POSITIONING AND MICROVIBRATION CONTROL
BY ELECTROMAGNETS OF AN AIR SPRING VIBRATION ISOLATION SYSTEM

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SUMMARY

Active positioning and microvibration control has been attempted by electromagnets equipped in a bellows-type, air-spring vibration isolation system. Performance tests have been carried out to study the effects. The main components of the system's isolation table were four electromagnetic actuators and controllers. The vibration isolation table was also equipped with six acceleration sensors for detecting microvibration of the table. The electromagnetic actuators were equipped with bellows-type air springs for passive support of the weight of the item placed on the table, with electromagnets for active positioning, as well as for microvibration control, and relative displacement sensors. The controller constituted a relative feedback system for positioning control and an absolute feedback system for vibration isolation control. In the performance test, a 1,490 kg load (net weight of 1,820 kg) was placed on the vibration isolation table, and both the positioning and microvibration control were carried out electromagnetically. Test results revealed that the vibration transmission was reduced by 95 %.

INTRODUCTION

Semiconductor manufacturing is making quantum leaps in recent years, consequent to which higher precision is sought in semiconductor manufacturing systems and detection devices. In line with this, measures against vibration occurring in such systems and devices are becoming factors of utmost importance.

Conventional measures against such vibration had mainly consisted of passive type vibration isolation systems featuring air and coil springs. However, such passive systems are inappropriate for $mgal$ order vibration levels and low frequency regions. Moreover, swift vibration isolation control is necessary for not only vibration that is transmitted from the floor on which the system is installed, but also for vibration that is generated by the system itself, including vibration that is caused by disturbance in the periphery of the system, such as the vibration from an air conditioner. To satisfy such conditions, studies have been in recent years actively carried out on pneumatic actuators[1], linear motors, piezoelectric actuators, and electromagnetic actuators[2][3], as part of the research and development of active-control vibration isolation systems. A paper by the authors on the development of a vibration isolation system, featuring electromagnetic actuators which use both air springs and electromagnets, had been submitted previously[2]. This paper discusses microvibration isolation achieved by PID control of the air spring pressure, positioning control in the vertical direction, and control of the electromagnetic force by absolute velocity feedback. Such control made it possible to reduce the vibration transmissibility, under a load of 1,250 kg, for both horizontal and vertical directions, by 90%. However, as no positioning control was being done in the horizontal direction, and due to influences by the center-of-gravity position when the load was increased, by the property of rubber used for the air spring, and by the unbalanced force generated by the bias electric current, the precise position (attitude) of the table in relation to the floor could not be determined. In the worst case, no control was able to be made due to contact, as the clearance between the electromagnet poles and the yoke was small. Table positioning constituted an important factor when attempting control by electromagnets.

The following discusses active positioning and microvibration control by electromagnets equipped in a bellows-type, air-spring vibration isolation system. The system comprised an isolation table, electromagnetic actuators and controllers. The electromagnetic actuators were equipped with bellows type air springs for passive support of the weight of the item placed on the table, with electromagnets for active positioning, as well as for microvibration control, and displacement sensors for detecting the relative displacement between the table and the floor. The controller constituted a relative feedback system for positioning control and an absolute feedback system for vibration isolation control. Detected relative displacement and absolute acceleration data was applied for feedback control of electromagnets. A performance test was carried out, in which a 1,490 kg load (net weight of 1,820 kg) was placed on the vibration isolation table, and both the positioning and microvibration control were carried out electromagnetically. Test results revealed that the vibration transmission was reduced by 95 %.

Definition of symbols

G	Center of gravity
F_x , F_y , F_z	Actuator controlling force
F_{xi} , F_{yi} , F_{zi} (i=1 $\sim$ 4)	Controlling force of each actuator
M_α , M_β , M_γ	Controlling force moment
k_{xi} , k_{yi} , k_{zi} (i=1 $\sim$ 4)	Constant of each air spring
c_{xi} , c_{yi} , c_{zi} (i=1 $\sim$ 4)	Damping coefficient of each air spring
x , y , z	Isolation table displacement
α , β , γ	Rotational displacement of isolation table
u , v , w	Floor displacement
ξ , η , ζ	Rotational displacement of floor
m	Mass
I_α , I_β , I_γ	Moment of inertia for each axial rotation
W	Disturbance vector
k_e	Constant determined by number of coil windings, magnetic permeability, & cross-sectional area of magnetic pole
i_1 , i_2	Excitation current of both opposing electromagnets
h_1 , h_2	Void of both opposing electromagnets
h_0 , i_0	Void under balanced condition (center of void) & bias current
x_r , i_c	Micro-fluctuating part under balanced condition

