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Modeling and Control of a Four Mount Active Micro-vibration Isolation System

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ABSTRACT

Micro vibration isolation, typically originated from ground, is always a prime concern for the nano-measurement instruments such as Atomic Force Microscopes. A four mount active vibration isolation system is proposed in this paper. Modeling and control of such a four mount system was analyzed. Combined active-passive isolation principle is used for vibration isolation by mounting the instrument on a passively damped isolation system made of Elastomer along with the active stage in parallel that consists of very soft actuation system, the Voice Coil Motor. The active stage works in combination with the passive stage for working as a very low frequency vibration attenuator.

Key Words : Seismic vibration, Elastomer, Hybrid active passive vibration isolation

1. INTRODUCTION

Ground-borne (seismic) vibration has always been considered to be prime source of disturbance in nanotechnology, especially when resolution of measurement instrument is in nanometer scale seismic vibration with amplitude around submicron region and with very low frequency around 1.0 Hz, is of great concern. Many researches have so far been carried out in the field of isolation of instruments from such micro vibration in order to make the measurement result more trustworthy. Main principle is to provide stiffness and damping in the path of vibration from ground to instrument such that the isolator works like a low-pass filter which attenuates relatively high frequency disturbances.

Among previous researches there are Minus K technology which uses passive isolation such as negative stiffness mechanism to isolate vibration. ELITE 3 system implements series active passive approach to isolate both vertical and ground vibration. This system is modular and three such modules need to be used to isolate all six rigid body modes. Halcyonics benchtop six dof isolator imple-ments spring mounts as passive isolators and four vertical and four horizontal electrodynamic actuators to control the six rigid body modes dynamic vibration in the upper rigid plate.

This paper proposes a hybrid active-passive vibration isolator to eliminate such micro vibration transmitted to the instrument in all six degrees of freedom. The reason behind this hybrid solution is to take advantage of both the passive as well as active stage. Both the natural frequency as well as high frequency response is first improved through the dynamics of the passive system which provides high attenuation rate and decouples the active stage from static sag. The active stage then works on the system to reduce natural frequency so that low frequency attenuation is achievable as well as the resonance amplitude is controllable.

2. MODELING OF THE FOUR MOUNT PASSIVE SYSTEM

Passive system is realized by four elastomers supporting an upper rigid plate. This plate can support a static load up to 150 Kg having six rigid body modes. Active control system can be implemented

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consisting of seismic accelerometers mounted on the upper rigid plate along with six voice coil motors (three vertical and three horizontal) designed and mounted such that they can control all six rigid body modes in low frequency region from 0.1 to 100 Hz [1], [3]. Sensors are mounted such that they can pick up any disturbance accelerations in the upper plate transmitted through the passive system and then can provide feedback signal to the actuators. Complete system thus looks as shown in Fig. 1:

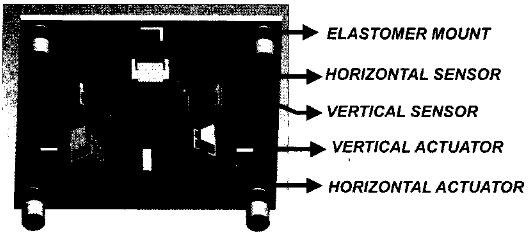

Fig. 1. Complete system conceptual design.

Elastomer is assumed to have stiffness and damping properties typically represented by spring in parallel to a damper such that each spring representing a value of principal stiffness is paralleled by idealized viscous damper, each damper representing principal damping [4]. Elastomer mounts can be modeled as follows using principal axes (X, Y, Z) stiffness and damping. Fig. 2 shows two elastomers mounts connected to upper plate.

