Experimental Evaluation of Ultra Hard Mount Vibration Control Systems

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Abstract

In this paper the performance of an ultra-hard mount active vibration control system is evaluated. High stiffness mounts offer position stability and high force disturbance rejection, at the cost of sensitivity to indirect disturbances. Active vibration control can be used to overcome this, using feedback to dampen the resonance and feedforward for lowered transmissibility. This paper evaluates different feedback strategies and compares experimentally on a single-axis piezo-based ultra hard mount system. It was found that good damping performance can be achieved, reducing the output vibrations by almost 60% with various feedback methods. Furthermore, using straightforward stiffness compensation feedforward, the influence of indirect disturbances was shown to be reduced significantly. The influence of indirect disturbances was reduced by 94% with feedforward when compared to the uncontrolled case.

Keywords

Active vibration control, Vibration isolation, Disturbance feedforward control, Disturbance rejection, Hard mount, Piezoelectric stack

Introduction

The performance of high-precision machines and scientific instruments depends on the disturbances acting on them. While the nature and dynamics of such disturbances are highly case-specific, they can be divided into two classes. Indirect disturbances are commonly floor vibrations and are thus often referred to as such. Direct disturbances may be caused by the forces acting directly on a system, forces transmitted through cables, and motion of systems components [1].

Coping with the disturbances is often the role of the mount between the device and the machine's frame or the floor. When the floor vibrations are the primary concern, the mount can be designed as a soft vibration isolation system [2]. The low stiffness and the corresponding low resonance frequency of the system (typically below 5Hz) are advantageous in this context, as above the resonance, the transmissibility of vibration is attenuated. This, however, comes at the cost of problems with levelling, sagging and increased force disturbance sensitivity [3]. Hard mounts have been proposed to address these issues [1]. Using a higher stiffness mount leads to much-decreased sensitivity to direct disturbances [4] and a higher resonance frequency of the system, reaching 35 Hz [3]. In consequence, the system is more susceptible to indirect vibrations.

In this paper, we focus on applications where the direct disturbances acting on the system are large and position stability is especially important. To assure it, an *ultra-hard* mount based on piezoelectric stack actuators is proposed. Thanks to their capability of exerting high forces, an active solution can be created to deal with both direct and indirect disturbances effectively. Additionally, piezoelectric stacks are proven technology in the high-tech industry, have low energy consumption in static operation, and produce no magnetic fields that can interfere with sensitive equipment [5].

The high stiffness of the piezoelectric stack prevents the excitation of the structure by direct disturbances. The performance is further improved by active damping of the resonance peak. The most common method is 'sky-hook' damping, utilizing velocity feedback to dampen resonance modes [6]. This technique can also be applied with different sensor types like accelerometers or force sensors [1] [7]. When position-related measurements are available, Positive Position Feedback [8] can be used to avoid differentiation and noise amplification. While the active damping methods are well-developed and commonly used, extensive studies comparing them to each other are missing.

Because of the high resonance frequency, the system's passive structure is ineffective in isolating the floor vibrations. However, the influence of indirect vibrations can be cancelled actively using feedforward techniques based on the measurement of the incoming floor vibrations [9]. Feedforward requires an accurate dynamic model of the system which can be difficult to acquire in practice [10], as such work has focused on adaptive feedforward techniques [3] [11]. So far, the feedforward techniques have been applied in hard mounts, but their effectiveness in high-stiffness piezoelectric systems has not been investigated.

The objective of this work is to experimentally evaluate the performance of ultra hard mounts in the vibration control context. The problem under study is formally presented in Section 2, together with preliminary information. Section 3 presents the results obtained from the experiments. The paper is concluded in Section 4.