TEST APPARATUS

Microvibration Isolation System

Figure 1 shows the configuration of the microvibration isolation system, while Table 1 shows the specifications of the same. The main components shown are the isolation table, on which a device subject to vibration isolation is placed, four electromagnetic actuators and controllers. The table has an overhanging configuration due to its relation with the center-of-gravity of the device which would be placed on it. A total of six acceleration sensors are installed on the table for detecting vibration on the table.

Figure 2 shows the analog control system used for vibration isolation. Displacement sensors detect relative displacement, in the horizontal and vertical directions, between the isolation table, supported by air springs, and the floor. There are two feedback systems. One is a relative feedback system by which relative displacement data is fed back for positioning of the isolation table. The other is an absolute feedback system by which absolute acceleration data, detected on the isolation table, is fed back for vibration isolation control in the horizontal and vertical directions. Electromagnets are used for the control. The absolute acceleration speed is detected by six acceleration sensors installed on
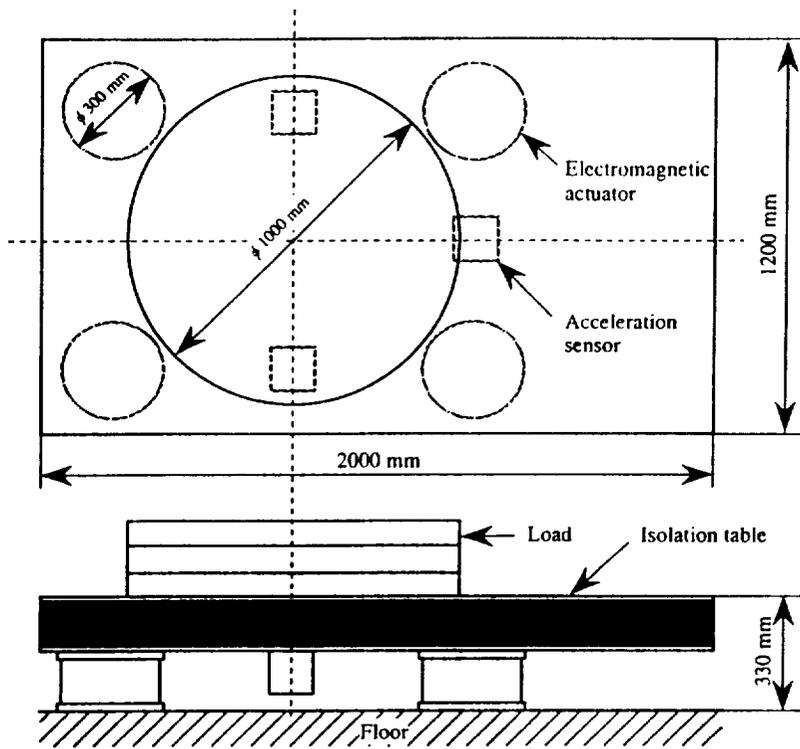
Figure 1. Configuration of microvibration isolation system.
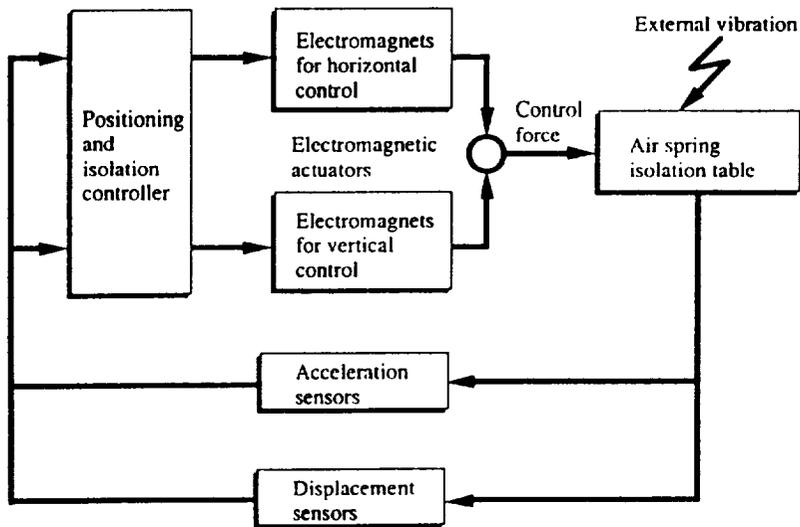
Figure 2. Components of control system.

