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CONTROL DESIGN FOR HARD MOUNT VIBRATION ISOLATION IN HIGH-PRECISION MACHINERY

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Abstract
Hard mounted vibration isolation systems can provide a stiff connection between a machine and its base. To achieve adequate vibration isolation from floor vibrations by such a hard mount system, a modified preconditioned FxLMS based adaptive feedforward controller is studied in this paper [1],[3]. It is shown that such a controller can be applied to poorly damped systems, but only in combination with a feedback controller that provides sufficient damping. Simulation results for a single axis setup demonstrate that the vibration isolation performance can be tuned using an error weighing filter [5].

INTRODUCTION

Vibration isolation systems are present in almost any high-precision machine to reduce acceleration levels in the machine. These accelerations are induced by floor vibrations and disturbance forces acting directly on the machine, e.g. acoustics, cables and internal stage motions. These vibration isolation systems provide a low-pass filter for floor vibrations, providing good isolation above the first resonance frequency (typically 1 Hz), which is referred to as the suspension mode. In many systems, additional active components are used to add (skyhook) damping near the suspension mode frequency. Due to their low stiffness, these systems are referred to as soft mount systems.

Unfortunately, their low stiffness leads to relatively large amplitudes of motion when the machine is subjected to static and low frequent direct disturbance forces. Moreover, settling times to disturbances are relatively long, due to the low suspension mode frequency, even when the suspension mode is actively damped. These drawbacks are becoming increasingly pronounced considering the ongoing demand for better resolution and increased throughput, for example in the semiconductor industry.

In this research project, we investigate the application of an active hard mount concept for vibration isolation in high-precision machinery. Hard mount systems provide a connection between the machine and the floor with a stiffness which is typically two orders of magnitude higher compared to soft mount systems. As a result, the aforementioned drawbacks can be eliminated. However, the suspension mode frequency of hard mount systems is higher due to their larger stiffness and, consequently, the passive isolation from floor vibrations is reduced. Hence, active components are now required to improve the isolation from floor vibrations as well as provide additional damping of the resonance modes.

Our objective is to determine design approaches for active hard mount vibration isolation systems for use in high-precision machinery. Hence, the mechanical design of these mounts as well as the control system design should be considered. However, in this paper we merely discuss the chosen control strategy, applied to a single axis laboratory setup with floor vibrations acting as the input disturbance.

Outline

Firstly, we present the control objectives and our control strategy. In the second section, this strategy is discussed in more detail. The third section is dedicated to the simulation and implementation of the control scheme, for the single axis laboratory setup. The paper concludes with a summary and some remarks on future work.

CONTROL OBJECTIVES AND STRATEGY

Figure 1 shows a simplified model of a machine supported by an active mount, which is represented by a suspension spring k_{susp} and a force actuator F_a. The machine model has an internal resonance mode due to the structural stiffness k_{struct}. Such structural modes are common in high-precision machines and are often poorly damped. The machine is subjected to a direct disturbance force $F_d(t)$ and floor vibrations $\ddot{x}_0(t)$. The actuator in the mount is controlled using feedback of the error signal $\ddot{x}_1$ as well as feedforward of the floor vibrations $\ddot{x}_0$. Note that acceleration sensors are used here.

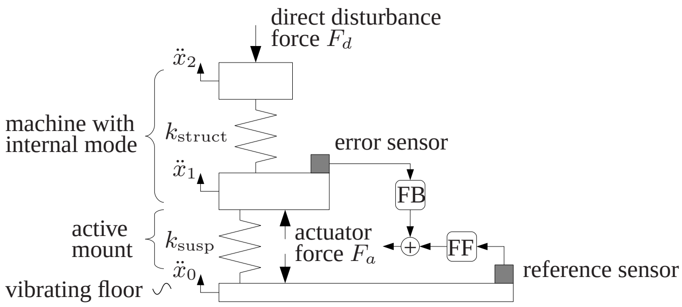


Figure 1: Simple representation of a machine with an internal resonance mode, supported by an active mount. The machine is subjected to a direct disturbance force and floor vibrations. The actuator is controlled using feedback (FB) and feedforward (FF).

Based on the foregoing discussion, three general control objectives can be formulated for active hard mount vibration isolation systems[1]:

- **Response to direct disturbances:** the robustness of the passive hard mount system to direct disturbance forces must be preserved in the controlled system.
- **Response to floor vibrations:** the transmissibility of floor vibrations (defined as $\ddot{X}_1(s)/\ddot{X}_0(s)$) must be similar to the low-pass characteristics of a soft mount system.
- **Damping of resonance modes:** the controlled hard mount system should increase the damping of the suspension mode as well as the structural resonance modes compared to the passive system.

