Retrofitting A Passive Vibration Isolation System with Zero-Compliance Modules

Takeshi MIZUNO*, Yuji ISHINO, Masaya TAKASAKI*****

** Department of Mechanical Engineering, Saitama University, Shimo-Okubo 255, Sakura-ku, Saitama 338-8570, Japan (Tel: +81-48-858-3455; e-mail: mizar@mech.saitama-u.ac.jp)*
*** Department of Mechanical Engineering, Saitama University, Shimo-Okubo 255, Sakura-ku, Saitama 338-8570, Japan (Tel: +81-48-858-3453; e-mail: yishino@mech.saitama-u.ac.jp)*
**** Department of Mechanical Engineering, Saitama University, Shimo-Okubo 255, Sakura-ku, Saitama 338-8570, Japan (Tel: +81-48-858-3451; e-mail: masaya@mech.saitama-u.ac.jp)*

Abstract: A vibration isolation module with zero-power magnetic suspension is developed to retrofit a passive vibration isolation system to improve the performance against direct disturbance. In the module, the zero-power magnetic suspension mechanism is connected in series with a normal spring through a middle mass. This structure can generate zero compliance to direct disturbance while the transmitted vibration is still reduced. Such zero-compliance module was actually designed and fabricated. The characteristics of the fabricated module were studied experimentally. A passive vibration isolation system using air springs were retrofitted with four modules. It was demonstrated that the performance against direct disturbance was improved significantly.

Keywords: vibration, magnetic suspension, magnetic bearings, active compensation, passive suspension, active vehicle suspension, disturbance rejection, linearization.

1. INTRODUCTION

Vibration isolation is an important technology in semiconductor manufacturing and high-precision measuring (Yoshioka *et al.*, 2001). There are two kinds of vibration that must be reduced by a vibration isolation system: vibration transmitted from the ground through the suspension (transmitted vibration) and vibration caused by disturbances acting on an isolation table directly (direct disturbance) (Yasuda & Ikeda, 1993). Soft suspension is suitable for reducing the transmitted vibration because dynamic coupling between the vibration source and the isolation table is weakened. Therefore, air spring is mostly used to suspend the isolation table in common passive vibration isolation systems (Racca, 1996; Rivin, 2003). However, such soft suspension makes the system weak against direct disturbance; the vibration of the isolation table is easily induced by disturbance acting on the table.

In this paper, a vibration isolation module is developed to retrofit such a passive system. The module uses a zero-power magnetic suspension mechanism to produce negative stiffness, which is connected with a normal spring through a middle mass. This structure can generate *infinite stiffness*, or *zero compliance* to direct disturbance while the transmitted vibration is still reduced (Mizuno, 2001; Mizuno *et al.*, 2006).

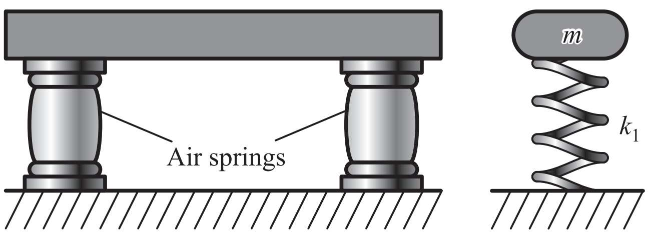
A zero-compliance module is designed and fabricated based on this concept. The characteristics of the fabricated module are studied experimentally. A passive vibration isolation system using air springs is retrofitted with four modules to improve the performance.

2. PRINCIPLES

2.1 Passive Vibration Isolation System

Figure 1 illustrates a typical passive vibration isolation system where an isolation table m is supported by air springs. Its simplest single-degree-of-freedom model is also shown in Fig.1b where the air springs are approximated as a single spring k. It is well known that the vibration transmitted to the table from ground is reduced only at frequencies higher than $\sqrt{2m/k}$ (Rivin, 2003). It implies that less stiffness (smaller k) is better for reducing the transmitted vibration. Therefore, soft air springs are used in most passive vibration isolation systems. However, such soft suspension makes the system weak against direct disturbance; the vibration of the isolation table is easily induced by disturbance on the table, for example, by putting a weight on the table.

