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Single Axis Vibration Isolation System Using Series Active-passive Approach

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ABSTRACT

To control the vibration transmitted to the precision instruments from ground has always been of great interest among the researchers. This paper proposes a single axis vibration isolation system which can be used as a module for a table top six axis isolator for highly precise measurement and actuation system. The combined active-passive isolation principle is used for vertical vibration isolation by mounting the instrument on a passively damped isolation system made of Elastomer along with the active stage in series which consists of very stiff piezo actuator. The active stage works in combination with the passive stage for working as a very low frequency vibration attenuator. The active stage is isolated from the payload disturbance through the passive stage and thus modularity in control is achieved. This made the control algorithm much easier as it does not need to be tuned to specific payload.

1. Introduction

Seismic(ground-borne) vibration has long been considered to be prime source of disturbance in nanotechnology. When resolution of measurement instrument is in nanometer scale, seismic vibration with amplitude around submicron region and very low frequency around 0.5Hz, is of great concern. Control of such vibration is necessary for precision instruments to work precisely. Many researches have so far been carried out in the field of isolation of instruments from ground borne vibration. 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. Commercial isolation systems use approaches such as passive, active or semi- passive. This paper proposes a series active-passive structure that can provide vertical isolation in single axis with frequency as low as 0.5Hz and can work as a stable mount for instruments with weight around 50Kg. Passive stage is realized by elastomer flexure. Active stage consists of a very stiff piezo actuator and a feedback sensor. This stage uses feedback control through sensors mounted in an intermediate mass between active and passive stage to de stiffen the active stage as well as to provide inertial damping to the system.

2. Objective of vibration isolation

Required isolation objective is to attenuate transmitted vibration from base to payload. Here payload is instrument which needs to be isolated [1,3,4]. To perform such action vibration transmission path from base to payload is designed to provide necessary damping and stiffness such that base disturbance is perfectly attenuated. Single axis basic isolator is modeled as follows:

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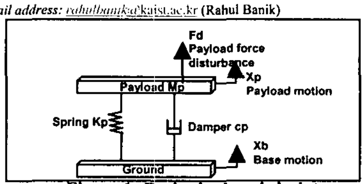

Figure 1: Basic single axis isolator

Transfer function of the system from payload(x_p)to base disturbance(x_b) is:

$$\frac{x_p}{x_b} = \frac{s + k_p/c_p}{s^2 + \frac{c_p}{M_p}s + \frac{k_p}{M_p}} = \frac{2d\omega_n s + \omega_n^2}{s^2 + 2d\omega_n s + \omega_n^2} \quad (1)$$

Normalized transmissibility curve for passive system with different damping coefficient is:

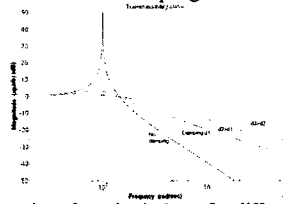

Fig. 2:Transmiss. of passive isolator for different damping

It is evident that natural frequency (ω_n) of an attenuator such as an isolator should have very low value to effectively attunuate low frequency disturbances[4]. Minimum isolator natural frequency,f_n is: $f_n = \frac{f_d}{\sqrt{2}} Hz$. For transmissibility T, the relation is:

$$f_n = \frac{f_d}{\sqrt{1 + \frac{1}{T}}} Hz$$

So two most important characteristics of a vibration isolator are resonance effect at natural frequency which gives rise to very high amplitude and high frequency roll-off which implies the attenuation rate[2,4]. To overcome resonance, damping is necessary. But increase of

This damping is called relative damping as it is proportional to relative velocity between payload and ground. Most passive isolators use relative damping to provide isolation which has always been a trade off between increase of amplitude at resonance and high frequency roll off. So to choose appropriate passive isolator two most important characteristics should be damping coefficient and transmissibility at resonance.

Effect of payload disturbance on natural frequency:

For a practical system to work as passive isolator, it has to handle different payloads varying in weight and structure. For simplicity we consider payload to be rigid body with variation of mass only. This variation adversly affects dynamics as well as disturbance attenuation. To make payload stable, absolute payload velocity V_p needs to be minimized[1]. Transmissbility of absolute payload velocity(V_p) to payload force(F_d)(fig.1) is

$$\frac{V_p}{F_d}=\frac{s/M_p}{s^2+\frac{c_p}{M_p}s+\frac{k_p}{M_p}}=\frac{s/M_p}{s^2+2d\omega_n s+\omega_n^2} \quad (2)$$

Transmissibility of payload velocity to base disturbance velocity is:

$$\frac{V_p}{V_b}=\frac{\frac{c_p}{M_p}s+\frac{k_p}{M_p}}{s^2+\frac{c_p}{M_p}s+\frac{k_p}{M_p}}=\frac{2d\omega_n s+\omega_n^2}{s^2+2d\omega_n s+\omega_n^2} \quad (3)$$

Figure3 compares normalized transmissibility curves(eqs.2 and 3). It shows that improved rejection of base disturbace came at the expense of payload disturbance rejection and vice-versa.

