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# Semi-active vibration absorber based on real-time controlled MR damper

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ABSTRACT

A semi-active vibration absorber with real-time controlled magnetorheological damper (MR-SVA) for the mitigation of harmonic structural vibrations is presented. The MR damper force targets to realize the frequency and damping adaptations to the actual structural frequency according to the principle of the undamped vibration absorber. The relative motion constraint of the MR-SVA is taken into account by an adaptive nonlinear control of the internal damping of the MR-SVA. The MR-SVA is numerically and experimentally validated for harmonic excitation of the primary structure when the natural frequency of the passive mass spring system of the MR-SVA is correctly tuned to the targeted structural resonance frequency and when de-tuning is present. The results demonstrate that the MR-SVA outperforms the passive TMD at structural resonance frequency by at least 12.4% and up to 60.0%.



## 1. Introduction

The tuned mass damper (TMD) is a mass spring system with oil damper that is installed on the vibrating primary structure at anti-node position to reduce vibrations [1]. The classical design of the passive spring and passive viscous damper for the chosen mass of the TMD is given by the formulae according to Den Hartog [1]. Due to simplicity of TMDs, these passive damping devices have been employed in many different types of structures [2–4]. The vibration reduction of the primary structure due to the TMD is usually seen to be sufficient when the primary structure vibrates at the target resonance frequency to which the TMD was designed. However, the primary structure may also vibrate at other frequencies due to:

(a) *Forced vibrations*: The disturbing force excites the primary structure at a frequency that differs from the target resonance frequency.
(b) *Other resonance frequencies*: Another resonance frequency of the primary structure is excited than the resonance frequency that was used for the design of the TMD.
(c) *Time-varying target resonance frequency due to environmental impacts*: The resonance frequency of the target mode differs from the resonance frequency that was used for the design of the TMD due to temperature effects and/or life loads on the structure.

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In all these cases de-tuning between the TMD properties and the modal parameters of the primary structure is present that ends up in deteriorated vibration reduction efficiency [5–7].

This drawback of TMDs triggered the development of a vast variety of controllable tuned mass dampers of which some examples of active and semi-active solutions are briefly described subsequently; a comprehensive overview can be found in [8,9]. Active tuned mass dampers are based on different types of actuators [10]. The active solution allows realizing both the precise frequency and damping tunings at the same time but the power consumption might be significant. To overcome this drawback, many semi-active tuned mass damper (STMD) concepts have been developed [11–15]. These systems rely on actively controlled spring systems, shape memory alloys, Piezo stacks and other actuators that primarily target to adjust the frequency of the STMD to the dominant frequency of the primary structure. Controllable friction dampers have also been implemented in STMDs to control their energy dissipation or to increase the relative motion of STMDs for higher efficiency [16,17]. Another class of STMDs is based on the use of magnetorheological (MR) dampers [18–26]. In these devices, the real-time controlled MR damper enables to adjust the frequency and damping of the mass damper to the actual frequency of the primary structure. The reported improvement compared to the passive TMD as benchmark damper depends on the level of de-tuning and is up to 50% [18,24].

Whereas the natural frequency of TMDs and STMDs, respectively, is slightly below the target structural frequency according to the selected mass ratio, the natural frequency of the undamped dynamic vibration absorber (VA) is chosen to be equal to the frequency of the harmonic disturbing force that excites the primary structure [27,28]. Since the undamped VA, which was invented by Frahm in 1909 [27] and is well described in [1], is a mass spring system without damper, the vibration reduction performance of the undamped VA is very sensitive to frequency de-tuning which triggered the development of various controlled VAs. Actively controlled VAs are based on the integration of an actuator that enables precise frequency tuning of the active VA [29–35]. Adaptive VAs have also been realized on the basis of MR elastomers [36–40], electrorheological fluids [41], shape memory alloys [42], MR dampers to control the damping in the VA [43] and hybrid systems [44,45].

In this paper, a new type of a semi-active vibration absorber with MR damper (MR-SVA) is presented where the MR damper controls both the frequency and damping of the MR-SVA according to the actual frequency of the primary structure. A nonlinear damping control approach is formulated that takes the maximum relative motion constraint of the MR-SVA into account to secure that the MR-SVA can be installed on real structures where space is limited. The MR-SVA is numerically and experimentally validated for harmonic excitation and different levels of de-tuning between the MR-SVA and the primary structure. The structure of the paper is as follows: first, the main principles of the MR-SVA are explained and the experimental set-up is described. Then, the control algorithm of the MR-SVA is given. Sections 4 and 5 describe the numerical and experimental results and a summary closes the paper.

## 2. System description

### *2.1. Semi-active vibration absorber based on real-time controlled MR damper*

#### *2.1.1. Goal*

The undamped VA as described in Section 3.2 of Den Hartog's book [1] was invented by Frahm in 1909 [27]. It is a mass spring system without damper whose main characteristics are:

1. The natural frequency of the undamped VA is equal to the frequency of harmonic disturbing force.
2. The damping in the undamped VA is zero.

Due to these characteristics, the spring force of the undamped VA is at all instants equal and opposite to the disturbing force whereby the motion of the primary structure is fully cancelled. The goal of this paper is to describe a new type of a semi-active VA with real-time controlled MR damper (MR-SVA) that targets to realize the characteristics of the undamped VA *as close as possible* considering the semi-active constraint of the MR damper force. The MR-SVA is designed and validated for harmonic excitation [46].

#### *2.1.2. Main principle of MR-SVA*

The desired control force to be tracked in real-time with the MR damper is composed of a desired stiffness force and a desired friction force (Fig. 1(a)). The desired positive or negative stiffness force augments or diminishes the passive spring stiffness and thereby realizes the real-time frequency tuning of the MR-SVA. The desired friction force is minimized to generate as close as possible the target behaviour of the undamped VA. An adaptive nonlinear damping control algorithm guarantees that the relative motion amplitude of the MR-SVA mass does not exceed its maximum value due to spatial limitations.

The MR-SVA prototype is built in a rack to easily attach the MR-SVA to the primary structure (Fig. 1(b)). The MR-SVA mass $m_2 = 26.325$ kg is supported by four compression springs (Table 1). The resulting natural frequency $f_2 = 3.10$ Hz corresponds to the correct frequency tuning of the passive TMD according to Den Hartog [1]. The MR damper under consideration is a rotational type with a maximum torque of 45 N m at maximum current of 2 A. It is bolted to the roof of the rack and thereby acts on $m_2$ in parallel to the passive springs. The controlled MR damper torque is applied to the MR-SVA mass by a 150 mm