



Active Vibration Isolation Using a Voice Coil Actuator with Absolute Velocity Feedback Control

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Abstract: This paper describes the active vibration isolation using a voice coil actuator with absolute velocity feedback control for highly sensitive instruments (e.g., atomic force microscopes) which suffer from building vibration. Compared with traditional isolators, the main advantage of the proposed isolation system is that it produces no isolator resonance. The absolute vibration velocity signal is acquired from an accelerator and processed through an integrator, and is then input to the controller as a feedback signal. The controller output signal then drives the voice coil actuator to produce a sky-hook damper force. In practice, the phase response of the integrator at low frequencies (2~6 Hz) deviates from 90 degree which is the exact phase difference between the vibration velocity and acceleration. Therefore, an adaptive filter is used to compensate for the phase error. Analysis of this active vibration isolation system and comparison of model predictions to experimental results indicate that the proposed method significantly reduces transmissibility at resonance without incurring increased transmissibility at higher frequencies.

Keywords: active vibration isolation; proportional control; adaptive filter; sky-hook damper; absolute vibration velocity; voice coil actuator

Introduction

The isolation of vibration-sensitive equipment from floor vibrations is usually accomplished using elastomeric isolators [1-3]. However, there is a trade-off between low and high frequency isolation performance depending on the damping of this resilient isolator [1]. Traditional passive isolators can efficiently isolate high frequency vibrations, but suffer from a critical defect in regards to low frequency resonance, which is a frequency at which passive isolators amplify vibrations rather than isolate them. Choosing a high damping ratio results in superior resonant frequency control but poor high-frequency isolation. However, as we decrease the damping ratio, we begin to trade off the resonance control for high-frequency isolation.

To solve the trade-off problem, active isolation solutions such as skyhook damping [4] have been proposed. Such solutions are usually based on elastic mounts and controllable actuators. Details on the theory and application of active vibration controls can be found

in [5]. For active vibration control at micron or nanometer scale displacement ranges, piezoelectric actuators are often used to obtain good positioning accuracy and external perturbation control due to their high accuracy and stiffness [6-9]. On the other hand, because the piezoelectric actuator being significantly more stiff than the spring, the isolation performance of a spring-supported platform is reduced when actuators and springs are arranged in parallel. Therefore, application of the electromagnetic actuator [10-15] instead of piezoelectric actuator in active vibration isolation has been a popular research topic in recent years.

Inertial actuators which do not need to react off a base structure have been used in active vibration isolation systems [10], so they can be used as modules for direct installation on a vibrating structure. To achieve stable skyhook damping with an inertial actuator, the natural frequency of the actuator must be below the first resonance frequency of the structure under control and the actuator resonance should be well damped. The active isolation of a system containing a distributed



parameter isolator using absolute velocity feedback control with four electromagnetic actuators was presented in 2012 [11]. The study investigated and compared two techniques for stabilizing systems which are suffering from instability due to isolator resonances and flexibility of the base. The first technique involves the addition of mass to the base structure, while the second involves the use of an electronic lead compensator.

Based on the skyhook damping concept, the absolute velocity of the upper plate is directly fed back into the controller to generate the control signal that drives the voice coil actuator to produce an anti-vibration force; the response of the upper plate at the resonance frequency is attenuated without compromising high frequency performance. This system is simple and straightforward to implement, but obtaining the correct absolute velocity is crucial determining the vibration isolation performance. In practice, the absolute vibration velocity signal is acquired from an accelerometer and is processed through an integrator. The phase response of the integrator at low frequencies such as 2~6 Hz deviates from 90 degrees, which is the exact phase difference between the vibration velocity and acceleration, and results in the phase error of the absolute vibration velocity signal.

The proposed method uses an adaptive filter to compensate for this error. The following section establishes the theoretical equation which governs the system's vertical vibration and the theoretical transmissibility. Compared with the passive system, the theoretical transmission of the active isolation system at resonance frequency can be reduced below 1, and isolation at high frequencies is not influenced by this active control.