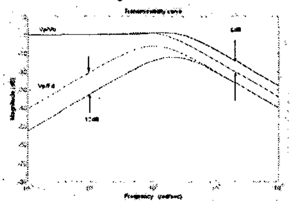

Figure 3: Comparison of payload to base velocity transmissibility(V_p/V_b) with payload absolute velocity to payload disturbance force transmissibility (V_p/F_d)

Reducing natural frequency by order of two, high frequency base disturbance reduces 6dB, but payload force disturbance rejection reduces 12dB.

Effect of Inertial damping:

To overcome effect of reduction of high frequency asymptotic roll-off after resonance and to achieve transmissibility less than one at resonance, inertial damping is provided [1]. Inertial damping is proportional to payload velocity and so it eliminates zero from eqn. 1. Transfer function of system becomes:

$$\frac{x_p}{x_b}=\frac{k_p/c_p}{s^2+\frac{c_p}{M_p}s+\frac{k_p}{M_p}}=\frac{\omega_n^2}{s^2+2d\omega_n s+\omega_n^2} \quad (4)$$

To provide inertial damping, most efficient way is active approach. Providing active damping physical and practical limitations of passive damping can be overcomed.Among many approaches, most simple and effective approach is mounting payload on a soft spring in parallel with a soft actuator that provides theoretically zero stiffness. This architecture is shown below:

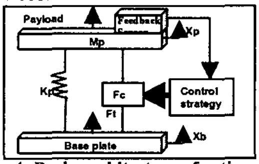

Figure 4: Basic architecture of active soft isolator

Since this structure can have very low natural frequency and also can provide active inertial damping, it is best way to implement an isolator. But since this architecture has very low stiffness, stability of such system, when it is in earth, is difficult to achieve. This form of architecture is well suited for space borne vibration isolator where there is no gravity effect. Open loop trasnfer function of such system will be same as eqn(4) when $F_c=-c_p sX_p$, is the force generated by soft actuator proportional to velocity of the payload measured by a feedback sensor. This form of damping is called skyhook damping [4].

3. Smart isolator design

Major requirement for ground-bourne vibration isolator is to attenuate low frequency vibration typically starting from around 0.5Hz as well as to support payload under gravity effect. Considering these effects a combined active-passive structure is the probable candidate where active stage will attenuate low frequency vibration and high frequency attenuation will be achieved by using a passive isolator. Among the hybrid active-passive systems, following two approaches are most common. Figure5(a) is parallel and figure5(b) is series approach.

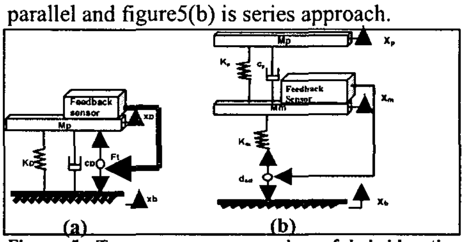

Figure 5: Two common approaches of hybrid active-passive vibration isolation system.

In parallel approach, a soft actuator with almost no added stiffness, works in parallel with the soft passive isolator which is typically modeled as a spring and a dashpot. Feedback provides necessary inertial damping to the system along with the passive damping. Passive stage supports the payload mass as well.

In series approach, payload is mounted on passive stage below which stiff active stage is added. This form of architecture has many advantages compared to its parallel counterpart. It takes advantage of passive as well as active stage. Passive stage handles gravity sag and also works to isolate the feedback sensor from payload disturbance thus reducing burden of tuning control algorithm for specific payload. Moreover when active stage is not working, passive stage can provide necessary isolation. Active stage is designed with knowledge of passive isolator. It is typically an actuator with very high stiffness that works for both providing mechanical strength against gravity and also providing flexibility and modularity of control. Since actuator stage is very rigid, with active isolation turned off, any disturbance in base will pass effectively to the intermediate mass and so feedback sensor can measure the disturbance which is not affected by any payload disturances. Also disturbance in base cannot cause resonance due to compliance in the active stage and thus the control bandwidth is not limited by resonance. Active stage thus provides feedback to actively nullify motion of intermediate mass. To do this, control algorithm must effectively soften active stage stiffness. With only passive stage, isolation starts at f_p, conmer frequency of passive stage. Active stage softens actuator stiffness over mid frequency bandwidth(from f_a to f_p) and so overall system has very low natural frequency. Passive stage transfer function is:

