Analysis and Design of Magnetic Levitation and Mechanical Spring Compound Vibration Isolation Control System

Weihua Dong¹, Mingda Zhai¹ and Xiaolong Li*¹

¹College of Intelligence Science and Technology, National University of Defense Technology, Changsha, Hunan, 410073, China

*Corresponding author’s e-mail: 13787786254@163.com

Abstract. In this paper, the magnetic levitation and mechanical spring compound vibration isolation control system is designed by applying maglev active control on the basis of mechanical vibration isolator. The system can effectively solve the problems of fixed structure parameters and poor environmental adaptability of passive vibration isolators, and can be applied to the anti-vibration protection of precision instruments in complex environments. This paper analyses the mechanism and characteristics of maglev vibration isolation, deduces the dynamic model of the system, discusses the design of the active vibration isolation control strategies, and carries out several simulation experiments of vibration isolation in different environments. The results show that the design scheme of the system is reasonable and feasible.

1. Introduction

The traditional vibration isolation mainly uses damping materials or mechanical devices. This technology has been widely used in various fields and has achieved relatively obvious vibration isolation effects. However, for the system with higher requirements such as precision instrument protection, the vibration isolation effects of traditional technology are usually unsatisfactory, and the main problems are as follows:

1) The structural parameters of the system are fixed and cannot be adjusted dynamically[1], so it is difficult to adapt to random vibration environments;

2) There is a contradiction between the bandwidth and stiffness of the system, that is to say, the positioning accuracy and vibration isolation effect cannot be satisfied at the same time[2].

To solve above problems, it is a good idea to use maglev technology to achieve active vibration isolation control[3]. However, as a result of the limitation of system energy consumption, active control is seldom used alone[4]. If passive vibration isolation is combined with active control, the resonance peak of vibration can be reduced by active control, so as to improve the isolation performance of low-frequency vibration. Meanwhile, the passive isolator is used to support the gravity of the isolated object and to limit the position. In this way, the bandwidth can be increased, and the system will be more secure and reliable. Therefore, this paper designs a compound active-passive integrated vibration isolation control system based on magnetic levitation and mechanical spring, which can meet the needs of complex environments and greatly improve the vibration resistance of precision instruments.

2. Principle of compound vibration isolation system and design of electromagnet actuator

2.1. Introduction to the working principle of the system
The structure of the magnetic levitation and mechanical spring compound vibration isolation control system is shown in Figure 1. The system is mainly composed of vibration table, electromagnets, mechanical springs, inner and outer metal frames, the controller and so on. The mechanical spring is used as passive vibration isolation device and the electromagnet is used as active vibration isolator.

The vibration table is fixed on the ground to provide vibration excitations to the experimental device. Two electromagnets are separately mounted on the upper and lower inner surfaces of the outer frame and two pieces of armature are symmetrically mounted on the upper and lower outer surfaces of the inner frame. An air gap of 8-12 mm is formed between electromagnet and armature. The vibration isolation object is placed in the inner frame. In active vibration isolation control, the difference of suction between the upper and lower electromagnets generates a control force of equal magnitude with external vibration excitation and opposite to the direction of it, so as to counteract the transmission of external vibration. The assembly of the compound vibration isolation device is shown in Figure 3.

An eddy current displacement sensor is installed between the electromagnet and armature to detect the air gap spacing between them. Two acceleration sensors are installed on the inner and outer frame respectively to detect the vibration parameters of the system. The velocity and displacement of the measured object relative to the inertial reference system can be obtained by integrating the acceleration signal. The air gap spacing and the acceleration of inner and outer frames are input into controller as feedback signals, and the control quantity is calculated according to the control algorithm. The control is realized by Speedgoat’s hardware-in-the-loop rapid control development system based on MATLAB/Simulink Real-Time module. After being isolated by the optocoupler, the PWM wave generated by the controller controls the IGBT to output variable current, which drives the electromagnet to produce the acting force, so as to complete the closed-loop control of active vibration isolation.

