HOUSING VIBRATION CONTROL OF A LARGE ROTOR SUSPENDED BY MAGNETIC BEARINGS

Kai Zhang, *Lei Zhao, Hongbin Zhao

Department of Engineering Physics, Tsinghua University, 100084

*Institute of Nuclear and New Energy Technology, Tsinghua University, 100084

ABSTRACT: This paper introduces a test rotor system suspended by active magnetic bearings for HTR-10 setup, whose mechanical structure is hard to be firm because of space limit and compact design requirements. The system has two severe resonances at a middle frequency band. Housing vibrations coming from the structure resonances can destroy the stability of the system. Special controller designs are needed to restrain them, or even static suspension stability cannot be achieved. A non-parametric frequency domain method was used to identify the system model. The frequency of one of the structure resonances varied markedly with controller stiffness. Modeling the resonances was complex, and a convenient method based on an iterative process of identification and controller parameter tuning was used to deal with the resonances. In controller designs of the process, zero-pole pair method and μ-synthesis method were applied respectively. Their effects were proved by experiments, and their advantages were compared. Such methods are useful in restraining the housing vibrations and are valuable for further work of HTR-10 setup.

KEY WORDS: High Temperature Gas Cooled Test Module Reactor (HTR), Active Magnetic Bearing (AMB), Vibration Control

0. INTRODUCTION

The 10MW high temperature gas-cooled reactor (HTR-10) has been constructed by INET at Tsinghua University of China. It is the first module high temperature gas-cooled test reactor in the world. The second phase of HTR-10 is to set up a direct helium cycle to replace the current steam cycle. The Active Magnetic Bearings (AMB) is chosen to support the shaft due to its advantages such as no wear, longer life, no lubrication and no irradiation in the primary loop, etc.

In rotor system designs, influences of structure resonances of system housing should be avoided. They can lead to undesirable performance. Generally, system housing is mechanically designed to be stiff enough to avoid unwanted low natural frequencies. In the large rotor system suspended by active magnetic bearings, the mechanical structure of system is not so firm because of space limit and the stator size, which is adjacent to the rotor size, and the housing vibrations are easy to be excited. It should be considered carefully. Alternatively, the structure resonances can destroy the stable suspension of the rotor. The test rotor system mentioned in this paper is used to verify housing vibration control ability of AMBs for HTR-10 setup. The system has two severe resonances at a middle frequency band. Without careful controller design to restrain them, even static suspension stability cannot be achieved.

The non-parametric frequency domain method is applied to identify the system model. Two methods using zero-pole pairs and μ synthesis theory respectively are studied and applied to the system. The frequency of one of the structure resonances obviously varies with controller stiffness. To deal with it, the controller design becomes an iterative process of identification and controller parameter tuning. The effects of the two methods in restraining the vibrations of the housing are proved by the experiments, and their advantages are compared. Such methods are useful in dealing with the housing vibrations.

1. MODEL

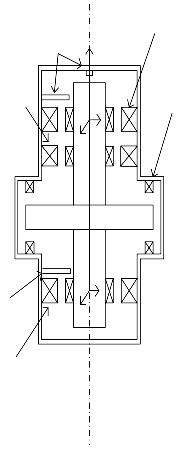

FIG 1. Schematic structure of the rotor system

The schematic structure of the rotor system is shown in FIG 1. Some rotor parameters are as follows:
Mass: 120 kg; *Bearings*: 5-DOF magnetic bearings; *Running speed*: 2400 RPM; x_1: x direction of the upper radial bearing; y_1: y direction of the upper radial bearing; x_2: x direction of the lower radial bearing; y_2: y direction of the lower radial bearing.

Coupling of the rotor's axial movement and radial movement was ignored. The axial AMB of the rotor was controlled by an analog PID controller. The work concentrates on the radial AMBs.

2. SYSTEM IDENTIFICATION AND CONTROLLER DESIGN

To effectively design a suitable controller, identification of the model is necessary. Since AMBs are open-loop unstable, all experiments must be performed in closed-loop. A controller that can stabilize the rotor should be found first. The normal PID controller without considering the housing vibration was tried. In static suspension experiments, the housing vibrations were excited. The stiffness of the controller had to be decrease. At last, the PD controller with very low stiffness was used and suspended the rotor stably, and the identification could be carried out then.