Table 1. Specification of microvibration isolation system

Mass		Total	Table	Load	Actuator
		1820kg	210kg	1490kg	120kg
Size		Total			Actuator
		900$mm \times$ 1200$mm \times$ 150mm			ϕ 300$mm \times$ 150mm
Electromagnet			Max. magnetic foce		Current
	Hrizontal		177N		3.8A
	Vertical		1209N		10A
Air spring		Standard load capacity		Natural frequency	
	Hrizontal	—		—	
	Vertical	530kg (0.5Mpa)		2.7Hz	

the isolation table, and is coordinate-transformed into acceleration signals for six degrees of freedom.

Electromagnetic Actuators

Figure 3 shows a cross-section view of the electromagnetic actuator. It contains a bellows-type air spring for supporting the item placed on the table; DC electromagnets which generate magnetism in the vertical and horizontal directions, for positioning the isolation table supported by an air spring, also for isolating vibration; displacement sensors for detecting relative displacement between the installation floor and the isolation table. The air spring, compact yet capable of excellent support, was installed in the center of the actuator, around which were installed electromagnets.
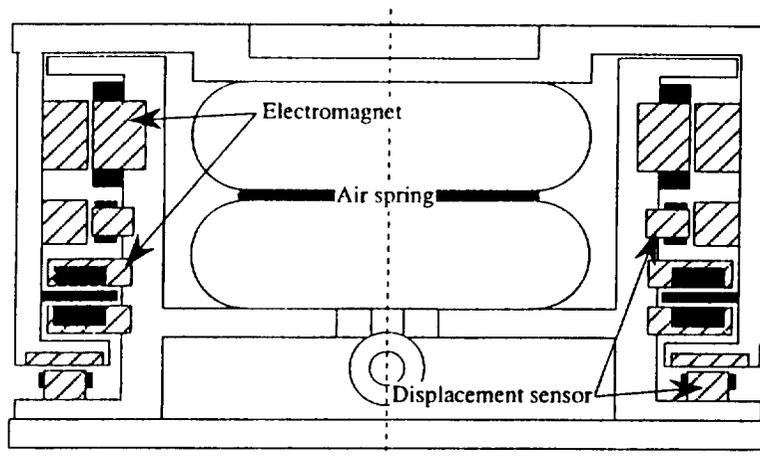
Figure 3. Cross-sectional view of electromagnetic actuator.
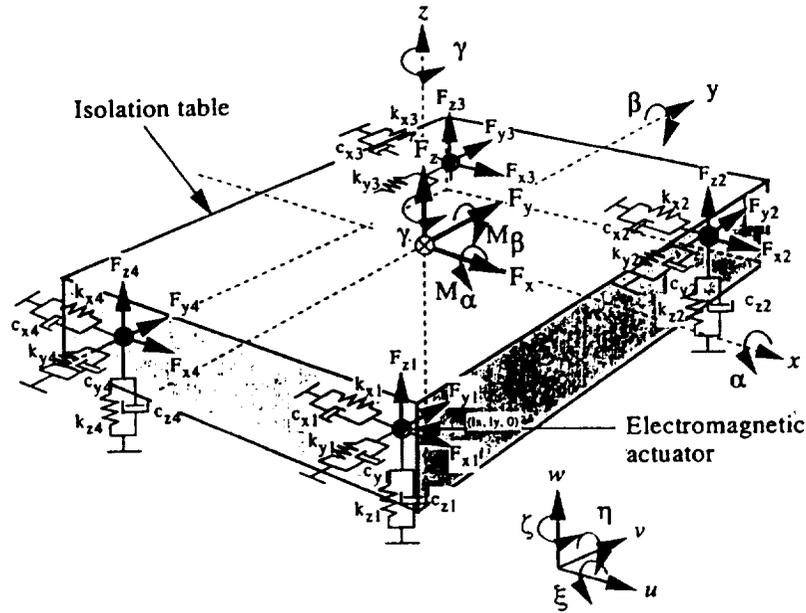
Figure 4. Model of microvibration isolation system.

As for the characteristics of the air spring and electromagnets, detailed studies had already been made by the authors[2].