$$\frac{x_p}{x_m} = \frac{c_p s + k_p}{M_p s^2 + c_p s + k_p} \tag{5}$$

Transfer function of active stage with no feedback:

$$\frac{x_m}{x_b} = \frac{\frac{k_m}{M_m}\left[s^2 + \frac{c_p}{M_p}s + \frac{k_p}{M_p}\right]}{s^4 + \left[\frac{c_p}{M_p} + \frac{c_p}{M_m}\right]s^3 + \left[\frac{k_m}{M_m} + \frac{k_p}{M_m} + \frac{k_p}{M_p}\right]s^2 + \frac{k_m c_p}{M_m M_p}s + \frac{k_m k_p}{M_m M_p}} \tag{6}$$

Choice of passive stage:

To choose passive stage for vertical isolation, requirements are[2]:

-- Minimum static sag under gravity.

-- Natural frequency around 20-25Hz for payload weight upto 50Kg.

-- Dynamic amplification less than 12dB at resonance.

First most important consideration is minimum static sag. We considered payload weight, M_p = 50kg. and sag, ds = 0.5mm. Static stiffness(K_p) and natural frequency(f_p) of passive stage that can provide 0.5mm deflection at 50kg load are:

$$K_p = \frac{M_p * g}{ds} = 980\,N/mm$$

$$f_p = \frac{1}{2\pi}\sqrt{\frac{K_p}{M_p}} = 22.28\,Hz$$

Second damping factor (c_p)was choosen and to do so the amplification at resonance was chosen. Transmissibility at resonance is: $T_{max} = \frac{1}{2d}$

where d is the damping coefficient.

Again we know from equation (5), $d = \frac{1}{2}\frac{c_p}{\sqrt{k_p M_p}}$

so

$$T_{max} = \frac{\sqrt{980 * 10^3 * 50}}{c_p}\ \text{dB} \tag{7}$$

From commonly used materials we choose Elastomer as passive isolator with d = 0.05 and T_{max} = 10dB from equation (7). Moreover Elastomer is flexible and can be formed into any shape with high axial stiffness and low bending stiffness.It also has equal dynamic spring stiffness in all direction. Transfer function of the passive system becomes:

$$\frac{x_p}{x_m} = \frac{14s + 19600}{s^2 + 14s + 19600} \tag{8}$$

Choice of the active stage:

To choose active stage we need to assume some parameters. Since only the feedback sensor is connected to the middle mass, the mass(M_m) is considered to be typically 100 times smaller than the payload mass(M_p) i.e M_m = M_p/100. The actuator natural frequency should be well away from the active control bandwidth (0.2-200Hz) to avoid resonance condition. So natural frequency of actuator, f_a = 1500Hz was choosen. Such hard actuation is possible with typically piezoelectric actuator. Stroke length of this actuator is typically small, but since we are concerned with vertical ground vibration isolation with amplitude around submicron region, PZT can be chosen. Further it should have enough mechanical strength to overcome the total load of the payload combined with sensor assembly. Here feedback signal F_t provides enough displacement d_{act} to nullify displacement(x_m) at mass M_m. Considering all assumptions, transfer function of active stage without feedack is:

$$\frac{x_m}{x_b} = \frac{8.88*10^2 s^2 + 1.24*10^9 s + 1.74*10^{12}}{s^4 + 1415s^3 + 9.08*10^7 s^2 + 1.24*10^9 s + 1.74*10^{12}} \tag{9}$$

Transmissibility of passive as well as active stage without feedback is as follows:

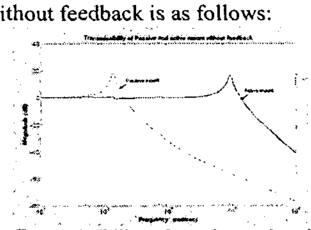

Figure 6: Transmissibility of passive and active mount without feedack

Feedback control:
Block diagram shows how control algorithm works:

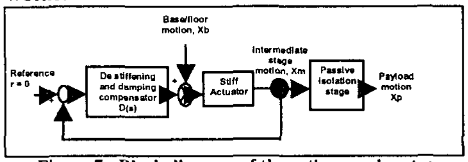

Figure 7 : Block diagram of the active-passive stage.