Active Vibration Isolation using the Sky-Hook Damper Concept

The absolute vibration velocity signal acquired from an accelerometer and processed through an integrator is input to the controller as a feedback signal, and the controller output signal drives the voice coil

actuator to produce a sky-hook damper force. The single-degree-of-freedom mechanism of this system is shown as Figure 1.

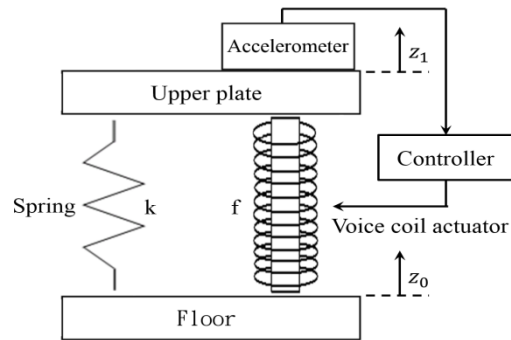


Figure 1. System mechanism.

The voice coil actuator is installed in parallel with four steel-spring isolators, which are placed between a rigid upper platform of mass m and a rigid base platform as a passive weight support mechanism. The four passive isolators are considered to have a total stiffness k , and both mass and the damping constant are neglected. Usually, such a system features six-degree-of-freedom rigid-body motions of the upper platform but, for the sake of convenience, the heave mode of the upper platform is the only motion considered in the theoretical derivation. The active control theory of vibration along another two horizontal axes is the same. Environmental vibration displacement on the foundation is z_0 , and vibration displacement on the upper platform is z_1 . The active control for suppressing the resonant oscillation of the passive vibration isolator is achieved by electromagnetic force f induced by the voice coil carrying electric current through a magnetic field. The theoretical equation which governs the system vibration can be written in following form.

$$m\ddot{z}_1 + k(z_1 - z_0) = f, \tag{1}$$

where electromagnetic force f is

$$f = C_{amp} \times C_{voice} \times V, \tag{2}$$

in which C_{amp} is the response function of the power amplifier, and C_{voice} is the response function of the voice coil actuator. When active control is implemented, the voltage signal V from the controller output must first pass through a power amplifier to provide enough electric current for the voice coil. Following the concept of sky-hook damping, the actuator force is proportional to the absolute velocity of the upper platform. Therefore, the voltage signal V is also proportional to this absolute velocity, as in the following equation if both of C_{amp} and C_{voice} are linear responses.

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$$V = -K_p \times (\dot{z}_1 \times S) \tag{3}$$

where K_p is the proportional coefficient of controller, and S is the sensitivity of the sensors including the accelerometer and integrator. After combining the above three equations and merging C_{amp} 、 C_{voice} 、 S as a parameter C^* , the final governing equation can be obtained as follows

$$m\ddot{z}_1 + k(z_1 - z_0) = -C^* \times K_p \times \dot{z}_1. \tag{4}$$

The excitation (building vibration at the floor) is

$$z_0 = a_0 e^{i\omega t}, \tag{5}$$

where a_0 is the excitation amplitude and ω is the circular excitation frequency. The system response is given by

$$z_1 = a_1 e^{i\omega t}. \tag{6}$$

Substituting (5) and (6) into (4), the transmissibility can be obtained as

$$\frac{a_1}{a_0} = \frac{k}{(k - m\omega^2) + i\omega k_p C^*}. \tag{7}$$

Numerical Simulations

Numerical simulations were conducted to verify the aforementioned theoretical model as applied to an active vibration isolation system. The upper platform has mass $m = 55$ kg. Four passive isolators have a total stiffness $k=13600$ N/m and parameters $C_{amp} = 1$, $C_{voice} = 0.18$ N/V, $S = 393.7$ V·s/m. These parameters are substituted into equation (7), and transmissibility of the active vibration system is obtained as different values of K_p . As shown in Figure 2, when K_p exceeds 16, the theoretical transmission of the active isolation system at resonance frequency can be reduced below 1, and the isolation at high frequencies is not influenced by this active control.

