INTRODUCTION
Nano precision instruments with nanometer resolution are susceptible to disturbance transmitted mostly from ground vibration and mechanical vibration due to movements of the measuring instrument itself. These kinds of vibrations which are submicrometer level and low frequency distract both precision and accuracy of precision instruments.
Six degree of freedom active vibration isolation system is required for these kinds of vibration. In this system, control algorithms are very important factors of performance improvement. The system of this sort has coupling and system uncertainties due to changing of environment.
The objective of this work is to design controllers which are PI controller and H-infinity controller for six degree of freedom active vibration isolation system and compare two controllers through experiments.

VIBRATION

FIGURE 1. Ground vibration levels.

Ground vibration levels illustrate in Figure 1. Sub-micrometer level ground vibrations exert a bad effect to precision instruments. Figure 2 describes vibration criterion defined by ISO.

OVERVIEW OF THE ACTIVE VIBRATION ISOLATION SYSTEM
In this paper, six degree of freedom active vibration isolation system composed by passive isolation to attenuate high frequency vibration and active isolation to attenuate low frequency vibration.

FIGURE 2. Generic vibration criterion (VC) curves for vibration-sensitive equipment.

Generally precision measuring and actuating system as AFM (Atomic force microscopy) and lithography instruments follow VC-E criterion. Recently, more tight criterions are required. Ground vibration isolation is very important issue to satisfy criterion.
FIGURE 3 describes six degree of freedom active vibration isolation system used for experiment. This system uses four rubber mounts for supporting payload and passive isolation, six VCM (Voice Coil Motor) actuators for active isolation by actuating horizontal and vertical directions, and six accelerometers for sensing and feedback six directional acceleration signals of upper plate.

PASSIVE ISOLATION SYSTEM MODELING
Passive mount assumed to have stiffness and damping properties typically represented by spring in parallel to a damper.

FIGURE 4 shows conceptual diagram of passive isolation system both XY and XZ plane.

\[
\begin{bmatrix} M \end{bmatrix} \dddot{x} + \begin{bmatrix} C \end{bmatrix} \dot{x} + \begin{bmatrix} K \end{bmatrix} x = \begin{bmatrix} F_s \end{bmatrix} + \begin{bmatrix} F_d \end{bmatrix} + \begin{bmatrix} F_u \end{bmatrix}
\]

\[
\begin{bmatrix} M \end{bmatrix} \dddot{x} + \begin{bmatrix} C \end{bmatrix} \dot{x} + \begin{bmatrix} K \end{bmatrix} x = \begin{bmatrix} F_s \end{bmatrix} + \begin{bmatrix} F_d \end{bmatrix} + \begin{bmatrix} F_u \end{bmatrix} + \begin{bmatrix} F_0 \end{bmatrix}
\]

\[
\begin{bmatrix} M \end{bmatrix} \dddot{x} + \begin{bmatrix} C \end{bmatrix} \dot{x} = \begin{bmatrix} F_s \end{bmatrix} + \begin{bmatrix} F_d \end{bmatrix} + \begin{bmatrix} F_u \end{bmatrix} + \begin{bmatrix} F_0 \end{bmatrix}
\]

FIGURE 4. Conceptual diagram of passive isolation system.

Passive mount assumed to have stiffness and damping properties typically represented by spring in parallel to a damper.

To effectively model the system consisting of viscoelastic material, some assumptions were made:

- The principal axes of inertia of the payload are, respectively, parallel to the principal elastic axes of the mounts, so product moment of inertia \( I_{xx} = I_{yy} = I_{zz} = 0 \).

- Consider the upper plate and the payload as one rigid body.

- The forces \( (F_x, F_y, \text{ and } F_z) \) and moments \( (M_{y}, M_{z}, \text{ and } M_{x}) \) are applied directly to the upper plate.

- Ignore rotational stiffness and damping, because the motions of payload and lower plate make compressions and tensions much more than angular motion for passive mounts.

This system can be modeled as following equation.

\[
\begin{bmatrix} M \end{bmatrix} \dddot{x} + \begin{bmatrix} C \end{bmatrix} \dot{x} + \begin{bmatrix} K \end{bmatrix} x = \begin{bmatrix} F_s \end{bmatrix} + \begin{bmatrix} F_d \end{bmatrix} + \begin{bmatrix} F_u \end{bmatrix} (1)
\]

M, C and K are mass, damping and stiffness matrices, \( F_u \) is input force, and \( F_u \) is direct disturbance force respectively. P is six axis directional motion of the payload, and B is six axis directional motion of the ground vibration.