Our control strategy is a combination of (adaptive) feedforward control using floor vibrations as the reference signal and feedback control of the remaining machine motion. The benefit of floor vibration feedforward control is that, in theory, it has no influence on the system response to direct disturbances. Moreover, we expect to be able to achieve better vibration isolation, because feedforward control allows for a larger control bandwidth. Feedback control is used to add artificial damping to the suspension mode and the structural mode, which basically requires velocity feedback in the relevant frequency range [4].

ADAPTIVE FEEDFORWARD VIBRATION ISOLATION FOR A FEEDBACK CONTROLLED PLANT

Our feedforward control strategy is based on the Filtered-reference Least Mean Square (FxLMS) adaptive algorithm for finite-impulse-response (FIR) filters. By using an adaptive controller, model uncertainty, nonlinearities, time-varying system parameters and non-stationary signals can be coped with better. More importantly, no knowledge is required of the transfer function between the vibration source and the measured error signal, which may be very difficult to obtain. Moreover, FIR filters are unconditionally stable and the FxLMS algorithm for this type of filters is linear-in-the-coefficients, which leads to an efficient numerical implementation.

An in-depth discussion of the FxLMS algorithm is beyond the scope of this paper, but many excellent references are available in literature, see e.g. [2],[6] and the references therein. Here, we only discuss the very slow convergence of the algorithm for poorly damped systems and the methods that we use to improve the convergence speed. In general, the FxLMS algorithm is not suitable for poorly damped systems, for two reasons:

- The convergence speed reduces for a larger eigenvalue spread of the autocorrelation matrix of the filtered reference signal. This eigenvalue spread, in turn, depends on the reference signal's spectral density and the dynamics of the plant under control. For poorly damped systems, this leads to an ill-conditioned autocorrelation matrix of the filtered reference signal and, hence, a large eigenvalue spread.

[1]Note that specific control objectives depend heavily on the application (e.g. required accuracy, disturbance characterisation)

- A large number of filter coefficients (L) is required, as the filter coefficient vector must estimate the (very long) impulse response of the optimal filter. Not only is the convergence speed greatly reduced for large L, implementation of the controller is likely to be impossible due to computational limitations.

Actively increasing the damping in the system is the solution to these issues for this field of application. As mentioned before, a feedback controller should provide additional damping. Then, the feedforward controller acts on a well-damped closed loop system. As a result, the required number of feedforward filter coefficients can be reduced by a factor 10–100. Hence, feedback control is crucial to the application of adaptive feedforward FIR controllers in hard mount vibration isolation systems.

The problem of slow convergence due to the plant dynamics still remains. To overcome this problem, we combine the preconditioned FxLMS [3],[6] with the modified FxLMS [1]. Figure 2 shows a block diagram of this modified preconditioned FxLMS algorithm acting on a system $S(z)$ which is controlled by a feedback controller $C(z)$. $W(z)$ denotes the feedforward FIR filter.

The gray blocks represent the mechanical system (see figure 1). Note that influence of the direct disturbance force F_d is omitted here. The *primary path* $P(z)$ is defined as the transfer from the floor vibrations $\ddot{x}_0$ to the disturbance acceleration $\ddot{x}_{1d}(k)$. The *secondary path* $S(z)$ is defined as the transfer from the actuator force $F_a(k)$ to the acceleration $\ddot{x}_{1c}(k)$.

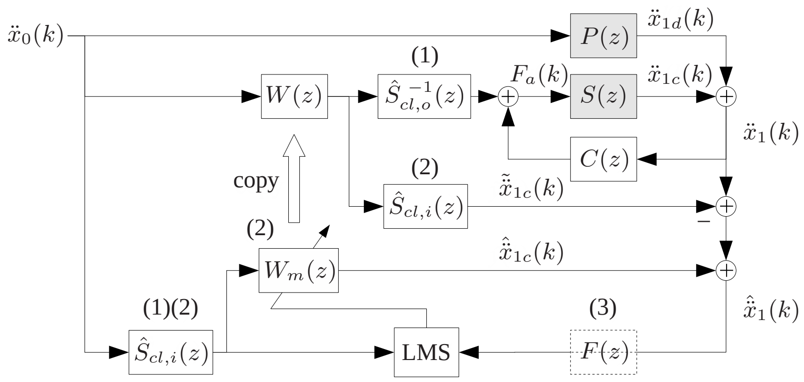


Figure 2: Modified preconditioned FxLMS-based feedforward vibration isolation control scheme for a feedback controlled system $S(z)$, with possible error weighing $F(z)$. $C(z)$ denotes the feedback controller. $W(z)$ denotes the feedforward FIR filter.

