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# DYNAMIC CHARACTERISTICS OF THE MAGNETIC BEARING SYSTEM WITH THE MAGNETIC DAMPER

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## ABSTRACT

The magnetic damper is introduced into the ordinary magnetic bearing system. Dynamic characteristics of the magnetic bearing system with the magnetic damper are investigated by the theoretical calculation, the experimental modal analysis, and the actual operation of the system, respectively. Compared with the ordinary magnetic bearing system, the modal damps on the 1st and 2nd bending critical speeds have increased a lot, the vibration of the rotor has been reduced, and the system can get across the first two bending critical speeds, and operate on 16,800 rotations per minute safely.

## INTRODUCTION

Compared with the traditional bearings, the magnetic bearing system has many advantages, such as non-contact supporting and active control capability. However, it is still difficult to apply the magnetic bearing into the actual flexible rotor system, because the supporting damp is so light that the vibration of the rotor can not be restrained, especially when the system operates near or over the bending critical speeds [1].

Many methods are introduced to solve the problem so far. In the literature [2], mode separation control method and modal balancing method are adopted to design the control system, and the AMB-supported rotor is successful to pass the 3rd bending critical speed safely. On-line closed-loop active balancing facility is adopted for a supercritical rotor on AMBs in the literature [3]. LOQ is adopted for a small experiment system in the literature [4], and the system has passed the first bending critical speed.

Differing from the above, the magnetic damper is introduced into the ordinary magnetic bearing system in this paper, in order to increase the supporting damp and help the system to pass the bending critical speeds safely. To avoid redundant restraint, the magnetic damper can support only damp but no stiffness.

If no control is adopted, the magnetic bearing has negative stiffness, and the system is not stable. Only when suit control strategy and suit control parameters are adopted, the magnetic bearing can support suit positive stiffness and damp. And the system is stable and has better dynamic characteristics. This is the ordinary magnetic bearing system which has been investigated in the related literatures. To restrain the vibration of the rotor, another magnetic bearing is introduced into the ordinary magnetic bearing system, and suit control parameters are adopted to ensure that the stiffness of the magnetic bearing is equal or near equal to zero in our work. Therefore the magnetic bearing becomes a magnetic damper.

The magnetic bearing system with the magnetic damper is developed in the paper. Dynamic characteristic of the system are firstly investigated by the theoretical calculation, the experimental modal analysis, and the actual operation of the system. Then, the analytic results are compared with that of the ordinary magnetic bearing system. The experimental setup is shown in Fig. 1. The rotor weighs about 7.35 kilogram and is 828mm long. The distance between 1# radial bearing and the magnetic damper is 184mm, while the distance between 2# radial bearing and the magnetic damper is 519mm.

## SYMBOLS AND NOTATIONS

$k_{rs}$: displacement stiffness coefficient
$A_s$: plus coefficient of sensor segment
$T_s$: time constant of displacement segment
$A_p$: plus coefficient of amplifier segment
$T_p$: time constant of power amplifier segment
$k_{\mathrm{p}}$: proportional coefficient of controller
$k_{\mathrm{i}}$: integral coefficient of controller
$k_{\mathrm{d}}$: differential coefficient of controller
$T_{\mathrm{d}}$: differential time coefficient of controller
$K_{\mathrm{eq}}$: equivalent damp of damper

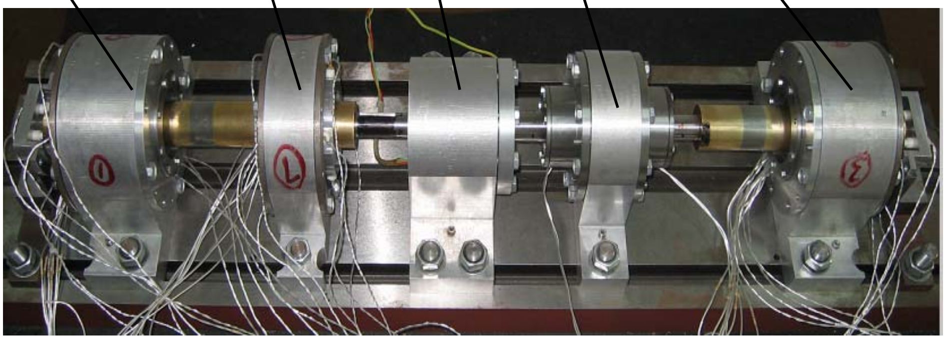


Fig. 1 The experimental setup.