DESIGN OF CONTROLLERS
The purpose of active control is to effectively decouple the isolator frequency from the static sag requirement so that it can improve the low frequency isolation.

Two kinds of control method, PI control and H-infinity control, are proposed and designed to experiment on the six degree of freedom active vibration isolation system and compare performance of two controllers.

FIGURE 5 shows block diagram of the modal decoupled closed loop acceleration feedback PI control. It has modal matrix for modal decoupling and kinematics of actuator and sensor which are obtained by their geometrical position. PI controller must be designed six decoupled mode and conform to required transmissibility for the system performance.

FIGURE 6. Block diagram of six modal PI controller

Relationship of each mode and PI controller can be simply represented as FIGURE 6. Using this relationship, equation (2) is derived and PI controller is can be designed using this equation.

\[
(M_m + K_p) s^3 + (C_m + K_i) s^2 + K_m s = 0
\]
The controller gains $K_p$ and $K_i$ can be tuned using pole placement technique considering the closed loop system transmissibility.

**FIGURE 7. H-infinity feedback control loop.**

FIGURE 7 shows block diagram of closed loop acceleration feedback H-infinity control. $\Delta$, $K_{oo}$ and $W$ are system uncertainty, H-infinity controller and weighting function respectively. Purpose of this control is to minimize the effect of $z$ by the external input $d$. It can be represented as equation (3).

$$
\begin{bmatrix}
W_1S \\
W_2K_\infty S
\end{bmatrix} < \gamma
$$

(3)

When $S$ is $(I + GK_{oo})^{-1}$

Two weighting functions must be defined properly. $W_1$ is guideline of system sensitivity function which is determined by required transmissibility. $W_2$ prevent saturation of actuator and is defined to ensure rank condition solving matrix equation of controller. Then minimum $\gamma$ and controller $K_{oo}$ which are satisfied equation (3) can be got. H-infinity controller is obtained by using robust control toolbox of the MATLAB.

**EXPERIMENTS**

To verify the performance of two controllers, experimental system setup as illustrated in FIGURE 8 and two controllers are applied to the system.

**FIGURE 8. Experimental setup.**

To compare performance of ground vibration isolation and robustness of two controllers, experiment carried out with two conditions which are condition without system uncertainties and condition with system uncertainties. Time response and transmissibility are checked on each condition.

**FIGURE 9. Time response of PI and H-infinity controls against the ground vibration without system uncertainties.**

**FIGURE 10. Time response of PI and H-infinity controls against the ground vibration with system uncertainties.**

As the results displayed in FIGURE 9 and FIGURE 10 show, PI and H-infinity controllers attenuate the ground vibration effectively in condition without the system uncertainties. But, time response of the PI controller get worse on condition with the system uncertainties. On the other hand H-infinity controller shows same attenuation performance.

These kinds of results appear on the transmissibility from the ground vibration to the
payload as illustrated in FIGURE 11 and FIGURE 12.

(a) Horizontal direction transmissibility

(b) Vertical direction transmissibility

FIGURE 11. Comparison of transmissibility from the ground vibration to the payload between PI and H-infinity control without the system uncertainties. (solid – passive, dot – PI control, dash – H-infinity control)

(a) Horizontal direction transmissibility

(b) Vertical direction transmissibility

FIGURE 12. Comparison of transmissibility from the ground vibration to the payload between PI and H-infinity control without the system uncertainties. (solid – passive, dot – PI control, dash – H-infinity control)

These situations come from inaccuracy of modal decoupling of the PI control by the system uncertainties. The system uncertainties affect to the system parameter values of inertia, damping and stiffness. On the other hand, H-infinity control based on robust control shows better performance with the system uncertainties.

CONCLUSION

In this work, modeling of the six degree of freedom active vibration isolation system was carried out. For system control, PI controller which is easy to design and H-infinity controller which is robust to uncertainties and disturbances are designed and proposed to control active isolator and applied to six degree of freedom active vibration isolation system. And then, performance of the PI controller is compared with the H-infinity controller.

In conclusion, the H-infinity controller is more compatible with six degree of freedom active vibration isolation system which has coupled axis and the various system uncertainties.

REFERENCES