In contrast, higher stiffness (larger k) is better for being



(a) Typical vibration isolation system (b) Basic model

Fig.1 Passive vibration isolation system robust to direct disturbance because it reduces the displacement of the isolation table from its desired position. In conventional passive-type vibration isolation systems, therefore, a trade-off between lower and higher stiffness is inevitable, so that performance is limited (Yasuda and Ikeda, 1993).

Active-type vibration isolation systems do not suffer from such performance limitations in principle (Rivin, 2003). One of the fundamental control strategies is *skyhook damping* (Fuller *et al.*, 1996); the absolute velocity of the isolation table is fed back. The performance limitations of passive-type vibration isolation systems can be effectively overcome by two-degrees-of-freedom control (Yasuda *et al.*, 1996; Yoshioka *et al.*, 2001). Various other control methods, such as repetitive (Daley *et al.*, 2006) and active acceleration (Zhu *et al.*, 2006) control have also been applied. However, most active vibration isolation systems use high-performance sensors to detect the vibration of an isolation table with high sensitivity in a low-frequency domain. Since these sensors are costly, active systems are more expensive than passive systems, which represents a critical obstacle to expanding their fields of application.

2.2 Vibration Isolation Using Negative Stiffness

To overcome the above-mentioned performance limitation of conventional passive systems and high cost of active ones, vibration isolation systems using negative stiffness have been proposed and developed (Mizuno, 2001; Mizuno *et al.*, 2006; Hoque *et al.*, 2006a; 2006b, 2010). This concept is briefly described here. Two springs having spring constant of k_1 and k_2 are connected in series as shown by Fig.2a. The total stiffness k is given by

$$k = \frac{k_1 k_2}{k_1 + k_2}. \tag{1}$$

If one of two springs has negative stiffness that satisfies

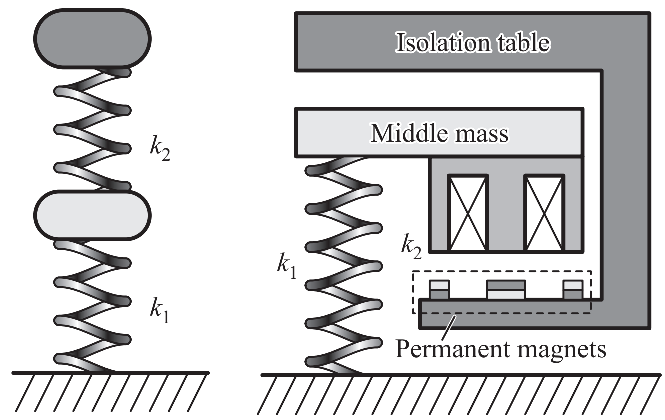
$$k_2 = -k_1, \tag{2}$$

the total stiffness becomes infinite, that is

$$|k| = +\infty. \tag{3}$$

It indicates that infinite stiffness can be generated by connecting a soft (low-stiffness) negative spring with a soft positive spring in series and equalizing their amplitudes of stiffness. In the proposed vibration isolation systems, this principle of generating infinite stiffness or zero compliance is used to make the vibration isolation system robust to direct disturbance.

Figure 2b shows the configuration of one of the proposed vibration isolation systems (Mizuno, 2001). A zero-power magnetic suspension system is used as a negative spring. A middle mass is connected to the base through a spring k_1 that works as a conventional vibration isolator. An electromagnet for zero-power magnetic suspension is fixed to the middle mass. The part of an isolation table facing the electromagnet is made of permanents magnet for producing bias flux and ferromagnetic material (steel) for confining the magnetic flux to desired magnetic paths. This is referred to as the reaction part. This system can reduce vibration transmitted from ground by making k_1 small and infinite stiffness can be produced simultaneously to counteract direct disturbances by setting the amplitude of negative stiffness equal to k_1.