3. ACTIVE CONTROL OF VIBRATION

For active control implementation, the coupled system dynamics was first decoupled using state space

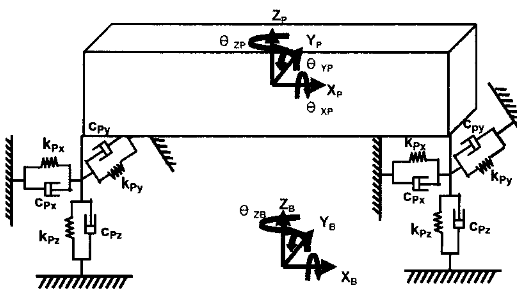

Fig. 2. Passive system basic model.

approach. System model with two inputs (the control signal input, $F_u(s)$ and vibration input, b(s)) contributing to the output variable, p(s) of the plant, is as follows

$$
\begin{aligned}
&([M]s^2+[C]s+[K])p(s)=[C]sb(s)+[K]b(s)+F_u(s)\\
&\Rightarrow p(s)=\frac{1}{([M]s^2+[C]s+[K])}F_u(s)+\frac{([C]s+[K])}{([M]s^2+[C]s+[K])}b(s)
\end{aligned}
\tag{1}
$$

$$
\Rightarrow p(s)=\begin{bmatrix} G_{c11}(s) & 0 & 0 & 0 & 0 & G_{c16}(s) \\ 0 & G_{c22}(s) & 0 & 0 & G_{c25}(s) & 0 \\ 0 & 0 & G_{c33}(s) & 0 & 0 & 0 \\ 0 & 0 & 0 & G_{c44}(s) & 0 & 0 \\ 0 & G_{c52}(s) & 0 & 0 & G_{c55}(s) & 0 \\ G_{c61}(s) & 0 & 0 & 0 & 0 & G_{c66}(s) \end{bmatrix} F_u(s)+\begin{bmatrix} G_{t11}(s) & 0 & 0 & 0 & 0 & G_{t16}(s) \\ 0 & G_{t22}(s) & 0 & 0 & G_{t25}(s) & 0 \\ 0 & 0 & G_{t33}(s) & 0 & 0 & 0 \\ 0 & 0 & 0 & G_{t44}(s) & 0 & 0 \\ 0 & G_{t52}(s) & 0 & 0 & G_{t55}(s) & 0 \\ G_{t61}(s) & 0 & 0 & 0 & 0 & G_{t66}(s) \end{bmatrix} b(s)
$$

where M, C and K are mass, damping and stiffness matrix respectively.

Independent transfer function for both Gc(s) and Gt(s) were obtained which can be used to design control system for each rigid body modes. $Gc_{ii}(s)$ are individual components of transfer functions of control input $F_u(s)$ to the output p(s) and $Gt_{jj}(s)$ are the individual components of transmissibility components of disturbance input b(s) to output p(s).

Modal decoupling:

Decoupled controller can be designed using SISO (Single Input Single Output) model by decoupling the model using modal decomposition technique and then using the feedback compensator for individual modes to control [2]. For modal decomposition, the damping matrix was considered to be diagonal as well.

Considering scaled normalized eigenvector [φ] and multiplying the equation of motion (1) we get

$$
\begin{aligned}
&[\varphi]^T[M][\varphi]\{\ddot{p}\}+[\varphi]^T[C][\varphi]\{\dot{p}\}+[\varphi]^T[K][\varphi]\{p\}\\
&\quad=[\varphi]^T[C][\varphi]\{\dot{b}\}+[\varphi]^T[K][\varphi]\{b\}+[\varphi]^T[F_u]
\end{aligned}
\tag{2}
$$

Now considering scaled modal mass, stiffness and damping matrices:

$[\varphi]^T[M][\varphi]=[M^*]$, $[\varphi]^T[C][\varphi]=[C^*]$
and $[\varphi]^T[K][\varphi]=[K^*]$

Fig. 3 shows modal decoupled open loop transmissibility for six rigid body modes:

Control system can be implemented now to control only these six rigid body modes using six individual SISO(Single Input Single Output) feedback control

