

Structure design and simulation research of active magnetic bearing for helium centrifugal cold compressor

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Abstract. Helium centrifugal cold compressors are utilized to pump gaseous helium from saturated liquid helium tank to obtain super-fluid helium in cryogenic refrigeration system, which is now being developed at TIPC, CAS. Active magnetic bearing (AMB) is replacing traditional oil-fed bearing as the optimal supporting assembly for cold compressor because of its many advantages: free of contact, high rotation speed, no lubrication and so on. In this paper, five degrees of freedom for AMB are developed for the helium centrifugal cold compressor application. The structure parameters of the axial and radial magnetic bearings as well as hardware and software of the electronic control system is discussed in detail. Based on modal analysis and critical speeds calculation, a control strategy combining PID arithmetic with other phase compensators is proposed. Simulation results demonstrate that the control method not only stabilizes AMB system but also guarantees good performance of closed-loop behaviour. The prior research work offers important base and experience for test and application of AMB experimental platform for system centrifugal cold compressor.

1. Introduction

With the application of superconducting magnets in large particle accelerators, nuclear fusion device and other great science projects, it is crucial to acquire liquid helium by large helium cryogenic system [1]. The many large devices of scientific experiment all over the world -- Tore Supra in French, Continuous Electron Beam Accelerator Facility (CEBAF) in America, Large Hadron Collider (LHC) for European Organization for Nuclear Research (CERN) in Switzerland, Experimental Advanced Superconducting Tokamak (EAST) in China, are all equipped with helium refrigeration system [2-5]. In the refrigeration system, utilizing cold compressor to pump gaseous helium from saturated liquid helium tank is the most common method to obtain super-fluid helium [6].

As the rotation speed of the spindle for cold compressor now being developed at Technology Institute of Physics and Chemistry, Chinese Academic Sciences (TIPC, CAS), is 60000 rpm, fairly high, it's rotor and support assembly deserve extra-care. In the previous application cases, there are mainly three specialized designed kinds of bearing—ceramic ball bearings, aerostatic bearings and active magnetic bearings used to support the rotor of the centrifugal compressor in cryogenic refrigeration system [7]. However, active magnetic bearings are regarded as the best support assembly than other bearings because of its multitudes of advantages, such as zero frictional wear and working in harshen environment



like ultra-speed rotating, elimination of the lubrication and negative pressure system [8]. Especially in recent years, with the availability of components for power electronics and information processing and especially the appearance of advanced control theories, AMB system shows great applying potentiality and is slowly replacing other kinds of bearings.

Although many advantages of AMB with application to cold compressor, the structure and control strategy of AMBs are often very different and need customize according to the different work condition. AMBs applied in cold compressor also need special design, which is a difficult problem. The paper aims at structure design and control simulation for AMB, it's the preliminary work for practical application to cold compressor. The important content in this thesis is arranged as follows: section two depicts the structure and characteristics of cold compressors, section three introduces the hardware components of AMB system. Section four explains the structure parameters design of axial and radial magnetic bearings. Modal analysis and control simulation of AMB is demonstrated in section five and section six.

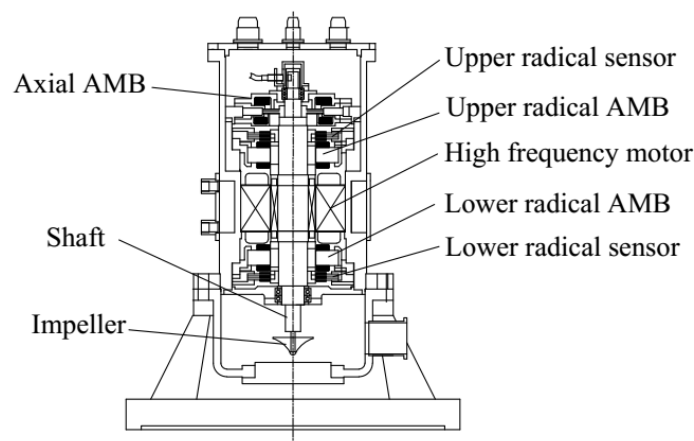


Figure 1. Structure components of the helium cold compressor.