The preconditioned FxLMS algorithm uses the inner-outer factorisation[2] of $\hat{S}_{cl}(z)$ to precondition the plant with the inverse of the outer factor, $\hat{S}^{-1}_{cl,o}(z)$. Here, $\hat{S}_{cl}(z)$ represents the closed loop model of the feedback controlled secondary path. This model can be obtained indirectly by identifying the open loop secondary path and closing the loop for a given feedback controller $C(z)$, or by directly identifying the closed loop secondary path.

[1]The inner-outer factorisation of a discrete-time system $G(z)$ is defined as $G_i(z)G_o(z) = G(z)$, such that the outer factor $G_o(z)$ has a causal, stable (right) inverse and the inner factor $G_i(z)$ is an all-pass system [7]

Using this precondition filter $\hat{S}_{cl,o}^{-1}(z)$, the reference signal only needs to be filtered by the inner factor $\hat{S}_{cl,i}(z)$, which is an all-pass system. This results in a faster convergence of the FxLMS algorithm, because the eigenvalue spread of the autocorrelation matrix of the filtered reference signal is significantly reduced. The additional blocks for the preconditioned FxLMS algorithm are indicated by (1) in the figure.

The modified preconditioned FxLMS algorithm uses an internal model $\hat{S}_{cl,i}(z)$ to reformulate the standard preconditioned FxLMS problem into a general adaptive filter problem for an auxiliary feedforward controller $W_m(z)$. The filter coefficients of $W_m(z)$ are then copied to the actual controller $W(z)$ at each sample instant k. This general adaptive filter problem generally provides faster convergence than the FxLMS algorithm. The additional blocks for the modified FxLMS algorithm are indicated by (2) in the figure.

An additional error weighing filter $F(z)$ is indicated by (3) in the figure. In the next section we will use this filter to shape the performance of our control system. See also [5], where this technique is referred to as residual noise shaping. When using error weighing, the inner and outer factors $\hat{S}_{cl,i}(z)$ and $\hat{S}_{cl,o}(z)$ must be calculated from the series connection of the error weighing filter and the closed loop model of the secondary path: $F(z)S_{cl}(z)$.

SIMULATION AND EXPERIMENTAL RESULTS

A schematic drawing of our single axis laboratory setup is shown in figure 3. The setup consists of horizontally linear guided masses which model a machine and its supporting floor, equivalent to figure 1. The floor mass is excited by a shaker. The “machine” is connected to the floor mass by a hard mount, which consists of a prestressed piezoelectric actuator, in series with an elastic element to obtain a suspension mode frequency of 35 Hz. Charge accelerometers are used to measure the acceleration of each mass. A dSPACE DS1005 system is used for real-time control and data acquisition, at a sample rate of 2 kHz.

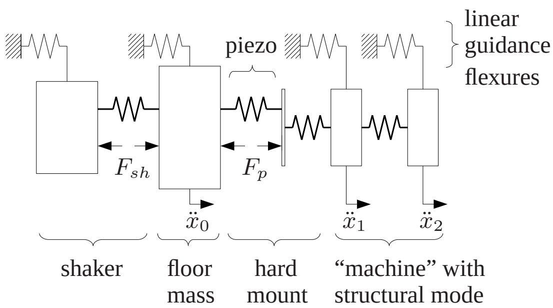


Figure 3: Schematic drawing of the single axis laboratory setup

The shaker steering signal is used as the reference signal for the feedforward controller instead of the floor mass acceleration, because the steering signal is conveniently generated in the controller. The measured error signal is the acceleration $\ddot{x}_1$. We have obtained models of the primary and secondary paths (now $\ddot{X}_1(z)/F_{sh}(z)$ and $\ddot{X}_1(z)/F_p(z)$ respectively) using subspace model identification (SMI) [8]. To obtain the identification data sets, the primary and secondary paths are excited sequentially using pseudo-random binary sequences.

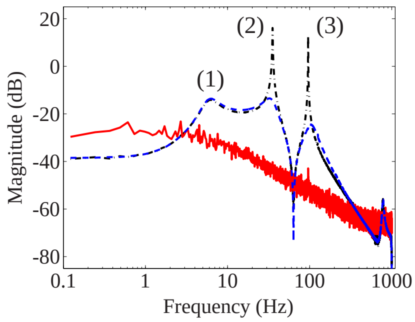
Controller Design & Simulation

The feedback controller was designed by loop shaping of the secondary path. The designed controller is a lag filter, combined with 2^{nd} order high-pass and low-pass filters. By closing the feedback loop, the damping of the "machine" suspension mode is increased from 0.4% to 18% and the damping of the structural mode is increased from 0.1% to 15%. As a result, the number of feedforward filter coefficients can be reduced from 22000 (!) to 1000, thus allowing implementation of the control scheme on the dSPACE system.

