Rotor Dynamic Analysis on the Bearing-rotor System of Magnetic Centrifugal Compressor

Yusheng Hu [a,b], Fang Zhang [a,b] ,Xiaobo Zhang [a,b],Gao Gong [a,b],Jiuzhang Su [a,b],Chao Zhang [a,b]

[a] State Key Laboratory of Air-conditioning Equipment and System Energy Conservation, Jinji West Road, Zhuhai City, Guangdong Province, China

[b].Gree Electric Appliaces. Inc. of Zhuhai, Jinji West Road, Zhuhai City, Guangdong Province, China,zhangfang1900@163.com.

***Abstract*—Modal frequency and modal vibration mode of a centrifugal compressor rotor-bearing system, which equipped with magnetic bearings, are significant target parameters to insure stable operation of the compressor. Meanwhile, calculations and optimizations of those parameters are also necessary processes of compressor structure designing. In this paper, basing on the basic theories of rotor dynamics, ANSYS is used to analyze vibration mode and modal frequency of a compressor rotor-bearing system. The correction coefficient of modulus of elasticity of laminated and basic excitation sources caused by impellers in pneumatic section and magnetic bearings' stiffness parameters are considered when doing rotor dynamics modeling simultaneously throughout the simulation. Influences on critical speed of the rotor-bearing system caused by rotor speed is analyzed through the simulation. It has provided a reference to designing shafts in long axis system of magnetic high-speed centrifugal compressor by obtaining the simulation results.**

I. INTRODUCTION

Magnetic bearing has the characteristics of no contact, no lubrication and no friction. Besides, active control of supporting performance can achieve active control of rotary shaft vibration, and these characteristics of magnetic bearings make it an ideal substitute for traditional bearings in high-speed rotating machinery.

The application of magnetic bearing to centrifugal compressor has a significant advantage for further improving compressor running speed, improving compressor energy efficiency and reducing compressor noise and vibration[1] [2].

In this paper, a rotor-bearing shaft system of a magnetic levitation compressor with a power 200KW and a cooling capacity of 400RT(ton of refrigeration) is taken as an example. On the basis of rotor dynamics analysis and ANSYS software, the natural frequency and vibration mode of the rotating shaft system are solved. By analyzing the characteristics of the rotor system, the reference is provided for the design of the long axis rotating shaft of the maglev high speed centrifugal compressor.

II. INTRODUCTION OF ROTATING SHAFT SYSTEM OF MAGNETIC LEVITATION COMPRESSOR

The rotating shaft of a magnetic levitation centrifugal compressor is a typical long axis system in Fig.1, which is mainly composed of two stage impellers, radial bearing rotor, motor rotor and thrust bearing, the main parameter is shown in table I.

TABLE I. THE MAIN PARAMETER OF ROTOR-BEARING SHAFT SYSTEM

	Parameter	Value
d_r	Motor rotor diameter	135 mm
di	Impeller diameter	220mm
P	Rated power	200KW
M	Total mass	80Kg
L	Total length	978mm

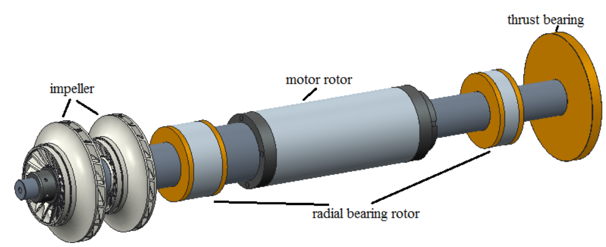

Figure 1. A typicai centrifugal compressor rotor-bearing system

III. RESEARCH AND SIMULATION ANALYSIS OF BEARING ROTOR SYSTEM

A. Dynamic analysis of rotor

Compared with the traditional sliding bearings, the magnetic bearing support reduces the stiffness of the structure, and the critical speed of the shafting also decreases. The dynamic characteristics of the shafting play an important role in the accuracy of suspension and the stability of control.

