Theoretical Investigation on Dynamic Behavior of AMB-Rotor System subject to Base Motion Disturbances

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Abstract: A theoretical investigation on the dynamic behavior of an AMB-rotor system subject to base motion disturbances was carried out in the present paper. The rotor responses of the AMB-rotor system under the rigid and flexible base supporting conditions were compared, which will provide theoretical basis on the base disturbance rejection performance research of AMB-rotor systems. The results show that under the rigid base supporting condition, the capacity of the AMB's controller to suppress the base disturbances is very limited. In order to attenuate the disturbance responses, a flexible base with specified mass and stiffness should be adopted in the AMB-rotor system.

Keywords: Active Magnetic Bearing, Rotor Rigid Base, Flexible Base, Base Disturbances, Dynamic Behavior

Introduction

Active magnetic bearing (AMB) provides several advantages over conventional bearings in practical industrial applications. The advantages include elimination of a lubrication system, friction-free operation, low power loss, and controllability of bearing dynamic characteristics [1]. However, to use an AMB in some special work environment with bad conditions such as aircraft engines and rotating machinery on ships, many problems need to be solved. These include the high temperature AMB, the small-scale control system and the highly reliable retainer bearing, as well as the dynamic behavior of the AMB-rotor system on rotating machinery subject to various base motion disturbances.

Previous literature indicated that researches on AMB systems mostly focus on the AMB itself. There are few references on the influence of base motion disturbances on AMB systems. Changsheng Zhu [2] did experimental research on the influence of the single-frequency horizontal base disturbance on the AMB-rotor system's dynamic characteristics. He pointed that capacity of the AMB's controller for suppressing the base transverse vibration is relatively weak, and suggested base vibration be considered in the controller design process. Fangzhen Song, etc. [3] built a mathematical model of a rigid base-AMB-rotor system for a stability analysis under different excitation conditions. They found that in some cases, the base vibration may possibly lead to the failures of AMB system. Suzuki [4] developed a control strategy based on acceleration feedback to reduce the rotor imbalance response caused by base vibration, without affecting the main control performance. Kasarda et al [5] experimentally studied the influence of base excitation on dynamic characteristics of the AMB-rotor system. However, the gyroscopic effect of the rotor was not considered. Cole et al [6, 7] proposed a control strategy to effectively suppress the shock and other forms of vibration on the base. But the research on the basic characteristics of the system is not done.

In this paper, the most commonly mechanical "mass-spring-damper" system model is used to perform a theoretical investigation on dynamic response of AMB-rotor system under different base supporting and base motion disturbance. The aim of the work is to explore new ideas and new methods to improve the base disturbance rejection performance of AMB-rotor systems.

Dynamic model

In case of decoupling conditions, the AMB-rotor system can be handled as "mass-spring-damper" system [8], whose equivalent stiffness and damping can be given by the parameters of the control system. Therefore, the "AMB-rotor-base" systems supported by two kinds of bases, can be simplified respectively as the models shown in Figure 1 and Figure 2 in the direction of a single degree of freedom. In which m is the mass of the rotor, k, d represent the equivalent stiffness and damping of the AMB, x_o, x_o' are the rotor displacements in the vertical direction, x_i, x_i' are the displacement excitations on the bases in the vertical direction, and $f(t)$, $f'(t)$ are the imbalance excitations on the rotor, respectively. In Figure 2, the flexible base is equivalent to "mass - spring" unit, without considering the damping effect, where M represents of the mass of the base, K represents the equivalent stiffness of the base and x_2' is the displacement response of the base in the vertical direction.

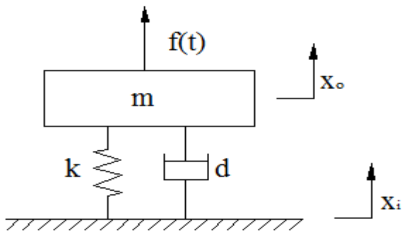

Fig.1. Simplified model supported by rigid base

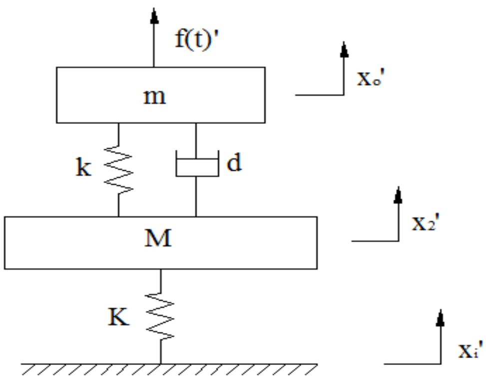

Fig.2. Simplified model supported by flexible base

According to the dynamic relationship, the motion equations of the two systems supported by different bases are described as follows:

$$m\ddot{x}_o + k(x_o - x_i) + d(\dot{x}_o - \dot{x}_i) = f(t). \tag{1}$$

$$\begin{cases} m\ddot{x}_o + k(x'_o - x'_2 + x'_i) + d(\dot{x}'_o - \dot{x}'_2 + \dot{x}'_i) = f'(t) \\ M(\ddot{x}_2 - \ddot{x}_i) - k(x'_o - x'_2 + x'_i) - d(\dot{x}'_o - \dot{x}'_2 + \dot{x}'_i) + K(x_2 - x_i) == 0 \end{cases}. \tag{2}$$

