

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.

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. 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 150Kg having six rigid body modes. Active control system can be implemented 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 figure 1:

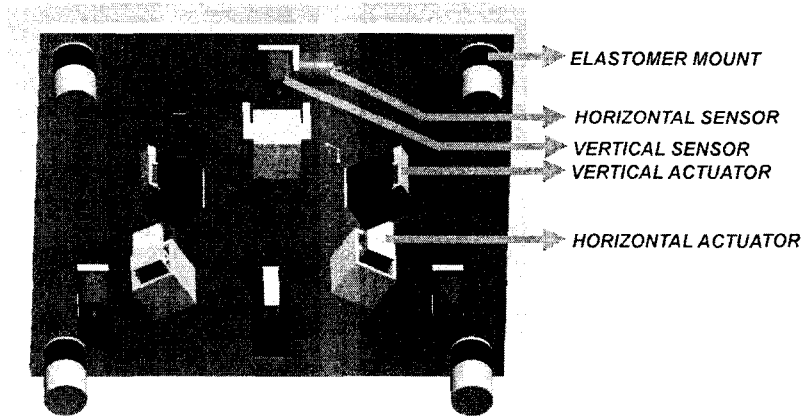


Fig. 1: Complete system conceptual design

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. Figure 2 shows two elastomers mounts connected to upper plate.

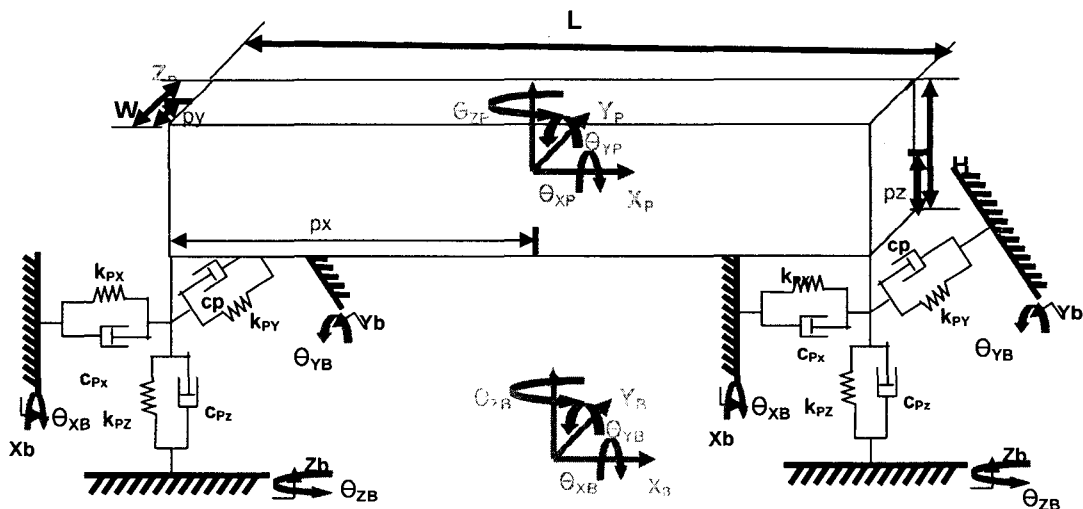


Fig. 2: Passive system basic model

3. active control of vibration

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

$$([M]s^2 + [C]s + [K])p(s) = [C]sb(s) + [K]b(s) + F_u(s)$$

$$\Rightarrow p(s) = \frac{1}{([M]s^2 + [C]s + [K])} F_u(s) + \frac{([C]s + [K])}{([M]s^2 + [C]s + [K])} b(s)$$

$$\Rightarrow p(s) = \begin{bmatrix} G_{11}(s) & 0 & 0 & 0 & 0 & G_{16}(s) \\ 0 & G_{22}(s) & 0 & 0 & G_{25}(s) & 0 \\ 0 & 0 & G_{33}(s) & 0 & 0 & 0 \\ 0 & 0 & 0 & G_{44}(s) & 0 & 0 \\ 0 & G_{52}(s) & 0 & 0 & G_{55}(s) & 0 \\ G_{61}(s) & 0 & 0 & 0 & 0 & G_{66}(s) \end{bmatrix} F_u(s) + \begin{bmatrix} G_{11}(s) & 0 & 0 & 0 & 0 & G_{16}(s) \\ 0 & G_{22}(s) & 0 & 0 & G_{25}(s) & 0 \\ 0 & 0 & G_{33}(s) & 0 & 0 & 0 \\ 0 & 0 & 0 & G_{44}(s) & 0 & 0 \\ 0 & G_{52}(s) & 0 & 0 & G_{55}(s) & 0 \\ G_{61}(s) & 0 & 0 & 0 & 0 & G_{66}(s) \end{bmatrix} b(s)$$

