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A 3-DOF Modular Vibration Isolation System Using Zero-Power Magnetic Suspension with Adjustable Negative Stiffness

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***Abstract*-This paper presents a 3-DOF vibration isolation system combining three vibration isolation modules. Each vibration isolation module is constructed by connecting a positive stiffness spring in series with a negative stiffness spring. The positive and negative spring is realized by an active controlled magnetic suspension. In the previous system, conventional zero-power control system has been used to generate 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 the adjustment of stiffness is necessary. On the other hand, the suspension system was used in the vertical and horizontal directions, which made the system complicated. To overcome the above problems, a vibration isolation system is developed with three modules connected by parallel mechanism. Some experiments have been carried out to measure the efficacy of the control system, as well as the vibration isolation system.**

I. INTRODUCTION

The design of vibration isolation system concerns many fields of research and applications, such as semiconductor industries, high precision measurement, etc. There are two kinds of vibrations that may be considered before designing vibration isolation system. They are direct disturbance on the table and ground vibration. Both vibration disturbances can be suppressed by using passive technique or using active control. Usually vibration absorbers extract kinetic energy from the vibrating host system and the active system introduces opposing forces in the structure to affect compensation. Passive techniques can be used to isolate vibration of mechanical systems with low cost. However, passive techniques need the trade-off between the isolation elements where higher stiffness of suspension are used for suppressing the direct disturbance, and suspension with lower stiffness are used for ground vibration isolation [1,2]. Passive systems cannot adapt with changing conditions. On the contrary, active techniques can overcome such limitations because it can adapt with changing condition, and the performances are better than passive techniques as well [3-7]. The conventional active control systems use high-performance sensor such as servo-type accelerometer, which is rather costly and makes the system costlier. This is one of the hurdles to develop vibration isolation system using active control. Recently the authors have proposed a vibration isolation systems using active zero-power controlled magnetic suspension in the view of reducing system development costs as well as maintenance costs [8,9]. In the proposed system, eddy-current relative displacement sensors can be used for displacement feedback. Again, the control current converges to zero for the zero-power control system. Therefore, the developed system becomes rather inexpensive than the conventional active systems.

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].

A unique characteristic of the zero-power control is that it realizes negative stiffness [12]. Therefore, a very high (theoretically infinite) stiffness of a vibration isolation system can be achieved by combining a mechanical spring in series with a zero-power control. However, the amplitude of negative stiffness realized by zero-power control is solely dependent of magnetomotive force or gap-force coefficient of the permanent magnets. The amplitude of negative stiffness is fixed for a particular magnet, and the gap between suspended object and the permanent magnet, does not have any capability to change stiffness. Hence realizing 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 barrier 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 three-degree-of-freedom (3-DOF) vibration isolation system is developed by connecting three modules in parallel. Each module is constructed by connecting a mechanical spring in series with the modified zero-power control. Middle tables are introduced in each module 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. Some experimental results are presented to show the efficacy of the control system, as well as the vibration isolation system.

II. ZERO-POWER CONTROL

This section presents the realization of negative stiffness by actively controlled zero-power magnetic suspension. The basic model, controller and the characteristic of the zero-power control system is described below.

A. Basic Model

In this work, negative stiffness is realized by using hybrid magnet consists of permanent magnets and an electromagnet. 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. 1. The equation of motion is given by

$$m\ddot{x} = k_s x + k_i i + w\,, \tag{1}$$

where x : displacement of the suspended object, k_s : gap-force coefficient of the permanent magnet, k_i : current-force coefficient of the permanent magnet, 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 simplicity, the transfer function representation of the dynamics described by (1) becomes

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

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

B. Realization of Zero-Power and 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 very stable and a fundamental control law in magnetic suspension, zero-power control is realized from PD control in this research using

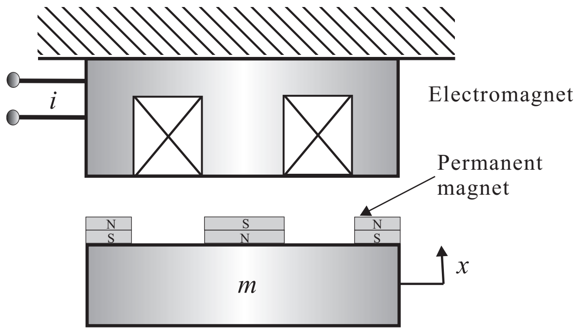

Figure 1. Basic model of zero-power control

the second approach. In the current controlled magnetic suspension system, PD control can be represented as

$$I(s) = -(p_d + p_v s)X(s), \tag{3}$$

where p_d : proportional feedback gain, p_v : derivative feedback gain. Fig. 2 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

$$I(s) = -(p_d + p_v s)X(s) + \frac{p_z}{s} I(s)\,, \tag{4}$$

where p_z : integral feedback gain in the minor current loop. From (2) to (4), it can be written as

$$\frac{X(s)}{W(s)} = \frac{(s - p_z)d_0}{s^3 + (b_0 p_v - p_z)s^2 + (b_0 p_d - a_0)s + a_0 p_z}, \tag{5}$$

$$\frac{I(s)}{W(s)} = \frac{-s(sp_v + p_d)d_0}{s^3 + (b_0 p_v - p_z)s^2 + (b_0 p_d - a_0)s + a_0 p_z}. \tag{6}$$

