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A NEW MULTI-DEGREE-OF-FREEDOM ACTIVE VIBRATION ABSORBER

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1. INTRODUCTION

Dynamic vibration absorbers have long been successfully used in industry to attenuate structural vibration for both narrow and broad-band excitations [1]. More recently, the performance of these passive devices has been enhanced through the addition of an active component. This active component consists of a self-balanced force pair applied between the absorber mass and the structure to be controlled. These hybrid passive/active dynamic vibration absorbers provide improved response attenuation over that achieved by the passive system at the expense of the energy added to the system via the control force [2]. The required control effort is a function of both the actuator mass and the tuning. Consequently, a smaller actuator mass leads to larger control forces, and a detuned actuator requires larger forces than a tuned one. In some applications, these drawbacks limit the implementation of hybrid absorbers due to constraints in size, weight, and/or required control force magnitudes.

In this paper, a new multi-degree-of-freedom hybrid dynamic vibration absorber configuration is proposed. This new passive/active device requires smaller active forces than traditional reaction-mass actuators to achieve the same structural vibration attenuation [3]. The performance of the proposed actuator is evaluated experimentally by controlling a fixed-free beam excited with white noise. The results are compared to those obtained using a traditional single reaction-mass actuator like those presently used in industry.

2. DUAL-MASS ACTUATOR

A diagram of the proposed actuator concept is shown in Fig. 1a. The new actuator consists of two reaction masses, m_{a1} and m_{a2}, attached to the primary structure which is to be controlled via coupling impedances, and of the self-balanced control forces, $F_c(t)$, applied between the reaction masses. This configuration, referred as a dual-mass (DM) actuator, should be contrasted to the single-mass (SM) actuator in Fig. 1b, where the control forces are applied between the reaction-mass and the primary structure. The proposed actuator concept can be extended to more than two masses.

It is easy to show that both the SM and DM passive dynamic absorbers increase the input impedance seen by the force, $F(t)$, when the excitation frequency is at or near the resonance(s) of the absorber. The control forces, $F_c(t)$, can then be interpreted as actively increasing the input impedance over a wider frequency range. Analytical work has demonstrated that the proposed DM actuator requires smaller control forces than the SM actuator to increase this input impedance [4]. The purpose of this work is to demonstrate this fact experimentally.

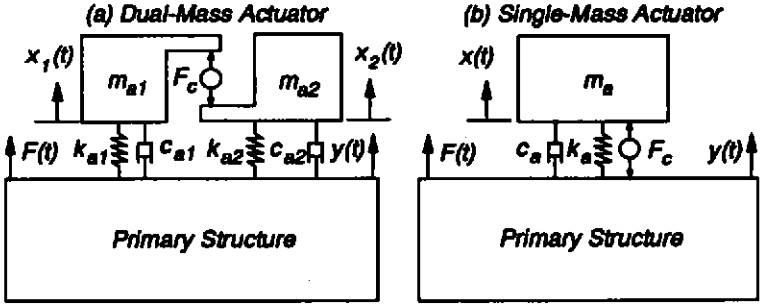

Fig. 1 Schematic of DM and SM Passive/Active Actuators.

3. EXPERIMENTAL SETUP AND PROCEDURE

The performance of the proposed DM actuator was tested using the experimental setup shown in Fig. 2. A steel beam (25 x 38 mm cross section and 1.15m length) was clamped to a granite table. The beam was excited using a Ling Dynamic Systems shaker driven with white noise over the [0-200 Hz] band. The shaker was attached 0.64 m from the base of the beam. A PCB force transducer was used to measure the excitation force from the shaker. The error signal to be minimized was obtained from a PCB accelerometer mounted at the free end of the beam. The actuator was also mounted at the free end of the beam.

The DM actuator consisted of two steel masses attached to metal plates that were then mounted to a rigid housing. The thickness of the metal plates dictated their spring constants. The control forces were implemented with a voice coil/rare-earth magnet [3]. For convenience, the SM actuator was implemented by replacing one of the metal springs with

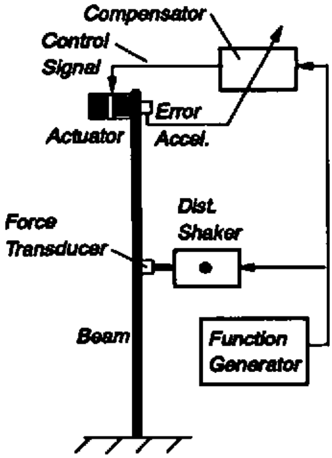

Fig. 2 Schematic of Experimental Setup.

a thicker plate which effectively locked one of the reaction masses to the primary structure. Although this halved the SM absorber mass, the effect of a reduced reaction mass on the required control force is only important at the absorber's resonance [4].