(a) Serial connection of two springs (b) Vibration isolation system using zero-power magnetic suspension

Fig.2 Vibration isolation system using negative stiffness

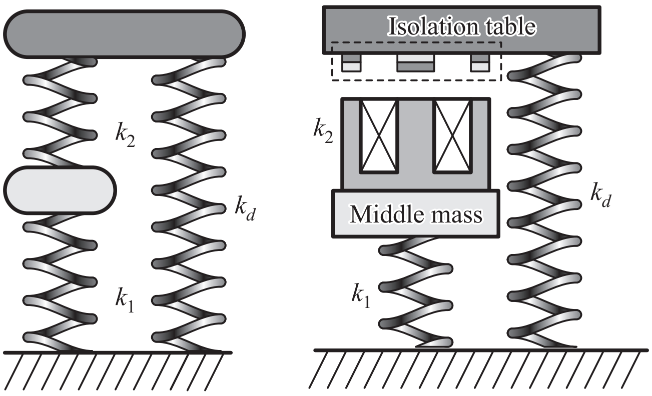
2.3 Weight Support Mechanism

In the system shown by Fig.2b, the whole weight of the isolation table is supported only by zero-power magnetic suspension. When the isolation table is large, therefore, a lot of permanent magnets are needed to suspend its weight, which will raise the cost of system. Another problem that can be expected in putting the proposed system to practical use is that the reaction part must be installed under the middle mass, because the electromagnet can produce only attractive force usually. This makes the structure of the vibration isolation system rather complex.

These problems have been overcome by introducing an auxiliary suspension for supporting the weight of the isolation table (Mizuno *et al.*, 2006). A spring k_d is inserted in parallel with the series-connected positive and negative springs as shown by Fig.3a. The total stiffness $\widetilde{k}_c$ is given by

$$\widetilde{k}_c = \frac{k_1 k_2}{k_1 + k_2} + k_d. \tag{4}$$

Under the condition (2), the resultant stiffness becomes infinite for any finite value of k_d. Figure 3b shows a configuration of vibration isolation system with weight



(a) Introduction of weight support mechanism

(b) Modified configuration of vibration isolation system

Fig.3 Vibration isolation system with weight support mechanism

support mechanism. A spring k_d is installed between the isolation table and the base. When the spring is set to produce upward force greater than the gravitational force in the equilibrium state, the zero-power magnetic suspension is operated to produce downward force. Since the reaction part is installed above the middle mass, the modified structure is more suitable for design and production than the original. It should be commented that the use of a soft weight-support spring enables the system to keep the isolation performance from ground vibration.

2.4 Retrofit of Passive Vibration Isolation System

The configuration shown by Fig.3 suggests a retrofit of a conventional passive suspension system. Figure 4 demonstrates the concept of the system retrofitted by introducing a zero-compliance module consisting of a series connection of positive and negative springs. The weakness of the passive system to direct disturbance can be overcome by the module effectively. From a view from the opposite side, a weight support mechanism shown in Fig.3 is replaced by an air spring that is often used in conventional passive systems. Considering that such passive systems have been already used widely in laboratories and industries and the requirement to higher-performance vibration control is increasing, the proposed retrofit will be promising not only technically but also commercially.

3. REALIZATION OF NEGATIVE STIFFNESS

3.1 Zero-Power Control

The key technology in the proposed vibration isolation system is the realization of a suspension (spring) with negative stiffness without instability. The original method is the zero-power control (Mizuno, 2001). It has been

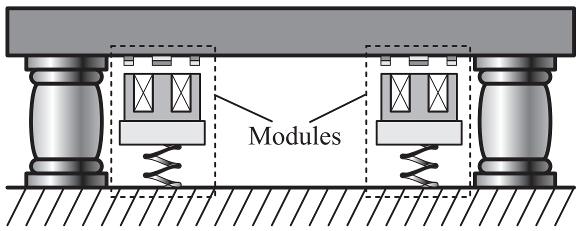


Fig.4 Retrofit of a passive system

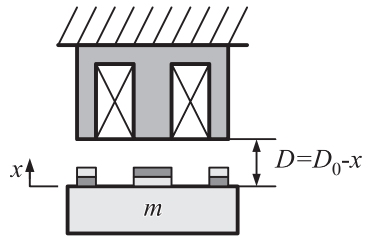


Fig.5 Model of a magnetic suspension system

generalized to be applicable to linear actuators (Mizuno *et al.*, 2003). In this work, the original method is adopted.