The dynamic characteristics of the shafting play an important role in the accuracy of suspension and the stability of control. The purpose of rotor characteristic analysis is to determine the critical rotational speed of the rotor support system, and to adjust the distribution of these critical speeds according to the experience or bearing characteristics of the magnetic bearing. It can deviate properly from the working speed of the machine so as to obtain reliable rotor dynamic performance. Therefore, it is of great significance to analyze the structural characteristics of magnetic bearing rotor components.

Generally speaking, the differential equations of motion for mechanical systems are：

$$M\ddot{x}+C\dot{x}+Kx=F \quad (1)$$

Where M、C、K are the mass, damping and stiffness matrixs of the system, X is generalized coordinate vector for system, F is the generalized external force acting of the system.

Because of rotating machinery rotor has gyration effect, an anti symmetric gyroscopic matrix appears in the motion equation of the system. The differential equations of motion are as follows:

$$M\ddot{x}+\left(C+G\right)\dot{x}+Kx=F \quad (2)$$

With the development of rotor dynamics, modern computing methods can be divided into two main categories: transfer matrix method and finite element method. The two methods have their own advantages and disadvantages. The accuracy of transfer matrix method is not higher than that of the finite element method, but the transfer matrix method is faster than the finite element method.

In the design process, combined with the advantages of the two methods, the rotor dynamics is used to estimate the characteristics of the maglev centrifuge, and the dynamic design of the system is carried out with the finite element software.

B. Dynamic analysis of rotor based on ANSYS

In this paper, the ANSYS software is used to analyze the modal analysis of the rotor system of the compressor bearing.

Because of the motor rotor is laminated by stacking structure, the motor rotor should be set up for anisotropic material calculation. The radial and axial elastic modulus of the anisotropic material is, E、E' the shear modulus is G、G', and the Poisson's ratio is v、v'. The ratio of modulus of elasticity to the modulus of elasticity in the vertical direction is defined as the modulus of superposed elastic modulus.

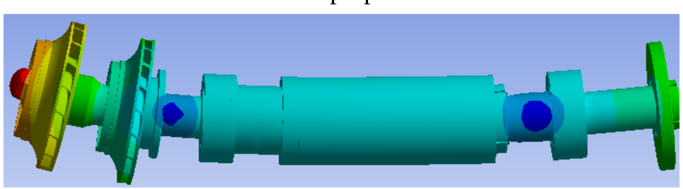

Figure 2. The vibration mode of the 450RT centrifugal compressor rotor-bearing system

Through the simulation, the modal shape of the 450RT compressor shaft is shown in Fig. 2. Meanwhile, we measured the compressor shaft and obtained the first order solid frequency and its formation as shown in Fig. 3.

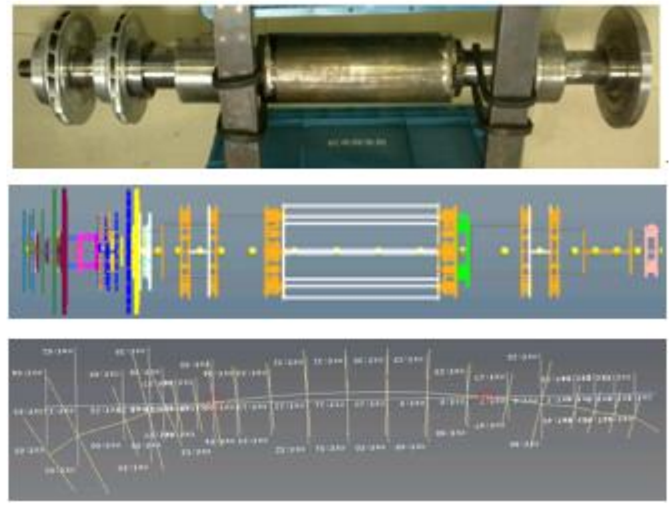

Figure 3. The test vibration mode of the 450RT centrifugal compressor rotor-bearing system

The accuracy of the parameter setting of the motor which is considered the correction coefficient of modulus of elasticity of laminated is the key effect on the accuracy of the fixed frequency simulation.

TABLE II. THE COMPARE BETWEEN MEASURE AND SIMULATION OF MODAL FREQUENCY

	Modal frequency（Hz）	
	First order flexible frequency	second order flexible frequency
Measured value	341	621
Simulation value	348	605
Ratio error	2%	2.60%

By comparing the simulation results with the fixed frequency value of the rotating shaft which is shown in Table II., the accuracy of the parameter setting of the rotor overlay structure is verified.

