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Investigation into a nonlinear vibration neutralizer that incorporates magnets

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Vibration neutralizers are devices designed to reduce vibration levels within a host structure caused by external sources. Unlike tuned vibration absorbers, which target a resonance frequency of the host structure, vibration neutralizers are specifically tailored to suppress vibration at a harmonic frequency of the external excitation source. The design and application of these devices are critical, especially given the challenges posed by low-frequency vibration control, where either a large mass or a small stiffness is required. However, large masses are undesirable in practical applications, due to increased energy consumption, while low stiffness can lead to significant displacements, resulting in additional engineering challenges. In this study, a vibration neutralizer incorporating permanent magnets was constructed and tested. Different configurations of magnetic poles were explored to minimize the addition of mass to the system using a preload created by magnetic forces. The impact of static preload was investigated, showing that it provides a softening effect. A further investigation was carried out into alternative permanent magnet configurations to enhance the performance of the neutralizer by introducing hardening or softening effects, expanding the range of low frequency vibration control. The characterization of the nonlinear neutralizer shows that it could be a promising vibration control device.

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1. INTRODUCTION

Vibration neutralizers are devices used to manage vibration levels of a host structure due to external excitation. Typically comprising three key elements, a spring, a mass, and a damper, these devices are used to mitigate vibration and are often finely tuned to suppress vibration at the frequency of an external harmonic force. Over many years and with technological advancements, several strategies have been proposed to enhance the robustness of these devices, that is, their ability to maintain effective performance despite variations in the excitation frequency. One such strategy involves exploring nonlinear effects.

Recent advances in computational tools and the growing demand for improved structural performance have motivated an interest in incorporating nonlinearity, particularly stiffness nonlinearity, into engineering systems. Nonlinear absorbers have been widely investigated for passive vibration mitigation and motion isolation. Qian and Zuo[1] presented a comprehensive study on nonlinear vibration absorbers for vibration control, focusing on the nonlinear dynamics of a simply supported beam coupled to a spring–mass–damper type nonlinear energy absorber. Some studies discuss the incorporation of cubic nonlinearity into vibration neutralizers. This approach has been shown to improve robustness to mistuning and increase bandwidth without requiring additional mass[2]. Others have investigated the beneficial effects of geometric stiffness nonlinearity in case studies involving common structures[3].

In the context of geometric nonlinearity, some studies have investigated nonlinearity caused by magnets in a cantilever beam, where excitation is provided by an electrodynamic shaker, resulting in responses ranging from quasi-linear to strongly nonlinear[4]. Understanding nonlinear systems is crucial, as they can exhibit jump phenomena in the vibration response at certain frequencies under harmonic excitation. These jumps, which are characteristic in systems which have softening or hardening stiffness nonlinearities, occur when the system's nonlinearity and damping exceed specific thresholds[5]. In such cases, the jump-down frequency depends on both damping and nonlinearity, while the jump-up frequency is mainly governed by the nonlinearity. Aiello and Gatti carried out an experimental investigation into the effectiveness of a nonlinear neutralizer with hardening stiffness on the motion of a linear single-degree-of-freedom host structure[6]. The neutralizer was designed to exhibit cubic stiffness through a specific geometric arrangement. Their results show that at low excitation levels, the system behaves linearly, but as the excitation increases, the frequency response reveals characteristic nonlinear phenomena such as peak bending, jump phenomena, and response instabilities.

To further address the limitation of narrow frequency bandwidth, recent studies have proposed adaptive tuned vibration absorbers (ATVAs), which incorporate variable stiffness elements that can be adjusted in real time[7]. This approach enables the absorber or neutralizer to remain effective across a range of excitation frequencies by automatically tuning its dynamic properties.

Other systems exhibit complex behaviour, for example a beam-spring two-degree-of-freedom magnetically coupled bistable energy harvester that combines the benefits of multistable and multimodal harvesters[8]. Another energy harvester system introduces a centrifugal softening effect (CSE) and magnet-induced nonlinearity[9]. In this paper, a nonlinear vibration neutralizer involving a beam and permanent magnets is developed and tested. The magnets modify the effective stiffness of the beam, lowering the tuned frequency without adding significant mass. While most previous studies have focused on geometric stiffness or bistable configurations solutions that often require additional mass, complex mechanisms, or external power the present work introduces a fully passive alternative that exploits magnetic interaction to provide tunable nonlinear stiffness. This enables efficient vibration mitigation over a wider frequency range and highlights the potential of magnet-based devices as lightweight and adaptive solutions for engineering structures.

2. DESIGN OF A VIBRATION NEUTRALIZER

Figure 1 depicts the vibration neutralizer considered in this work. The device is shown in the left part of the figure and consists of a case (base) which can be attached to a host structure. Inside the case, there is a beam (elastic element) connected by one of its ends. At the free end of the beam there is a mass consisting of a support and three permanent magnets, one magnet is positioned at the beam centre and two at each side. On the case, near the free end of the beam, there is space for three magnets (Position P1, P2 and P3), one at the centre (P1) and two at each side (P2 and P3). The magnets can be configured in different ways according to their polarity, attracting or repelling each other.