MODELS

Isolation System Model

Figure 4 shows the isolation system model. A balanced situation is shown, with the isolation table being supported by four electromagnetic actuators. The directions of each electromagnet force matches the coordinate axes, the working points being inside the x and y plane surfaces (plane surfaces including the center of gravity), identical to those of the air springs.

This model was made assuming that an ideal rigid body was completely non-coupled supported, making it possible to design a control system for controlling each degree of freedom independently. Consequently, the model featured the following:

1. Inertial coupling was eliminated by matching the coordinate axes with the inertial main axes of the system.

2. The center of elasticity and elastic main axis of the four air springs were matched with the center of gravity and the coordinate axes.

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From the above, the equation of motion for the isolation system is the following Equation 7, by way of Equations 1 - 6.

$$m\ddot{x} + C_x\dot{x} + K_x x = F_x + W_x \tag{1}$$

$$m\ddot{y} + C_y\dot{y} + K_y y = F_y + W_y \tag{2}$$

$$m\ddot{z} + C_z\dot{z} + K_z z = F_z + W_z \tag{3}$$

$$I_\alpha\ddot{\alpha} + C_\alpha\dot{\alpha} + K_\alpha \alpha = M_\alpha + W_\alpha \tag{4}$$

$$I_\beta\ddot{\beta} + C_\beta\dot{\beta} + K_\beta \beta = M_\beta + W_\beta \tag{5}$$

$$I_\gamma\ddot{\gamma} + C_\gamma\dot{\gamma} + K_\gamma \gamma = M_\gamma + W_\gamma \tag{6}$$

$$M\ddot{X} + C\dot{X} + KX = F + W \tag{7}$$

$$X = [x\ y\ z\ \alpha\ \beta\ \gamma]^T$$

$$M = diag[m_x\ m_y\ m_z\ I_\alpha\ I_\beta\ I_\gamma]$$

$$C = diag[C_x\ C_y\ C_z\ C_\alpha\ C_\beta\ C_\gamma]$$

$$K = diag[K_x\ K_y\ K_z\ K_\alpha\ K_\beta\ K_\gamma]$$

$$F = [F_x\ F_y\ F_z\ M_\alpha\ M_\beta\ M_\gamma]^T$$

$$W = [W_x\ W_y\ W_z\ W_\alpha\ W_\beta\ W_\gamma]^T$$

Model of Electromagnets

The magnetic attraction force of the controlling electromagnets is expressed in Equation 8. If this force is linearized in the proximity of a balanced condition, it becomes as expressed in Equation 9.

$$f = k_e\left\{\left(\frac{i_1}{h_1}\right)^2 - \left(\frac{i_2}{h_2}\right)^2\right\} \tag{8}$$

$$i_1 = i_0 + i_c\ ,\ i_2 = i_0 - i_c$$

$$h_1 = h_0 - x_r\ ,\ h_2 = h_0 + x_r\ ,\ x_r = x - x_0$$

$$i_0 \gg i_c\ ,\ h_0 \gg x_r$$

$$f = k_e\left\{4\frac{i_0^2}{h_0^3}X_r - 4\frac{i_0}{h_0^2}i_c\right\}$$

$$= k_u x_r + k_c i_c \tag{9}$$

Thus, the controlling force, vector F, becomes

$$F = K_u(X - X_0) + K_c I_c \tag{10}$$

$$K_u = diag[K_{ux}\ K_{uy}\ K_{uz}\ K_{u\alpha}\ K_{u\beta}\ K_{u\gamma}]$$

$$K_c = diag[K_{cx}\ K_{cy}\ K_{cz}\ K_{c\alpha}\ K_{c\beta}\ K_{c\gamma}]$$

$$I_c = diag[I_x\ I_y\ I_z\ I_\alpha\ I_\beta\ I_\gamma]^T$$

The controlling electric current is distributed to each of the 12 electromagnets. If the control matrix is set as $B(6 \times 12)$

$$I_c = B \, I_{ca} \tag{11}$$

$$I_{ca} = diag \left[i_{z1} \; i_{y1} \; i_{z1} \; i_{z2} \; i_{y2} \; i_{z2} \; i_{z3} \; i_{y3} \; i_{z3} \; i_{z4} \; i_{y4} \; i_{z4} \right]^T$$

DESIGNING OF THE CONTROL SYSTEM

The control system was designed so as to achieve both the positioning control and vibration-isolation control of the vibration isolation system. The control system for increasing the vibration control performance, versus direct disturbance, featured the same design as that for the vibration isolation performance.

