machining accuracy and the support stiffness.

2. The initially designed rotor

The maglev centrifugal blower rotor is shown as Fig.1, the structure of sealing section(SS), blower wheel section(BWS) and motor section(MS) can't be easily altered because of the power, the design aim is to determine the diameter and the length of the radial magnetic bearings section(MBS), and the length of the enameled wire section(EWS).

The structure of a radial 12-pole magnetic bearing is shown as Fig.2, assuming side pole width t equals pole-to-pole distance and the half width of main pole, we determined the diameter d_m of MBS, the inner diameter d_0, the medium diameter d_1, the outer diameter d_2 of the magnetic bearings, the magnetic bearings length L_r, a pole height h and the number of turns N per pole by the geometric relationship, the air gap c_0 and the maximum load force, the initial design result of a radial magnetic bearing is shown as Table 1.

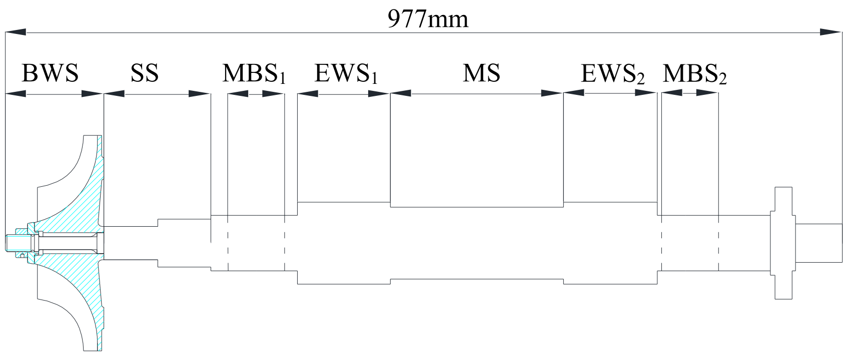


Fig.1 The structure of the maglev centrifugal blower rotor

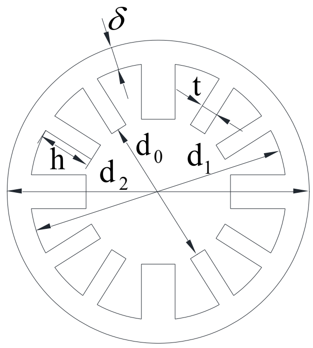


Fig.2 The dimensions of the magnetic bearings

Table 1. The design result of a magnetic bearings.

t(mm)	N	h(mm)	d_0(mm)	d_1(mm)	d_2(mm)	c_0 (mm)	L_r(mm)
11	120	35	96.7	167	190	0.2	102

To avoid magnetic saturation, the shaft diameter of magnetic bearings section is set to 65mm from experience, which is less than d_0-2t-2c_0 , the length of 104mm is set to 2mm longer than L_r considering assembly errors.

3. The critical speeds of the rotor without silicon steels

According to the theory of elastic mechanics, the rotor was discretized into finite elements, dynamical differential equations of rotor are established as Eq. (1).

$$M\ddot{x}+\Omega G\dot{x}+Kx=F \tag{1}$$

where M is mass matrix, K is stiffness matrix, G is gyroscopic matrix, Ω is angular velocity, x is generalized displacement, F is generalized force.

The ANSYS was employed to do dynamics analysis of the rotor finite element models(FEM). The element solid273 was adopted to model axisymmetric rotor, which is helpful to reduce calculation complexity and guarantee calculation precision, magnetic bearings were modeled as the element combin14. To avoid the local vibration of blades, the blower wheel was simplified as a cylindrical boss, the mass, the polar rotary inertia, the diameter rotary inertia, the length and the gravity center are all same before and after the simplification, the mesh and the boundary constraint of the rotor FEM model are shown as Fig.3.

Support stiffness has a considerable influence on modal shapes and critical speeds, the rotor axis modal shapes are shown as Fig.4 and the Campbell diagram is shown as Fig.5 with support stiffness of 5×10^5 N/m, it indicates that the fourth modal corresponds to the first bending mode, the first bending critical speed of 23474.4 rpm is closed to the rated speed of 23000 rpm.

Support stiffness of magnetic bearings varies usually from 10^5 N/m to 10^7 N/m below that of traditional bearings, Fig.6 shows the critical speeds variation of the initial rotor along with support stiffness.