2. Structure and characteristics of cold compressors

In the Large Cryogenic Refrigeration System for Liquid Helium to Super-Liquid Helium being developed in TIPC, CAS, centrifugal compressors are applied to compress pure gaseous helium from 2.8kPa to 42kPa with mass flow rate of 30g/s. In addition, based on preliminary fluid simulation and calculation of compressors impeller runner, there are three-stage centrifugal compressors in series form. The pressure ratios of those three-stage compressors are designed as 0.65, 0.7 and 0.75 respectively. The biggest rotate speed, corresponding to the biggest pressure ratio of the rotor for compressors, is of 60000 rpm [9], which is the maximum continuous rated revolution of active magnetic bearings.

As is shown in figure 1, the cold compressor that we designed and manufactured is of vertical construction. The impeller was mounted at the bottom of the rotor, in the area of the compressor that remains at extreme low temperature of 3.4K. The motor was located in the centre of the two radial AMBs and the axial AMB stator was mounted at the top of the rotor. Both the motor and the bearing stators are at room temperature, in order to prevent heat shrinking and disconnection of coil because of the low-temperature environment. The performance of cold compressor depends highly on reduction of heat transfer from the room temperature parts to extreme low temperature parts. There are two ways to reduce heat transfer—increase length or hollow on inside of the rotor between the lower bearing end and the impeller. If the rotor is too long, it needs to pass through critical speed domain and the controller for AMB system would be complicated, considering the rather high rotor speed. In order to induce the difficulty of hermetic seal, all components of compressor, including impeller, rotor, bearings and motor, are sealed at the same negative pressure condition.

In addition, because discharging voltage of helium gas is low, a low voltage type motor and AMBs were applied in this helium cold compressor.

3. Electronic control system of AMBs

Active magnetic bearings are typical mechatronics products. The main components of the electronic control system include the radial and axial AMBs, position sensors, DSP controllers, host computers, power amplifiers and so on. The simplified working principle, presented in the figure 2, can be explained as: a sensor measures the displacement of the rotor from its reference position, a microprocessor as a controlled devices a control signal from the measurement, a power amplifier transforms this control signal into a control current, and the control current generates a magnetic field in the actuating magnets, resulting in magnetic forces in such a way that the rotor remains in its hovering position [8]. The controller is responsible for the stability of the hovering state as well as the stiffness and the damping of such a suspension.

The controllers of AMBs were developed by digital signal processor (DSP/TMS320C6713), with a powerful floating-point operation of 1800 MFLOPS and one 300MHz ARM MPU core. The A/D converter AD7357 has two channels with 4.25MSPS rate and fourteen-bit precision, while the D/A converter AD5644R has four channels with fourteen-bit precision. The type of host computers is standard PXI industry computer. Based on host computer and LABVIEW (NI), monitoring system was developed. Position sensors are of induction type, made with silicon steel sheets in order to reduce eddy current losses. Switch amplifiers are applied there to reduce loss and increase efficiency. In addition, the type of auxiliary bearings are ceramic bearings of SKF. The system can achieve five channels displacement signals, ten current signals and speed signal acquisition, processing and display, current control signals calculation and amplification, diagnosing the states of the AMB operation and emergency treatment.

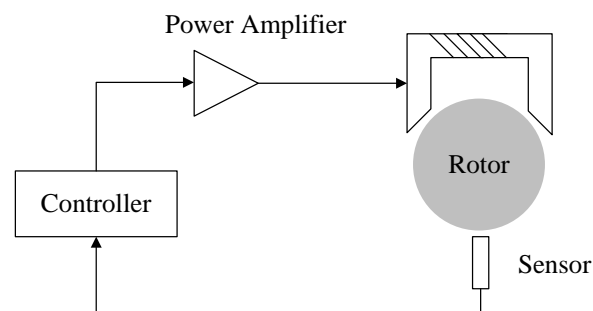


Figure 2. Schematic of the working principle of the AMB.