The mechanical spring is placed vertically in parallel with the maglev isolator, which can ensure that the electromagnet and armature will not have a mechanical contact when the external vibration excitation overloads or the electromagnetic actuator fails to respond in time due to malfunction. In the initial state, the spring generates a certain elastic force by pre-compression to support the vibration isolated object in the equilibrium position. Using mechanical spring to support the weight of isolated object can effectively reduce the power consumption of active isolator and facilitate the differential control. In high frequency vibration, mechanical vibration isolator and controllable maglev vibration isolator work together to suppress the external vibration.
2.2. Design of the maglev actuator isolator

The maglev actuator can easily change the magnetic field intensity by changing current of the electromagnet, so its equivalent stiffness and damping can be changed by adjusting the system parameters\cite{5,6}, which can provide a simple and feasible condition for active vibration control. The force output by electromagnets is nonlinear with the acting distance. Therefore, the maglev isolator will have better effects on nonlinear vibrations and can be adapted to more variable excitations. Moreover, the structure of maglev actuator is simple, safe and reliable, so it can be effectively applied to the design of vibration isolation control system.

Considering the limitation of actuator installation space and weight, the electromagnet framework is designed as E-shape structure, as shown in Figure 4. Under vibration environments, electromagnet current will fluctuate continuously. So, in order to reduce the heating, both electromagnet core and armature iron are stacked with silicon steel sheets of 0.35mm thickness. The coil adopts $\Phi 2.5$ mm enameled copper round wire and the number of winding turns is 325. The coil skeleton is assembled with F-grade epoxy fiberglass board and the coil window area is 90mm×30mm. The designed actuator can generate an electromagnetic suction over 900N at a distance of 8 mm, thus ensuring the control requirements of the vibration isolation system.

3. Analysis of system structure and design of the controller

3.1. Analysis and modeling of system structure

In order to describe the compound vibration isolation system more clearly and to design the active vibration isolation controller, the structure modeling and analysis of the system should be carried out at first. On the basis of actual physical system, we establish the dynamic model of the compound vibration isolation system, deduce the electromagnetic-mechanical correlation equations of the maglev actuator, and linearize the system model.

The force analysis of the compound vibration isolation system is shown in Figure 2. We ignore small vibrations in other directions and only study the motion in the vertical direction. Downward is set as the positive direction of force and motion, then the dynamic equation of system is established:

\begin{equation}
m \ddot{z} = mg + F_{e1} - F_{e2} - k \cdot (z + \Delta z - r) - q \left( \frac{dz}{dt} - \frac{dr}{dt} \right)
\end{equation}

\begin{equation}
mg = k \cdot \Delta z
\end{equation}

In formula (1), m is the mass of the isolated object; $F_{e1}, F_{e2}$ are electromagnetic suctions on the vibration isolated object by maglev actuator; $r, z$ refer to the amplitude of vibration outside the
system and the vertical displacement of the vibration isolated object; k and q refer to the stiffness and damping of mechanical spring. Without the maglev actuator, the equivalent stiffness and damping of system completely depend on the characteristic parameters k and q of mechanical vibration isolator, which cannot be dynamically adjusted. \( \Delta z \) represents the initial compression of the spring. Formula (2) indicates that in the initial state, the elastic force generated by pre-compression of the mechanical spring will support the vibration isolated object in the equilibrium position. In above dynamic model, the electromagnetic force is:

\[
F_{el} = \frac{\mu_0 SN^2}{8} \frac{i_1^2}{h_1^2}, \quad F_{ez} = \frac{\mu_0 SN^2}{8} \frac{i_2^2}{h_2^2}
\]  

(3)

Among which, \( \mu_0 = 4\pi \times 10^{-7} \, N / A^2 \) is the vacuum permeability; \( S \) is effective magnetic conduction area of the electromagnet. \( N \) refers to coil turns of the electromagnet winding; \( i_1, i_2 \) refer to the current in coils; \( h_1 = r - z + h_{10} \), \( h_2 = z - r + h_{20} \) are air gaps between electromagnet and armature. For a certain maglev actuator, \( SN \) and \( \mu_0 \) are fixed values. Let \( K = \mu_0 SN^2 / 8 \), then the electromagnetic force can be transformed to:

\[
F_{el} = K \left( \frac{i_1}{h_1} \right)^2, \quad F_{ez} = K \left( \frac{i_2}{h_2} \right)^2
\]  

(4)