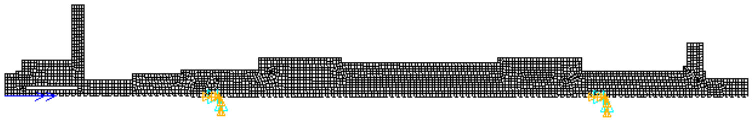
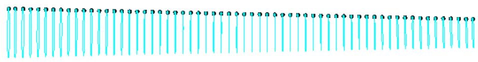


Fig.3 The FEM model of the initial rotor without silicon steel sheets

a. Axial translation mode



b. Radial swing mode

c. Radial swing mode

d. the first bending mode

Fig.4 The modal of the initial rotor without silicon steel sheets

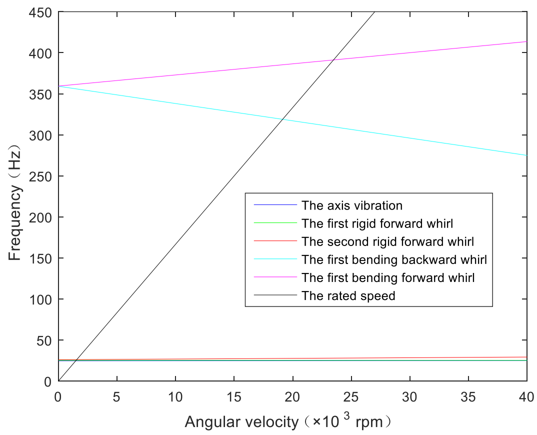


Fig.5 The Campbell dialogue of the initial rotor without silicon steel sheets

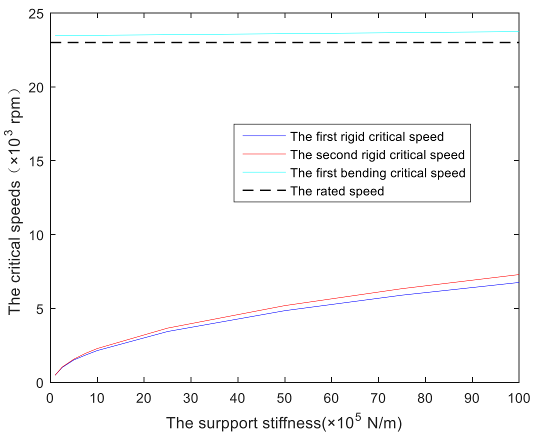
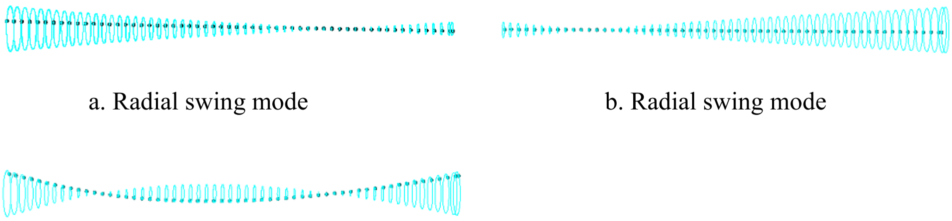


Fig.6 The critical speeds variation of the initial rotor without silicon steel sheets

We can see that the first rigid critical speed, the second rigid critical speed and the third rigid critical speed almost linearly increase and are far lower than the first bending critical speed, and the first bending critical speed is closed to the rated speed of 23000 rpm which changes slightly, so the initially designed rotor is necessarily revised.

4. Rotor revision

Enhancement of rotor stiffness helps improve the critical speeds, the usual approach is to shorten rotor length or enlarge rotor diameter. After the shaft diameter of magnetic bearings section increased 7mm, which is still 2.3mm less than d_0-$2t$-$2c_0$, besides, the length of the left and the right enameled wire section were reduced respectively by 40mm and 20mm, the axis modal shapes of the revised rotor are shown as Fig.7, the critical speeds vary along with support stiffness as Fig.8.