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

$$W(s) = \frac{F_0}{s}, (\, F_0 : \text{constant}). \tag{7}$$

From (6) and (7)

$$\lim_{t\to\infty} i(t) = \lim_{s\to 0} sI(s) = 0. \tag{8}$$

It indicates that control current always converges to zero in the zero-power control for any load. Again, the steady displacement of the suspension, from (5) and (7), is given by

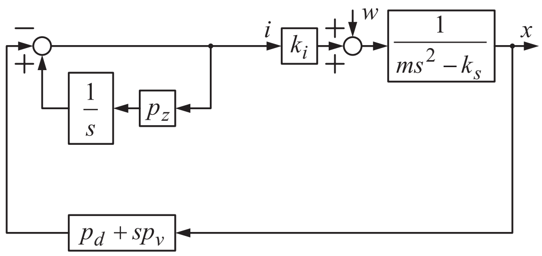

Figure 2. Block diagram of the zero-power controller

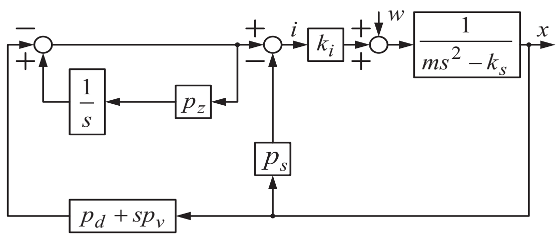

Figure 3. Block diagram of the modified zero-power controller

$$\lim_{t\to\infty} x(t) = \lim_{s\to 0} sX(s) = -\frac{d_0}{a_0}F_0 = -\frac{F_0}{k_s}. \tag{9}$$

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

$$k = -k_s. \tag{10}$$

C. Stiffness Adjustment of Zero-Power Control

It is seen from (10) that the stiffness realized by zero-power control is constant, because there is no other gain included in it. 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 [14]. In this work, stiffness variation capability of zero-power control is realized by the first approach. Fig. 3 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

$$I'(s) = -(\frac{p_d s}{s - p_z} + p_v s + p_s)X(s), \tag{11}$$

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

The transfer-function representation of the dynamics shown in Fig. 3 is given by

$$\frac{X(s)}{W(s)} = \frac{(s - p_z)d_0}{s^3 + (b_0 p_v - p_z)s^2 + (b_0 p_d - a_0)s + a_0 p_z + b_0 p_s}. \tag{12}$$

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

$$\lim_{t\to\infty} x(t) = \lim_{s\to 0} sX(s) = -\frac{d_0 p_z}{a_0 p_z + b_0 p_s}F_0$$
$$= -\frac{p_z}{k_s p_z + k_i p_s}F_0$$
$$= -\frac{F_0}{k_s + k_i p_s / p_z}. \tag{13}$$

Therefore, the stiffness of the system becomes

$$k' = -k_s - k_i \frac{p_s}{p_z}. \tag{14}$$

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

III. Concept 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

$$k_c = \frac{k_1 k_2}{k_1 + k_2}. \tag{15}$$

The above basic system has been modified by introducing a secondary suspension to avoid some limitations for system design and supporting heavy payloads [15]. The concept is demonstrated in Fig. 4. A spring k_3 is added in parallel with the serial connection of positive and negative springs. The total stiffness $\widetilde{k}_c$ is given by

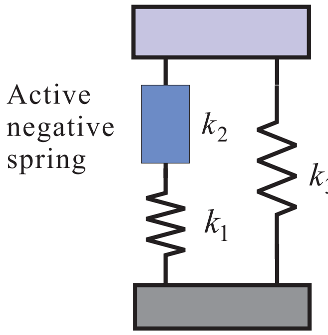

Figure 4. Principle of vibration isolation system using zero-power control

$$\tilde{k}_c = \frac{k_1 k_2}{k_1 + k_2} + k_3 . \tag{16}$$

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

$$k_1 = -k_2 , \tag{17}$$

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

$$\left|\tilde{k}_c\right| = \infty. \tag{18}$$

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 vibration isolation performance as well.

IV. Experimental Apparatus

A three-axis vibration isolation system is developed combining three vibration isolation modules. Each module can be considered as a single-degree-of-freedom vibration isolation system as shown in Fig. 5. The developed 3-DOF vibration isolation table is shown in Fig. 6.