As for positioning control, the magnetic poles were positioned in the center of void of the electromagnets, and attitude control was carried out for the isolation table. This control system was constituted as a relative feedback system, by which relative displacement data, of detected displacement between the installation floor and the isolation table, were fed back by a PID controller. The vibration isolation control system was constituted as an absolute feedback system, by which absolute speed on the isolation table was fed back. The controllers are thus

$$G_r(s) = [g_{rij}(s)] \; , \; g_r = - \left(k_P + \frac{k_I}{s} + k_D(s) \right) \tag{12}$$

$$G_a(s) = [g_{aij}(s)] \; , \; g_a = -k_a(s) \tag{13}$$
$$(i = 1 \sim 6 \; , \; j = 1 \sim 6)$$

Regarding Equation 7 as a relative system, let us consider a servo system controlled by controller $G_r(s)$ and which follows up command value X_{ref}. The servo performance becomes better along greater $G_r(s)$, as the transfer function of the closed loop is

$$\frac{X_r}{X_{ref}} = \frac{K_c \, G_r(s)}{Ms^2 + Cs + K - (K_u + K_c \, G_r(s))} \tag{14}$$

Considering an absolute feedback system which includes a controller $G_r(s)$ for positioning control, regarding disturbance vector W in Equation 7 as the disturbance caused by floor vibration X_0, we get

$$M \ddot{X} + C \dot{X} + K X = F + C \dot{X_0} + K X_0 \tag{15}$$
$$X_0 = [u \; v \; w \; \xi \; \eta \; \zeta]^T$$

Accordingly, the equation of movement, when Laplace transformed, becomes as Equation 16, and the vibration transmissibility becomes as Equation 17. The G_r(s) and G_a(s) were designed so that the desired vibration isolation performance could be obtained, as much as possible from the low frequency side. This involved making the controller operate at extremely low frequencies, and eliminating differentiating elements ($k_D = 0$).

$$
\begin{aligned}
Ms^2 X(s) + Cs X(s) + K X(s) &= Cs X_0(s) + K X_0(s) + F(s) \\
&= Cs X_0(s) + K X_0(s) + [K_u (X(s) - X_0(s)) \\
&\quad + K_c \{G_r(s)(X(s) - X_0(s)) + G_a(s) X(s)\}] \qquad (16)
\end{aligned}
$$

$$
\frac{X(s)}{X_0(s)} = \frac{Cs + K - (K_u + K_c G_r(s))}{Ms^2 + Cs + K - \{K_u + K_c (G_r(s) + G_a(s))\}} \qquad (17)
$$

It is acknowledged here that the $G_r(s)$ and $G_a(s)$ constitute a trade-off relationship, for satisfying both the positioning and the vibration isolation performance simultaneously.

PERFORMANCE TESTS

Figure 5 (a), (b), and (c) show displacement data of the isolation table for directions X, Y, and Z, respectively. Each case represents a start with no positioning control and a duration where positioning control was applied. The isolation table was able to be positioned within 50 μm of the void center of the electromagnets by this positioning control, justifying the validity of the attitude control.

Figures 6 and 7 show the isolation performance and vibration transmissibility, respectively, of cases where control was being made. It is acknowledged from these figures that vibration of the isolation table (for directions X, Y, and Z) was able to be reduced to 1/20 of that on the installation floor, and that a performance almost equal to that indicated in the simulation was attained.

Figures 8 (a) and (b) show the vibration control performance versus impulse disturbance. The case of 8 (a), where passive vibration isolation was implemented, indicated a slow transient damping characteristics. In contrast, the case of 8 (b), where active vibration isolation was implemented, indicated a convergence within a short time.