Considering only the influence of base disturbances on the rotor is concerned in the present paper, the effects caused by the imbalance excitations on the rotor (i.e. $f(t)$, $f^{'}(t)$ =0) is ignored. Using the Laplace transform to equations(1),(2) and respectively using x_o, x_o' as input and x_i and x_i' as output, the transfer functions of the two systems supported by rigid and flexible bases are given as following:

$$G(s) = \frac{X_0(s)}{X_i(s)} = \frac{k + ds}{ms^2 + ds + k}. \tag{3}$$

$$G'(s) = \frac{X'_0(s)}{X'_i(s)} = \frac{K(ds + k)}{mMs^4 + (m + M)ds^3 + (mk + MK + Mk)s^2 + dKs + kK}. \tag{4}$$

PID control is the most commonly used control strategy of AMB systems. Its equivalent stiffness and damping can be represented as

$$\begin{cases} k = ck_i K_p - k_x \\ d = ck_i (K_d - \dfrac{K_i}{w^2}) \end{cases}. \tag{5}$$

where K_p is the proportional coefficient, K_i is the integral coefficient, K_d is the differential coefficient, k_i is the stiffness coefficient of current, k_x is the stiffness coefficient of displacement, c is the total gain coefficient for other sectors and w is the frequency of external disturbances.

There are many forms of base disturbances, such as step, shock, sine wave, rectangular wave, seismic waves, etc. The most common sine wave was chosen as the input for the analysis, which can be defined as

$$x_o = a\sin(wt). \tag{6}$$

where a stands for the amplitude of base disturbances and w is the frequency of base disturbances.

To simplify the calculation, we assume the frequency of external disturbances in PID control is the same as the frequency of base disturbances. Taking equation (6) for the Laplace transform, and together with equation (5) for the substitution into (3), (4), the vibration response of the rotors can be represented as the following equations when the systems are excited by base disturbances

$$X_0(s) = \frac{aw[ck_i(K_d w^2 - K_i)s + (ck_i K_p - k_x)w^2]}{(s^2 + w^2)[mw^2 s^2 + ck_i(K_d w^2 - K_i)s + (ck_i K_p - k_x)w^2]}. \tag{7}$$

$$X'_i(s) = awK[(K_d w^2 - K_i)ck_i s + (ck_i K_p - k_x)w^2] / (s^2 + w^2) / \{mMw^2 s^4 + (K_d w^2 - K_i)k_i \cdot (m+M)cs^3 + [(m+M)(ck_i K_p - k_x) + MK]w^2 s^2 + (K_d w^2 - K_i)ck_i K s + (ck_i K_p - k_x)w^2 K\} \tag{8}$$

From equations (7) and (8), we can see that the factors that affect the vibration response of the rotor include not only the system structural parameters, and control parameters, but also the amplitude and frequency of base disturbances. Equations (7) and (8) can be solved in time domain by Laplace inverse transform with the help of the commercial mathematical software Maple. In this paper, the used AMB-rotor system model parameters are shown in Table 1. Substituting them into equation (5), k and d are therefore:

$$k = 2.2e6[N/m], d = 3.1e6(0.025 - \frac{3}{w^2})[N \cdot s/m]. \tag{9}$$

Table 1. Parameters of AMB-rotor system

Parameter	Symbol	Value
Mass of rotor	m	16.29[kg]
Air Gap	x_o	0.3[mm]
Area of magnetic pole	A	180[mm^2]
Number of coils	N	160
Bias current	I_o	2.4[A]
Stiffness coefficient of current	k_i	1.2e6[N/m]
Stiffness coefficient of displacement	k_x	154.3[N/A]
Control parameters	K_p, K_d, K_i, c	1.1, 0.025, 3, 2e4

Result analysis

Rigid base supporting. When the structural and control parameters of an AMB-Rotor system are determined, the vibration response of the rotor is affected only by the base disturbances. From equation (7), it can be seen that the vibration response of the rotor is proportional to the amplitude of the base disturbance. Therefore the larger the amplitude of the base disturbance is, the greater the vibration response of the rotor will be. This law also applies to flexible base supporting.