Figure 5 shows a basic model of magnetic suspension with a hybrid magnet. The hybrid magnet consists of a permanent for generating the bias flux and an electromagnet for controlling the suspension force. In this model, the permanent magnet is attached to the suspended object that is referred to as "floator". The attractive force generated by the permanent magnet balances the gravitational force at the equilibrium position. The equation of motion is given by

$$m\ddot{x}(t) = F(t) - mg + w(t)\,, \tag{5}$$

where x : displacement of the floator from the the equilibrium position, F : attractive force produced by the hybrid magnet, w :disturbance force acting on the floator. The magnetic force F is approximately given by

$$F = K\frac{(I_0 + i)^2}{(D_0 - x)^2}\,. \tag{6}$$

where K : characteristic coefficient of the hybrid magnet, I_0 : equivalent constant current determined from the magnetmotive force of the permanent magnet, D_0 : steady-state gap between the floator and the electromagnet including the thickness of permanent magnet. In the neighbourhood of the equilibrium points, it can be approximately linearized as

$$F \cong F_0 + k_s x + k_i i\,, \tag{7}$$

where

$$F_0 = K\frac{{I_0}^2}{{D_0}^2}(= mg)\,, \tag{8}$$

$$k_s = K\frac{{I_0}^2}{{D_0}^3}\,, \tag{9}$$

$$k_i = K\frac{I_0}{D_0{}^2}. \tag{10}$$

From Eqs.(5) to (8), we get

$$m\ddot{x}(t) = k_s x + k_i i + w(t). \tag{11}$$

There are various methods of achieving the virtually zero-power state where the control current i converges to zero in the steady-state (Mizuno & Takemori, 2002). One of the simple methods is to feed back the integral of current in addition to PD control as

$$i(t) = -(p_d x(t) + p_v \dot{x}(t)) + q_I \int i(t)dt. \tag{12}$$

The transfer function representation of the dynamics described by Eqs.(11) and (12) becomes

$$X(s) = \frac{s - q_I}{t_c(s)} W(s), \tag{13}$$

$$I(s) = -\frac{s(p_d + p_v s)}{t_c(s)} W(s), \tag{14}$$

where

$$t(s) = (ms^2 - k_s)(s - q_I) + k_i s(p_d + p_v s). \tag{15}$$

It is assumed that the feedback gains p_d, p_v and q_I are selected for the closed-loop system to be stable. When a constant force W_0 is applied to the floator, that is

$$W(s) = \frac{W_0}{s}, \tag{16}$$

the control current converges to zero, that is

$$\lim_{t\to\infty} i(t) = \lim_{s\to 0} sI(s) = 0. \tag{17}$$

Therefore, the zero-power control is achieved. In addition,

$$\lim_{t\to\infty} x(t) = \lim_{s\to 0} sX(s) = -\frac{W_0}{k_s}. \tag{18}$$

The negative sign appearing in the right-hand side verifies that the zero-power control system behaves as if it has negative stiffness.

3.2 Nonlinear Compensation

The stiffness of the zero-power suspension system is given by $-k_s$ as shown above. It indicates that the amplitude of stiffness varies according to the gap (see Eq.(9)). It causes undesirable displacement of the isolation table when large constant direct disturbance acts on the table. An effective way of compensating such nonlinearity is to feed back the higher-order terms of the displacement in addition to the basic zero-power control given by Eq.(12) (Hoque *et al.*, 2006b; 2010). It is derived by considering higher terms regarding to the displacement instead of Eq.(7) as

$$F \cong F_0 + k_i i + k_s(x + c_2 x^2 + c_3 x^3 + \cdots), \tag{19}$$

where

$$c_2 = \frac{3}{2D_0}, \quad c_3 = \frac{4}{2D_0^2}. \tag{20}$$

The control current for cancelling the second-order term can be found from (19) as

$$i(t) = -(p_d x(t) + p_v \dot{x}(t)) + q_I \int i(t)dt - q_s x^2(t), \tag{21}$$

where

$$q_s = \frac{k_s c_2}{k_i} = \frac{3I_0}{2D_0{}^2}.$$

4. ZERO-COMPLIANCE MODULE

Figures 6 and 7 show a photo and a cross-section diagram of a developed zero-compliance module. It is to be noted that

