A Design of Robust Servo Controllers for an Unbalance Vibration in Magnetic Bearing Systems

F. MATSUMURA, M. FUJITA, and C. OIDA

Department of Electrical and Computer Engineering,
Kanazawa University, 2-40-20 Kodatsuno, Kanazawa 920, Japan

Summary

This paper deals with the problem of an unbalance vibration in magnetic bearings. An efficient damping control scheme for the vibration is proposed. First, the state equation of a magnetic bearing system in which unbalance of a rotor is taken into consideration is shown. Next, the control system achieving the rotation around the geometrical axis is proposed. Here we formulate the control problem within a framework of the so-called Robust Servomechanism problem, and we design the control law using the Selective Turn-over method. Finally, the effectiveness of the control system is actually confirmed by several experimental results.

1. INTRODUCTION

In order that a rotor supported by magnetic bearings rotates around a certain axis smoothly, the rotor must be symmetric to the axis. But there exist residual unbalance on the rotor. By this unbalance, a shaft vibration is arisen when the rotor rotates at high speed. In this paper, based on modern control theory, we design a control system achieving a reduction of the unbalance vibration.

The following is discussed for a magnetic bearing combining radial control and thrust control [1], however, it can be applied to general magnetic bearing systems also.

2. PROBLEM STATEMENT

The state equation and the output equation of the magnetic bearing system in which an unbalance of a rotor is taken into consideration can be written as follows.

$$\dot{x} = Ax + Bu + D_1 w_1 + D_2 w_2 \tag{1}$$
$$y = Cx \tag{2}$$

where w_1 is a vector of sinusoidal disturbances caused by unbalance and w_2 is a vector of constant disturbance to this system. Each disturbance is not included in the equation given in [1]. These disturbances are arranged as follows.

$$\dot{w} = Fw \tag{3}$$

Above equations show that when a rotor rotates around the geometrical axis, sinusoidal disturbances arise a shaft vibration.

3. DESIGN OF ROBUST CONTROLLER

We introduce robust control [2] in this system, with which our problem can be solved. This system is augmented with a necessary number of servocompensator, which is described as the following equation,

$$\dot{x}_c = A_c x_c + B_c e \tag{4}$$

in cascade with each input line. The equation of the augmented system is as follows.

$$\begin{bmatrix} \dot{x} \\ \dot{x}_c \end{bmatrix} = \begin{bmatrix} A & 0 \\ B_c C & A_c \end{bmatrix} \begin{bmatrix} x \\ x_c \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u \tag{5}$$

where

$$A_c = \text{block diag}[F, F, \ldots, F]$$
$$B_c = \text{block diag}[K, K, \ldots, K]$$

$$F = \begin{bmatrix} 0 & p & 0 \\ -p & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \qquad K = \begin{bmatrix} 1 \\ 0 \\ 1 \end{bmatrix} \qquad \text{p: angular frequency}$$

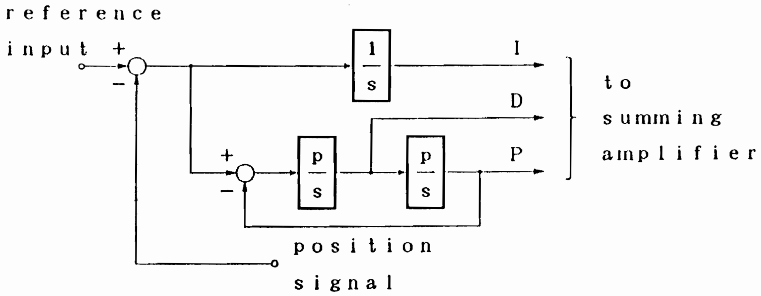


Fig.1. servocompensator

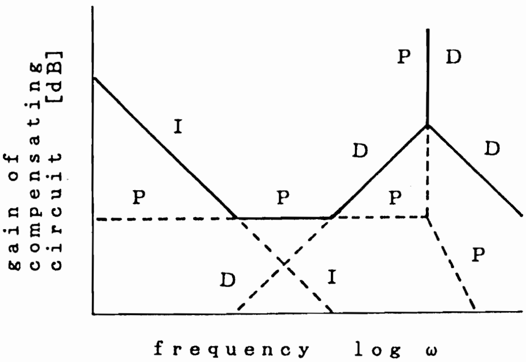


Fig.2. Frequency characteristic

The structure of servocompensator is shown in Fig. 1. This compensator consists of an integrator corresponding to internal model of step type disturbance and an oscillator corresponding to internal model of sinusoidal disturbances. Fig. 2 shows the frequency characteristic of this compensator. The line P shows the response of second order system with sharp resonance, the line D the differential of it and the line I the response of the integrator. The synthesized response is shown by solid line. This line and inertia stiffness make the frequency characteristic of stiffness. A sharp peak is realized at resonance angular frequency p.