Figure 1 also shows a simplified version of the linear model of the neutralizer on the right part of the figure, considering a rigid base. The base has mass m_b and the beam tip mass (support plus the permanent magnets) has mass m_t. The elastic element is the beam which has stiffness k. The amplitude of the base acceleration A_b and beam tip acceleration A_t can be related to the force applied at the base, in the frequency domain, using eq. (1)

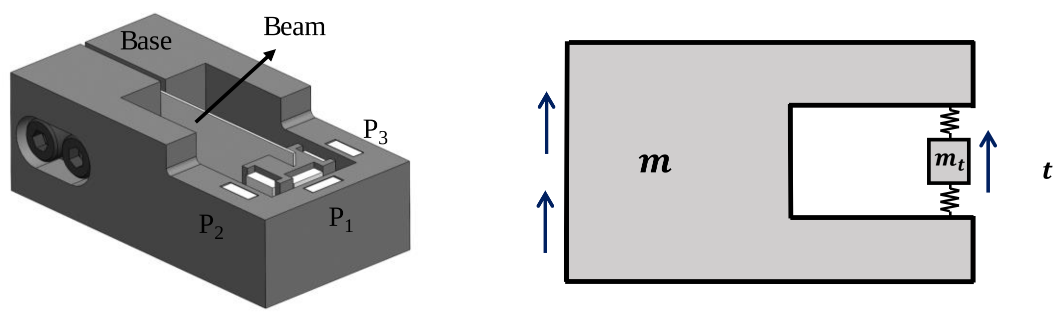

Figure 1. 3D representation of the neutralizer (left), where the beam represents the bending stiffness. Permanent magnets are positioned at the beam tip and at positions P1, P2, and P3 on the base. Simplified linear model considering a rigid base (right), where the springs represents the beam stiffness k and m_t the tip mass, including the support and the three attached magnets.

$$\begin{Bmatrix} A_b \\ A_t \end{Bmatrix} = -\omega^2 \begin{bmatrix} k(1+\mathrm{j}\eta)-\omega^2 m_b & -k(1+\mathrm{j}\eta) \\ -k(1+\mathrm{j}\eta) & k(1+\mathrm{j}\eta)-\omega^2 m_t \end{bmatrix}^{-1} \begin{Bmatrix} F_b \\ 0 \end{Bmatrix} \tag{1}$$

where, ω is the forcing frequency and η is the loss factor, used to introduce material damping in the model. A harmonic force with amplitude F_b is assumed to be applied to the base of the device. The behaviour of the neutralizer is discussed in the following sections.

3. EXPERIMENTAL EVALUATION

An experimental procedure to characterize the parameters of the neutralizer was first carried out without any magnets attached to the base. The base of the neutralizer was excited with a

SPEKTRA APS 400 long-stroke shaker, driven by a power amplifier model APS 124 and controlled via the VCS 403 Vibration Control System, and both the force applied and the responses at the base and the tip of the beam were recorded. A mechanical impedance sensor Model 288D01 (sensitivity 10.2 mV/m/s² and 22.4 mV/N) was used to measure the applied force and the acceleration of the base. To prevent mass loading of the beam by an accelerometer, a laser vibrometer (Polytec VibroGo) was used to measure the velocity at the tip of the beam, subsequently converting it to acceleration for comparison with the base response. Figure 2(a) shows a photograph of the experimental apparatus.

Two different excitation signals were employed to characterize the neutralizer. First, a broadband random signal was generated using the same VCS 403 system to characterize the neutralizer in terms of linear behavior, with low amplitude to avoid excessive displacement of the beam tip. Then, a frequency sweep (swept sine) signal with constant amplitude was used to characterize the neutralizer in terms of nonlinear behavior, with three amplitude levels tested: 0.002 mm, 0.005 mm, and 0.008 mm. The Acquisition System Sirius System was used to record the force, acceleration, and velocity signals with a sampling frequency of 500 Hz. These signals were subsequently transformed to the frequency domain. The frequency response functions were calculated using the H_1 estimator, resulting in a frequency resolution of approximately 0.002 Hz.