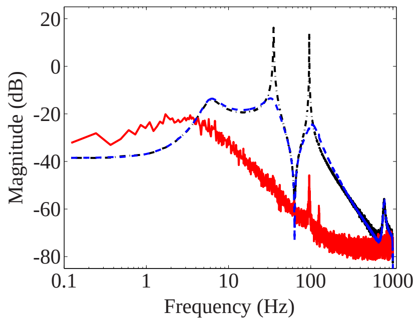
In figure 4(a), simulation results are shown for the feedback controlled system (dotted) and the combined feedforward and feedback controlled system (solid). The uncontrolled system response is also plotted for comparison (dash-dotted). The most important resonance modes of the setup are indicated in the figure:

(1) shaker resonance mode (6 Hz)
(2) machine suspension mode (35 Hz)
(3) machine structural mode (95 Hz)

The feedback controller reduces the measured acceleration level from 85 mm/s^2 (rms) by 15 dB to 16 mm/s^2 (rms). The feedforward controller combined with feedback converges in approximately 2 seconds and achieves 33 dB reduction (2 mm/s^2 rms) after convergence.



(a) Without error weighing



(b) With error weighing

Figure 4: Simulation results – amplitude frequency responses of $\ddot{X}_1(z)/F_{sh}(z)$ for the modified preconditioned FxLMS combined with feedback: a) without error weighing, and b) with error weighing. The solid line shows the feedforward and feedback controlled system. For comparison, the feedback controlled system (dashed) and the uncontrolled system (dash-dotted) are also plotted.

Even though the reduction of vibration energy is remarkable, there are two points which need improvement. Firstly, the transmissibility only drops by 20 dB/decade starting from 5 Hz, whereas soft mount isolation systems generally achieve 40 dB/decade roll-off. Secondly, floor vibrations are amplified below 3 Hz.

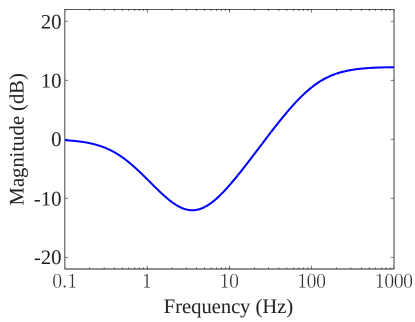


Figure 5: Amplitude frequency response of the error weighing filter which is used to achieve more high-frequency suppression

To improve the isolation performance at high frequencies, an error weighing filter is designed (figure 5). Figure 4(b) shows the performance of the controller with this error weighing filter. Now, the transmissibility drops by approximately 40 dB from 10 Hz to 100 Hz and the overall reduction becomes 30 dB. Note that the amplification at low frequencies has become slightly worse.

Experimental results

Straightforward implementation of the control scheme on the dSPACE hardware results in actuator saturation. To reduce the actuator signal to acceptable limits, two changes are made to the control strategy. Firstly, the summed squared filter coefficients are also weighed in the minimisation criterion, resulting in a leaky FxLMS algorithm [2]. In addition, the outer factor $\hat{S}_{cl,o}(z)$ is regularised (see [6] for details).

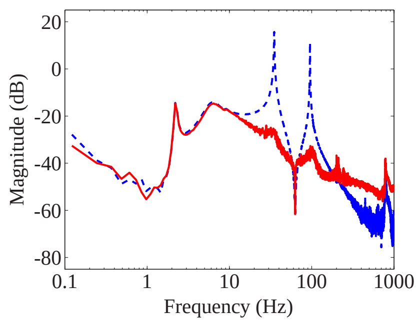


Figure 6: Experimental results – amplitude frequency responses of $\ddot{x}_1$ to F_{sh} for the modified preconditioned FxLMS with error weighing combined with feedback, with regularised outer factor and leaking filter coefficients. The solid line shows the controlled system, the dashed line shows the uncontrolled system.

Of course, these changes result in a degraded performance, which is shown in figure 6. The desired 40 dB/decade roll-off is clearly not achieved. Moreover, with these changes, an overall reduction of 20 dB is obtained. Note that the amplification of low frequent vibrations is eliminated using this approach. The resonance at 2.2 Hz, which is not identified by the SMI routine, is the setup suspension mode. Below this frequency, measurement noise is the dominant factor in the acceleration signal.

SUMMARY

Adaptive feedforward control based on the FxLMS algorithm can be applied for vibration isolation of poorly damped systems, but only in combination with feedback control that provides sufficient damping. To achieve an acceptable convergence speed, a modified preconditioned FxLMS algorithm is proposed. To shape the performance of the control system, error weighing can be applied, which is shown in simulation for a single channel laboratory setup.

To prevent actuator saturation in the actual implementation, the control scheme must be changed, which causes a significant loss of performance. Future work will focus on these implementation issues. We expect that model reference adaptive control, using the floor mass acceleration signal as a reference signal instead of the shaker steering signal, will solve these problem.

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