



PHASE COMPENSATOR FOR HYPERSTABLE HYBRID MASS DAMPER

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ABSTRACT

In this paper, we propose and validate a simple control law, dedicated to hybrid mass dampers in order to improve stability and performance. A particular phase compensator is added to the original velocity feedback to correct the dynamics of the actuator face to the one of the controlled structure. The resulting system is hyperstable theoretically. The main interest of this kind of devices is its fail-safe property which is essential for aerospace applications. Theoretical analysis and experimentation illustrate this hybrid control device and its performances.

1 INTRODUCTION

Usually, when inertial actuators are used to actively control structures, the resonance frequency of the actuator is much lower than the fundamental resonance frequency of the controlled structures. The resulting device is called Active Mass Damper (AMD) and many control strategies have been developed [1, 6, 9]. Some approaches consider the problem of the tuning and the possible vicinity of the actuator resonance frequency to the one of the main structure. A compensator in the feedback loop [4, 8] is introduced to actively soften the actuator. But the pole-zero cancellation principle on which they are based presents some known dangers. Another class of dampers called Hybrid Mass Damper (HMD), or Hybrid Vibration Absorber (HVA) have recently appeared, trying to combine passive [5] and active vibration control. The objectives are: (i) to increase the performance, (ii) reduce the consumption on the considered bandwidth and (iii) to ensure a fail-safe behavior [2, 7].

In this contribution, we propose a simple control law previously theoretically introduced in [3]. We show that it improves the performance of classical hybrid dampers based on decentralized velocity feedback techniques. Actually, a compensator is introduced in the control loop to correct the phase of the actuator in order to become stable at the considered frequency. The resulting system is hyperstable [3] and fail-safe.

2 THE α -HYBRID MASS DAMPER

The section briefly presents the basic principles of the α -Hybrid Mass Damper. More details can be found in [3].

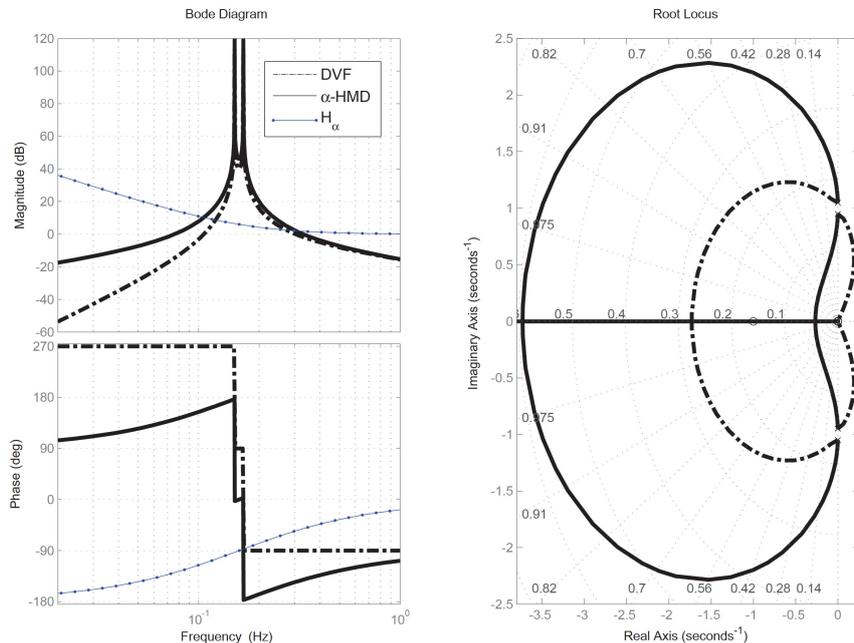


Figure 1: (a) Bode and (b) root locus plots of sensor-actuator open loop transfer function for Direct Velocity Feedback (black dashed line) and for α controller using $\alpha = \omega_0$ (continuous black line). Transfer function $H_\alpha(s)$ in blue dotted line.

Consider a system with a resonant frequency of $\omega_0 = \sqrt{k_1/m_1} = 1 \text{ rad.s}^{-1}$ without damping. To this system, a classical dynamic vibration absorber is associated (mass ratio $\mu = 1\%$). Usually, AMD are used with Direct Velocity Feedback (DVF) law. A control law proportional to the measured velocity of the main structure is generated to drive the actuator.

The open loop transfer function and the root locus are plotted in fig 1 (black dashed lines). By analyzing the stability margins, we see that the system is stable only at very low feedback gain. One sees also on the root locus that the lower frequency pole goes immediately in the right half plane, leading to instability. The closed loop system will always be marginally stable. This is mainly due to the absence of zero between the pole of the TMD and the pole of the structure.