Because the different parts of the system are coupled, theoretically modeling the housing vibrations was difficult. The non-parametric frequency domain method [2, 3] was chosen to identify the open loop model of the system. The identification frame is shown as FIG 2, and the following relationships are obtained.

$$T_{ur} = u/r,\ T_{yr} = y/r \tag{1}$$

$$G = y/r = T_{yr}/T_{ur} \tag{2}$$

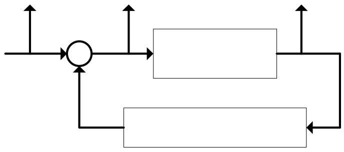

FIG 2. Identification Frame

The closed-loop system identification procedure is divided into two steps. T_{ur} and T_{yr} are gotten respectively by sine sweep first. Then data of the plant G is achieved through the formula (2). Such a procedure can avoid noise biasing problems. In the identification, the magnets were used as exciters and the sine sweep signal as r was directly added to the control signal of the closed-loop system. Data of u and y was recorded simultaneously.

In the identification, models of the 16 channels, such as G_{x1x1}, G_{x1x2}, G_{x1y1}, G_{x1y2}, G_{x2x1}, G_{x2x2}, G_{x2y1}, G_{x2y2}, etc., are to be identified. To simplify controller design, the controllers discussed here are all based on SISO models, identification work is focused on G_{x1x1}, G_{x2x2}, G_{y1y1} and G_{y2y2}. Furthermore, G_{x1x1} is similar with G_{y1y1}, and G_{x2x2} is similar with G_{y2y2}. The identification result of G_{x1x1} is shown as FIG 3.

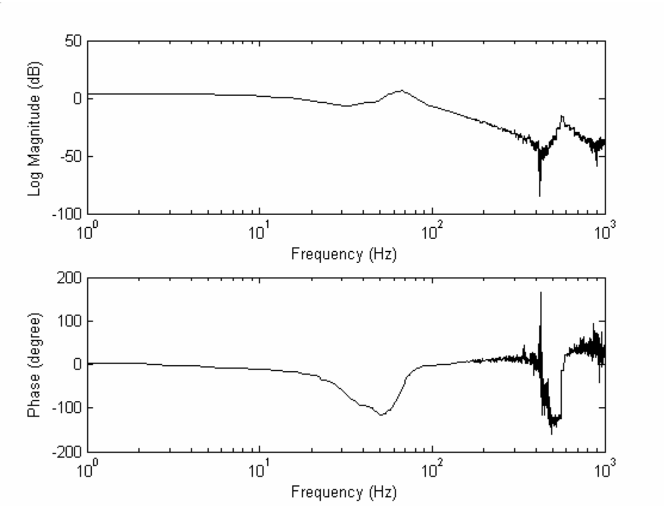

FIG 3. Identification result of G_{x1x1}

It is seen that there are 2 resonance peeks at about 78 Hz and about 590 respectively. With preliminary analysis of the models and the experimental results, it is found that the resonance of 590 Hz mainly comes from the base vibration of the rotor system, the origin of the resonance of 78 Hz is much more complex and it depends on different factors. The connection of the stator with the housing and the closed-loop stiffness between the rotor and the stator are both influenced its resonance frequency. Building a precise mathematical model was difficult, and a simple controller design method was used to deal with the problem. It is an iterative process. G_{x1x1} is selected as the controlled object. The controller designs of the other channels as G_{x2x2}, G_{y1y1}, and G_{y2y2} are similar with the controller designs of G_{x1x1}.

The iterative process is as follows: 1. A controller with slightly higher stiffness is designed based on the model identified from the closed-loop system with the PD controller described in the preceding section. 2. The model controlled with the new controller is identified again. 3. the resonance frequency of 78 Hz increases and a new controller with higher stiffness can be designed based on the new model. 4. The iteration continues until a controller with suitable stiffness is obtained. The controller design process is shown as FIG 4. The two different methods are used respectively in the iterative process to design controllers that can stabilize the rotor with suitable stiffness.