Each module 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 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

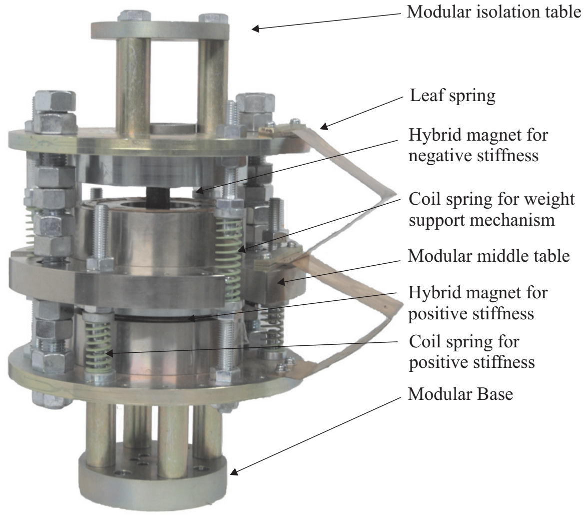

Figure 5. A vibration isolation module

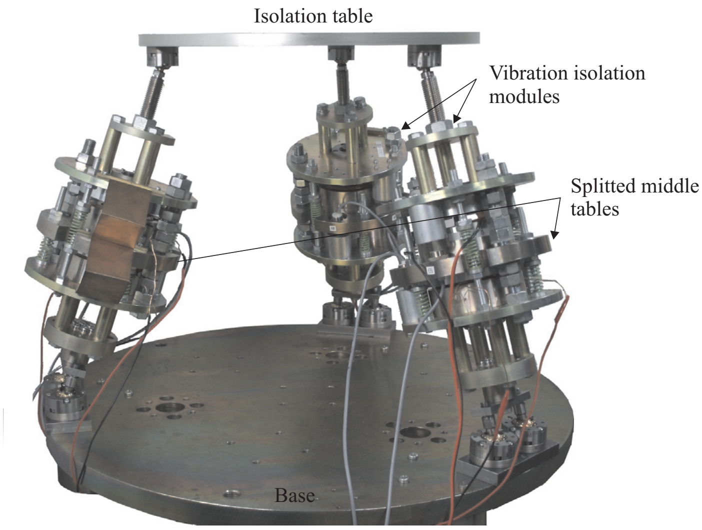

Figure 6. Photograph of the developed 3-DOF vibration isolation system

the relative displacement between middle table to isolation table and a separate 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

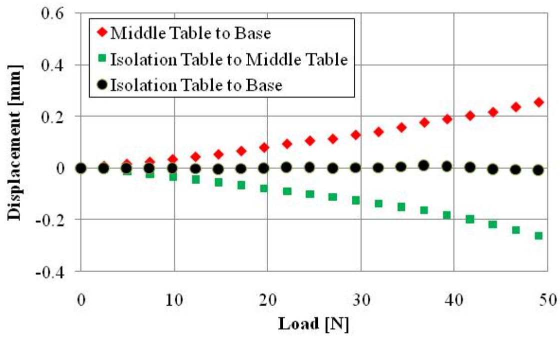

Figure 7. Static response of the isolation table (vertical direction)

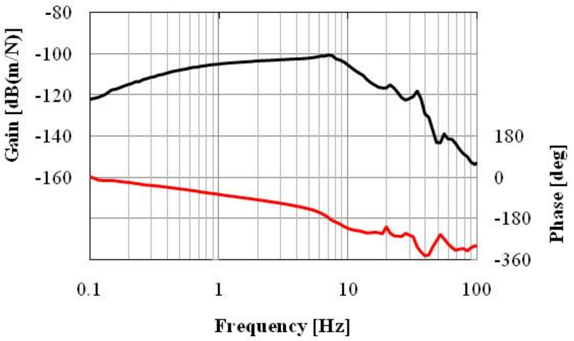

Figure 8. Frequency response of the isolation table to dynamic direct disturbance

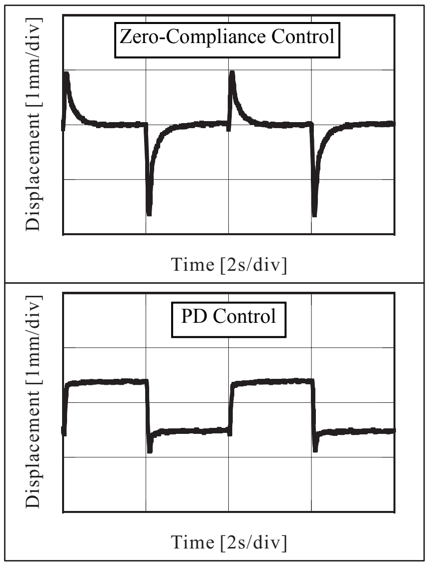

Figure 9. Step response of the isolation table (vertical direction)

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.

