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Robust hybrid mass damper

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ABSTRACT

In this paper, the design of a hybrid mass damper (HMD) is proposed for the reduction of the resonant vibration amplitude of a multiple degree-of-freedom structure. HMD includes both passive and active elements. Combining these elements the system is fail-safe and its performances are comparable to usual purely active systems. The control law is a revisited direct velocity feedback. Two zeros are added to the controller to interact with the poles of the plant. The developed control law presents the particularity to be simple and *hyperstable*. The proposed HMD is compared to other classical control approaches for similar purpose in term of vibration attenuation, power consumption and stroke.

1. Introduction

Tuned mass dampers (TMD) or dynamic vibration absorber (DVA) are small oscillators, appended to a primary structure to dissipate the vibrational energy of a specific resonance. The main reduction of the primary structure oscillation amplitude is achieved by the spring force (or the inertia force) of the TMD and the viscous damper in the TMD makes the rest and also makes the TMD insensitive to changes in the disturbing frequency (see [1–5]). Since their first invention, these practical and robust devices have found numerous applications ranging from civil structures to precision engineering. The beauty is that the same design rules can be used to damp nanometer vibrations or meter vibrations. The most popular is the so-called Den Hartog's method [6]; other interesting rules can be found in [7,8]. Only the technology changes. The major drawback of TMD is that they are tuned to damp only one specific resonance. To some extent, broadband dampers (e.g. mass mounted on a layer of elastomer) may damp several resonance modes, with a limited efficiency. A more efficient method to treat several resonances is to introduce an actuator which controls the force that the reaction mass applies on the structure. The resulting device is often called an active mass damper (AMD). In order to add viscous damping to the structure, the natural way is to drive the actuator with a signal which is proportional to the velocity. However, for large values of the control gain, the AMD poles tend to be destabilized, even when the structure velocity is measured near the AMD. It is believed that this is the reason why many elaborated control strategies have been introduced to control AMDs (although it is rarely openly confessed): classical tuning [9], fuzzy controllers [10–12] pole placement [13] Lyapunov's method [14] optimal control [15–18], μ-synthesis [19], H-infinity [20] or sliding mode control [21].

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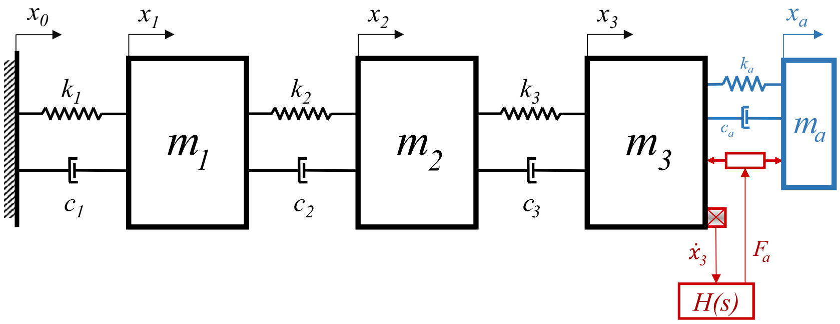

Fig. 1. Simplified model of a flexible structure with an AMD/HMD. (Blue: passive part, red: active part). (For interpretation of the references to color in this figure caption, the reader is referred to the web version of this paper.)

On the other hand, a novel class of AMD has appeared which are trying to combine several objectives at the same time. These devices are gathered under the common name of hybrid mass damper (HMD), or hybrid vibration absorber (HVA) even though the objective pursued may significantly differ from one HMD to the other. For example, in [18], a HMD is presented, where an optimal control is used to combine structural damping with a restricted stroke of the actuator. In [22] a H-infinity optimal control is used to minimize both the response and the control effort.

Among the HMD, several of them have the property of being fail safe, which means that they still behave as TMD even when the feedback control is turned off. In [23] the control relies on the pole placement technique. In [24], a dual loop approach is preferred to increase the stability margins. In [25], a two degree of freedom system is studied, which can behave as an active mass damper to suppress the vibrations induced by small earthquakes, and as a tuned mass damper to suppress the vibrations of a targeted mode excited by a big earthquake. In the active configuration, a linear quadratic regulator is used with a large gain on the structural velocity. The control is switched off above a threshold value of the structure displacement, i.e. when the actuator cannot deliver the requested force anymore.

The controller proposed in this paper is fail safe and unconditionally stable. The strategy consists of placing a pair of zeros adequately in the controller in order to obtain interacting poles and zeros in the open loop transfer function. Besides these assets, the control approach proposed requires a low consumption and is extremely efficient under harmonic excitation, and is extremely simple compared to controller found in the literature.

The paper is organized as follows. Section 2 presents the simplified flexible structure which is used to illustrate the controller efficiency. Section 3 contains the analysis of the stability of active/hybrid mass damper. Sections 4 and 5 present the concept of the proposed controller and its design laws. Section 6 discusses the performances and limitations of the new controller and compares it to others. Section 7 draws the conclusions.

2. Description of the simplified flexible structure

Consider the system shown in Fig. 1. It represents a flexible structure, excited from the base x_0. The mass suspended has been divided in three parts in order to include three resonances. In the remaining of the paper, the following numerical values have been used: $m_1 = m_2 = m_3 = 100$ kg, $k_1 = k_2 = k_3 = 4 \cdot 10^6$ N/m; leading to the following eigen-frequencies $f_1 = 14.1$ Hz, $f_2 = 39.7$ Hz, $f_3 = 57.3$ Hz. Dashpot constants c_1, c_2, c_3 are tuned in order to provide a modal damping ratio of $\xi = 0.01$ to all modes.

A HMD is appended on m_3, as shown in Fig. 1. The passive part of the HMD (in blue) consists in a mass m_a, a spring with a coefficient k_a and a damper with a viscous damper coefficient c_a. The values of these parameters depend on the operating mode (see Section 3). The active part (in red) consists in an absolute velocity sensors mounted on m_3, a controller $H(s)$ and an actuator F_a between m_3 and m_a.

3. Stability analysis of active/hybrid mass damper

Consider the active mass damper (AMD) as shown in Fig. 1. Typically, the natural frequency of the device (m_a, c_a, k_a) is chosen significantly lower than the frequency of the first mode of the structure. The following numerical values have been used: $m_a = 9.2$ kg, $k_a = 2861$ N/m, $c_a = 197.8$ N s/m. Under these conditions, the AMD behaves as a nearly perfect force generator in the whole frequency range containing the dynamics of the structure. In order to damp the structural resonances, the actuator is driven by a force proportional to the absolute velocity of the structure measured on mass m_3

$$F_a(s) = H(s)\dot{x}_3(s) \tag{1}$$

where $H(s)$ is a simple gain.