Influence of frequency of base disturbances on rotor vibration response. Amplitude of the base disturbance of $a = 1\mu m$ and frequency of base disturbances as 20,100,400Hz were defined for all the calculations in this section. The vibration response of the rotor was then studied. The results in Fig.3 show that the frequency of the base disturbance and the rotor response is the same under rigid base supporting. When the frequency of the base disturbance is low (close to the natural frequency of the system 58Hz), the amplitude of rotor response is almost the same as the one of the base disturbance. However, when the frequency of the base disturbance is far from the natural frequency of the system, the rotor vibration response is slightly suppressed, and the amplitude changes about 10%.

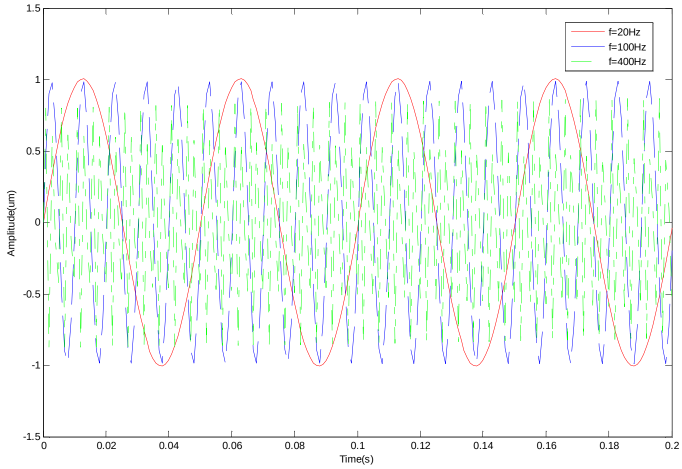

Fig.3. Rotor response under different frequency of base disturbances on rigid base

Influence of stiffness of AMB on rotor vibration response. As for the same circumstance of base motion disturbance, we can study the rotor vibration responses under different stiffness and damping by changing the control parameters of the AMB-rotor system. Fig.4 shows that the rotor response changes little when the stiffness varies in the regular scope of AMB (2.2e5-7 N/m) and the only reduced amplitude (about 40%) occurs when the stiffness is very small (2.2e4 N/m), which can hardly be achieved in practical applications because small stiffness would cause great difficulties on system parameter identification. Based on this result, it can be seen that the capacity of the AMB-Rotor system itself for suppressing base disturbances is very limited. When the base disturbance is large, the shaft may hit the retainer bearings and cause damage to the AMB-Rotor system.

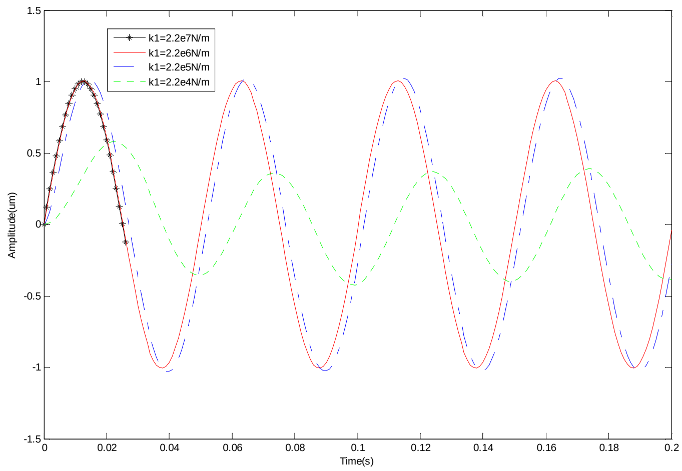

Fig.4. Rotor responses under different stiffness of AMB subject to base disturbances on rigid base (Frequency of the base disturbance is 20Hz)

Flexible base supporting. Compared to the rigid base supporting, the flexible base supporting is essentially equal to a rigid base added to an isolator with mass and stiffness. From the mechanical point of view, the rotor response can be affected by changing the transmission path of force and displacement.

Influence of stiffness of base supporting on rotor vibration response. First of all, the circumstance of f=20Hz, M=50kg is studied. The results in Fig.5 shows a little difference in rotor response between rigid base supporting and flexible base supporting with stiffness larger than 1e7N/m. The divergence phenomenon occurs when the stiffness is equal to 1e6N/m (close to the magnetic bearing stiffness). The rotor response is suppressed by 60% to 90% when the stiffness is between 1e5N/m to 1e4N/m. Thus, it can be inferred that by installing a soft base with low stiffness (such as rubber, air cushion, etc.) can greatly improve the base disturbance rejection performance of AMB-rotor systems with low frequency.