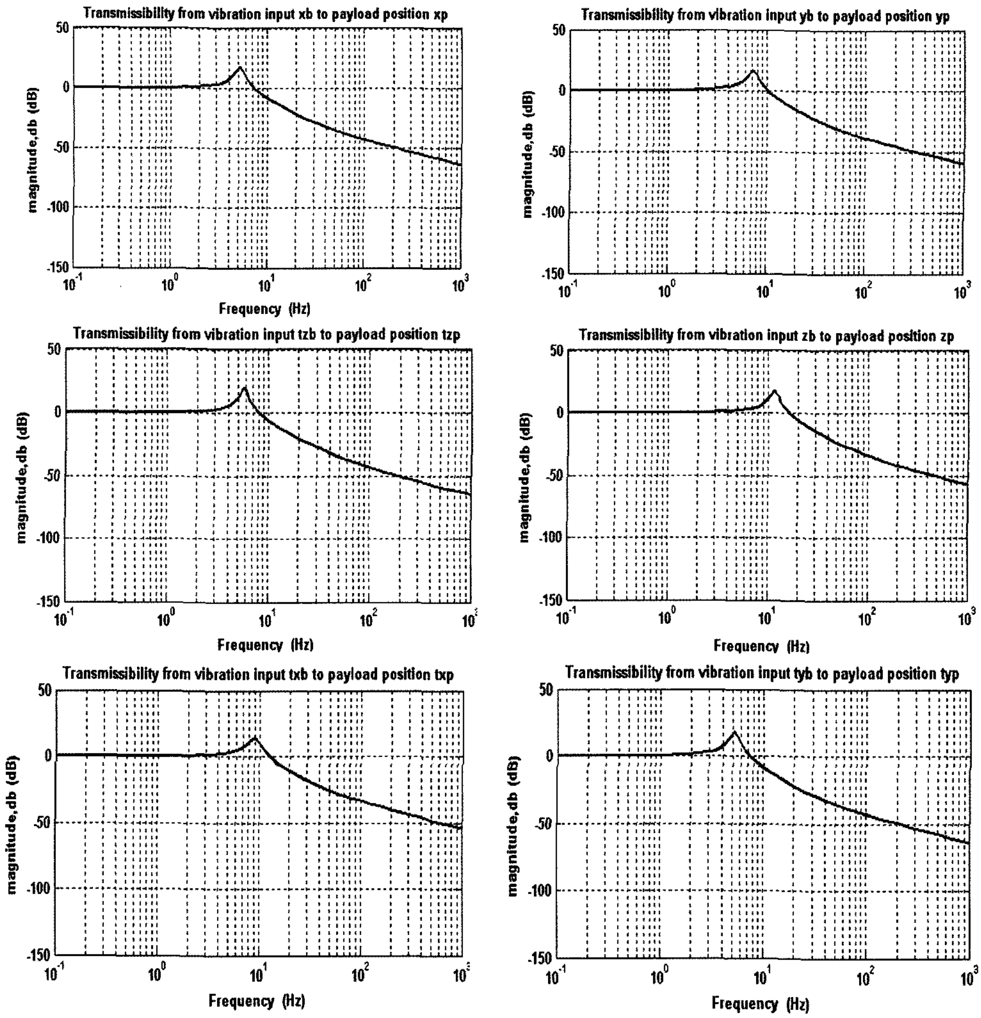

Fig. 3. Open loop transmissibility of modal decoupled modes.

system. Block diagram of the typical feedback control system can be as follows:

The compensator can be designed depending on typical feedback control algorithm such as position feedback PD(Proportional and Derivative) control. The compensator then becomes:

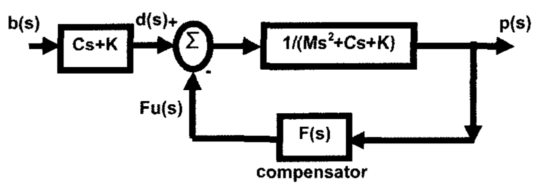

Fig. 4. Control system block diagram.

$$F = K_P + K_D s$$

$$\text{So } Fu(s) = F(s)p(s) = (K_P + K_D s)p(s) \tag{3}$$

Absolute position can be obtained by integrating the absolute acceleration of the payload measured by the accelerometers mounted on the upper plate. The closed loop transmissibility considering position feedback PD control becomes:

$$\frac{p(s)}{b(s)} = \frac{Cs + K}{Ms^2 + (C + K_D)s + (K + K_P)} \tag{4}$$

As can be seen from the transmissibility, feedback gains effectively modify the damping and stiffness of the closed loop system. Now the closed loop isolation

