

DYNAMIC BEHAVIOR OF A TUNABLE MAGNETIC VIBRATION ABSORBER

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Summary A magnetic vibration absorber (MVA), completely relying on magnetic forces, is used to reduce the displacement of a vibrating structure. The distinctive feature of this absorber is the ability of tuning its linear stiffness together with its nonlinear cubic and quintic stiffnesses. Repulsive and corrective magnets are used to finely tune the values of these stiffness coefficients. A modelisation, relying on a multipolar expansion of the magnetic fields of each magnets, is used to predict the values of the stiffnesses from the geometry. Using only three geometrical parameters the MVA can be passively designed either as a nonlinear tuned vibration absorber (NLTVA), a nonlinear energy sink (NES), or a bi-stable absorber with negative linear stiffness.

INTRODUCTION

Due to its passive feature, the vibration mitigation of mechanical structures using vibration absorbers is an important solution for engineering issues. Linear vibration absorbers like the well-known Tuned-Mass Damper (TMD) [1, 2] have been studied for the vibration reduction of both linear undamped [3] and damped [4] primary structures (PS). However, the main drawback of the TMD is related to its narrow bandwidth of optimal control. To overcome this limitation a Nonlinear Energy Sink (NES) with an essentially nonlinear restoring force can be used. Having no natural frequency, the NES can thus adapt itself to the frequency of the PS [5, 6]. Another idea is to use the nonlinearity of the absorber in order to control the nonlinearity of the PS and has led to the concept of the nonlinear tuned vibration absorber (NLTVA). Recently, bi-stable vibration absorber devices have been investigated to make smaller the energy barrier required to activate the energy transfer between the PS and a NES [7]. If numerous experimental devices have been proposed to experiment these absorbers they are often designed for a specific application [8, 9, 10].

The proposed magnetic vibration absorber (MVA) has the ability of properly tuning its linear and nonlinear characteristics. Thus, this flexible device can be used either as an NES, an NLTVA, or a bi-stable vibration absorber, all these tunings being realized passively with a simple change in the geometry of the system. The experimental realisation of the MVA is presented in this paper. Then, preliminary results of the reduction of the PS vibration are presented when the MVA is tuned as an NLTVA, an NES and a bi-stable absorber.

THE MAGNETIC VIBRATION ABSORBER

The magnetic vibration absorber is shown in Fig. 1. It is composed of 7 permanent ring magnets. Their length, internal

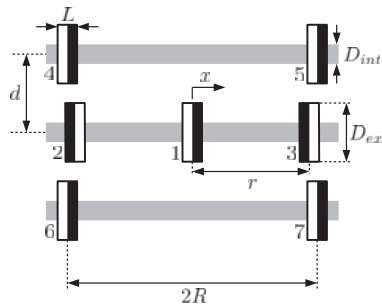


Figure 1: Diagram of the magnetic vibration absorber (MVA).

and external diameters are denoted L , D_{int} and D_{ext} respectively. Oscillating around the central position $x = 0$, the magnet 1 plays the role of the vibrating mass of the absorber. The other magnets (2, 3, 4, 5, 6 and 7) are fixed on plastic rods. The black and white parts of each magnet give the direction of their axial magnetisation. Magnets 2 and 3 are named the repulsive magnets since their magnetisation is the opposite of the magnetisation of the magnet 1. The amplitude of the repulsive force is tuned by modifying the geometric parameter r . Magnets 4, 5, 6 and 7 are named the corrective magnets since their magnetisation is the same that the magnetisation of the magnet 1. The amplitude of the corrective force is tuned by modifying

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the geometric parameters R and d . Therefore, the total force F_{Tot} applied on the mass of the MVA is tuned by the independent modifications of the geometric parameters r , R and d . Modeling the MVA using a multipolar expansion, this force can be written as

$$F_{\text{Tot}}(x) \approx -K_1 x - K_3 x^3 - K_5 x^5.$$

Modifying the geometric parameters r , R , and d , the linear, cubic and quintic stiffness terms can be tuned to change the properties of the MVA designing it as an NLTVA, an NES or a bi-stable absorber. More details about the design of the MVA can be found in [11].

VIBRATION MITIGATION

The MVA is then used to reduce the displacement of a plate selected as a PS vibrating with a large amplitude around one of its modal frequencies. Two opposite sides of the rectangular plate are clamped, whereas the other two are free. The device is attached to the PS and tuned successively as an NLTVA, an NES and a bi-stable absorber. Figure 2 shows the measured maximum displacement of the PS in each case and when the MVA is removed from the plate. These results present the effects of the linear stiffness coefficient tuning.

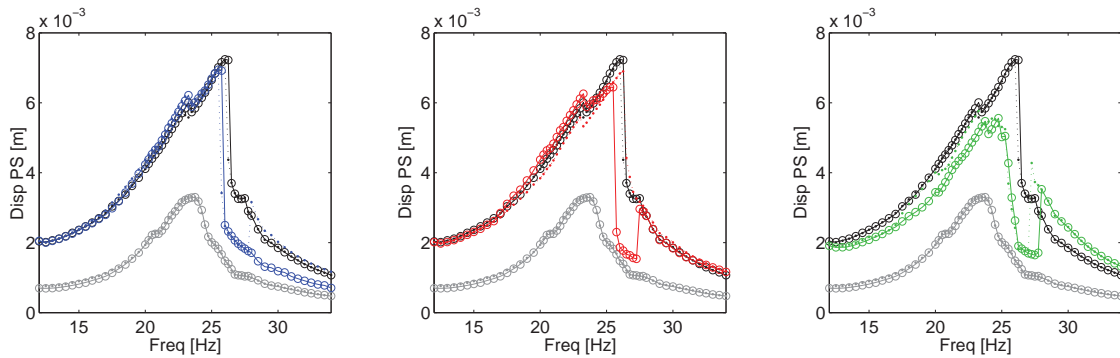


Figure 2: Maximum displacement of the primary structure without the MVA [ooo] and with the MVA tuned as an NLTVA [ooo] ($K_1 > 0$), an NES [ooo] ($K_1 \approx 0$) and a bi-stable absorber [ooo] ($K_1 < 0$). The gray circles [ooo] indicate the linear displacement of the primary structure for a lower excitation force. Circles and dots correspond to forward and backward sweeps respectively.

The measurements first show undoubtedly the effect of the vibration absorber on the nonlinear dynamics of the plate. Depending on the tuning, one can expect to control different parameters area in the frequency response curve. Current measurements are realised in order to analyze more deeply the advantages of each configurations as well as showing the best tuning that can be expected.

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