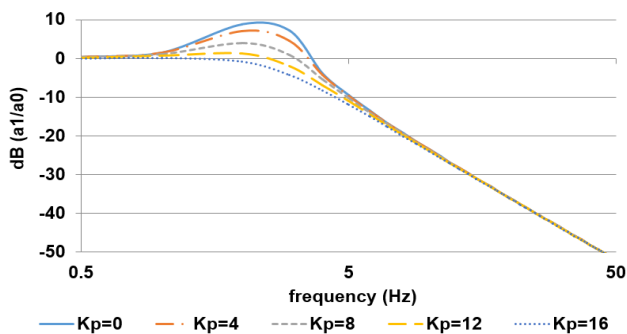


Figure 2. Theoretical transmission of the active vibration isolation system for different value of K_p .

Adaptive Algorithm

The numerical simulation results indicate that the control strategy employing the absolute velocity feedback control to activate the actuator with an appropriate gain of K_p can suppress the resonance of the passive isolation system at natural frequency. In practice, the absolute vibration velocity signal is acquired from an accelerometer and processed through an electric integrator, but the phase response of the integrator at low frequencies (e.g., 2~5 Hz) deviate from 90 degrees, which is the exact phase difference between the vibration velocity and acceleration. In this paper, an adaptive algorithm is designed to filter the feedback signal after integration to compensate for the phase error.

An adaptive algorithm consists of two parts: a digital filter which processes the expected output signals, an algorithm to adjust the weighting coefficients of the digital filter. One of two digital filters are usually applied to AVC, i.e., the finite impulse response filter (FIR filter) or the infinite impulse response filter (IIR filter). In this study, FIR filter is used as the system controller. Suppose a linear discrete-time FIR filter with length L , a series of weight coefficients, $w_l(n)$, $l=0, 1, \dots, L-1$, and a series of continuous reference inputs $\{x(n), x(n-1), \dots, x(n-L+1)\}$ are considered, where n is the time index, and the output signal is calculated as

$$y(n) = \sum_{l=0}^{L-1} w_l(n) x(n-l) = W^T(n) X(n), \tag{8}$$

where $X(n)$ is the reference input vector defined as

$$X(n) \equiv [x(n) \ x(n-1) \ \dots \ x(n-L+1)]^T, \tag{9}$$

and $W(n)$ is the weight vector defined as

$$W(n) \equiv [w_0(n) \ w_1(n) \ \dots \ w_{L-1}(n)]^T, \tag{10}$$

where the superscript T denotes transposition. The reference input $X(n)$ which is the velocity signal from the electric integrator has a phase error relative to the exact velocity. The filter is used to modify the phase error and obtain the exact velocity signal $y(n)$. Then, the LMS (Least Mean Square) algorithm is used to renew the weight coefficients as

$$W(n+1) = W(n) + \mu X(n) e(n), \tag{11}$$



where μ is a convergence factor that controls stability and the rate of convergence, and $e(n)$ is the residual vibration signal defined as

$$e(n) = d(n) - y(n), \tag{12}$$

where $d(n)$ is the exact velocity signal generated from the theoretical calculation. The weight coefficient $W(n)$ of the adaptive filter is obtained by off-line experimentation, and is used after the integrator to compensate for the phase error. As shown in Figure 3, this adaptive processing method is proposed to adjust the phase error. The acceleration signal $a(n)$ is a pure sinusoidal signal at 2.5 Hz, which is the natural frequency of the passive isolation system, and is processed through an electrical integrator to obtain the velocity signal $x(n)$. The adaptive filter is performed in a PC, and the sampling frequency is 1 KHz. Better results can be obtained by optimizing the length of the adaptive filter and the step size to update the adaptive filter. Optimum performance can be achieved using step size 0.01 and filter length 125. As shown in Figure 4, the phase between the vibration velocity and acceleration is compensated from 59.34° to 90° at 2.5 Hz. This adaptive proportional control method is proposed as the active control strategy, and the flowchart of this strategy is shown in Figure 5.

