

A VIBRATION ISOLATION SYSTEM USING STIFFNESS VARIATION CAPABILITY OF ZERO-POWER CONTROL

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ABSTRACT

This paper presents a vibration isolation system combining a positive stiffness spring in series with a negative stiffness spring. The negative spring is realized by an active zero-power control. The conventional zero-power control system yields constant negative stiffness, and the stiffness depends on the capacity of the permanent magnets or the gap-force coefficient of the magnets. This is one of the bottlenecks in the fields of application of zero-power control where adjustment of stiffness is necessary, such as developing vibration isolation system. To overcome the above problem, the basic zero-power control system is modified such that it can adjust stiffness by adding a minor proportional feedback of displacement to the zero-power control current. Some experiments have been carried out to measure the efficacy of the control system, as well as the vibration isolation system.

Keywords: Active Control, Vibration Isolation, Zero-Power Control, Zero-Compliance System, Magnetic Bearing.

1. INTRODUCTION

Vibration isolation system is widely used in many researches and precise manufacturing systems, such as semiconductor industries, high precision measurement, etc. There are two kinds of vibrations that may hamper desired operations. They are direct disturbance on the table and ground vibration. Both vibration disturbances can be suppressed by using passive technique or using active control. Vibration absorbers extract kinetic energy from the vibrating host system and the active system introduces opposing forces in the structure to affect compensation.

Vibration isolation of mechanical systems is achieved with passive technique because of its low cost. However, passive techniques need the trade-off between the isolation elements where high stiffness suspension are used for suppressing direct disturbance, and suspension with soft stiffness are used for ground vibration isolation [1,2]. On the other hand, active techniques do not have such limitations, and performances are better as well [3-7]. Recently vibration isolation systems have been developed using active zero-power controlled magnetic suspension in the view of reducing system development costs as well as maintenance costs [8,9]. The above systems use a combined system with positive and negative springs in series. A middle table is introduced in

this system between base to isolation table. Positive stiffness spring is used between base to middle table, and a suspension with negative stiffness is used between middle table to isolation table.

In this research, an active zero-power control is used to realize negative stiffness by using a hybrid magnet consists of electromagnet and permanent magnets. This control achieves the steady state in which the attractive force produced by the permanent magnets balance the weight of the suspended object, and the control current converges to zero. Since there is no steady energy consumption for achieving stable levitation, it has been applied to space vehicles [10], and to the magnetically levitated carrier system in clean rooms [11].

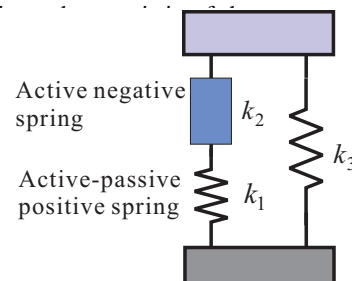


Fig 1. Principle of vibration isolation system using zero-power control

infinite stiffness by combing a mechanical positive spring with it is a very troublesome task, because a lot of mechanical springs are to be used to adjust both stiffnesses. This is one the obstacle to develop vibration isolation system using zero-power control.

Moreover, it can be noted that realizing negative stiffness can also be generalized by using linear actuator (voice coil motor) instead of hybrid magnet [13].

This paper demonstrates a control method of zero-power magnetic suspension system that has capability to adjust stiffness. This modified zero-power control introduces a proportional feedback of displacement to the original zero-power controller. The characteristics of the modified control system design are discussed analytically. Finally, a vibration isolation system is developed by connecting a mechanical spring in series with the modified zero-power control. Some experimental results are presented to show the efficacy of the control system, as well as the vibration isolation system.

2. PRINCIPLE OF VIBRATION ISOLATION

The vibration isolation system is developed to generate infinite (high) stiffness for direct disturbing forces and to maintain low stiffness for floor vibration. Infinite stiffness can be realized by connecting a mechanical spring in series with a magnetic spring that has negative stiffness [8, 9]. When two springs with spring constants of k_1 and k_2 are connected in series, the total stiffness k_c is given by

$$k_c = \frac{k_1 k_2}{k_1 + k_2}. \quad (1)$$

The above basic system has been modified by introducing a secondary suspension to avoid some limitations for system design and supporting heavy payloads [14]. The concept is demonstrated in Fig. 1. A spring k_3 is added in parallel with the serial connection of positive and negative springs. The total stiffness \tilde{k}_c is given by

$$\tilde{k}_c = \frac{k_1 k_2}{k_1 + k_2} + k_3. \quad (2)$$

However, if one of the springs has negative stiffness that satisfies

$$k_1 = -k_2, \quad (3)$$

the resultant stiffness becomes infinite for any finite value of k_3 , that is

$$|\tilde{k}_c| = \infty. \quad (4)$$

This research applies this principle of generating infinite stiffness against direct disturbance to the system. On the other hand, if low stiffness of mechanical springs for system (k_1, k_3) are used, it can maintain good ground