The relationships between control voltage and electromagnet current are expressed as follows:

\[
u_i = R \cdot i_1 + \frac{d}{dt} \left( L_1 \cdot i_1 \right) = R \cdot i_1 + \frac{2K}{h_1} \frac{d}{dt} \left( \frac{i_1}{k_1} \right) - \frac{2K}{h_1} \frac{d}{dt} \left( \frac{dh_1}{dt} \right)
\]

\[
u_z = R \cdot i_2 + \frac{d}{dt} \left( L_2 \cdot i_2 \right) = R \cdot i_2 + \frac{2K}{h_2} \frac{d}{dt} \left( \frac{i_2}{k_2} \right) - \frac{2K}{h_2} \frac{d}{dt} \left( \frac{dh_2}{dt} \right)
\]  

(5)

Where \( L_1, L_2 \) and \( R \) are the equivalent inductance and resistance in electromagnet coil.

The compound vibration isolation control system is a typical nonlinear system. There are complex relationships among the electromagnetic force, the instantaneous current in coil and the air gap [7]. If we want to use linear theory to design the vibration isolation controller, we need to linearize the nonlinear part of the system. The dynamics equation of the linearized system can be expressed as:

\[
m \frac{d^2 z}{dt^2} = (K_z \cdot i_1 + K_0 \cdot h_1) - (K_z \cdot i_2 + K_0 \cdot h_2) - k \cdot (z - r) - q \left( \frac{dz}{dt} \cdot \frac{dr}{dt} \right)
\]  

(6)

Where \( K_z = 2K_i / h_0^2 \), \( K = -2K_i / h_0^2 \).

Considering that the electromagnet vibrates in a small range near the equilibrium position, the electromagnet inductance can be approximated as a constant: \( L_u = 2K_i / h_0 \) [8]. Therefore, formula (5) can be simplified as:

\[
u_i = R \cdot i_1 + \frac{2K}{h_{10}} \frac{d}{dt} \left( \frac{i_1}{h_0} \right)
\]

\[
u_z = R \cdot i_2 + \frac{2K}{h_{20}} \frac{d}{dt} \left( \frac{i_2}{h_0} \right)
\]  

(7)

Finally, combining (2), (6) and (7), we obtain the linearized system model:
\[ \begin{align*}
  m \frac{d^2 z}{dt^2} &= K \left( i_1 - i_1 \right) + K_0 \left( \frac{r - z}{2} \right) - k \left( z + \Delta z - r \right) - q \left( \frac{dz}{dt} \right) + mg \\
  mg &= k \cdot \Delta z \\
  u_1 &= R \cdot i_1 + \frac{2K}{h_0} \frac{di_1}{dt} \\
  u_2 &= R \cdot i_2 + \frac{2K}{h_0} \frac{di_2}{dt}
\end{align*} \tag{8} \]

\(u_1, u_2\) are control voltages at both ends of the electromagnet. By designing the active vibration isolation algorithm to control the output of \(u_1\) and \(u_2\), different vibration isolation control strategies can be realized.

### 3.2. Design of active vibration isolation controller

The state variables, input variables and output variables of the system are selected as follows:

\[
  x = \begin{bmatrix} i_1 \\ i_2 \\ z \\ \dot{z} \end{bmatrix} ; \quad u = \begin{bmatrix} u_1 \\ u_2 \\ r \end{bmatrix} ; \quad y = \begin{bmatrix} i_1 \\ i_2 \\ z \end{bmatrix}
\]

According to above variables, we obtain the system state space equation:

\[
  \dot{x} = \begin{bmatrix} -R h_0 \\ 0 \\ 0 \\ 0 \end{bmatrix} - \frac{2K}{h_0} \begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} h_0 \\ 2K \\ 0 \\ 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix}
\]

\[
  y = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} \tag{10} \]

Substitute physical parameters into system state space equation, we get the eigenvalues of system matrix: \(\lambda_1 = 0.0000 \pm 33.4066i\), \(\lambda_2 = 14.5830 + 0.0000i\), which includes solutions \(\lambda_1\) and \(\lambda_2\) on the imaginary axis of the complex plane. From above analysis, we know that the system is critically stable, but the state of affairs is considered unstable in engineering. The controllability matrix and the observability matrix of the system are full rank, that means the system is both controllable and observable. Therefore, the controller can be designed to make the system stable.