A simple alternative to recover stability is to adequately place a pair of zeros at the right frequency. The controller is still a velocity feedback, however, a filter named α -controller is added in the control loop [3]:

$$H_{\alpha}(s) = g \frac{(s + \alpha)^2}{s^2} \quad (1)$$

The phase has been modified below the first resonant frequency (see its transfer function in fig 1 (a), blue dotted line). The parameter α is tuned to make the controller hyperstable. In this study, its value is $\alpha = \omega_0$. The rootlocus of the α -HMD is plotted in fig 1(b) (black continuous line). We can see that the whole root locus plot is in the left half plane, meaning an unconditional stability of the feedback system (infinite gain margin). More analysis and details can be founded in [3].

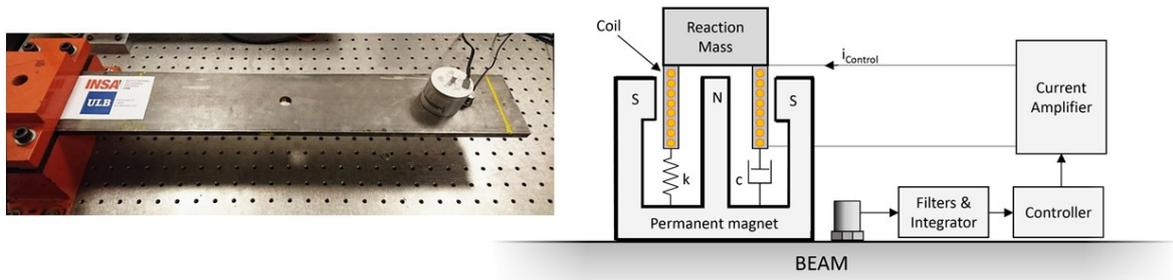


Figure 2: Picture of the experimental set-up used to test the proposed controller and the schematic of the control device

3 EXPERIMENTAL SET-UP

The structure and the control device are shown in Figure 2. The targeted mode is the first bending mode of the beam. The control device is Micromega products initially designed for purely active control (ADD-5N). Its reaction mass is 160gr, its frequency is around 21Hz and its damping of $\xi = 11.9\%$. The main structure used for the validation is a cantilever steel beam (Length: 58cm, width: 10cm, thickness: 1cm). An accelerometer is fixed nearby the actuator to feed the controller.

4 PERFORMANCES AND CONCLUSIONS

Figure 3 shows the effects of the proposed HMD in terms of FRF and integrated RMS value. The first one shows the damping introduced by the control on the targeted mode and the second graphic shows its wide range effect. Indeed, contrary to passive TMD, more than one mode is damped. Note that the damping on the first mode without any control is 0.24%; with the passive device it is around 9% due to the high mass ratio, and with the α -controller it reaches more than 16% for both resulting peaks.

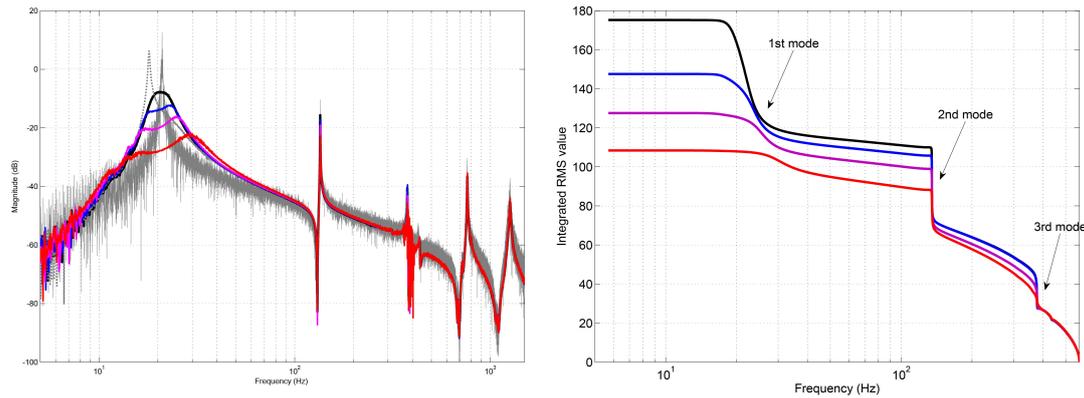


Figure 3: (a) Frequency response functions and (b) Integrated RMS value $[0 - 500] Hz$, without control (grey), with passive TMD (black), with DVF (grey dotted line) and with α -controller using various gain ($g = 1$ in blue, $g = 3$ in purple, $g = 9$ in red).

These figures validate the proposed robust hybrid mass damper. It combines two features: an unconditional stability, and a fail-safe characteristic by modifying the classical DVF law on a TMD.

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