Designing a compensator behaves according to variable rotational speed, sinusoidal disturbances can be reduced in a comprehensive range.

4. STABILIZATION

For the stabilizing of the augmented system (5), we use the Selective Turn-over method, with which we can allocate all the closed loop poles in the desirable region. As Fig. 3, any poles on the right space of the line can be folded back symmetrically with respect to an arbitrary straight line ($Re\lambda = -k$) parallel to the imaginary axis. Selecting the line reasonably, imaginary part of eigenvalue of closed loop system can be specified arbitrarily. Allocated whole poles in stable space, it is proved that a derived control system is optimal regulator.

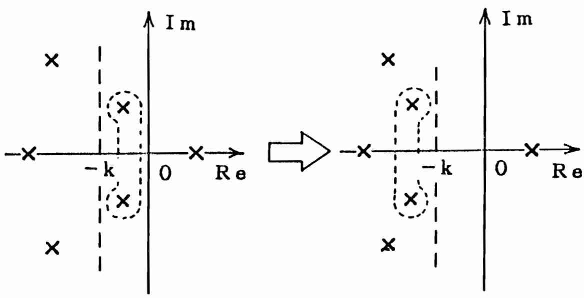


Fig.3. Selective Turn-over method

We stabilized this system according to the following procedure on the assumption that the rotational speed is 1900rpm.

(1) We select a line ($Re\lambda = -10$) for the poles of an original system and an integrator. In earlier study [1], stability have been affirmed by this design.

(2) Since the poles of vibration is far from real axis, a little movement of poles makes feedback gain large. This leads a saturation of a control circuit. Considering this circumstance, we select a line ($Re\lambda = -1$) for the poles of an oscillator.

The relationship of eigenvalues between an open loop system and its corresponding closed loop system is shown in Fig. 4. The pole ($\lambda = 0.0$) is a pole of an integrator, the poles ($\lambda = 0.0 \pm j199$) are those of an oscillator, the others are those of a magnetic bearing system.

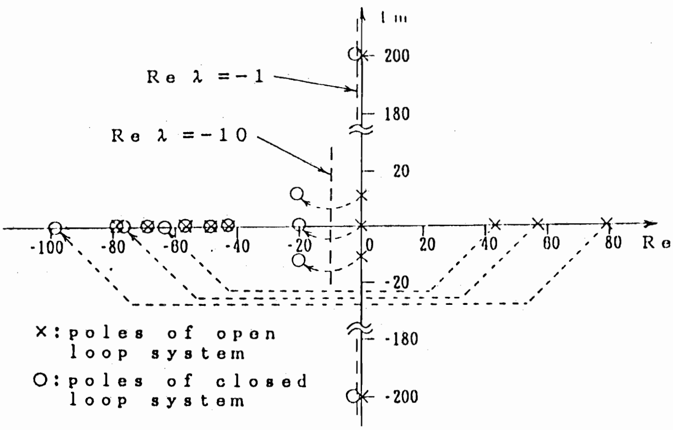


Fig.4. Poles of open and closed loop systems

5. EXPERIMENTAL MACHINE

The structure of the experimental machine is shown in Fig.5. There are a cage type induction motor at the center, electromagnets and radial position sensors on both sides, a dummy weight on the left side, and an axial position sensor on the right side.

Due to the staggered arrangement, radial and thrust direction control are achieved by only four pairs of magnet.

The rotor is 72mm in diameter at the electromagnets and the motor, 540mm in total length, 6.64kg in weight, and 4mm in gap length. The gap is wider than that of usual machines because only the rotor might be put in a vacuum with a glass tube. That is, this machine was designed for the purpose of an rotating anode X-ray tube.

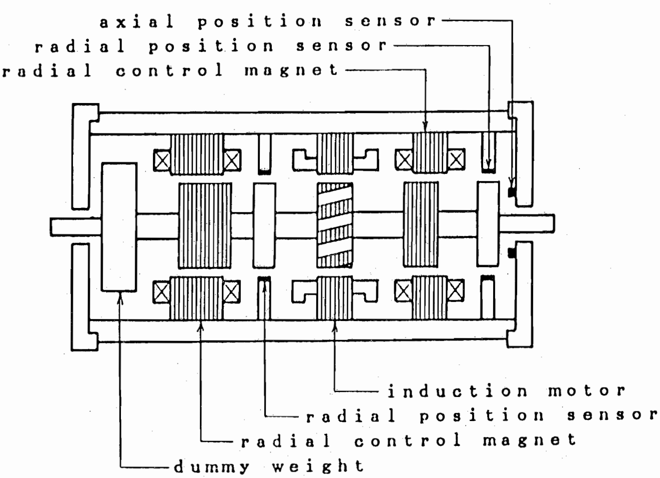


Fig.5. Structure of experimental machine

6. EXPERIMENTAL RESULTS

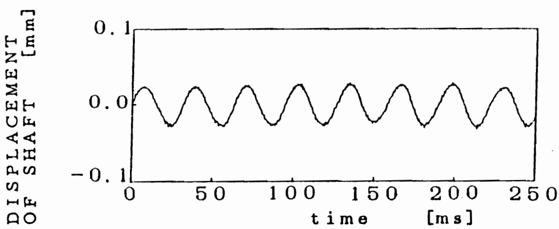
The following results are characteristics of the system designed for 1900rpm.