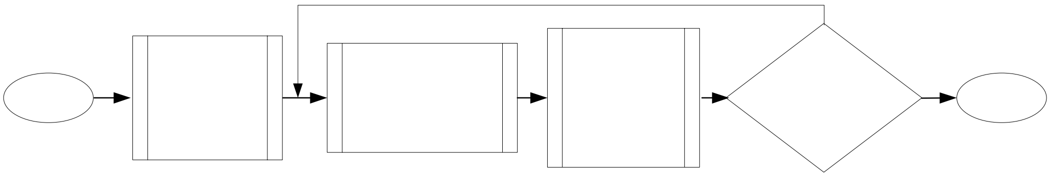

FIG 4. Controller design process

The first method stabilized the rotor by adding two zero-pole pairs to damp the two resonances of the housing respectively. The expression of zero-pole pair was $K(s\text{-}a_1\text{-j}b_1)(s\text{-}a_1\text{+j}b_1)/(s\text{-}a_2\text{-j}b_2)(s\text{-}a_2\text{+j}b_2)$. The two zero-pole pairs were parallel. In addition, they were inserted to D(s) channel of a PID controller and were connected in series with D(s). Parameters as k, a_1, b_1, a_2, and $b2$ were adjusted corresponding to the model identified. It was experimental and quite a few iterations were needed to do. The damping of the closed-loop system at the resonance frequencies was improved markedly. The final zero-pole pairs used are as the formula (3).

$$0.25\frac{s^2+400s+7.07\times10^5}{s^2+400s+1.05\times10^6}\quad,\quad0.75\frac{s^2+600s+1.20\times10^7}{s^2+600s+1.94\times10^7} \tag{3}$$

Define K_{pd} as the controller without the zero-pole pairs added, and define K_{zp} as the controller with the zero-pole pairs added. The bode diagram of K_{pd} and K_{zp} is shown as FIG 5.

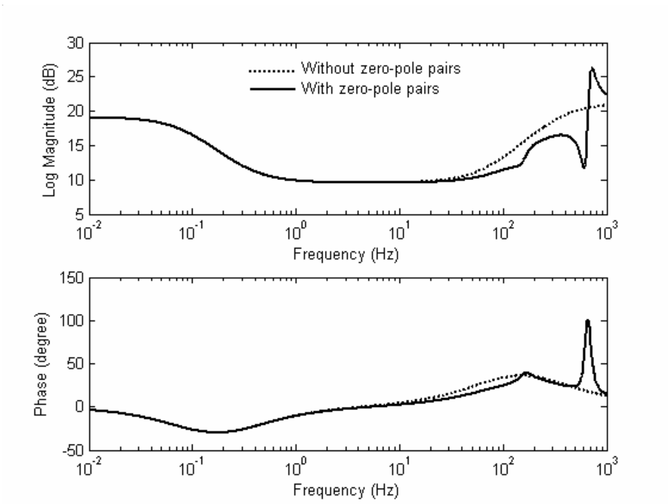

FIG 5. Bode diagram of Kpd and Kzp

After several iterations, the final identification result of G_{x1x1} with K_{zp} as the controller is shown as FIG 6. It is found that the original resonance frequency of 78 Hz has changed to about 105 Hz.

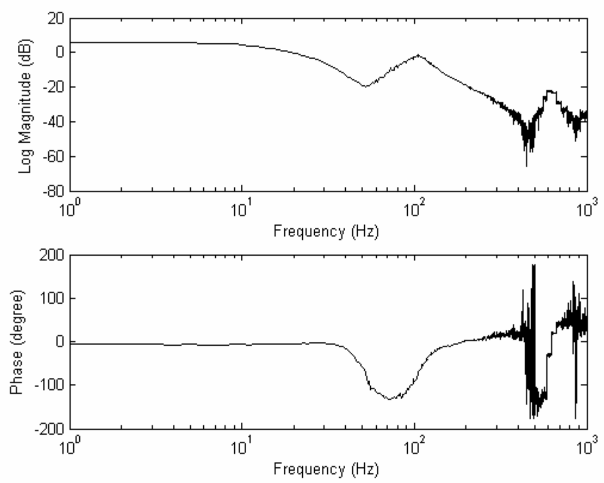

FIG 6. Final identification result of Gx1x1

The controller design method based on μ synthesis is used too. The μ controllers with good performance in the experiments were obtained. The nominal model G_n used was a simple second-order model, and it was based on theoretical analysis [1]. Its transfer function is $G_n = K_i K_s/(ms^2 + K_x)$, where K_i is the force-current factor of the magnetic bearings, K_s is the sensor gain, m is the mass of the rotor and K_x is force-displacement factor of the magnetic bearings. The bode diagram of G_n is shown as Line 1 in FIG 7.