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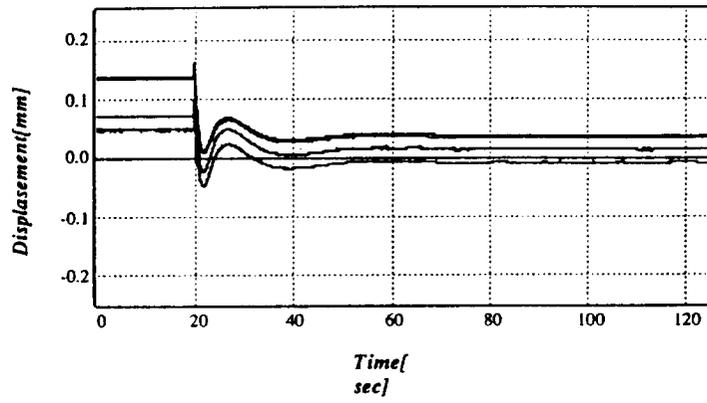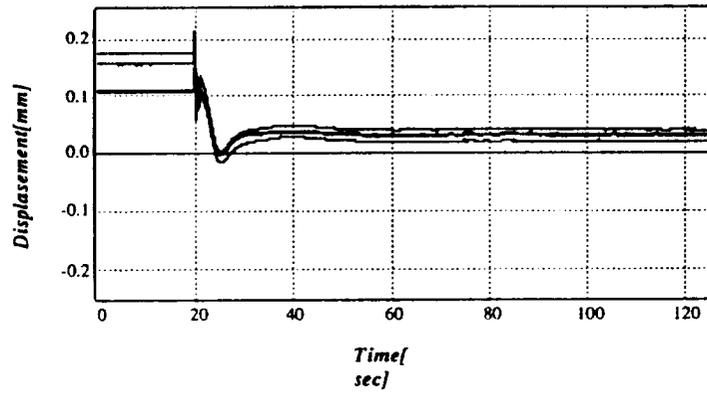
(a). X direction.

(b). Y direction.
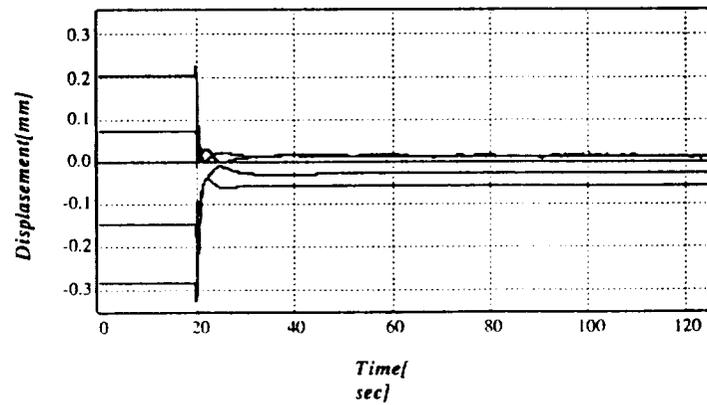
(c). Z direction.

Figure 5. Displacement of isolation table.
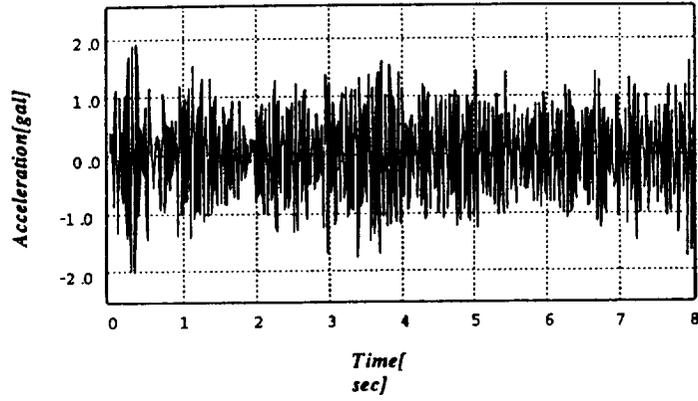
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(a). Acceleration of floor.
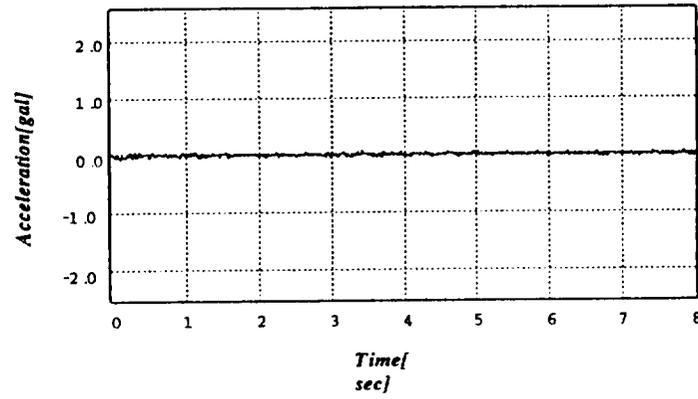
(b). Acceleration of isolation table (Simulation).
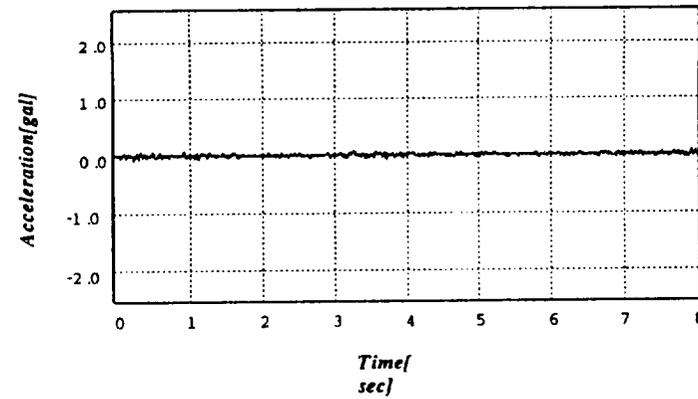
(c). Acceleration of isolation table.