It can be noted that six-axis vibration isolation table can be developed by connecting six modules in a similar way.

V. EXPERIMENTAL RESULTS

The experiments have been carried out to measure the response of the vibration isolation system for translational and rotational loading disturbances. Static characteristic of the isolation table in the vertical translational direction was measured as shown in Fig. 7. The table supported by three modules showed zero-compliance to static direct disturbance up to approximately 50N. The relative displacement shown between middle tables to base and that between table to middle tables were the average values of the three modules.

Figure 8 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 under 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 small displacement at the low frequency region. The displacement of the table at 0.1 Hz was -122 dB [m/N]. It confirms that the isolation table generated very high stiffness at low frequency dynamic direct disturbance.

Next, step response of the developed vibration isolation system in the vertical direction was measured as shown in Fig. 9. An electromagnet was employed to generate stepwise disturbance [0 to 10N] at 0.25 Hz. At first, the isolation table was suspended by PD control, and after that, it was switched to zero-compliance control when positive and negative stiffness were equal in magnitude. The results demonstrate that the displacement of the isolation table was large when PD control was used. However, the displacement of the isolation table was almost zero when zero-compliance control was used.

Next, the frequency response of the isolation table was measured along rotational direction (roll). The table was tilted along roll direction by applying torque using an electromagnet. In this case, the rotational angle was regarded as output for measuring the frequency response. Figure 10 shows the frequency response of the isolation table under PD control and zero-compliance control. It is seen from the figure that table displacement at 1 Hz under PD control was -68dB (degree/N.m) and it was almost same below 1 Hz. When the table was switched to zero-compliance control, the displacement of the table at 0.015 Hz reached to -97 dB. It reveals that the table can realize very high stiffness at low frequencies even for rotational mode.

Finally the isolation table was tilted at very low frequency (0.1Hz) by applying a sinusoidal torque. The angular response of the isolation table was measured again for PD control and zero-compliance control. The results show that the isolation was displaced from its original position when PD control was used. The isolation table remained almost in the same position and realized very high stiffness to rotational moment for zero-compliance control.

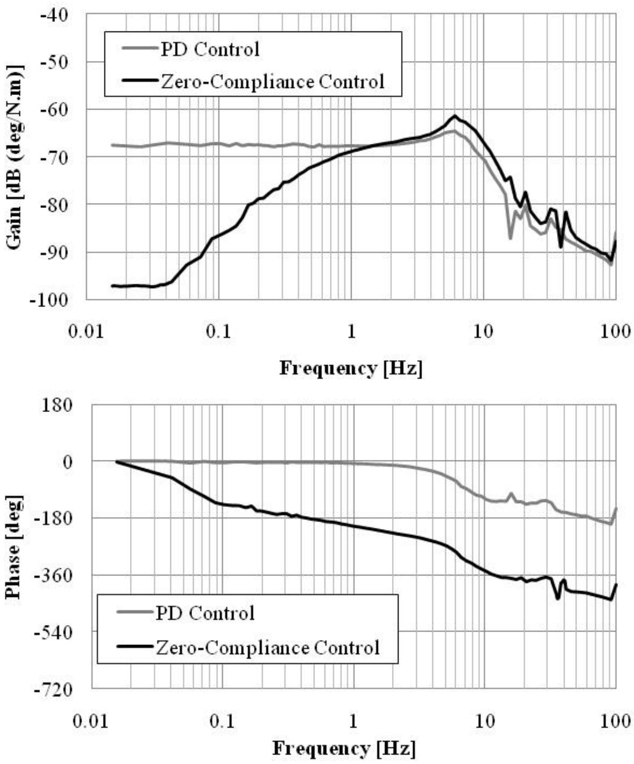

Figure 10. Frequency response of the isolation table to dynamic direct disturbance (roll)

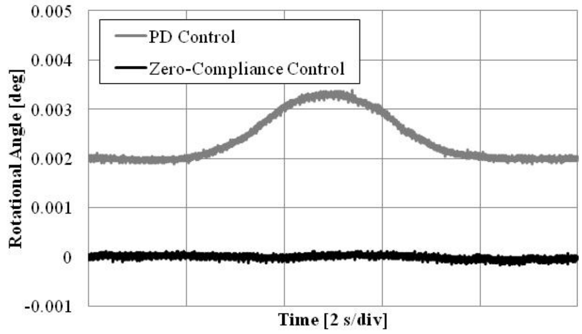

Figure 11. Response of the isolation table to low frequency dynamic direct disturbance (roll)

VI. 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 system can further be improved by providing minor current feedback to the integral of the zero-power control system.

A three-degree-of-freedom vibration isolation system has been developed using the fixed positive suspension and the modified zero-power controller. Infinite stiffness (zero-compliance) 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 applied in the vertical translational and rotational directions.

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