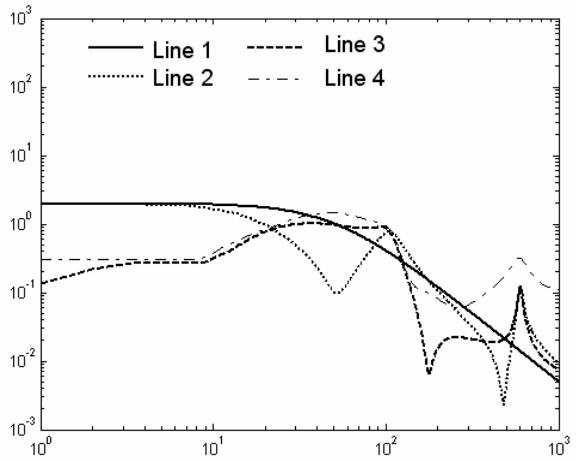

FIG 7. Bode diagram of Gnorminal, Gx1x1, etc.

Moreover, Line 2 in FIG 7 is the fitting for the identification data. Line 3 is the difference between Line 1 and Line 2. Line 4 is constructed based on Line 3 and represents the uncertainty of the nominal model. It is an additive uncertainty and is used in the μ controller designs. The model used to design the robust controller is as FIG 8.

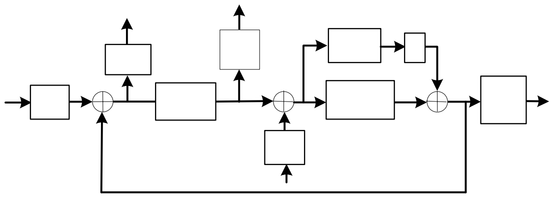

FIG 8. μ model used to design the robust controller

The weighting functions were chosen according to the system parameters and the performance requirements [4, 5]. W_{del} can be seen in FIG 7 and the other weighting functions are shown as below.

$$W_d = 80 \qquad W_y = 2.4 \times 10^{-3} \times \frac{1.6s + 20}{1.6s + 1} \qquad W_r = 8.2$$

$$W_u = \frac{12.2s^2 + 7.7 \times 10^4 s + 4.8 \times 10^8}{s^2 + 6.3 \times 10^5 s + 3.9 \times 10^{11}} \qquad W_e = 0.1$$

$$\begin{bmatrix} y'(s) \\ e'(s) \\ u'(s) \end{bmatrix} = \begin{bmatrix} W_y T(s) W_r & W_y S(s) G(s) W_d \\ W_e S(s) W_r & -W_e S(s) G(s) W_d \\ W_u C(s) S(s) W_r & -W_u T_i(s) W_d \end{bmatrix} \begin{bmatrix} r(s) \\ d(s) \end{bmatrix}$$

After three DK-iterations, the μ controller K_μ was achieved. Its bode diagram is shown as FIG 9. Its performance was tested by the experiments.

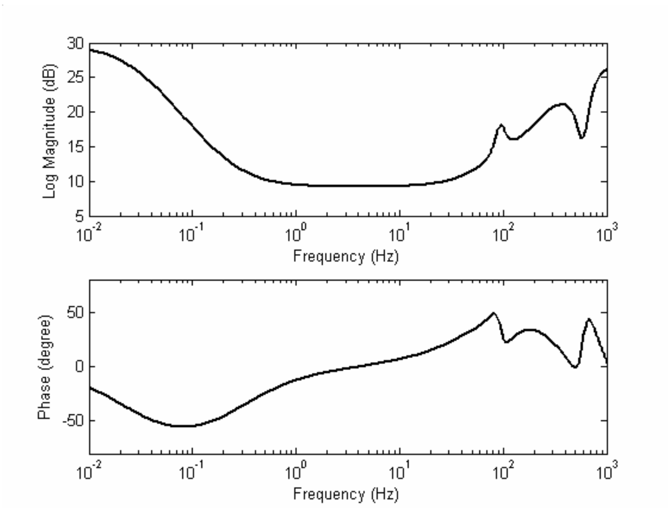

FIG 9. Bode diagram of Kμ

3. EXPERIMENTS

In the experiments, K_{zp} and K_μ were used to suspend the rotor statically respectively, and they could both well restrain the resonances of the housing. No resonance was observed in FFT analysis of captured data. Running experiments were carried out using the two controllers respectively. The speeds were both above 2400 rpm. The running status of the rotor suspended by K_μ is shown in FIG 10.