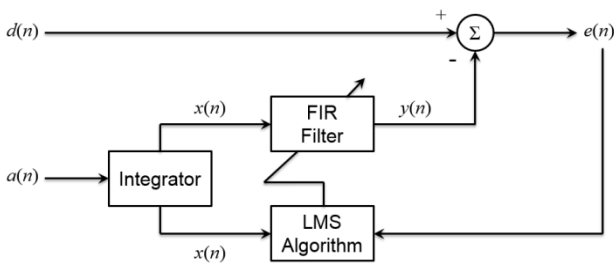


Figure 3. Flowchart of the adaptive processing method, $a(n)$ is the acceleration signal.

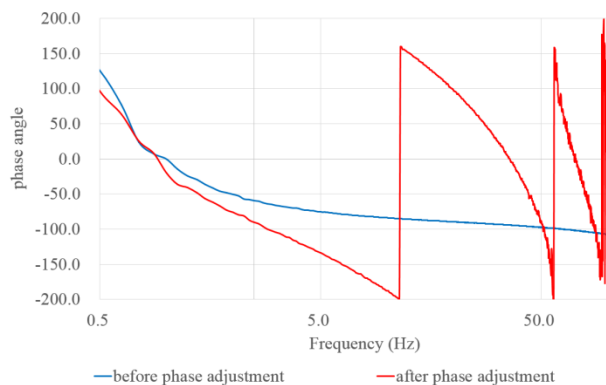


Figure 4. Phase angle between the velocity signal and the acceleration signal.

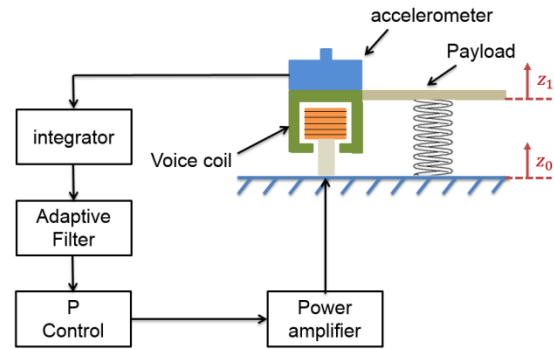


Figure 5. Flowchart for adaptive proportional control on vibration isolation.

Experimental Investigation

In addition to the numerical simulations, experiments were conducted to assess the performance of the proposed active vibration isolation system. Figure 6 shows the configuration of the isolation system in the experiments. A passive isolator consisting of a steel plate supported by four springs is located on a granite platform. Four accelerometers are installed on the side of the plate, and four voice coil actuators are placed below the plate opposite the accelerometer. In the experimental work, an FFT analyser (B&K pulse system 3560) is used to measure the response of the steel plate. The velocity signal is measured using an accelerometer (Wilcoxon Research 731A) and an electric integrator. The signal is fed back to the controller to implement the adaptive proportional control. The velocity signal is also captured by the analyser, so that the measured frequency responses can be compared to validate the theoretical method. Because the vibration level is very low, the velocity signal must be amplified using an analogue circuit before it is transmitted to the controller. Furthermore, there are many kinds of noise and high frequency signals in the measured signal, so a low-pass filter is required to purify the measured signal. In addition, the power of the control signal from the controller must be amplified, because the voice coil actuator needs enough input power to produce an electromagnetic force.

The control performance of the active vibration isolation system was investigated using transient response and transmissibility experiments. As shown in Figure 7, the experimental results indicate that the proposed method can quickly reduce resonance vibration while active control is activated. Before active control is activated, the peak value of the vibration velocity at the resonance frequency of passive isolator is about 70 $\mu\text{m/s}$. After control is activated, the vibration falls below 10 $\mu\text{m/s}$. The proposed method cancels 98% of the vibration energy at the resonant frequency, and the response time needed to effectively control vibration at the resonance frequency is less than 0.5 second.

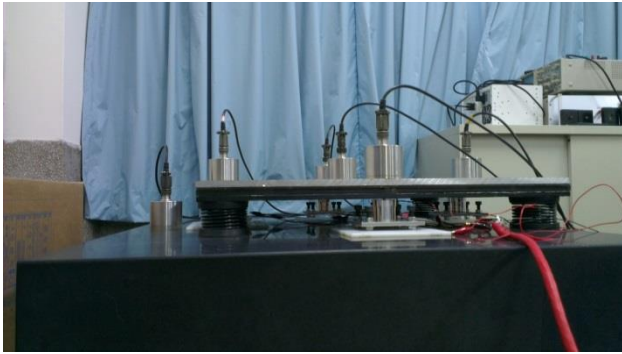


Figure 6. Experimental configuration.