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

Independent transfer function for both $G_c(s)$ and $G_r(s)$ were obtained which can be used to design control system for each rigid body modes.

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. Figure 10 shows decoupled open loop transmissibility for these six rigid body modes:

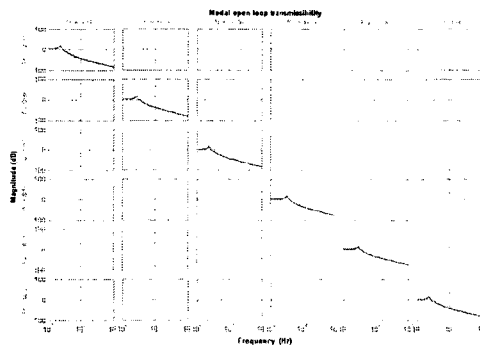


Fig.3: Open loop transmissibility of modal decoupled modes

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

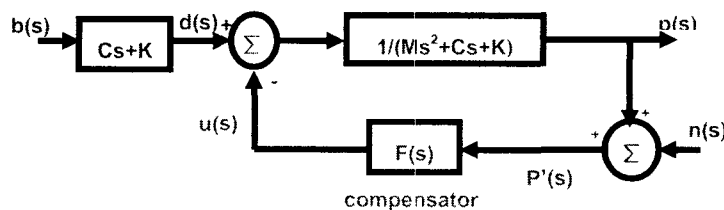


Fig.4: Control system block diagram

After tuning the PID feedback control, stable closed loop transmissibility of six diagonal

rigid body modes are:

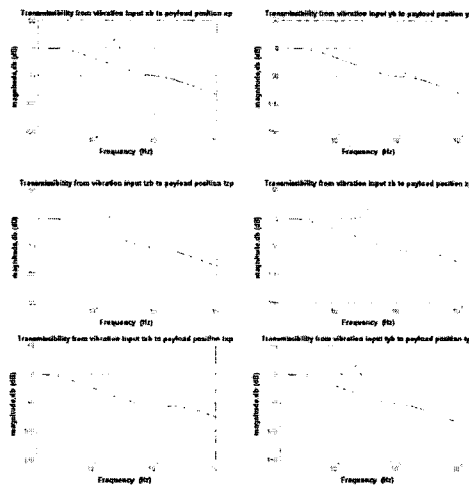


Fig.5: Comparison of open and closed loop transmissibility

4. Conclusion

Complete model of four mount six degree of freedom active vibration isolator was proposed along with the control system to implement. This isolator is well applicable for isolation of very low frequency ground vibration with typical amplitude in sub micrometer range and having frequency range around 01-100Hz.

Reference

- [1] Yoshiya Nakamura, Masanao Nakayama, Keiji Masuda, Kiyoshi Tanaka, Masashi Yasuda and Takafumi Fujita, "Development of active six-degree-of-freedom microvibration control system using giant magnetostrictive actuators" Smart Material Structure, vol.:9, year: 2000, pages:175-185.
- [2] Marcel Heertjes, Koen de Graaff, Jan-Gerard van der Toorn, Active Vibration Isolation of Metrology Frames: A Modal Decoupled Control Design" Journal of Vibration and Acoustics, vol.:127, year: June 2005, pages:223-233.
- [3] Hirokazu Yoshioka and Nobuyoshi Murai, "An Active Microvibration Isolation System" in Proceedings of the 7th International Workshop on Accelerator Alignment, Spring-8, 2002, pages: 388-401.
- [4] E.F. Gobel, Rubber Springs Design, Newnes-Butterworth, London: 1974 pages: 20-101.