Since two electromagnets have the same design parameters and the system structure is symmetrical, we only need to analyze one of them. The transfer function from the control voltage to the displacement of the isolated object can be obtained from system state space equation:

\[
  z(s) = \frac{\frac{2K h_0}{\left( 2 m^2 s^2 + 2 q s + 2 k + K_h \right)} \left( 2 K s + R h_0 \right)}{u(s)} = \frac{\frac{2K}{2 m^2 s^2 + 2 q s + 2 k + K_h} \frac{1}{R}}{s + \frac{2 K}{R h_0}} \cdot \frac{i(s)}{u(s)} \tag{12} \]

As can be seen from above formula, this system is a third-order system, which can be regarded as a second-order system in series with a first-order inertial link. The inertial link is the transfer function of
electromagnet control voltage to output current and its time constant is $T = 2K/(Rh) \approx 0.07$. Due to the resistance and inductance of electromagnet coil, there is a time delay between control voltage and output current [9], which can badly affect the control performance of the system. We introduce current feedback to reduce the time delay so that the electromagnet current can quickly respond to the control voltage. In addition, we can set the feedback coefficient to make the electromagnet approximate to a proportional link with a proportional coefficient of 1 in the frequency band of the system. Then the system is equivalent to a second-order nonlinear model.

When the frequency of external vibration is very low and the amplitude is large, we hope that the vibration isolated object can follow the outer frame to move. While when the system is subjected to high frequency mechanical vibrations, we need to reduce the vibration intensity to minimize the impact of external vibration on it. For this reason, we adopt the differential current-position dual closed-loop PID control to design two sets of active vibration isolation control algorithms for different frequencies vibration environments, in which the current loop is inner control and the position loop is outer control.

When the system is disturbed by low-frequency vibration, we introduce the following feedback control law:

$$u_1 = k_1 \left( k_p (h_1 - h_2) + k_i \int_0^t (h_1 - h_2) \, dt - k_d z \right)$$

$$u_2 = k_2 \left( k_p (h_1 - h_3) + k_i \int_0^t (h_1 - h_3) \, dt - k_d z \right)$$

(13)

where $h_1 - h_2 = 2(z - r)$.

When high frequency mechanical vibration occurs, transform the feedback control law into:

$$u_1 = k_1 \left( k_p (-z) + k_i \int_0^t (-z) \, dt - k_d z \right)$$

$$u_2 = k_2 \left( k_p z + k_i \int_0^t z \, dt - k_d z \right)$$

(14)

Where, $k_1$ is the gain from output voltage of the controller to the voltage at both ends of the electromagnet, and $k_2$ is the feedback coefficient of the current loop.

Substitute the control law into system model, we obtain the motion equation of system under the active vibration isolation control:

$$m \dddot{z} = \left( A \cdot k_p - B + k \right) \cdot \left( r - z \right) + \left( A \cdot k_d + q \right) \cdot \left( \ddot{z} - \ddot{r} \right)$$

(15)

Among which, $A$ and $B$ are constant terms:

$$A = \frac{\mu_0 SN^2 i_0}{2h_0^2}, \quad B = \frac{\mu_0 SN^2 i_0^2}{8h_0}$$

Above formula shows that the equivalent stiffness $(A \cdot k_p - B + k)$ and equivalent damping $(A \cdot k_d + q)$ of the compound vibration isolation system can be adjusted by setting values of $k_p$ and $k_d$ in control algorithm, and no longer solely depend on the characteristic parameters $k$ and $q$ of the mechanical spring. Therefore, the system environmental adaptability and comprehensive performance can be effectively improved.

4. Simulation experiment and result analysis

In order to verify the rationality of the design scheme and test the vibration isolation effect of the system, we bring the designed active vibration isolation control law into nonlinear model of actual system, then carry out several groups of simulation experiments for the vibration of different frequencies. The results are shown in Figure 5 and Figure 6.
Figure 5. Comparison of effects of tracking low-frequency vibration.