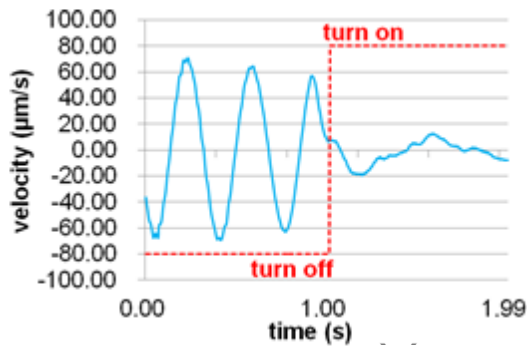


Figure 7. Time history of the displacement on payload with control.

As shown in Figure 8, the experimental results show that the proposed method can significantly reduce transmissibility from 24 dB to 0 dB at resonance without incurring increased transmissibility at higher frequencies. As shown in Figure 9, compared with the vibration criterion (VC) curves proposed by Gordon [16], the payload vibration after control can be reduced below the VC-E sufficient for the most demanding vibration-sensitive instruments. Control performance demonstrates that applying the proposed active control method to vibration isolators can effectively reduce the resonant vibration of passive isolators.

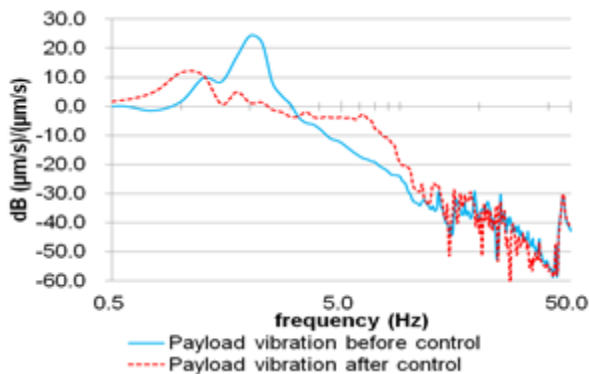


Figure 8. Experimental transmission of passive and active isolation systems.

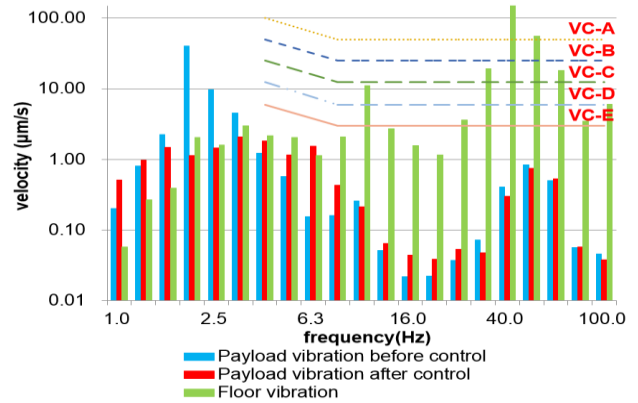


Figure 9. Comparison of one-third octave band measured vibration.

Conclusion

Active vibration isolation by adaptive proportional control is theoretically and experimentally investigated. Absolute velocity feedback control is implemented experimentally for active vibration control incorporating a passive vibration isolator to suppress the resonant oscillation of natural frequencies. In practice, conventional passive isolators suffer from an inherent trade-off between poor high-frequency isolation and amplification of vibration at the fundamental mounted resonance frequency. In this paper, the best isolation performance is achieved using an active system in combination with passive isolators, where the fundamental resonance can be actively controlled without degrading high-frequency performance. Theoretical analysis shows that the effect of active control is similar to a sky-hook damper in suppressing resonant oscillation. Experimental results show that the proposed method cancels 98% of the vibration energy at the resonant frequency. Control performance demonstrates that applying the adaptive proportional controller to actively control the vibration isolator can effectively reduce the resonant vibration of passive isolators.

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