4. Structure design of axial and radial AMBs

As is shown in figure 1, the two radial bearings have the same structure, consisting of a stator part and a rotor part. Stator part includes the stator casing and coils of power winding. These coils form 8 radial oriented electromagnets, namely, 8 poles, with the pole sequence N-S-S-N-N-S-S-N. The rotor part of radial bearing represents cylindrical magnetic circuit, made of Si-alloyed transformer steels on the rotor. Electromagnetic force produced by radial AMBs used to control horizontal displacement of the rotor. Its maximum load capacity is suggested at 200N, mainly counteracting vibration force caused by imbalanced mass.

At the top end of the rotating shaft, the axial bearing consists of a stator and a thrust disk, includes two annular electromagnets and each electromagnet represents annular magnetic circuit with slots where coils of power windings can be put. Rotor part mounted between electromagnets represents thrust disk fixed on the shaft. The maximum load capacity of axial bearing is suggested at 500N, more than twice load capacity of radial bearings. That is because vertical magnetic force produced by axial bearing, is used to balance three kinds of forces as follows: the gravity of rotor and impellers mass; axial force of centrifugal compressor; aerodynamic disturbance force occurring in the vicinity of the fluid seals and impellers [10]. It should be noted that considering the upper axial bearing should undertake all the weight of rotor, the upper axial bearing, with larger magnetic area and more coil turns of magnetic pole,

was designed a little bigger than the lower in order to decrease heat-up of spindle. More detail parameters of the magnetic bearing are listed in table 1.

Table 1. Main parameter of the AMBs.

Parameter	Value
Radial AMB	
Number of magnetic poles	8
Load capacity (N)	200
Interior/outer diameter of stator (mm)	60/120
Axial length (mm)	20
Maximum current(A)	6.5
Radial air gap between bearing and rotor (mm)	0.35
Radial air gap between auxiliary bearing and rotor (mm)	0.1
Axial AMB	
Load capacity (N)	500
outer diameter of stator/rotor (mm)	80
Interior diameter of the upper/lower stator (mm)	50/62.5
Maximum current (A)	6.5
Axial air gap between bearing and rotor (mm)	0.6/0.4
Axial air gap between auxiliary bearing and rotor (mm)	0.2

5. Rotor dynamic analysis for cold compressor

When designing a stable AMB system, two key points need to be considered: first, ensure more than 20% separate margin between the rated rotation frequency and the bending modal frequency [11]; then provide proper stiffness and damper rates within working speed range and even when in higher high-order critical speed [12]. Therefore, it is particularly important to calculate modal analysis and critical speed of rotor for the choice of AMB control strategies, relating to the stability of whole centrifugal cold compressor operating.

Assuming that there is no vibrational coupling between bending in radial direction and axial vibration, it's reasonable to ignore axial stiffness. For simplifying the analysis, those two radial active magnetic bearings are simulated as two stiffness spring assembly, whose stiffness is constant and damping is zero. Here in the analysis, stiffness was set to be 5×10^5 N/m, obtained from "natural" stiffness value with the same order of magnitude as the displacement stiffness [13].

A sketch of the rotor model is shown in figure 3. Material property of its four main materials used central spindle is given in table 2. And figure 4 gives the results of rotor dynamic analysis.

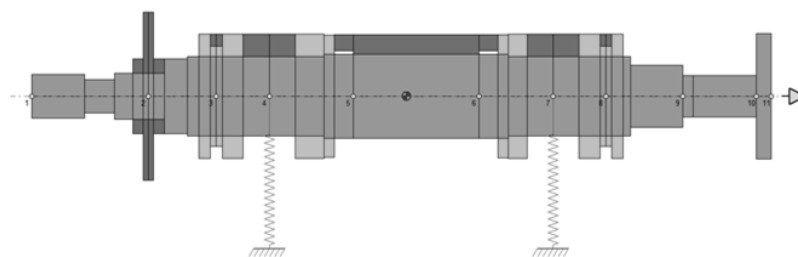


Figure 3. Model of the rotor.