The control approach used was the adaptive Filtered-X LMS algorithm where the output from the function generator was used as the reference signal. The compensator was implemented with a finite impulse response filter of order 255 and with a sampling frequency of 800 Hz. The first three natural frequencies of the beam were at 13, 86, and 242 Hz. The properties of both the DM and the SM actuators were selected to attenuate the second mode of the beam. To this end, the SM actuator was tuned to 82 Hz with a damping ratio of 3.6%. The tuning ratio of $f_a/f_2 = 0.95$ was nearly optimal for the actuator-beam mass ratio of 3% [1]. The DM actuator was designed to have natural frequencies below and above the beam's second resonance. The resonances of the DM actuator were at 61 and 96 Hz, with modal damping ratios of 4.5 and 2.6%, respectively. The total mass ratio for the DM actuator was 5.4%. In contrast to the SM actuator parameters, none of the DM actuator parameters were optimal. The optimization of the DM actuator is being presently investigated and will be presented in future publications.

4. RESULTS

The spectra of the error accelerometer for the beam with and without the actuator and configured in both the DM and SM modes were measured. The control voltage driving the actuator in each mode was also measured. The results were normalized by the input disturbance force.

Fig. 3a shows the normalized error signal from the original beam and from the beam with the passive actuators. This figure shows similar attenuation levels obtained with both passive devices. Fig. 3b shows again the error spectrum of the original beam and of the actively controlled beam-actuator systems, i.e. passive/active actuators. This plot demonstrates that both actuators successfully attenuated the error signal in the proximity of the second resonance of the beam. To quantify the achieved reduction over the [20-200 Hz] band, the root mean squared (RMS) value of the controlled error signal was calculated by integrating the spectrum over this frequency range. The response was attenuated by factors of 3.5 and 3.7 for the passive SM and DM actuators, respectively. Attenuation factors of 11.5 and 12.3 for the passive/active SM and DM actuators were obtained, respectively. Thus, the RMS value of the error signal was reduced by an additional 7% using the DM actuator as compared to the SM actuator.

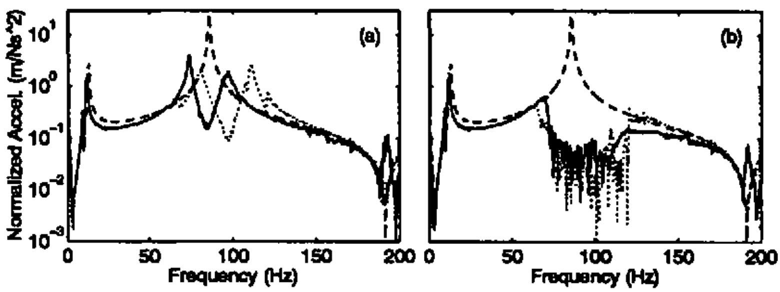

Fig. 3 Normalized Error Signals (- - - Original System —— SM Actuator · · · DM Actuator) for (a) Passive System and (b) Passive/Active System.

The spectra of the control signals required by each actuator, normalized by the shaker force, are shown in Fig. 4. The control force for the SM actuator was higher than that required by the DM actuator over almost the entire frequency range. Thus, the DM actuator provided better error signal reduction with less control force than the optimally tuned SM actuator.

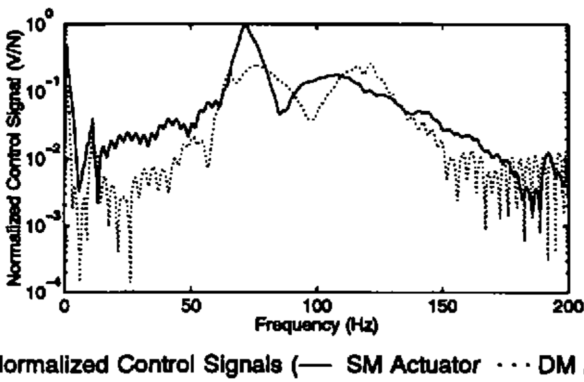

Fig. 4 Normalized Control Signals (—— SM Actuator · · · DM Actuator).