System Description

A single-axis experimental setup, presented in figure 1, is used to represent the ultra hard mount system. A platform with adjustable mass represents the payload to be supported. The main stack actuator (model *P-843.20*) has a stiffness of 27×10^6 N/m (k_2 in figure 1), and constitutes the ultra hard mount and connects the payload to the shaking base. This shaking base is actuated by another stack actuator (model *P-235.1S*), with a higher stiffness of 860×10^6 N/m (k_1 in figure 1). Motion of all elements of the setup is constrained to a single degree of freedom using flexures. The abstraction of the setup is also presented in figure 1. Accelerations $\ddot{x}_1$, $\ddot{x}_2$ are measured using *PCB Synotech 393B05* accelerometers. Absolute displacement of $m_1$is measured with a *PIseca D-510.021* and the absolute displacement of m_2 with a *D-050* capacitive sensor. Finally, relative measurements are taken with the integrated strain gauge sensors of the stack actuators. In the configuration used in this paper, the resonance frequency of the system is 103 Hz. This can be adjusted to a higher or lower frequency by adjusting the mass m_2.

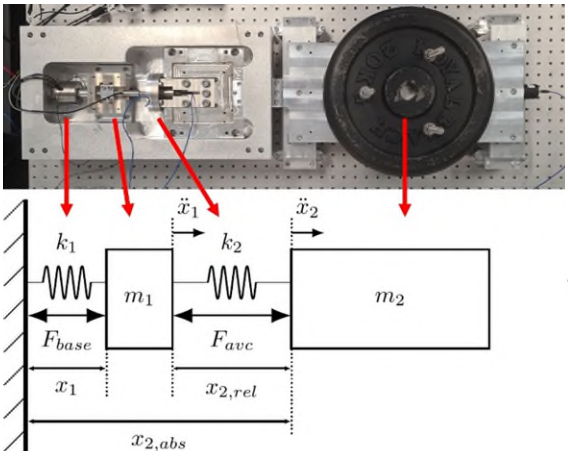

Figure 1. Top-down overview of the piezo-based experimental setup, with corresponding mass-spring damper model

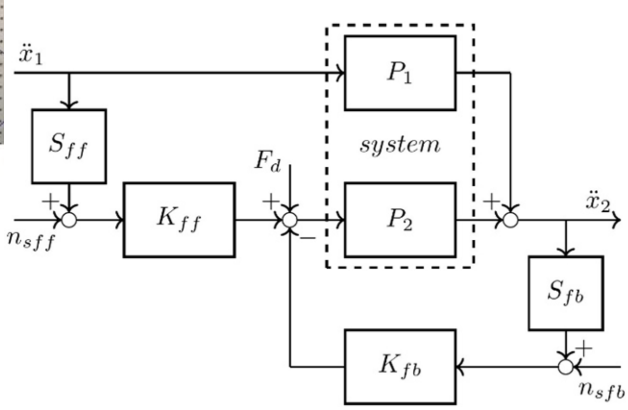

Figure 2. Block-diagram representation of an active vibration control system, showing both feedback and feedforward controllers.

To study the response of a system to floor vibrations, a vibration profile based on the VC-C curve from [12] is applied to the shaking base. Direct disturbances are applied to the main stack actuator, also used for active vibration control, with the force profile starting at 10 N at low frequencies and descending with -1 slope from 5 Hz onwards.

Figure 2 shows the block diagram of the implemented control system with acceleration measurements. When absolute and relative positions are measured, similar structures are used. Transfer functions P_1 and P_2 represent the transmissibility and compliance of the passive structure, respectively. Transfer function K_{fb} represents the feedback controllers for active damping and K_{ff} corresponds to the feedforward controller's dynamics.

The goal of vibration control is to minimize the motion of the mass $\ddot{x}_2$. To calculate the total error, the Power Spectral Density (PSD) of the signal is integrated to obtain the Cumulative Power Spectrum (CPS):

$$CPS(f) = \int_0^f PSD(v)\ \mathrm{d}v$$

The CPS visualizes the contribution to the total error at each frequency. The final value of the CPS is equal to the square of the root-mean-square (RMS) of the signal [13]. The reduction of the resonance peak by active damping is represented by the sensitivity function [14]

$$S = \frac{1}{1 + K_{fb} P_2}.$$

To prevent noise amplification and changing the dynamics of the system at other frequency regions, it should have a notch filter characteristic. Such characteristics are achieved with the triangular open loop gain $K_{fb}P_2$ [15]. The differences between the different controllers implemented are shown in figure 3.