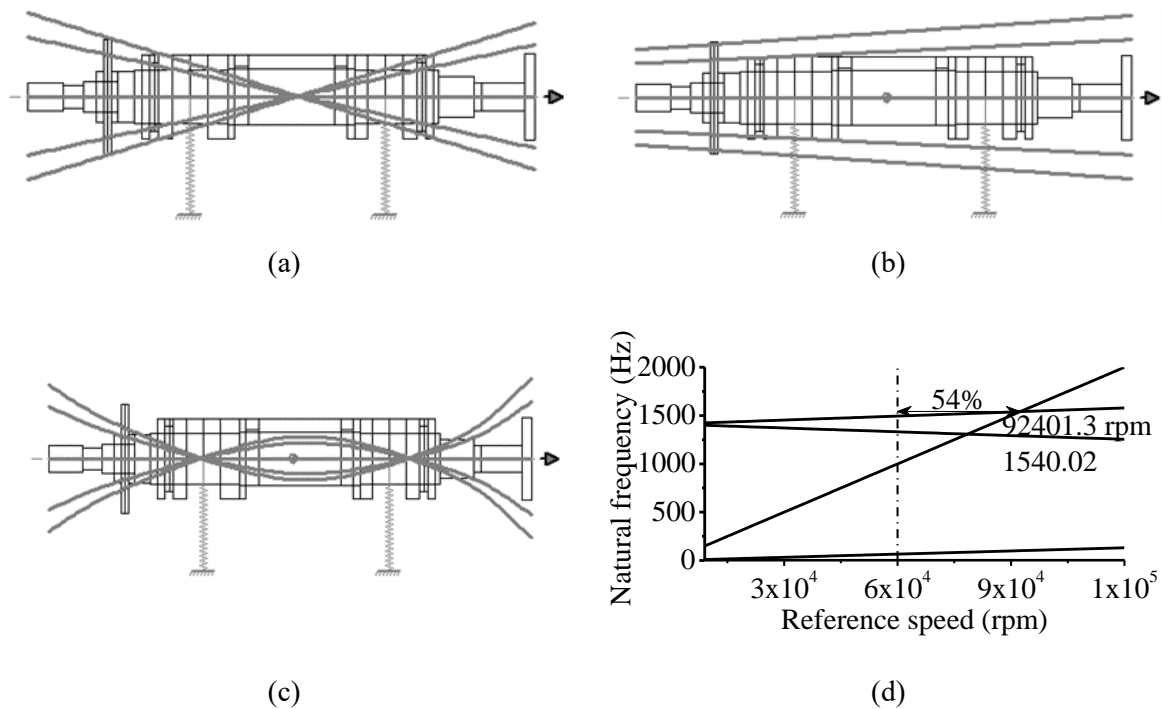


Figure 4. (a) 1st critical speed mode shape with modal frequency 63.98Hz; (b) 2nd critical speed mode shape with modal frequency 73.06Hz; (c) 3rd critical speed mode shape with modal frequency 1540.02Hz; (d) Campbell chart of the rotor.

From the results of rotor dynamic analysis, modal frequency, modal shape and critical speed can be well known and some conclusion can be drawn: (1) when the compressor is running at 1000Hz, vortex frequency of the first order bending mode is at 1540.02Hz, with 54% safe margin; (2) rotor can be considered as a rigid rotor with working speed no more than first order bending critical speed, conventional PID arithmetic, combining with phase compensation method, can control rotor well.

Table 2. Parameters of the rotor.

Density (kg/m ³)	7800
Young's modulus (N/m)	1.9E+11
Poisson ratio	0.28
shearing modulus (N/m ²)	7.90E+10

6. Simulation of rotor dynamic and AMB control

6.1. Mathematical model of AMB system

It's usual to linearized mathematical model of active magnetic bearing, whose analytical equation is quadratic and nonlinear in engineering application [14]. The linearized electromagnetic force equation is listed as (1). There, electromagnetic force labels f . In addition, the two important constants k_i (N/A) and k_x (N/m), are called current stiffness and displacement stiffness respectively. Considering measuring and controlling rotor lateral vibrations, take one of the radial magnetic bearings for an example, the build of mathematical model and design of control strategy is described in this section.