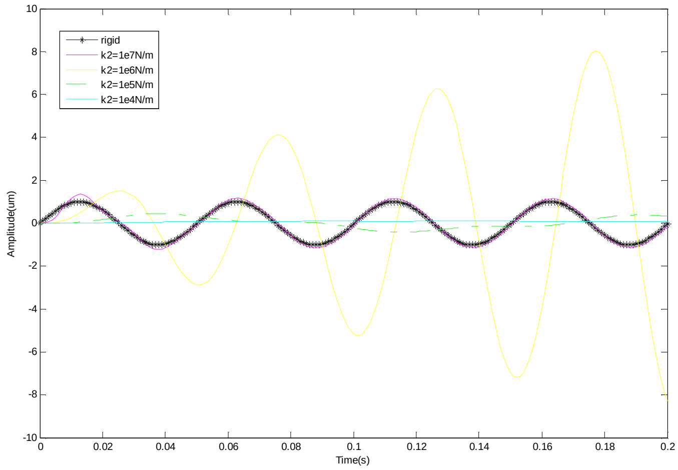

Fig.5. Low frequency response of the rotor under different stiffness of base supporting subject to base disturbances (Frequency of the base disturbance is 20Hz, M=50kg)

The circumstance of f=400Hz, M=50kg was then studied. Fig.6 shows that, the system has a overall better suppression to the base disturbance of high frequency comparing with the base disturbances of low frequency. The amplitude of rotor response changes more than 85% when the stiffness of base supporting is less than 1e7N/m. Therefore, it can be inferred that only rotor response of low frequency needs to be studied on suppression of base disturbances of AMB-rotor systems.

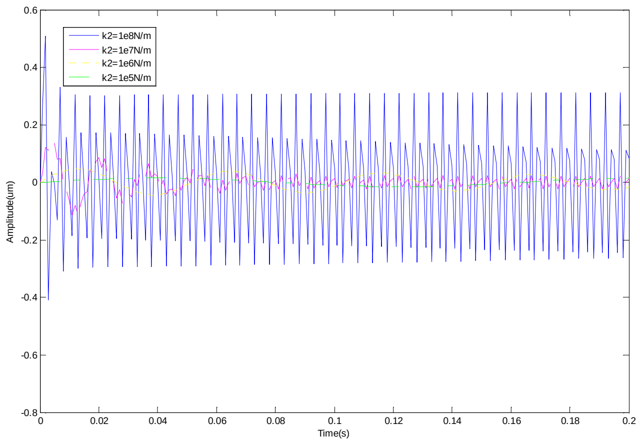

Fig.6. High frequency response of the rotor under different stiffness of base supporting subject to base disturbances (Frequency of the base disturbance is 400Hz, M=50kg)

Influence of base mass on rotor vibration response. In the present section, the influence of the mass of the base on the rotor response was studied. Assuming M=200kg, f=20Hz, Fig.7 shows that, comparing with M=50kg, the divergence is well suppressed when the stiffness of base is 1e6N/m and the amplitude begins to attenuate. When the stiffness is between 1e5-1e4N/m, the rotor vibration response is significantly reduced by over 80%. Thus, we can infer that, under the same circumstance of the stiffness of the base, the larger the mass of the base is, the better suppression of base disturbances will be. However, larger mass and lower stiffness mean that the base volume need be increased, which will affect the space design of the mechanical structure. Therefore, an optimal design of AMB-rotor systems by considering various factors of the system is required to achieve the optimal base disturbance rejection performance.

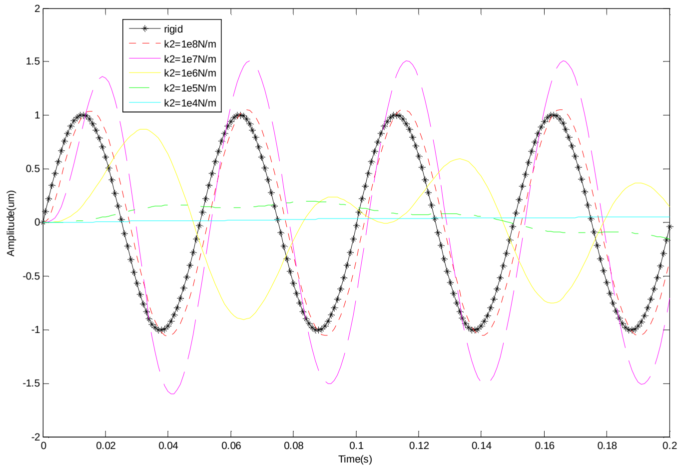

Fig.7. Low frequency response of the rotor under different stiffness of base supporting subject to base disturbances (Frequency of the base disturbance is 20Hz, M=200kg)

Conclusions

Under the rigid base supporting condition, the capacity to suppress the base disturbances of an AMB's controller by changing the stiffness and damping only is very limited. Large amplitudes of base disturbances may lead to great damage to the retainer bearings and the AMB-rotor system. However, using the flexible base supporting condition can effectively improve the base disturbance rejection performance of AMB-rotor systems without affecting the AMB-rotor system's operational characteristics. The flexible base supports shall be achieved by installing a soft base which have a suitable mass and stiffness calculated by structural optimization design method.

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