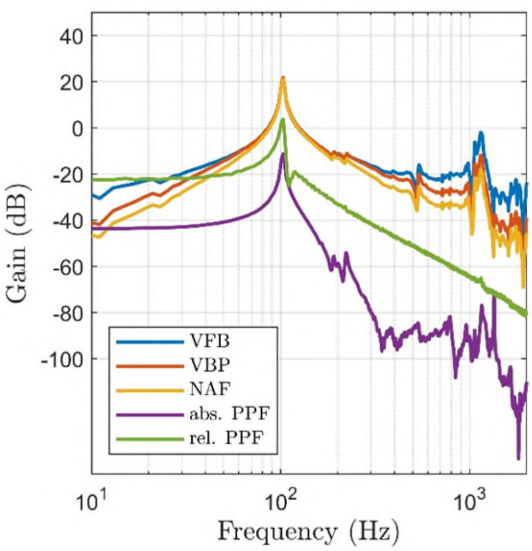

Figure 3. Open-loop gain of the different AVC strategies. An optimal shape is a triangular shape centered on the resonance frequency.

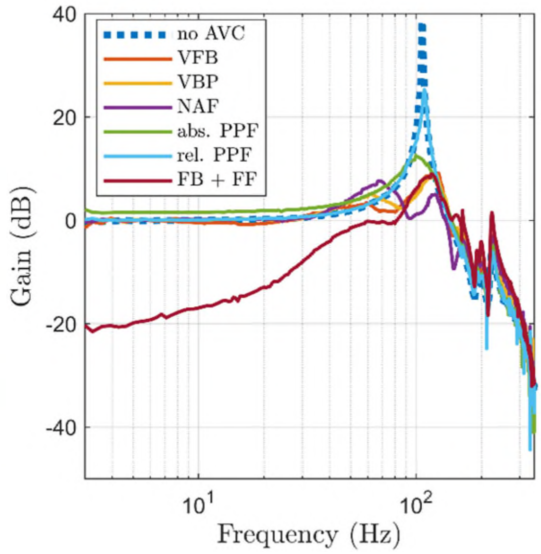

Figure 4. Transmissibility from x_1 to x_2 of the system with no AVC and with different strategies implemented, showing the damping performance of the different methods.

In the most common approach, velocity feedback (VFB) is used [6]. When accelerometers are used as sensors, the controller takes form of a low-pass filter

$$K_{vfb} = \frac{g_{vfb}}{s + \omega_v}$$

to avoid the drawbacks of using pure integrators. To decrease the influence of noise and excitation of high-frequency dynamics in the system a bandpass filter is added close to the target frequency to be damped, forming Velocity Band-Pass feedback (VBP) controller

$$K_{vbp} = \frac{g_{vbp}\ \omega_{c1}\ \omega_{c2}}{(s + \omega_{c1})\ (s + \omega_{c2})}.$$

To achieve a stronger effect, a controller with a resonance peak

$$K_{naf} = \frac{g_{naf}\ \ k\ \omega_{f,naf}^2}{s^2 + \zeta_{naf}\ \omega_{f,naf} s\ + \omega_{f,naf}^2}$$

is used to increase the gain in the vicinity of the resonance to be damped in the Negative Acceleration Feedback (NAF) strategy [16] [17].

When position-related measurements are available, Positive Position Feedback (PPF) controllers with dynamics

$$K_{ppf} = \frac{g_{ppf}\ k\ \omega_{f,ppf}^2}{s^2 + \zeta_{ppf}\ \omega_{f,ppf}\ s + \omega_{f,ppf}^2}$$

can be used to avoid differentiation of signals and noise amplification. This strategy can be used with both relative position between the isolated equipment and its mounts and absolute position, although the latter is much harder to measure in practice, requiring complicated separate measurement frames. The dynamics of the controller are the same for both relative and absolute PPF.