## MODEL OF MATHEMATICS

In the AMB rotor system, $x$, $y$, $z$, $\varphi$, $\psi$ are displacements in five degrees of freedom, respectively. For convenience, non-dimensional parameters are introduced as follows,

$$\bar{x} = x/d_0\ ,\quad \bar{y} = y/d_0\ ,\quad \bar{z} = z/d_0\ ,$$

$$\bar{\varphi} = \varphi\ ,\quad \bar{\psi} = \psi\ ,$$

where, $d_0$ is the average diameter. The rotor is divided into 79 nodes. The magnetic bearings are on the 6th and 74th node respectively, while the magnetic damper is on the 54th node. The vibration of the rotor in radial freedoms is measured by the sensors on the 3rd, 9th, 51st, 57th, 71st and 77th node. Non-dimensional equations of the system can be written as follows,

$$\begin{bmatrix} \mathbf{0} & \bar{\boldsymbol{M}} \\ \bar{\boldsymbol{M}} & \bar{\boldsymbol{C}} \end{bmatrix} \dot{\boldsymbol{R}} + \begin{bmatrix} -\bar{\boldsymbol{M}} & \mathbf{0} \\ \mathbf{0} & \bar{\boldsymbol{K}} \end{bmatrix} \boldsymbol{R} = 0 \qquad (1)$$

where, $\boldsymbol{R} = [\boldsymbol{R}_1, \boldsymbol{R}_2]^T$ , $\bar{\boldsymbol{M}}$ , $\bar{\boldsymbol{C}}$ and $\bar{\boldsymbol{K}}$ are non-dimensional matrixes of mass, damping, and stiffness, respectively. $\bar{\boldsymbol{C}}$ and $\bar{\boldsymbol{K}}$ are functions of $A_s$, $T_s$, $A_p$, $T_p$, $k_p$, $k_i$, $k_d$, $T_d$, $k_{eq}$, and $k_{rs}$.

Here,

$$R_1 = [\dot{\bar{x}}_1, \dot{\bar{y}}_1, \dot{\bar{\varphi}}_1, \dot{\bar{\psi}}_1, \dot{\bar{x}}_2, \dot{\bar{y}}_2, \dot{\bar{\varphi}}_2, \dot{\bar{\psi}}_2, \cdots\cdots, \dot{\bar{x}}_{79}, \dot{\bar{y}}_{79}, \dot{\bar{\varphi}}_{79}, \dot{\bar{\psi}}_{79}]^T$$

$$\boldsymbol{R}_2 = [\bar{x}_1, \bar{y}_1, \bar{\varphi}_1, \bar{\psi}_1, \bar{x}_2, \bar{y}_2, \bar{\varphi}_2, \bar{\psi}_2, \cdots\cdots, \bar{x}_{79}, \bar{y}_{79}, \bar{\varphi}_{79}, \bar{\psi}_{79}]^T$$

## CALCULATION RESULTS

The actual unbalance of the rotor can not be known. Suppose that the unbalance in each node is distributed in the same axes plane, and the values are as follows,

$$\boldsymbol{e}_j = 5\times10^{-6}\,(\text{m}) \qquad (j = 1,\ 2,\ \cdots\cdots,\ 79)$$

The control parameters are shown in Table 1.

TABLE 1: The control parameters

| | $k_p$ | $k_i$ | $k_d$ | $T_d$ (s) |
|---|---|---|---|---|
| Thrust Bearing | 2.9 | 16.6 | $4.9\times10^{-3}$ | $1.1\times10^{-5}$ |
| Radial Bearings | 2.6 | 16.6 | $6.9\times10^{-3}$ | $1.1\times10^{-5}$ |

The magnetic damper is introduced into the system, and the equivalent damp of the damper is $1.39\times10^{3}$ Ns/m. The unbalance responses of the 57th rotor node of the system with and without the damper between 0 and 20,000 rotations per minute can be calculated, as shown in Fig. 2.