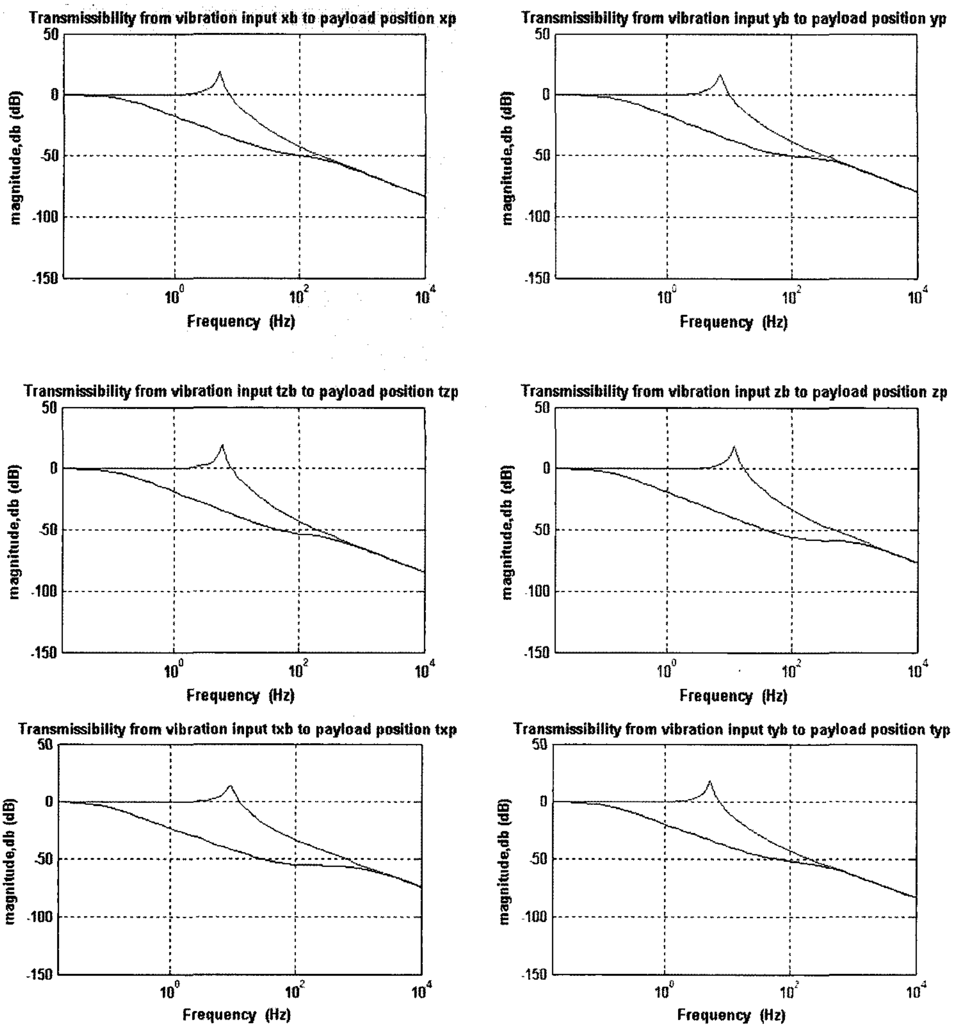

Fig. 5. Comparison of open (red) and closed (blue) loop transmissibility.

requirement for this active system is as follows:

- Closed loop resonance frequency : >= 0.1 Hz
- Closed loop damping at resonance : >= 1
- Stability

The PD gains (K_P and K_D) can be tuned using pole placement technique such that the closed loop system satisfies the above mentioned requirements. After tuning the PID feedback control, stable closed loop transmissibility of six diagonal rigid body modes are:

As can be seen from figure 5 that active tuning effectively modifies the low frequency response by starting isolation from low corner frequency as low as around 0.1 Hz and also improves damping at resonance without reducing the attenuation rate achieved by the passive elastomers. This system thus can isolate all six modal disturbance components starting from 0.1 Hz with sufficient damping and also can isolate high frequency disturbances using the passive system transmissibility.

4. CONCLUSION

Complete model of four elastomer mount six degree of freedom hybrid active passive vibration isolator was proposed. The control system that needs to be implemented to effectively isolate all six rigid body modes of vibration was also proposed. This isolator is well applicable for active isolation of very low frequency ground vibration with typical amplitude in sub micrometer range and having frequency range around 0.1-100 Hz and passive isolation afterwards.

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