The feedforward controller is used to decrease the transmissibility of vibrations from the base of the mount. By calculating the system's reaction to the measured indirect disturbance, their effect can be diminished with an opposing control force. The most basic approach is stiffness compensation feedforward. With this method only the stiffness is accounted for and combined with a low-pass filter to avoid feeding noise into the system:

$$K_{ff} = -k\frac{\omega_{lp}}{s + \omega_{lp}}.$$

Note that the position of the base (x_1) is used for feedforward signal generation. For best results, feedback for active damping and feedforward are used simultaneously.

Experimental Evaluation Results

In this section, we discuss the performance of the considered AVC strategies tested on the hard mount system. The controllers were tuned based on rules from literature and experimental observations. The corner frequency of velocity feedback ω_v was set at 10 rad/s. The corner frequencies of the velocity band-pass controller ω_{c1}, ω_{c2} were set at half the resonance frequency and three times the resonance frequency as it was found this gave a good trade-off between damping and low noise amplification. The tuning of the negative acceleration controller was based on positive position feedback tuning from [18]. Tuning of both positive position feedback controllers was based on the tuning by [19]. The damping coefficient for relative position feedback was halved as experiments showed this improved performance. For each controller, the optimum gain was found by performing a gain sweep on the experimental setup.

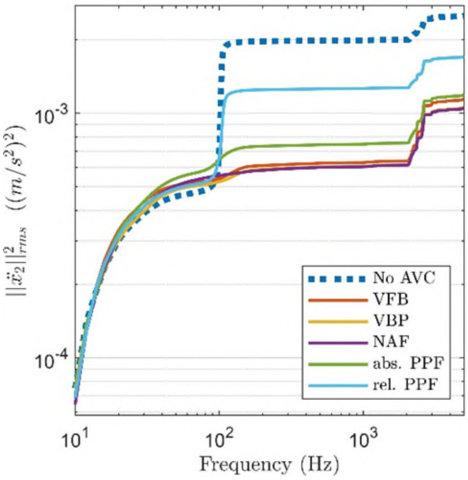

Figure 5. CPS results with different controller gains, showing that an increase in controller gain leads to increased damping, but also an increase in noise amplification and increased excitation of high-frequency modes.

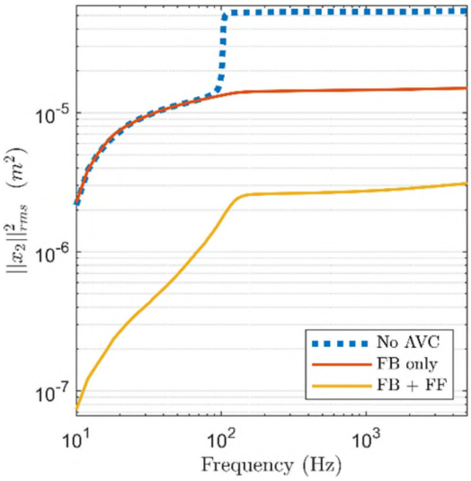

Figure 6. CPS results with different controller gains, showing that an increase in controller gain leads to increased damping. However, due to the open loop shape the low frequency noise amplification effect is strong, limiting the

The transmissibility with each strategy was measured by exciting the system and measuring the response from x_1 to x_2, with results plotted in figure 4. The response in the absence of control is characterized by a sharp resonance peak at 103 Hz. All the tested strategies achieve a significant reduction of the resonance peak, with the best results achieved using the acceleration-based methods (VFB, VBP, NAF). When PPF with absolute position measurement is used, base vibration transmission at low frequencies is amplified. A significant transmissibility reduction is obtained with the feedforward strategy. Due to the simplistic nature of the method used, it is mainly effective in a narrow range of frequencies. This, however, is sufficient to improve the performance of the system significantly, as will be demonstrated.