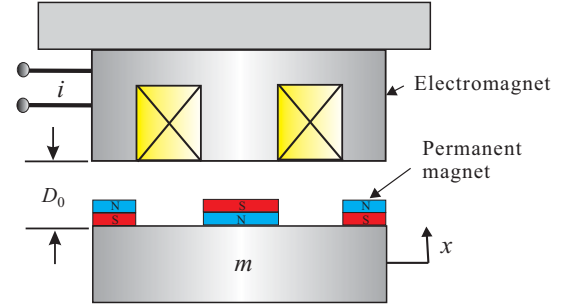


Fig 2. Basic model of zero-power control

vibration isolation performance as well.

3. ZERO-POWER CONTROL SYSTEM

Negative stiffness is generated by actively controlled zero-power magnetic suspension. The basic model, controller and the characteristic of the zero-power control system is described below.

3.1 Model

A basic zero-power controller is designed for simplification based on linearized equation of motions. It is assumed that the displacement of the suspended mass is very small and the nonlinear terms are neglected. The suspended object with mass of m is assumed to move only in the vertical translational direction as shown by Fig. 2. The equation of motion is given by

$$m\ddot{x} = k_s x + k_i i + w, \quad (5)$$

where x : displacement of the suspended object, k_s : gap-force coefficient of the permanent magnet, k_i : current-force coefficient of the electromagnet, i : control current, w : disturbance acting on the suspended object. The coefficients k_s and k_i are positive. When each Laplace-transform variable is denoted by its capital, and the initial values are assumed to be zero for

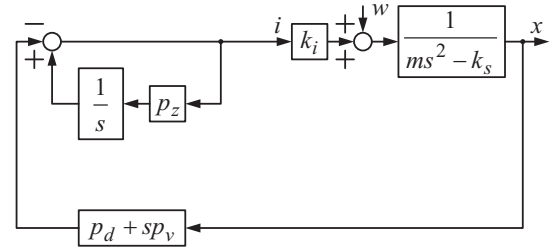


Fig 3. Block diagram of the zero-power controller

simplicity, the transfer function representation of the dynamics described by Eq. (5) becomes

$$\begin{aligned} X(s) &= \frac{1}{ms^2 - k_s} (k_i I(s) + W(s)) \\ &= \frac{1}{s^2 - a_0} (b_0 I(s) + d_0 W(s)), \end{aligned} \quad (6)$$

where $a_0 = k_s / m$, $b_0 = k_i / m$, and $d_0 = 1/m$.

$$\lim_{t \rightarrow \infty} i(t) = \lim_{s \rightarrow 0} sI(s) = 0. \quad (12)$$

3.2 Zero-Power Controlled Negative Stiffness

Zero-power can be achieved either by feeding back the velocity of the suspended object or by introducing a minor feedback of the integral of current in the PD (proportional-derivative) control system [12]. Since PD control is a fundamental control law in magnetic suspension, zero-power control is realized from PD control in this research using the second approach. In the current controlled magnetic suspension system, PD control can be represented as

$$I(s) = -(p_d + p_v s)X(s), \quad (7)$$

where p_d : proportional feedback gain, p_v : derivative feedback gain. Figure 3 shows the block diagram of a current-controlled zero-power controller where a minor integral feedback of current is added to the PD control. The control current of zero-power controller is given by

$$I(s) = -(p_d + p_v s)X(s) + \frac{p_z}{s}I(s), \quad (8)$$

where p_z : integral feedback gain in the minor current loop. From Eqs. (6) to (8), it can be written as

$$\frac{X(s)}{W(s)} = \frac{(s - p_z)d_0}{s^3 + (b_0 p_v - p_z)s^2 + (b_0 p_d - a_0)s + a_0 p_z}, \quad (9)$$

$$\frac{I(s)}{W(s)} = \frac{-s(p_v + p_d)d_0}{s^3 + (b_0 p_v - p_z)s^2 + (b_0 p_d - a_0)s + a_0 p_z}. \quad (10)$$