$$f = k_i i_x + k_x x \quad (1)$$

Transfer function of radial magnetic bearing are shown as (2), in which current stiffness and displacement stiffness is calculated by structural parameters of AMB stator listed in table 2.

$$G = \frac{k_i}{ms^2 + k_x} = \frac{28.4}{5.223s^2 - 264000} \quad (2)$$

The time delay of hardware in practical application cannot be avoided and it will cause phase lag of the whole system, which brings about the change of stiffness and damping. Therefore, it must be considered seriously in control simulation. The time delay caused by the processing time in order of magnitude of tens of nanoseconds, displacement sensor with a delay of half sampling rate due to zero-order holder, and power amplifier with about nanoseconds, was modeled by Pade approximation of third order based on MATLAB:

$$[\text{num}, \text{den}] = \text{pade}(1e^{-4}, 3) \quad (3)$$

In the radial direction decentralized system, a feedback control strategy—combine PID arithmetic with phase compensation methods, is prepared to be executed in the actual system. The principle of design controller is to adjust the gain and phase of open-loop transfer function, provide suitable stiffness with an order of magnitude the displacement stiffness and damping ratio with a range between 0.2 to 0.8, which guarantees the rotor stable and good performance. The two radial magnetic bearings have the same control strategy and take for example the upper AMB controller. The transfer function of PID and other filters is as (4) - (6).

$$F_{\text{PID}} = 1.5e^6 \cdot \frac{\left(1 + \frac{s}{2\pi \cdot 300}\right)\left(1 + \frac{s}{2\pi \cdot 400}\right)}{\left(1 + \frac{s}{2\pi \cdot 700}\right)\left(1 + \frac{s}{2\pi \cdot 800}\right)} + \frac{1}{1 + \frac{s}{2\pi}} \quad (4)$$

$$F_1 = \frac{1}{\left(1 + \frac{0.2s}{2\pi \cdot 1900}\right) + \left(1 + \frac{s^2}{2\pi \cdot 1900}\right)} \quad (5)$$

$$F_2 = \frac{1 + \left(\frac{s}{2\pi \cdot 5100}\right)^2 + \left(\frac{0.2s}{2\pi \cdot 500}\right)}{1 + \left(\frac{s}{2\pi \cdot 4950}\right)^2 + \left(\frac{0.2s}{2\pi \cdot 4950}\right)} \quad (6)$$

6.2. Simulation results

With the control strategy designed above, simulation results can be obtained. Bode diagram of the controller as well as the stiffness and damping ratio corresponding to different frequency are shown in figure 5. It presents that the controller can offer decent positive stiffness with the same order of magnitude as the displacement stiffness as well as proper damping. By further calculation, controller guarantees damping ratios of the parallel and conical mode with frequency of 104.41Hz and 144.90Hz at 0.34 and 0.32, within working rotation speed. The first bending mode frequency is at 1399.42Hz, with 39.94% safe margin and its damping ratio value is at 0.21. In addition, as is shown in sensitivity function plot of the AMB system in figure 6, according to the latest ISO standard for the assessment of AMB system robustness, the sensitivity function peak values are well no more than 3, the system with the control strategy described above can be considered a “Zone A” system, which is optimally robust to plant uncertainty [15]. To verify the dynamic performance of rotor, the response of the rotor displacement in the position of upper magnetic bearing to a step reference input of about 3.25A is shown in figure.7. It shows that the displacement in perpendicular direction reached its peak 0.06mm at time

0.0054s, and at last kept in the central location at 0.05mm. Simulation results demonstrate that the control method proposed in the paper not only can stable the AMB system but also achieved good performance in both transient and steady state periods.