The spectra presented in Fig. 3b and 4 correspond to fully converged compensators. The performance of the DM and SM actuators in various stages of the convergence process is summarized in Fig. 5. This figure shows the error signal RMS value integrated over the [20-200 Hz] band as a function of the control signal RMS value integrated over the same frequency range. For all error signal reductions, the DM actuator required a lower control effort than the SM actuator. For example, to reduce the error signal RMS value to 5.8 m/Ns^2, the DM actuator required 1.9 compared to 4.2 Volts/N (RMS) needed by the SM actuator, a reduction in required control signal of 6.9 dB.

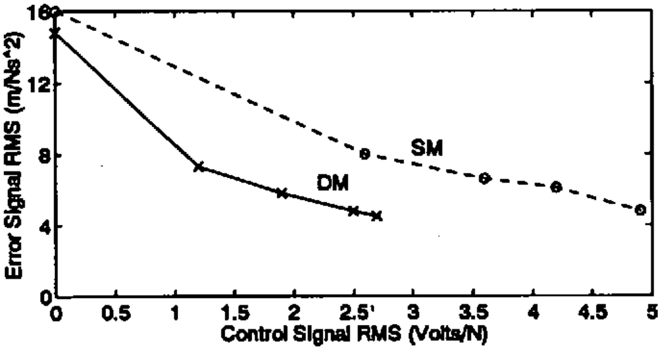

Fig. 5 Comparison of Achieved Error vs. Applied Control Signal Levels.

The time histories of the error and control signals were also recorded. Fig. 6 shows the error signal of the original beam. Fig. 7a and 7b show the error signal of the beam with the passive SM and DM absorbers, respectively. Both achieved similar error signal attenuation. Fig. 7c and 7d show the error signal with the passive/active SM and DM actuators, respectively, after full convergence of the compensator. Again, both actuators achieved similar attenuation of the error signal. Fig. 7e and 7f show the control signal required by the SM and DM actuators, respectively. The DM to SM control signal ratio was 0.52 for both peak and RMS values, showing that the SM actuator required nearly twice the control signal to achieve similar error signal attenuation.

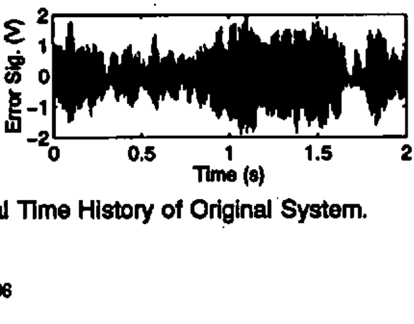

Fig. 6 Error Signal Time History of Original System.

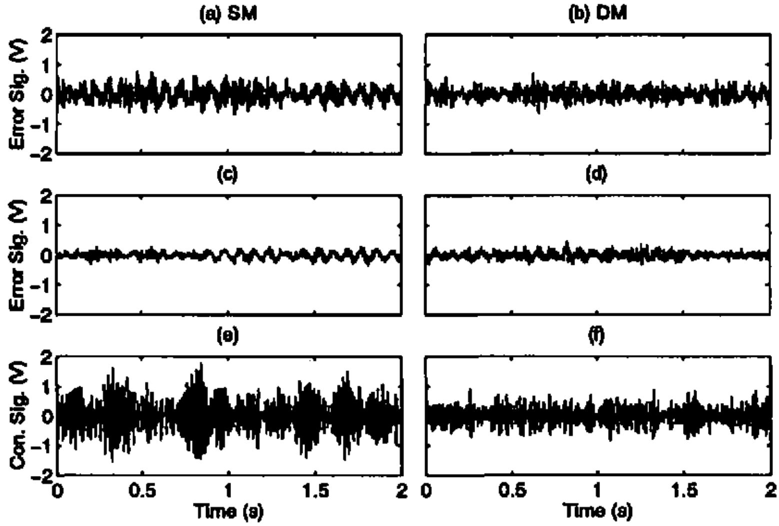

Fig. 7 Time Histories of Error and Control Signals.

5. CONCLUSION

A new active dynamic vibration absorber concept has been presented. The device consists of two or more reaction-masses with the control forces applied between them. The performance of this actuator was experimentally evaluated by controlling a fixed-free beam driven with band-limited white noise. An active single-mass absorber of the type presently used in industry was also tested and compared to the proposed dual-mass active absorber. The experimental results clearly demonstrated that the new dual-mass actuator produced better vibration attenuation with significantly less control effort, when compared to the single-mass actuator.

6. REFERENCES

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