Figure 6-1. Isolation performance of microvibration isolation system in horizontal(X) direction.
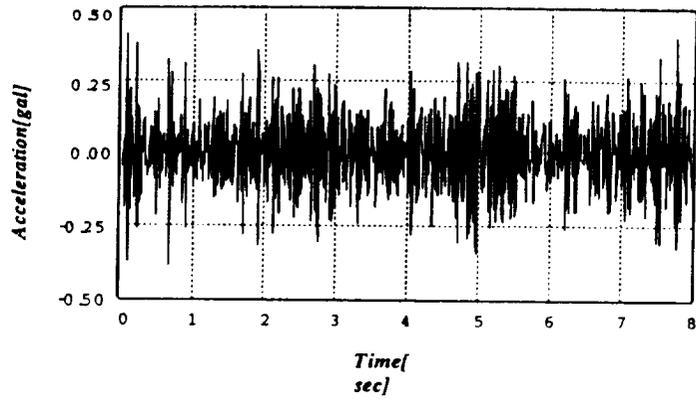
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(a). Acceleration of floor.
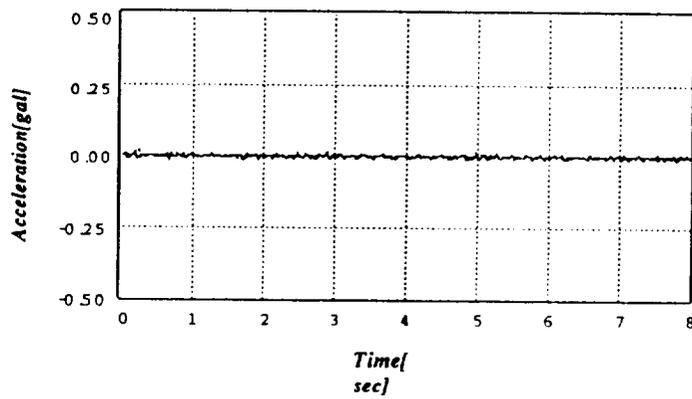
(b). Acceleration of isolation table (Simulation).
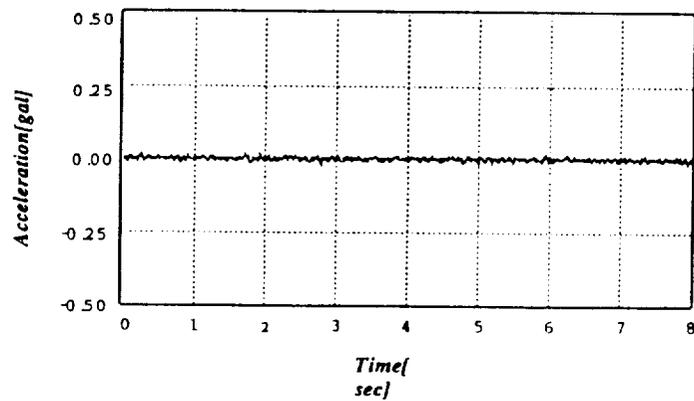
(c). Acceleration of isolation table.

Figure 6-2. Isolation performance of microvibration isolation system in vertical(Z) direction.