Figure 5 shows the CPS of the response of the system to combined direct and indirect disturbances. Without control, large contributions are caused by the resonance peak at 103 Hz and high-frequency modes of the system around 1100 Hz. With active damping, the contribution of the resonance is significantly decreased. However, visible from the increase before the resonance frequency compared to the open loop, the influence of the noise on the system is amplified. Furthermore, the high-frequency parasitic modes at around 1100 Hz are excited by the controllers, leading to a slight increase in vibrations. This shows the trade-off existing with these controllers: an increase in gain leads to both increased damping, amplified influence of noise and excitation of high-frequency modes. This behaviour limits the achievable reduction of vibrations.

The performance of the PPF using absolute position measurements approaches the performance of velocity feedback. This is achieved despite the amplification of the base-vibration transmissibility and smaller resonance peak reduction, thanks to smaller excitation of the high-frequency dynamics.

Table 1. Numerical results of experimental evaluation of different AVC feedback strategies showing the RMS of the acceleration of the isolated mass $\ddot{x}_2$ of each strategy and the percentage with respect to no AVC.

Method	RMS of $\ddot{x}_2$ (m/s^2)	% of No AVC
No AVC	2.5007 x10^{-3}	100 %
VFB	1.1426 x10^{-3}	45.69 %
VBP	1.0572 x10^{-3}	45.27 %
NAF	1.0489 x10^{-3}	41.94 %
abs. PPF	1.1877 x10^{-3}	47.49 %
rel. PPF	1.7018 x10^{-3}	68.05 %

Due the shape of the open-loop gain, visible in figure 3 relative PPF is not able to achieve similar performance. It can be seen that relative PPF has high open loop gain at low frequencies without a significant peak at the resonance frequency. As a result, it is not possible to dampen the resonance without also strongly amplifying low-frequency disturbances. The results are summarized in table 1 which shows the results numerically in terms of the RMS of $\ddot{x}_2$.

Table 2. Numerical results of experimental evaluation of indirect disturbance rejection with feedback and feedforward showing the RMS of the position of the isolated mass x2 and the percentage with respect to no AVC.

Method	RMS of x_2 (m/s^2)	% of No AVC
No AVC	5.3778 x10^{-5}	100 %
FB only	1.5021 x10^{-5}	27.93 %
FB + FF	3.0984 x10^{-6}	5.76 %

The position of the isolated mass x_2 is used as a performance indicator when evaluating feedforward since position feedforward was implemented. To show the influence of feedforward on the reduction of indirect disturbances, only these were used to excite the system. In figure 6 the obtained CPS are plotted. While the feedback control largely removes the amplification of vibration due to the resonance, it does not influence the low-frequency vibration transmission. When feedforward is used, the low-frequency contributions are reduced, which results in an almost 95% decrease in the final vibration magnitude. This shows that even straightforward stiffness compensation feedforward can greatly improve systems performance. These results can be found numerically in table 2.

Conclusion

This paper investigated the use of ultra hard mounts based on piezoelectric stack actuators in applications requiring high position stability. Thanks to the high stiffness of the actuators and the application of active damping for resonance peak reduction, the influence of direct disturbance forces on such a system is greatly diminished. Even though the system is characterized by high resonance frequency, the transmissibility of the base vibrations is reduced with the feedforward techniques.

The performance of feedback and feedforward techniques, developed for other systems and utilizing various sensor types, was evaluated by implementing them on a high stiffness single-degree-of-freedom experimental test setup.

It was found that high degrees of damping can be achieved with acceleration-based feedback controllers. The amount of damping is limited by amplification of disturbances, and excitation of high frequency modes. At the cost of more elaborate tuning NAF and VBP outperform VFB. If absolute position measurements are available PPF can be used for performance approaching that of VFB. However, the in practice easier to obtain, relative position measurement cannot reach similar damping.

To solve the indirect disturbance sensitivity drawback of ultra hard mount systems, feedforward was implemented. It was shown that even with straightforward stiffness compensation feedforward, the transmissibility can be lowered significantly.

The outcomes of the study motivate further research on the concept of ultra hard mounts.

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