Fig.6 Photo of fabricated module

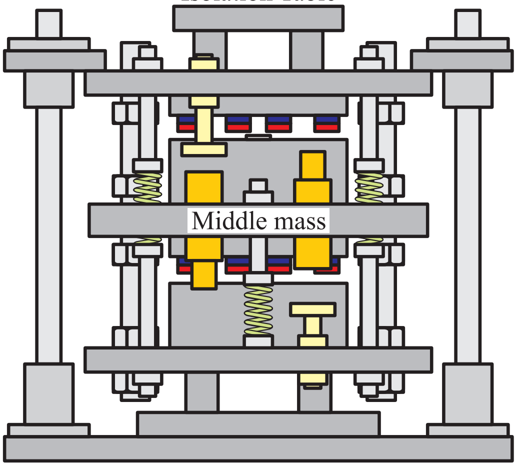


Fig.7 Schematic drawing of the module

this module can operate in the stand-alone mode as a vibration isolator. The height, diameter and mass of the apparatus are 200 mm, 100 mm and 15 kg, respectively. This module has an isolation table, a middle mass and a base. The mass of the isolation and the middle mass are 3 kg and 6 kg, respectively. Each of them is guided by a linear bearing to move only in the vertical direction. V-shaped leaf springs are used to prevent the rotation of the middle table and the isolation table about the vertical axes.

An electromagnet with a 150-turn coil is fixed to the middle mass. Six disc-shape permanent magnets made of NdFeB materials are attached on the ferromagnetic circular disc of the isolation table. They constitute a hybrid magnet for zero-power magnetic suspension. The middle mass is supported by three coil springs from the base. They work as a positive spring k_1. An electromagnet with a 180-turn coil is installed in parallel with them for adjusting the stiffness and adding damping.

To operate the module in the stand-alone mode, a weight support mechanism is incorporated in the module: the isolation table is supported by three coil springs from the base. In retrofitting, the weight support mechanism can be replaced by an air spring used as passive suspension in principle as shown in Fig.4. In the following experiment, however, the weight support mechanism will be used in parallel with the passive suspension mainly because of the difficulty in adjustment.

The relative displacement of the isolation table to the middle mass is detected by an eddy-current displacement sensor that is fixed to the middle mass. The relative displacement of the middle mass to the base is also detected with another eddy-current displacement sensor that is fixed on the middle mass.

5. EXPERIMENT

5.1 Performance of the Vibration Isolation Module

Zero–power control was achieved with a controller combining the PD control of displacement and the integral control of current. The weight-support springs and the auxiliary electromagnet were adjusted to satisfy Eq.(2). Figure 8 presents the measurement results when weights are put on the isolation table to produce static direct disturbance. The displacements of the isolation table and the middle mass are plotted to *downward* force produced by weights. When the force is in a range of 0 to $17\,\mathrm{N}$, the displacement of the isolation table is quite small so that high stiffness is achieved.

When the applied force is over about $14\,\mathrm{N}$, the negative stiffness of the zero-power magnetic suspension becomes lower because the gap between the isolation table and the electromagnet increases. As a result, the isolation table moves upwards.

Such nonlinear characteristic of the zero-power control system can be compensated by feeding back the square of the gap as described by Eq.(21). Figure 9 shows the load-displacement characteristics when the nonlinear compensation was applied. It is found from Fig.7 that the load-displacement relationship of the zero-power suspension