Figure 5 is a comparison diagram of tracking effects of mechanical spring, maglev vibration isolator and compound vibration isolation system under the vibration with frequency of 10Hz and amplitude of 5mm. It shows that when the system is subjected to low-frequency vibration disturbance, the mechanical spring cannot suppress the vibration, but even magnify it. While both the maglev vibration isolator and the compound device can make the isolated subject effectively track the low-frequency vibration. In another word, the improved system still has great stiffness in low-frequency vibration environment.

Figure 6. Comparison of vibration isolation effects for high frequency random disturbance.

Figure 6 compares the vibration isolation effects of three vibration isolation devices in 250Hz high-frequency random vibration with acceleration of 49m/s^2. Through the comparison, we know that the maglev isolator’s ability to attenuate the vibration intensity is better than that of mechanical spring. While the compound vibration isolation system has the best effect. For the vibration with above parameters, under the control of designed magnetic levitation and mechanical spring compound isolator, the amplitude attenuation ratio reaches 1/20 and the acceleration attenuation ratio reaches 1/60.

5. Conclusion

The results show that the design scheme of the magnetic levitation and mechanical spring vibration isolation control system is reasonable and feasible.

1) This paper effectively solved the problem that the structure parameters of traditional vibration isolation system are fixed and cannot be dynamically adjusted. Compared with mechanical or material vibration isolation, the magnetic levitation and mechanical spring compound vibration isolation control system can flexibly change the controller parameters to achieve satisfactory vibration isolation performance.

2) The contradiction between bandwidth and stiffness of the system is well solved in this paper. The feedback of gap acceleration signal feedback can be effectively used to make the system have both better stiffness and promising bandwidth.

3) For vibrations of different frequencies, the paper adopted differential current-position dual closed-loop PID to design 2 sets of active vibration isolation control laws respectively. By switching the control law, the system can not only track the low-frequency vibration, but also attenuate the high-frequency vibration effectively.

4) In this paper, the above conclusions are obtained through simulation analysis. However, the actual system is not a simple second-order model. There are many high frequency factors, such as the time constant of electromagnet coil, filters of sensors and so on. Even the elasticity of the connector pedestal may affect the experimental results. Therefore, the following work is supposed to build a test
platform according to the design scheme and conduct more experimental tests to optimize and improve the magnetic levitation and mechanical spring compound vibration isolation control system.

Acknowledgments
This work was supported by “National Key R & D Program of China” under Grant 2016YFB1200602.

References
[1] Long, Z.Q., Hao, A.M., Chen, G., Chang, W.S. (2003) The research of active isolation platform with magnetically levitated control. Journal of Astronautics, 24(5): 510-514.
[2] Daleya, S., Johnsonb, F.A., Pearsonc, J.B., Dixond, R. (2004) Active vibration control for marine applications. Control Engineering Practice, 12(4): 465-474.
[3] Liu, X.J., Hu, Y.F. (2009) The Performance Study on Two-Stage Vibration Isolation System with Active Magnetic Suspension Control. In: International Symposium on Digital Manufacturing. Seoul. pp. 150-155.
[4] Zhai, M.D., Li, X.L., Liu, S.K. (2014) Research on The Design of Mechanical and Electromagnetic Composite Vibration Isolation System. In: Prognostics & System Health Management Conference. IEEE. Zhang Jiajie. pp. 553-558.
[5] Mizuno, T., Takasaki, M., Kishita, D., Hirakawa, K. (2007) Vibration isolation system combining zero-power magnetic suspension with springs. Control Engineering Practice, 15(3): 187-196.
[6] Mizuno, T., Namai, M., Takasaki, M., Ishino, Y. (2017) Proposal of Magnetic Suspension with Elastic Ferromagnetic Substance for Vibration Isolation System. In: International Automatic Control Conference. pp. 1-6.
[7] Guo, Y.X. (2016) Research on the design of passive and active composite control device resisting impact and vibration based on the electromagnetic actuator. Xidian University, Xi An.
[8] Wang, F. (2018) Comprehensive investigation on active-passive hybrid isolation and tunable dynamic vibration absorption. China Ship Research and Development Academy, Bei Jing.
[9] Carrellaa, A., Brennana, M.J., Watersa, T.P., Shin, K. (2008) On the design of a high-static-low-dynamic stiffness isolator using linear mechanical springs and magnets. Journal of Sound and Vibration, 315(3): 712-720.