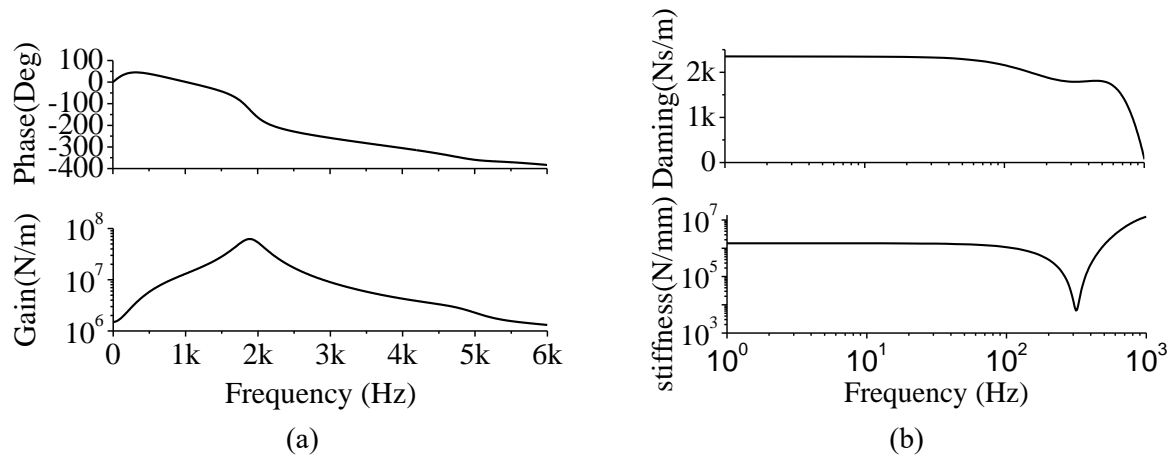


Figure 5. (a) Phase and gain in different frequency; (b) Damping ratio and stiffness in different frequency.

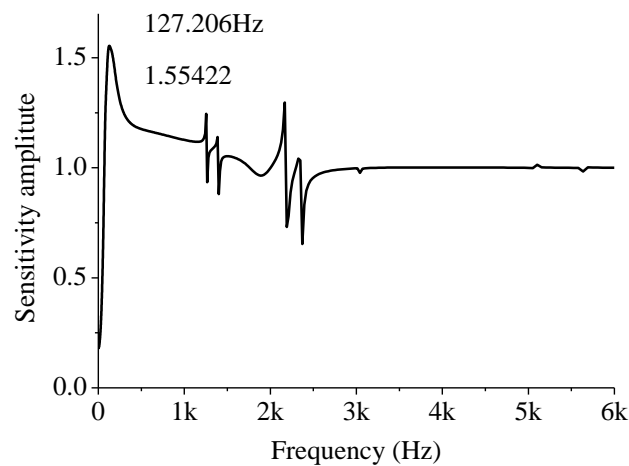


Figure 6. Sensitivity function of the AMB system.

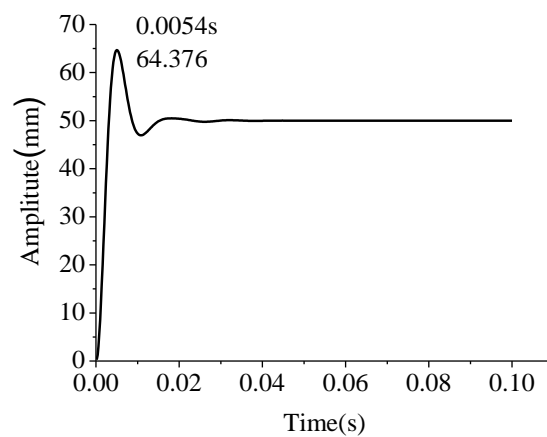


Figure 7. Response of rotor displacement to step signal.

7. Conclusion

AMB have many advantages over other traditional bearings and is considered as the best support assembly for cold compressor. This will bring opportunity and challenge for AMB applied in high-speed rotating machines of low-temperature refrigeration system. The paper discussed structure characteristics, hardware composition and control simulation of AMB system in detail, offer important theoretical and experimental foundation for the further application.

However, the AMBs is still in stage of design and manufacture, so calculation and simulation result is based on design results. Due to assembly and other error, deviation between design parameter and actual parameter of AMB will be inevitable. In order to design more accurate and proper controller based on the same control strategies depicted in the paper, it is necessary to do some extra experiment, for example, validation of current stiffness and displacement stiffness value by mechanical admittance method, identification of the AMB plant transfer function by frequency response test.

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