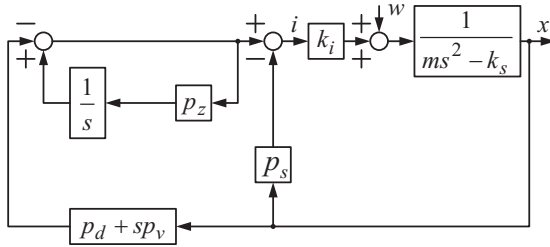


Fig 4. Block diagram of the modified zero-power controller that can adjust negative stiffness

To estimate the stiffness for direct disturbance, the direct disturbance, $W(s)$ on the isolation table is considered to be stepwise, that is

$$W(s) = \frac{F_0}{s}, \quad (F_0 : \text{constant}). \quad (11)$$

From Eqs. (10) and (11)

It indicates that control current always converges to zero in the zero-power control for any load. The steady displacement of the suspension, from Eqs. (9) and (11), is given by

$$\lim_{t \rightarrow \infty} x(t) = \lim_{s \rightarrow 0} sX(s) = -\frac{d_0}{a_0} F_0 = -\frac{F_0}{k_s}. \quad (13)$$

The negative sign in the right-hand side illustrates that the new equilibrium position is in the direction opposite to the applied force. It means that the system realizes negative stiffness. Assume that stiffness of any suspension is denoted by k . The stiffness of the zero-power controlled magnetic suspension is, therefore, negative and given by

$$k = -k_s. \quad (14)$$

3.3 Stiffness Adjustment

The stiffness realized by zero-power control is constant, as shown in Eq. (14). However, it is necessary to adjust stiffness of the zero-power control system in many applications, such as vibration isolation systems. There are two approaches to adjust stiffness of the zero-power control system. The first one is by adding a minor displacement feedback gain to the zero-power control current, and the other one is by adding a proportional feedback in the minor current feedback loop [15]. In this work, stiffness variation capability of zero-power control is realized by the first approach. Figure 4 shows the block diagram of the modified zero-power controller that is capable to adjust stiffness. The control current of the modified zero-power controller is given by

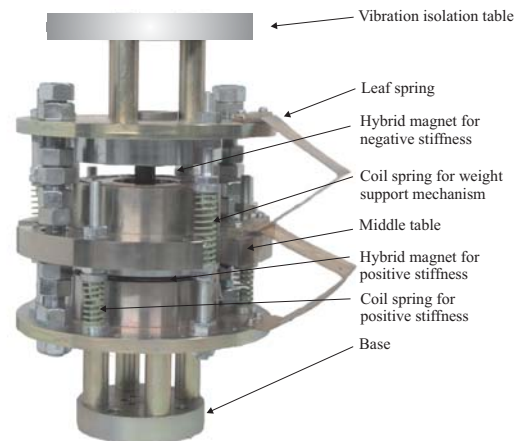


Fig 5. Photograph of the vibration isolation system

$$I'(s) = -\left(\frac{p_d s}{s - p_z} + p_v s + p_s\right)X(s), \quad (15)$$

where p_s : proportional displacement feedback gain across the zero-power controller.

The transfer-function representation of the dynamics shown in Fig. 4 is given by

$$\frac{X(s)}{W(s)} = \frac{(s - p_z)d_0}{s^3 + (b_0p_v - p_z)s^2 + (b_0p_d - a_0)s + a_0p_z + b_0p_s}. \quad (16)$$

From Eqs. (16) and (11), the steady displacement becomes

$$\begin{aligned} \lim_{t \rightarrow \infty} x(t) &= \lim_{s \rightarrow 0} sX(s) = -\frac{d_0p_z}{a_0p_z + b_0p_s} F_0 \\ &= -\frac{p_z}{k_s p_z + k_i p_s} F_0 \\ &= -\frac{F_0}{k_s + k_i p_s / p_z}. \end{aligned} \quad (17)$$

Therefore, the stiffness of the modified system becomes

$$k = -k_s - k_i \frac{p_s}{p_z}. \quad (18)$$

It indicates that the stiffness can be increased or decreased by changing the feedback gain p_s .