Objective of compensator is to reduce stiffness of active stage as well as to provide active inertial damping to overall system. Considering these facts, the compensator is choosen to be a PD controller with transfer function found after tuning as:

$$D(s) = \frac{50s^2 + 100s}{s}$$

Transfer function of feedback controlled active stage is:

$$\frac{x_m}{x_b} = \frac{8.88*10^7 s^2 + 1.24*10^9 s + 1.74*10^{12}}{s^4 + 4.44*10^9 s^3 + 7.12*10^{10} s^2 + 8.73*10^{13} s + 1.76*10^{14}} \quad (10)$$

Overall trasnfer function was as follows

$$\frac{x_p}{x_b} = \frac{1.24*10^9 s^3 + 1.76*10^{12} s^2 + 4.88*10^{13} s + 3.42*10^{16}}{s^6 + 4.44*10^9 s^5 + 1.33*10^{11} s^4 + 1.75*10^{14} s^3 + 2.8*10^{15} s^2 + 1.72*10^{18} s + 3.45*10^{18}} \quad (11)$$

Figure8 shows overall transfer function comparing passive and active stage with feedback:

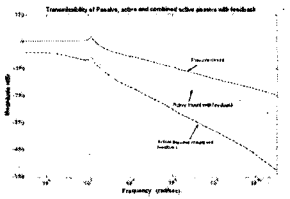

Figure 8: transmissibility of passive, active and combined active-passive stage with feedback

Combined active passive stage takes advantages of both feedback control as well as passive stage. Overall natural frequency of the system is found to be: f_n = 0.3Hz , which can successfully attenuate disturbing frequency as low as 0.43 Hz. Another consideration is Transmissibility at resonance. Damping factor was , d = 1.0. So T_{max} = 0.5 dB which is well below unity indicating better damping at resonance.

Comparison of Series and Parallel structure:

To compare performance of series active-passive vibration isolation system, a parallel system using feedback contrl with same natural frequency was simulated. Transmissibility of the parallel system tuned to same natural frequency as the series system (eqn 11) was found to be:

$$\frac{x_p}{x_b} = \frac{14.01s + 19620}{701.4s^2 + 9.824*10^5 s + 1.982*10^6} \quad (12)$$

Figure 9 shows the transmissibility curves of the passive mount (eqn 7), combined active-passive mount with feedback for both parallel (eqn 12) and series structure (eqn 11) with feedback gain tuned to same natural frequency.

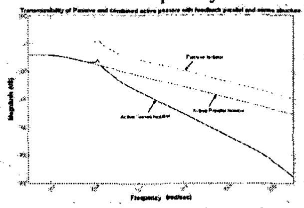

Figure 9: Transmissibility of passive and active-passive feedback compensated series and parallel mount.

As is evident from the figure after tuning gains for both configurations it is found that for same natural frequency, high frequency roll off rate for the series approach is better than the parallel approach. The parallel approach has high frequency roll-off rate aorund -40dB/decade whereas the series approach has high frequency roll-off rate around -78dB/decade. The reason behind this is in parallel approach the passive isolator implements the relative damping which is proportional to the difference between velocities of payload and base and so even though active stage compensates using active inertial feedback, parallel stage transmissibility actually follows passive isolator transmissibility after the corner frequency. On the other hand, series approach feedback sensor is isolated from payload through passive isolator and feedback actually implements the active inertial feedback which is much less affected by payload veloctiy.

4. Conclusion

From above simulation result it is found that the series active-passive stage is best candidate *for vertical ground vibration isolator. A single* axis isolator was designed and it's perfornace was verified with feedback control which shows that a stiff actuator with feedback combined with a passive soft isolator stage can effectively attenuate disturbance frequency as low as 0.4 Hz and also maximum transmissibility at resonance is also below unity. This modular stage can be used as a unit for a total six axis vibration isolation stage.

5. Reference

1. A.M. Beard, D.W. Schubert, A.H. von Flotow, "A Practical product *implementation of an active/Passive vibration isolation* system." SPIE vol. 2264; 38-49; Vibration monitoring and control 1994.
2. Eric H. Anderson, Donald J. Leo, Mark D. Holcomb, "UltraQuiet Platform for active vibration isolation." CSA Engineering, Inc.
3. Dino Sciulli, "Dynamics and control for Vination Isolation Design.", PhD Thesis, Virginia Polytechnic Institute and State University, 1997.
4. Ahmed Abu Hanieh, "Active Isolation and Damping of Vibrations via Stewart Platform", PhD Thesis, University Libre of Brussels, 2003.