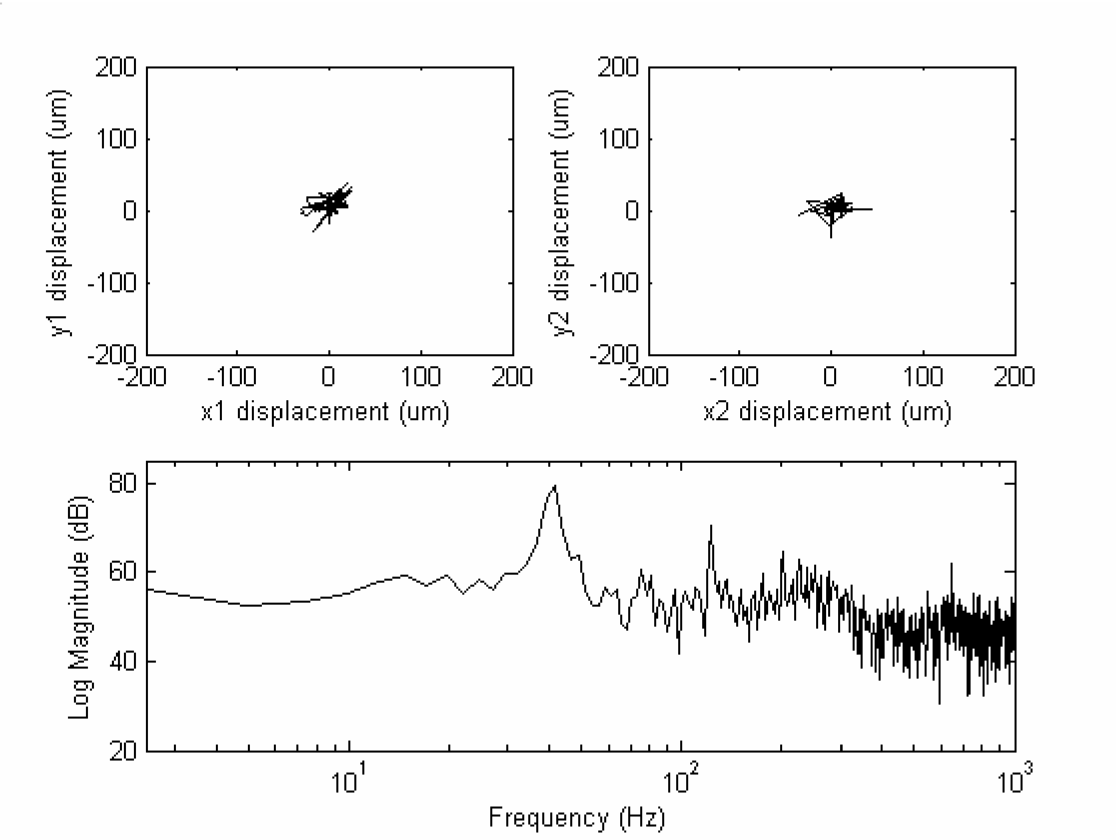

FIG 10. Running status of the rotor

In experiments, it was found that both of the controller design methods for the rotor suspension and the housing vibration restraining were effective. Parameter tuning of K_{zp} was much more experimental. Designers had to try many iterations before getting a suitable controller. However, the order of K_{zp} was much lower than K_{μ}. Generally speaking, the design method of K_{μ} was more convenient to use.

4. CONCLUSIONS

The test rotor suspended by AMBs for HTR-10 was studied and the housing vibrations of the rotor system greatly influenced the system stability. The low gain controller could suspend the rotor. Then the system model was obtained by the identification based on the non-parametric frequency domain method. One of the two structure resonance frequencies varied markedly with controller stiffness. A convenient method based on an iterative process of identification and controller parameter tuning was used to deal with the resonances. In controller designs of the process, zero-pole pair method and μ-synthesis method were applied respectively. Order of the controller designed by the zero-pole pair method was lower while the method was more experimental, whereas the μ-synthesis method was more convenient to use. Their effects were both proved by the experiments. The housing vibrations were well restrained. The study will be valuable for the further work of HTR-10 setup.

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AUTHOR INTRODUCTION

Kai Zhang, born in 1977, now is a PHD candidat of Department of Engineering Physics , Tsinghua University. He got his Bachelor degree in 2000 from Tsinghua University. He now researches on active magnetic bearing and control theory application. (zhangk00@mails.tsinghua.edu.cn)