a. Radial swing mode

b. Radial swing mode

c. the first bending mode

Fig.7 The modal of the revised rotor without silicon steel sheets

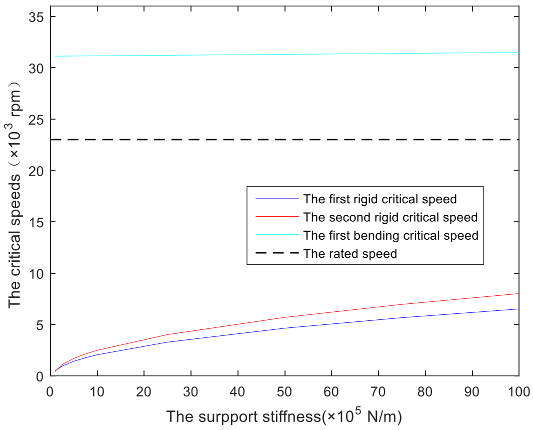
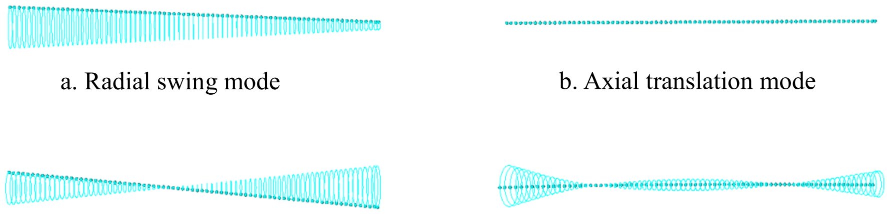


Fig.8 The critical speeds variation of the revised rotor without silicon steel sheets

We can see the second rigid critical speed vary within a range of 431.563~8004.405 rpm, and the first bending critical speed vary within a range of 31123.721~31484.338 rpm. The rated speed of 23000 rpm is 15% less than the first bending critical speed and 15% more than the second rigid critical speed, so the revised rotor achieves the design criterion.

The FEM model of the rotor with silicon steel sheets is shown as Fig.9, The elasticity modulus of the silicon steel sheets is one-tenth that of the silicon steel during the finite element calculation [7]. The modal shapes and the critical speeds are respectively shown as Fig.10 and Fig.11. we can see the second rigid critical speed vary within 666.59~6540.689 rpm, and the first bending critical speed vary within 32377.999~32440.483 rpm, the latter increases a little compared to the rotor without silicon steel sheets.

Fig.9 The FEM model of the revised rotor with silicon steel sheets



a. Radial swing mode　　b. Axial translation mode

c. Radial swing mode　　d. First bending mode

Fig.10 The modal of the revised rotor with silicon steel sheets

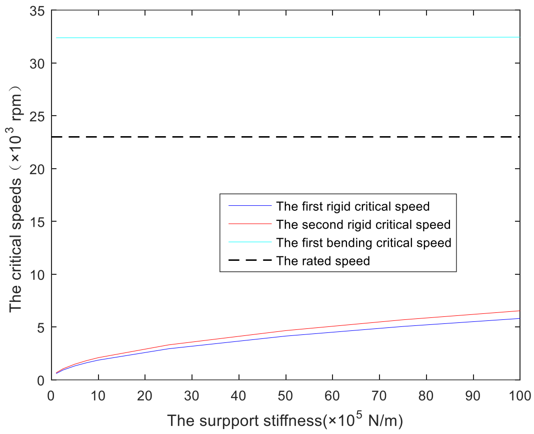


Fig.11 The critical speeds variation of the revised rotor with silicon steel sheets

5. Harmonic response of dynamic imbalance

In order to reduce the vibration of centrifugal blower and raise its stability and life, it's necessary to perform dynamic balance by adding or removing weights at two planes without precision requirement. The processing positions of the rotor are the impeller back and the right aluminum end ring of motor silicon steel sheets. Eccentricity distance and remaining unbalance mass follow as Eq.(2) and Eq.(3).

$$\mathrm{e} = \frac{G}{\omega} \tag{2}$$

$$\mathrm{m} = \frac{1}{2}\frac{eM}{r} \tag{3}$$

where G is dynamic balance grade (mm/s), ω is maximum service speed(rad/s), e is eccentricity distance(mm), m is permissible residual unbalance mass per plane(kg), M is rotor mass(kg), r is correction radius(mm).

Support damping coefficient follows as Eq.(4).

$$\mathrm{d}mp = 2\delta\sqrt{MK} \tag{4}$$

where $\mathrm{d}mp$ is support damping coefficient, δ is damping ratio, M is rotor mass, K is support stiffness.