4. EXPERIMENTAL SET UP

A single-axis vibration isolation system combining positive and negative stiffness suspension in series is shown in Fig. 5. It consisted of a circular base, a circular middle table and a circular isolation table. The height, diameter and weight of the system were 300mm, 200mm

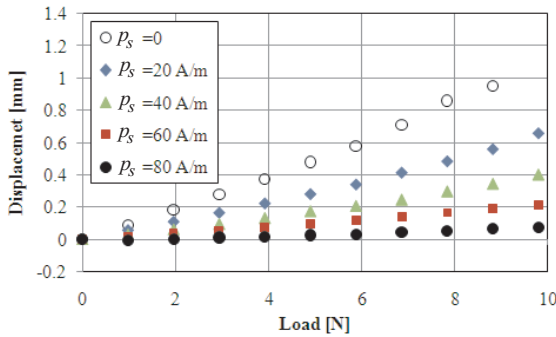


Fig 6. Load-displacement characteristics of the modified zero-power controller

and 20 kg, respectively. The relative displacement of the base to middle table was measured by an eddy-current displacement sensor, and positive stiffness was realized by a hybrid magnet consisted of an electromagnet (180-turns) that was fixed to the base, and four permanent magnets (15mm×2mm) attached to the

middle table. The permanent magnets are made of Neodymium-Iron-Boron (NdFeB). The middle table was also supported by three coil springs. The hybrid magnet and the coil springs were used in tandem to generate positive stiffness suspension. Another displacement sensor was used to measure the relative displacement between middle table to isolation table. A different hybrid magnet consisted of an electromagnet and six permanent magnets was used to realize negative stiffness. The isolation table was also supported by three coil springs as weight support mechanism.

The motion of the isolation table and that of the middle table were restricted to move only in the vertical direction. It was done by using a vertical shaft which was fixed to be base and passed through the center of the isolation table and middle table. The friction between the shaft and the isolation table and middle table were reduced to a minimum possible level by using suitable lubricant and ball bearings. Two additional threaded shaft and several hexagonal nuts were employed as stopper and limiter for the movement of the isolation table and the middle table. Two leaf springs were used to confine the rotational motions of the isolation table and middle table. These leaf springs were also behaved as damper for the both tables.

5. EXPERIMENTAL RESULTS

The experiments have been carried out to measure the performances both for the modified zero-power controller, as well as for the vibration isolation system. Figure 6 shows the load-displacement characteristics of the system with the modified zero-power controller (Fig. 4). When the proportional feedback gain, $P_s=0$, it can be considered as a conventional zero-power controller (Fig. 3). The result shows that when the payloads were put on the suspended object, the table moved in the direction of load application, and gap was widened. It indicates that the zero-power control realized

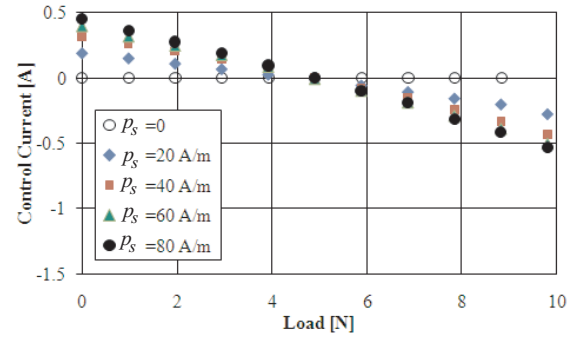


Fig 7. Load-current characteristics of the modified zero-power controller

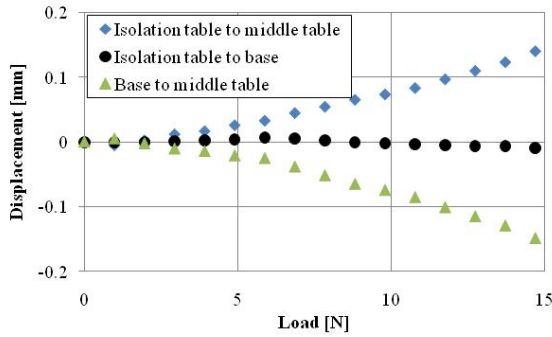


Fig 8. Zero-compliance characteristics of the isolation table to static direct disturbance using modified zero-power controller

negative displacement, and hence its stiffness is negative, as described by Eqs. (13) and (14). The conventional zero-power controller ($P_s=0$) realized fixed negative stiffness of magnitude -9.2 N/mm. When the proportional feedback gain, P_s was changed, the stiffness also gradually increased. When $P_s=80$ A/m, negative stiffness was increased to -103.1 N/mm. It confirms that proportional feedback gain, P_s can change the stiffness of the zero-power controller, as explained in Eq. (18).

Figure 7 shows the load-current characteristics of the zero-power controller. The control current necessary for the conventional zero-power controller ($P_s=0$) was always zero. It proves the analysis described in Eq. (12). When the gain P_s was increased, a minor control current was necessary. The maximum control current for $P_s=80$ A/m was ± 0.5 A up to 10 N load. However, the zero-power characteristic can be restored again by adjusting the weight support spring, once the system is stabilized.