Fig. 2 shows that, the magnetic damper is helpful to restrain the unbalance response of the rotor.

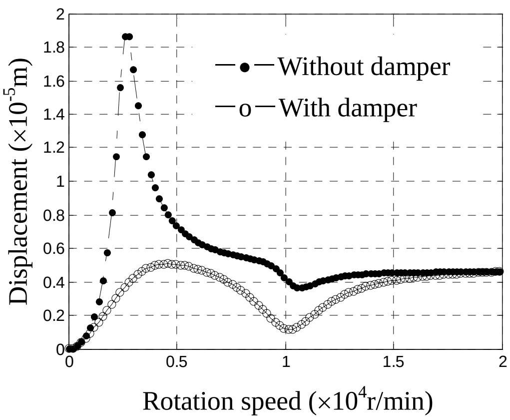


Fig. 2 Unbalance vibration of the 57th node

## EXPERIMENTAL MODAL ANALYSIS

Mode frequencies, modal damps and mode shapes can be obtained by the exciting tests on the static suspended rotor, and the dynamic characteristics can be predicted before the actual operation of the system.

The rotor is dealt with eleven positions as shown in Fig. 3, on which the pulse excitations are imposed separately. The vibrations of the 7th and 9th positions are measured by the two acceleration transducers synchronously.

The rotor is only suspended by the magnetic bearings, and the control parameters are shown in Table 1. By the exciting tests, the curves of the origin-point frequency response function in the 7th and 9th positions are shown in Fig. 4, and the curves of the cross-point frequency response function between the 7th position and the 9th position are shown in Fig. 5.

On the other hand, the rotor is suspended by the magnetic bearings and the magnetic damper. The equivalent damp of the damper is $1.39\times10^3$ Ns/m. By the exciting tests, the curves of the origin-point frequency response function in the 7th and 9th positions are shown in Fig. 6, and the curves of the cross-point frequency response function between the 7th position and the 9th position are shown in Fig. 7.