Fig.6 shows a displacement of the shaft (radial horizontal direction) rotating at 1900rpm. The case unbalance is not considered is also shown. As is evident from Fig.6, proposed control system reduces the unbalance vibration effectively.

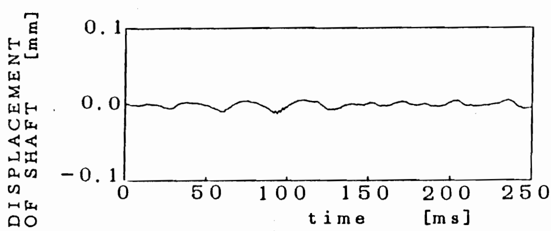
Fig.7 is one of the characteristics changing a rotational speed from 1600 to 2000rpm. As an example, the case of 1900rpm is shown. It is similar to the preceding result and at the other rotational speed are so. This reveals that the unbalance vibration is reduced efficiently according to the changing rotational speed.

Fig.8 shows transient responses for the 3N step type disturbance force in a radial horizontal direction. The result shows that the control system achieves asymptotic disturbance rejection. This means that proposed control theorem enables us to achieve the reduction of unbalance vibration and the stability.

Fig.9 shows a frequency characteristics of stiffness for radial disturbance force. The stiffness at rotational frequency and nearby 0Hz is extremely high. This reveals that both step type and sinusoidal disturbances are almost eliminated.

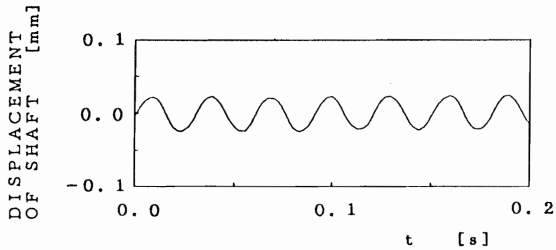


(a) Case of not realizing the damping of the vibration



(b) Case of realizing the damping of the vibration

Fig.6. Displacement of the shaft



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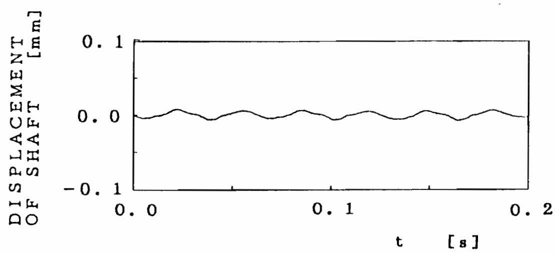

(b) case of realizing
the damping of the vibration

Fig.7. Displacement of the shaft

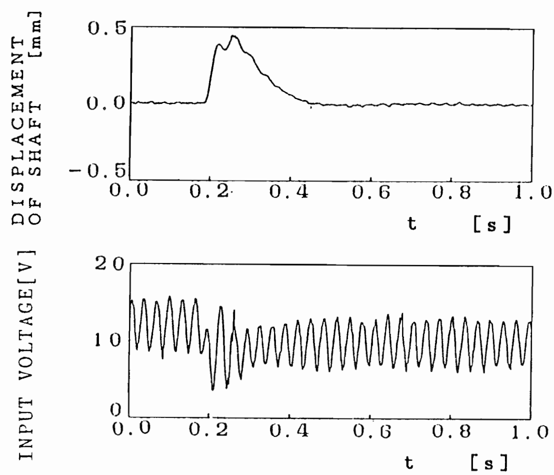

Fig.8. Response for step type disturbance

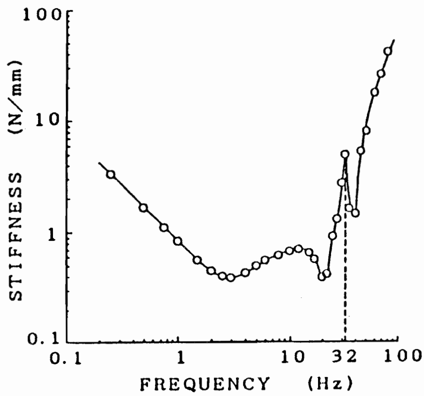

Fig.9. Frequency characteristic of stiffness

7. CONCLUSION

In order to reduce the unbalance vibration of a magnetic bearing systems, Robust Servo controller has been designed. The application of the Selective Turn-over method enables us to achieve the reduction of vibration and the stability. The effectiveness of the proposed controller has been demonstrated with several experimental results.

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