C. different support stiffness

In this chapter, the influence of the different support stiffness and damping on the modal of the rotating shaft is discussed, and the Campbell distribution diagram of the frequency of the rotating shaft is obtained [3].

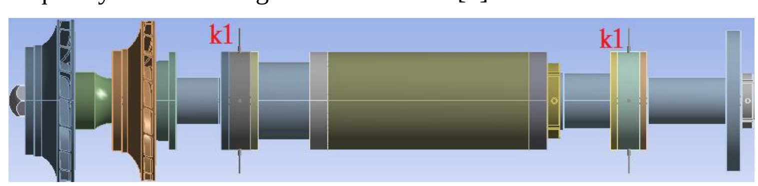

Figure 4. The support stiffness of the rotating shaft

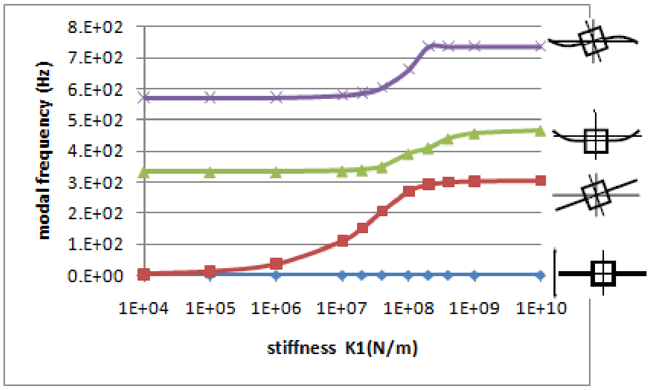

Figure 5. Modal frequency of different stiffness

The simulation results show that:

1) The bearing stiffness of the 450RT magnetic compressor is obtained is 4×10^7 N/m;

2) The translational mode is almost always 0, not affected by supporting stiffness;

3) The swing mode rises obviously with the support stiffness increasing between 1 x 10^6 N/m and 2 x10^8 N/m, which is basically unchanged after the supporting stiffness is greater than 2 x 10^8 N/m;

4) The first order flexible mode increases with the support stiffness between the support stiffness from 4 x 10^7 N/m to 1 x 10^9 N/m, which is basically unchanged after the supporting stiffness is greater than 1 x 10^9 N/m;

5) The two order flexible modal increases with the support stiffness between the supporting stiffness from 4 x 10^7 N/m to 2 x 10^8 N/m, and is basically unchanged after the supporting stiffness is more than 2 x 10^8 N/m.

In the actual operation of the rotor, the disk structure will deviate from the original plane and thus produce a gyroscopic moment that changes the deflection angle of the shaft, so that the critical speed value of the rotor is different from the critical speed of the gyroscopic effect[4].

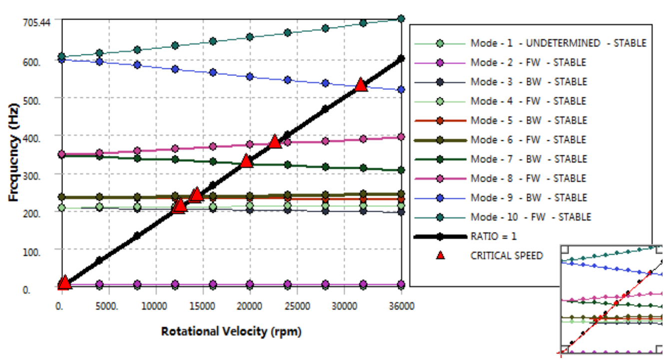

Figure 6. The Campbell distribution diagram of the frequency of the rotating shaft

Based on the ANSYS software, the Campbell distribution diagram of the rotor is obtained under the condition of support stiffness 4X10^7N/m, as shown in Fig. 6. The rotational speed corresponding to the intersection point of Campbell line and the whirl frequency line of each order is the critical speed of the rotor under this stiffness condition. From the Figure 6 we could see the effect of gyroscopic effect on the critical speed of the shaft: forward precession motion, gyroscopic effect can increase the critical speed; backward precession motion can depress the critical speed.