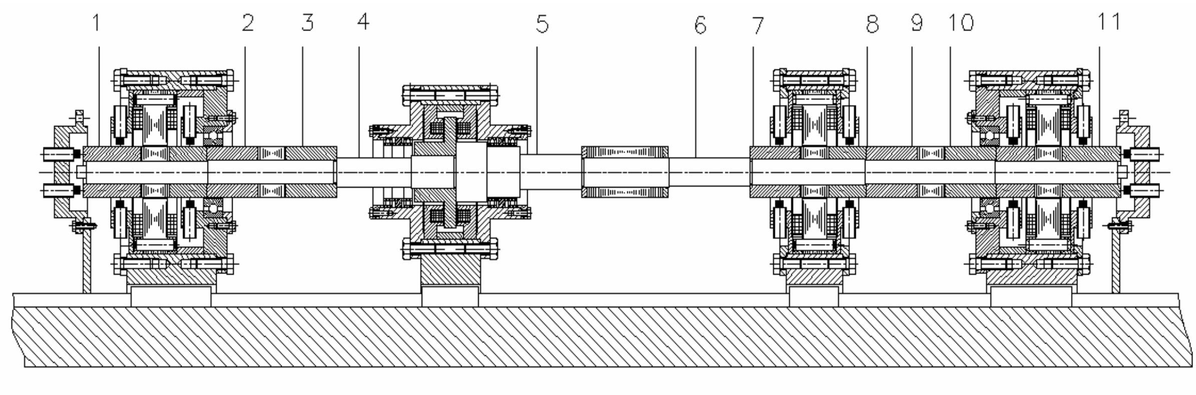


Fig. 3 Schematic illustration of the exciting locations

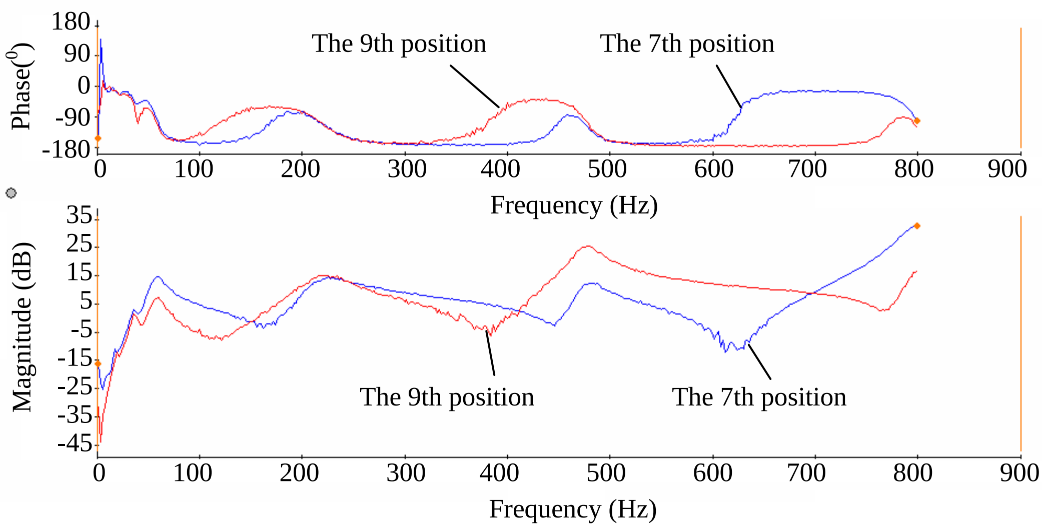


Fig. 4 The origin-point frequency response of the system without damper

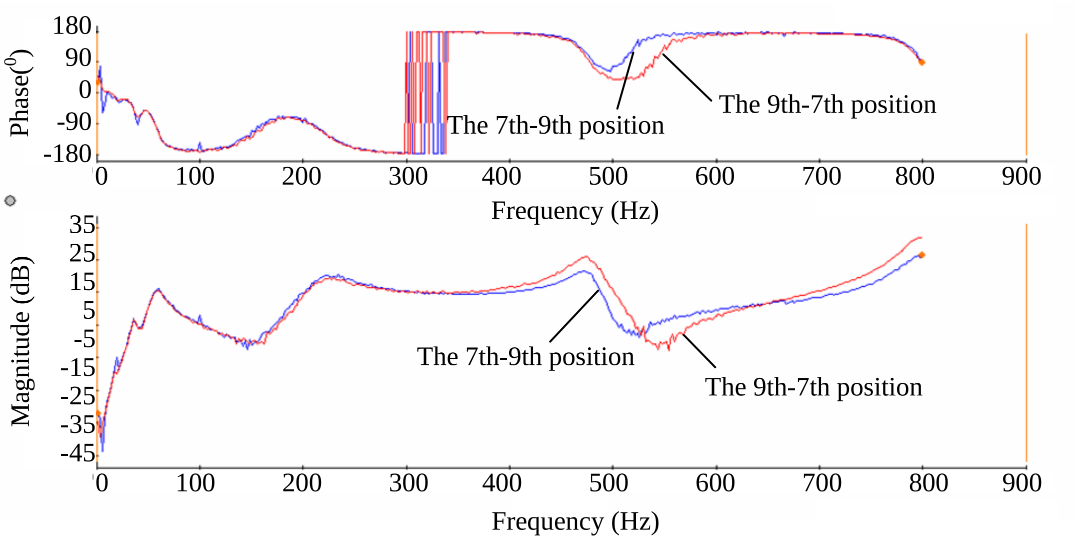

Fig. 5 The cross-point frequency response of the system without damper

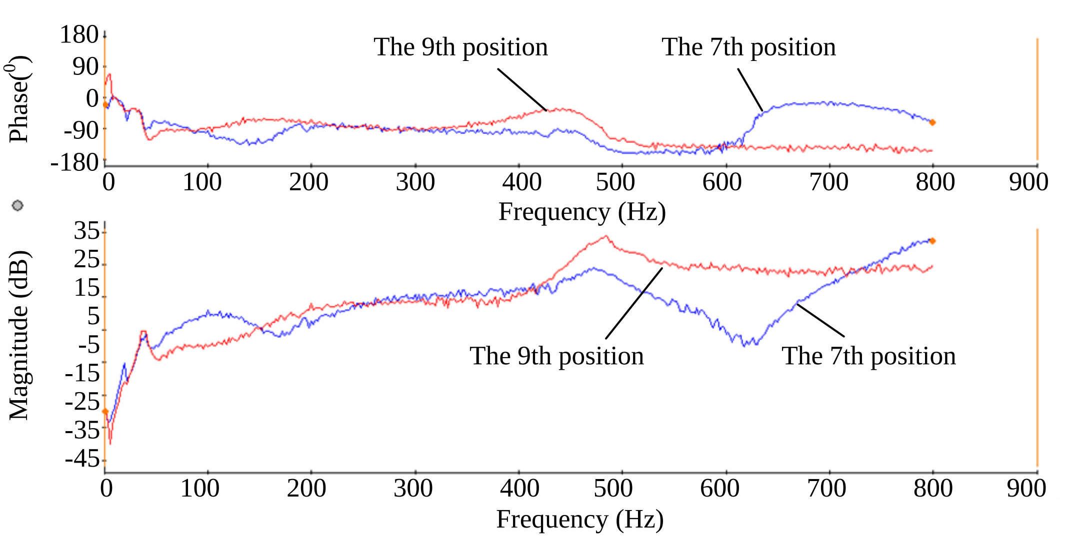

Fig. 6 The origin-point frequency response of the system with damper

The first five mode shapes can be obtained, as shown in Fig. 8. The values of the first five mode frequencies and the corresponding modal damping ratios of the system with and without the damper are shown in Table 2 and Table 3.

TABLE 2: The first five mode frequencies (Hz)

| | $N_1$ | $N_2$ | $N_3$ | $N_4$ | $N_5$ |
|---|---|---|---|---|---|
| without damper | 18.62 | 35.72 | 58.51 | 221.21 | 476.53 |
| with damper | 20.33 | 36.19 | 94.16 | 180.07 | 473.56 |

TABLE 3: The first five damping ratios

| | *1* | *2* | *3* | *4* | *5* |
|---|---|---|---|---|---|
| without damper | 6.47 | 7.37 | 12.02 | 9.94 | 2.61 |
| with damper | 7.38 | 7.88 | 30.02 | 20.87 | 4.15 |

The Experimental Modal Analysis shows that, the magnetic damper can increase the supporting damp of the system, especially when the system is on the 1st and 2nd bending mode frequencies. This is helpful for the system to get across the first two bending critical speeds safely.