To ensure the processing precision, the processing efficiency and the dynamic characteristics of the control system, dynamic balance grade and the damping ratio were respectively designed as G2.5 and 0.707[8].

The harmonic excitation FEM model is shown as Fig.12, the axis vibration amplitudes at magnetic bearings are shown as Fig.13 and Fig.14.

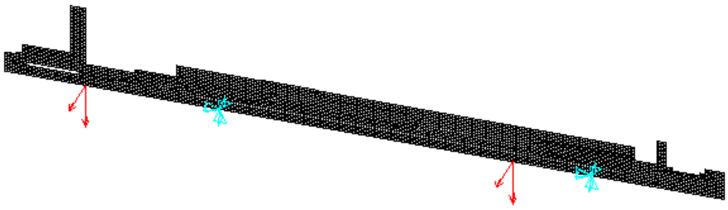


Fig.12 The harmonic excitation FEM model of the revised rotor with silicon steel sheets

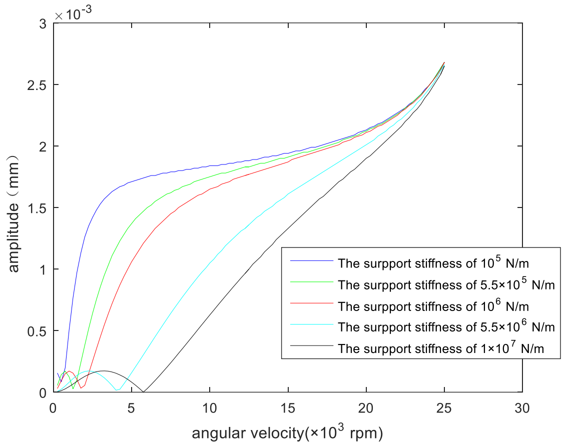


Fig.13 The MBS_1 vibration amplitude of the revised rotor with silicon steel sheets

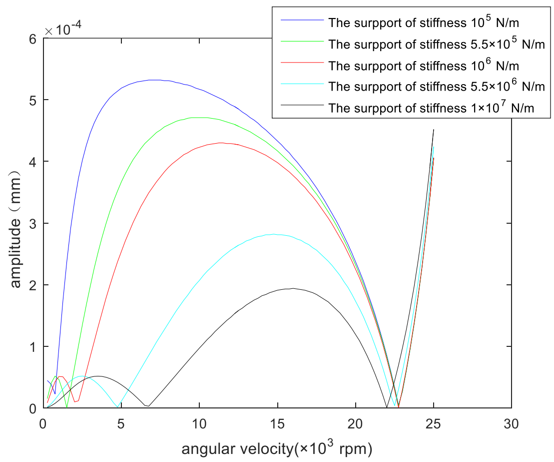


Fig.14 The MBS_2 vibration amplitude of the revised rotor with silicon steel sheets

It shows that the vibration amplitude increases as the support stiffness decreases, and are far less than the protective gap of 0.15mm, d*mp* increases with support stiffness and gyroscopic effect increases with angular speed, but gyroscopic effect influences more than d*mp* and support stiffness, the vibration amplitudes tend to be the same within the angular speed of 23000~25000 rpm, so the lower support stiffness such as 5×10^5 N/m can be adopted to reduce the design difficulty of control system, which has little effect on the machine vibration[9].

6. Conclusion

The rotor dynamics calculation shows the first bending critical speed of the initially designed rotor is closed to the rated speed of 23000 rpm, the empirical value of the diameter of magnetic bearings section was inappropriate. After the shaft diameter of magnetic bearings section increased 7mm, besides, the length of the left and the right enameled wire section were reduced respectively by 40mm and 20mm, the rotor conforms to the requirement that the rated speed of

23000 rpm has more than 15% margin to the first bending critical speed and the second rigid critical speed.

The rigid critical speeds increase almost linearly and the first bending critical speed changed little within support stiffness of 10^5 N/m~10^7 N/m.

Compared with the support damp and the support stiffness, gyroscopic effect influences gradually more on the vibration amplitude along with the angular speed increase, the vibration amplitudes equal nearly each other at the angular speed of almost 23000~25000 rpm, so the lower support stiffness such as 5×10^5 N/m can be adopted to reduce the design difficulty of control system. The maximum amplitude is still far less than the protective gap of 1.5mm, which ensures the centrifugal blower runs normally.

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