D. The influence of the aerodynamic force on the modal frequency of the impeller

The gap of impeller & labyrinth seal is small, and the additional stiffness and damping effect are generated during the process of fluid passing. This paper will take about the influence of additional stiffness and damping.

The impeller and seal structure are shown in Fig.7.

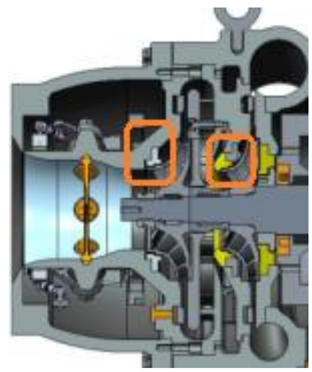

Figure 7. The matching diagram of the impeller and comb in the pneumatic section of the magnetic levitation compressor

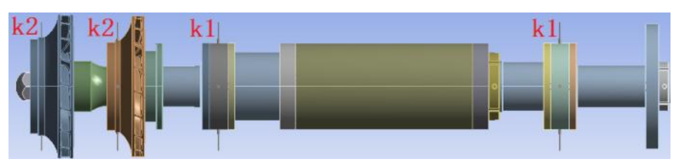

Figure 8. Think of the impeller and comb in the pneumatic section of the magnetic levitation compressor

Through simulation, the different stiffness of the impeller comb seal and the change of the critical speed are shown in Fig. 9.

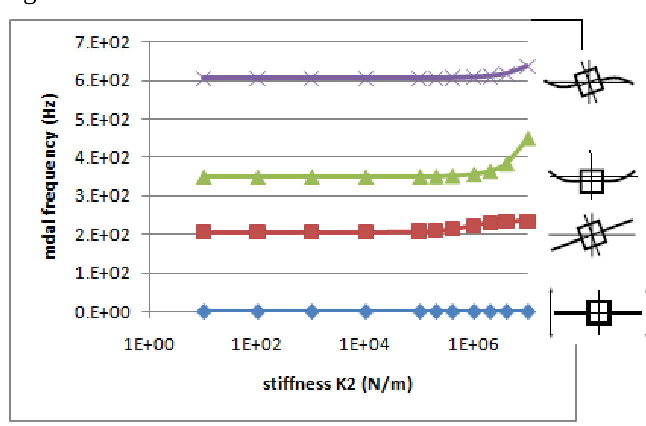

Figure 9. Think of the impeller and comb in the pneumatic section of the magnetic levitation compressor

IV. CONCLUSION

This paper established a method, which are considered the assembly relationship of the motor rotor and the correction coefficient of modulus of elasticity of laminated, for modal and fixed frequency of magnetic compressor rotor. It is verified that the error is within 3%. Besides, the influence of different stiffness support, gyroscopic moment and compressor pneumatic seal structure on the fixed frequency of 400RT magnetic compressor rotor is simulated and analyzed, which is coincide with the actual situation. The related research contents in this paper provide an effective method for the design of the shaft system of the magnetic centrifugal compressor.

REFERENCES

[1] Wen L. Soong, Gerald B.Kliman, Roger N. Johnson, Raymond A. White and Joseph E.Miller,"Novel High-Speed Induction Motor for a Commercial Centrifugal Compressor," IEEE Transactions on Industry Applications,2000, 36(3): 706-712.

[2] Ziyuan Huang, Yun Le, Bangcheng Han."Dsign of AMBs and Rotordynamics for a 30 kW, 60 000 rpm Permanent-magnet Machine,", Proceedings of the 15th International Symposium on Magnetic Bearings, F1A3-10053.

[3] Wroblewski, Adam C., Jerzy T. Sawicki, and Alexander H. Pesch, "Rotor model updating and validation for an active magnetic bearing based high-speed machining spindle," Journal of Engineering for Gas Turbines and Power, Vol.134, No.12 (2012), pp.122509.

[4] Li Dong, Yu Xiaoli, Yu Suyuan, Zhang Zhijian, "Principle Test of Active Magnetic Bearings for the Helium Turbomachine of HTR-10GT," ISMB12, PP:594-601.