Finally, experiments have been carried out with the developed vibration isolation system. Zero-compliance characteristics of the isolation table to static direct disturbance using modified zero-power controller is shown in Fig. 8. In this case, stepwise static direct disturbance in the vertical direction of the table was generated by putting payloads on the center of the table. The displacements of the isolation table to middle table and that of the middle table to base were measured by eddy-current displacement sensors. It is seen from the figure that positive stiffness was realized between base to

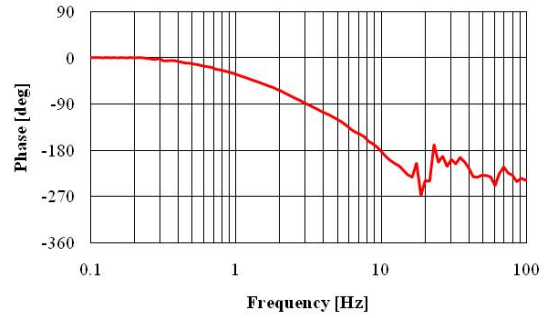
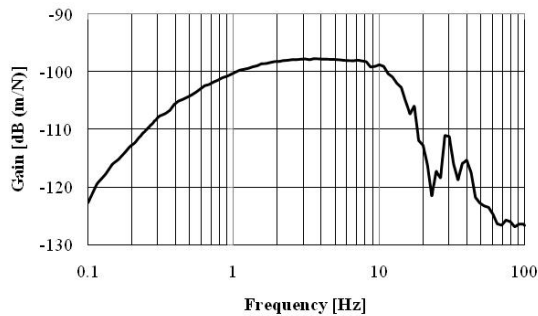


Fig 9. Frequency response of the isolation table to dynamic direct disturbance

middle table, and negative stiffness was realized between isolation table to middle table. Zero-compliance to direct disturbance of the isolation table to base was achieved when Eq. (3) was satisfied. In other words, an infinite stiffness of the isolation system was realized that confirms Eq. (4). At higher load, the zero-compliance characteristic may be lost due to the nonlinearity of the zero-power control system. A nonlinear compensation to zero-power controller can solve such problem [16].

Figure 9 shows the frequency response of the isolation table to dynamic direct disturbance. In this experiment, the isolation table was excited by an electromagnet which was fixed to the base above the isolation table. The displacement of the table was considered as output in this case. The frequency response was measured by a dynamic signal analyzer. The result shows that the table generated very low displacement at the low frequency region. The displacement of the table at 0.1 Hz was -122 dB [m/N]. It confirms that isolation table generated very high stiffness at low frequency dynamic direct disturbance.

Finally, step response of the developed vibration isolation system was measured as shown in Fig. 10. An electromagnet was employed to generate stepwise disturbance [0 to -10 N] at 0.25 Hz. The results demonstrate that the gap between isolation table to middle table was increased and the displacement of the middle table to base was decreased for downward disturbance on the table. However, the isolation table returned to the original position within 0.5s. The displacement and transient period to return to the original position can further be reduced by introducing feedforward control with the zero-power control system.

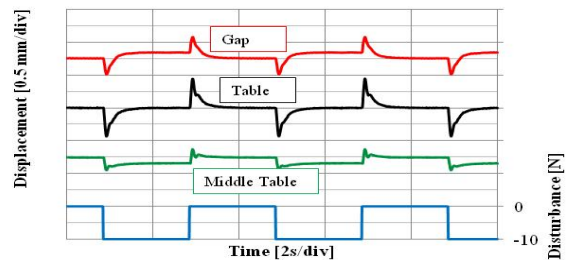


Fig 10. Step response of the isolation table

6. CONCLUSIONS

A vibration isolation system has been developed by combining an active zero-power control with mechanical springs. The zero-power control is modified by introducing a proportional displacement feedback to the zero-power control current. The modified zero-power controller yielded negative stiffness with the capability of adjusting negative stiffness. The concept was investigated by some experiments. The experimental results showed that the stiffness of the magnetic suspension system depends on the proportional gains of the displacement feedback. The modified zero-power control had wide stiffness range, and could be applied to develop vibration isolation system. The system can further be improved by providing minor current feedback to the integral of the zero-power control system.

A single-degree-of-freedom vibration isolation system has been developed using the fixed positive suspension and the modified zero-power controller. Infinite stiffness of the isolation system was realized when the negative stiffness was equal in magnitude with the positive stiffness. The static and dynamic response of the isolation table showed that the developed isolation system could effectively suppress the effect of direct disturbance.

7. ACKNOWLEDGMENT

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