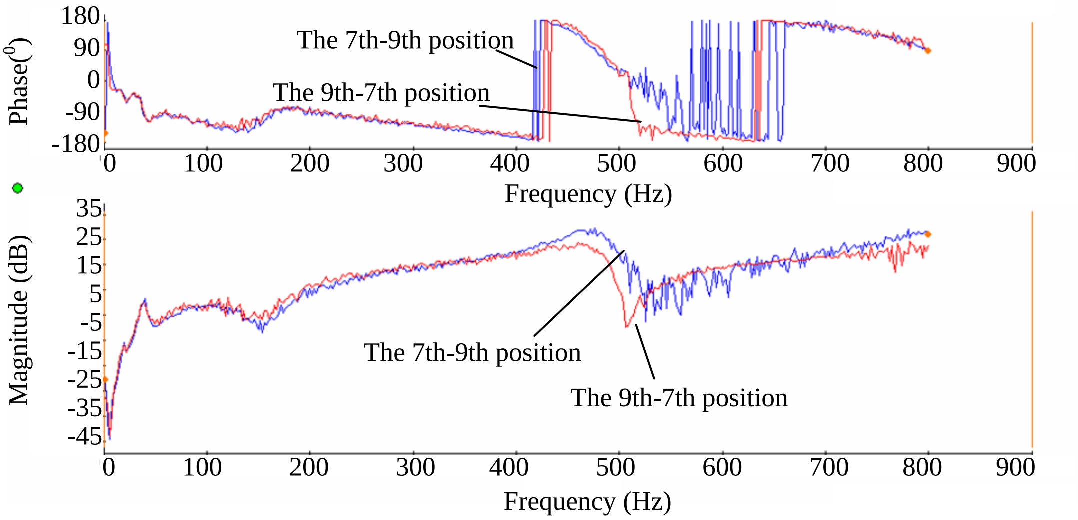


Fig. 7 The cross-point frequency response of the system with damper

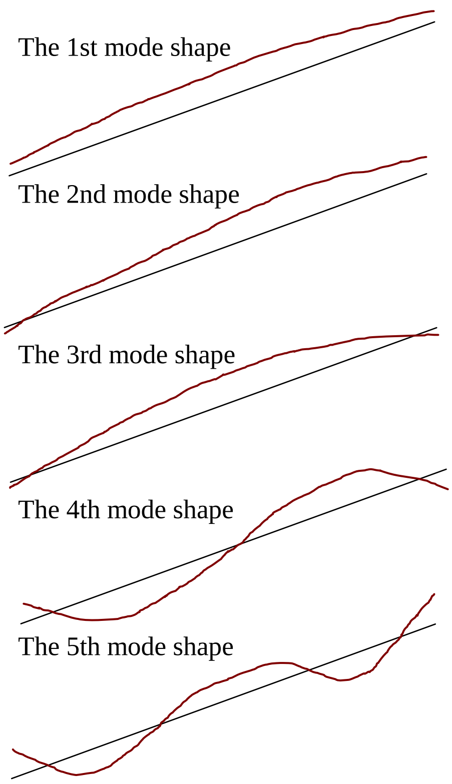


Fig. 8 The first five mode shapes

## ACTUAL OPERATION OF THE SYSTEM

The whole system is shown in Fig. 9. The control parameters are shown in Table 1. The rotor of the system with or without the magnetic damper is suspended stably and is driven by the built-in motor from 0 to 16,800 rotations per minute.

Fig. 9 Photo of the system

The vibrations of the 57th rotor node of the system with or without the magnetic damper are measured by the displacement sensor, and the same frequency vibrations as the rotation speed can be obtained by the Dynamic Signal Analyzer in real-time, as shown in Fig. 10.

Fig. 10 shows that, the system with the magnetic damper has enough supporting damp to restrain the vibration of the rotor, and the dynamic characteristics of the system have improved a lot.

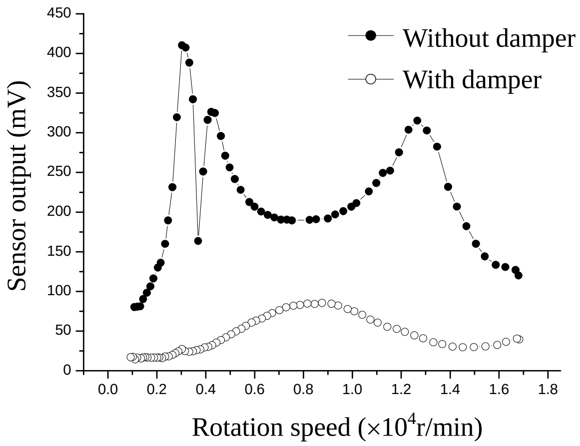


Fig. 10 Vibrations of the 57th rotor node

**CONCLUSION**

In this paper, dynamic characteristics of the system with the magnetic damper are first investigated by the theoretical calculation, the experimental modal analysis, and the actual operation of the system. Then the analytic results are compared with that of the ordinary magnetic bearing system. Main results are obtained as follows,

1) The magnetic bearing system with the magnetic damper can increase the supporting damp and restrain the vibration of the rotor, especially when the system operates on the 1st and 2nd bending critical speeds. The magnetic damper does not support stiffness in order to avoid redundant restraint. The system can get across the first two bending critical speeds, and operate on 16,800 rotations per minute safely.

2) Not only the magnetic bearing but also the magnetic damper can support the damp to the rotor, and restrain the vibration of the rotor. When the magnetic bearing can not support enough damp because of the system stability, the magnetic damper can be introduced into the system to increase the supporting damp, so long as the empty is enough. And this is easy for the system with a long flexible rotor.

**ACKNOWLEDGEMENT**

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