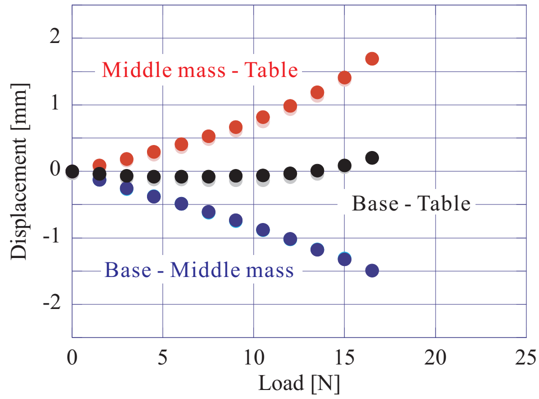


Fig.8 Load-displacement characteristics without nonlinearity compensation

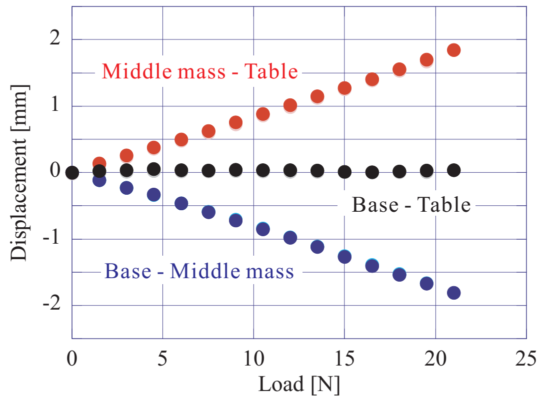


Fig.9 Load-displacement characteristics with nonlinearity compensation

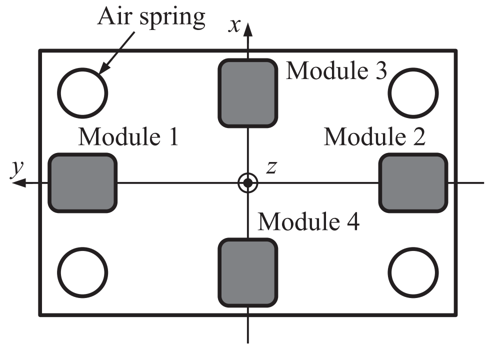


Fig.10 arrangement of modules (top view)

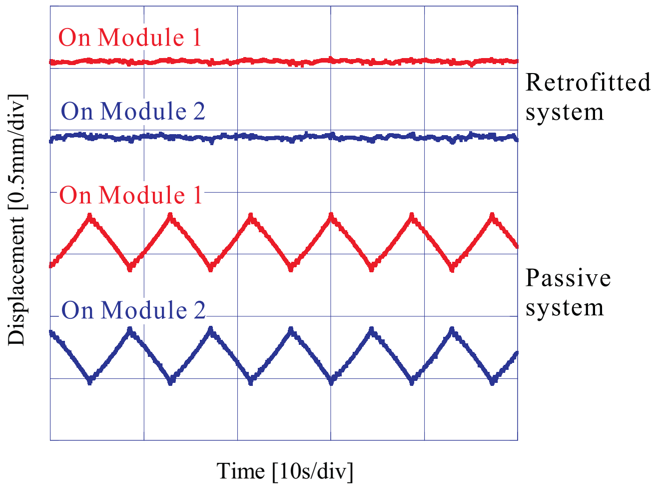


Fig.11 Response to a moving mass

becomes linear and as a result the displacement of the isolation table is kept to be small over 14 N .

5.2 Retrofitting a Passive Vibration Isolation System

Figure 10 shows a schematic diagram of a retrofitted vibration isolation system with four modules. It makes the system robust to direct disturbance in the vertical direction. To generate direct disturbance, a linear stage was fixed on the isolation table. The stage was programmed to move a mass of 1.0 kg to and fro on the table.

Figure 11 shows the displacements of the isolation table with and without the operation of the modules with the nonlinear compensation. It is found that the displacements of the table were significantly reduced by the operation of the modules. This result clearly demonstrates that the retrofitted system with the modules achieves higher stiffness to direct disturbance.

6. CONCLUSIONS

A vibration isolation modules with zero-power magnetic suspension was developed to retrofit a passive vibration isolation system. In the module, the zero-power magnetic suspension mechanism was serially connected with a normal (positive) spring through a middle mass. This structure can generate zero compliance to direct disturbance while the transmitted vibration is still reduced. The characteristics of the fabricated module were studied experimentally. A passive vibration isolation system using air springs was retrofitted with four modules. The performance to direct disturbance was improved significantly.

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