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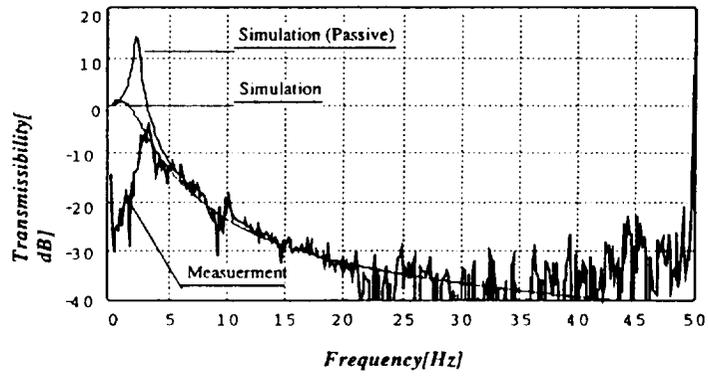
(a). Horizontal(X) direction.
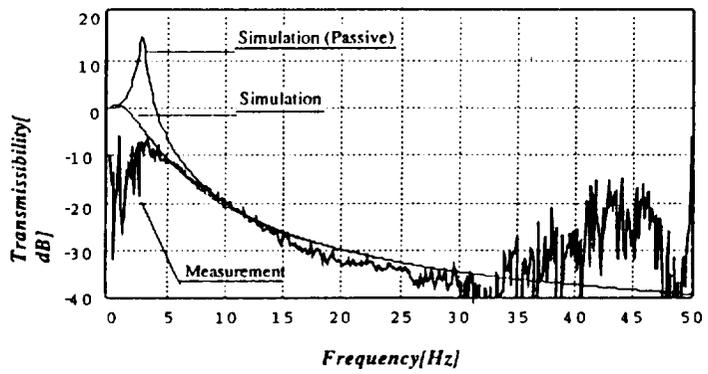
(b). Vertical(Z) direction.

Figure 7. Transmissibility of microvibration isolation system.

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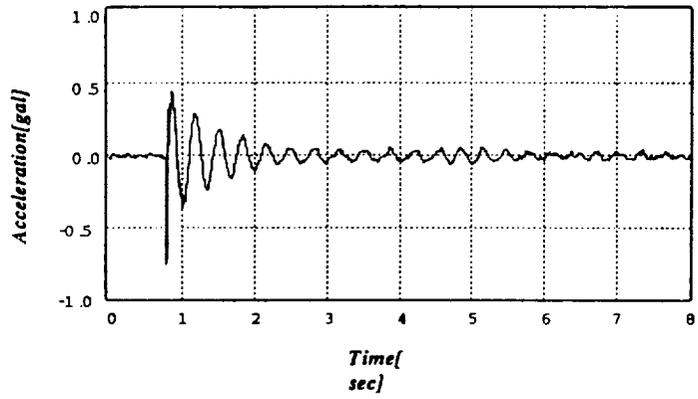
(a). Passive.
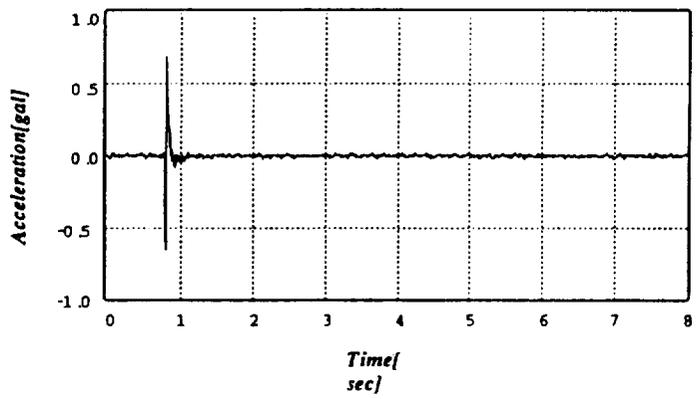
(b). Active.

Figure 8. Impulse response on the table.

CONCLUSION

Active positioning and microvibration control has been attempted by electromagnets equipped in a bellows-type, air-spring vibration isolation system. Performance tests have been carried out to access the control method and study the vibration isolation performance. The following are results obtained from this research.

1. An isolation table, passively supported by bellows-type, air-springs, was controlled by electromagnets, the result of which was an achievement of both stable positioning control and microvibration isolation.

2. Vibration transmissibility, in the horizontal and vertical directions, was able to be reduced to below 1/20 in performance tests. Vibration control was also possible for external disturbances that had direct effect.

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