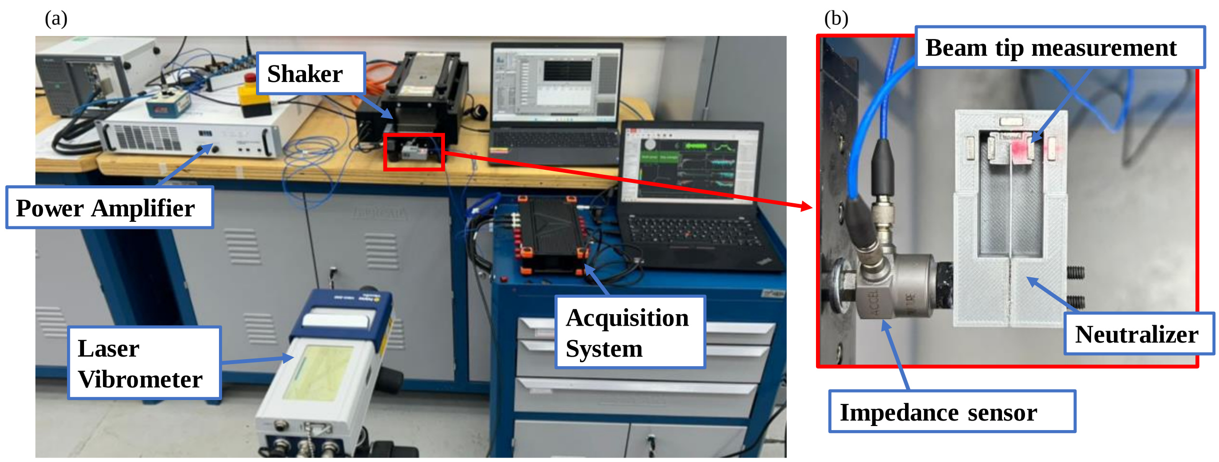

Figure 2. Photographs of the experimental setup. (a) A broad view of the experiment showing the laser vibrometer, shaker and power amplifier. (b) A view of the neutralizer mounted on the shaker, with the impedance sensor.

Figure 3(a) illustrates a comparison between the experimental accelerance and the accelerance of the two-degree-of-freedom model obtained using equation (1). Similarly, Figure 3(b) shows the accelerance at the beam tip.

The parameters used in the model were $m_b = 32$ g, $m_t = 2.2$ g, $k = 2.2$ Nm^{-1} and $\eta = 0.005$. These parameters were derived through a process involving measuring masses on a scale and iteratively adjusting stiffness and loss factor until achieving a qualitatively good agreement with experimental results regarding the first resonance frequency.

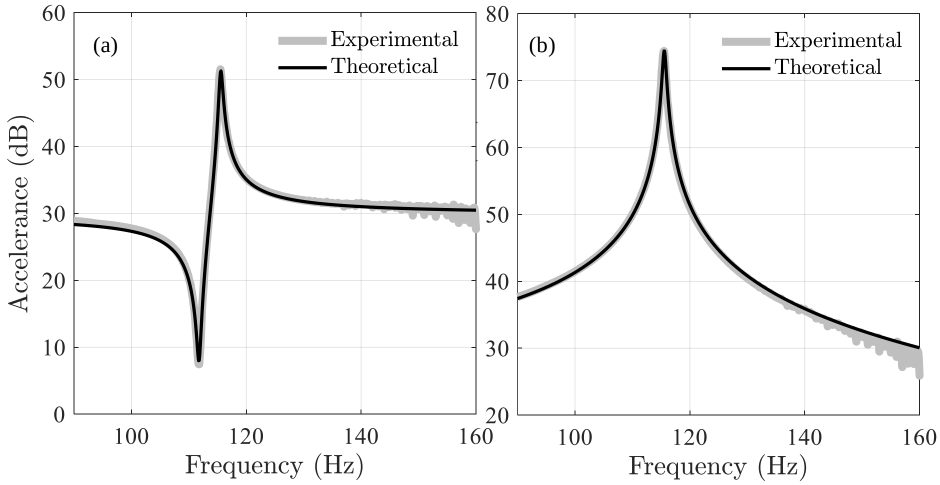

Figure 3. (a) Comparison between the experimental point accelerance and the point accelerance of the two-degree-of-freedom model obtained using equation (1). (b) same as (a), but the transfer accelerance between the force and the acceleration at the tip of the beam. (dB ref. 1 m/s²/N).

Introducing a permanent magnet at position P1 (refer to Fig. 1), configured to repel the center magnet at the free tip of the beam, creates a compressive axial force on the beam, thereby reducing its bending stiffness. Employing this setup, the accelerances of the neutraliser were measured as before, with low-amplitude excitation to prevent excessive motion of the beam tip. The accelerances of both the original configuration (without the magnet) and the new configuration are compared and depicted in Fig. 4, with (a) representing base accelerance and (b) illustrating beam tip accelerance.

Introducing a permanent magnet at position P1, shown in Fig. 1, configured to repel/atract the center magnet at the free tip of the beam, creates a compressive axial force along the beam, thereby reducing its bending stiffness. This modification added 0.4 grams to the base mass, which was considered negligible. Employing this setup, the accelerances of the neutralizer were measured as before, using a low-amplitude random signal to avoid excessive motion of the beam tip. The point accelerances of both the original configuration (without the magnet) and the new configuration are compared and depicted in Fig. 4, in which the magnets are configured to (a) repel each other, (b) no magnets at the base, and (c) attract each other.

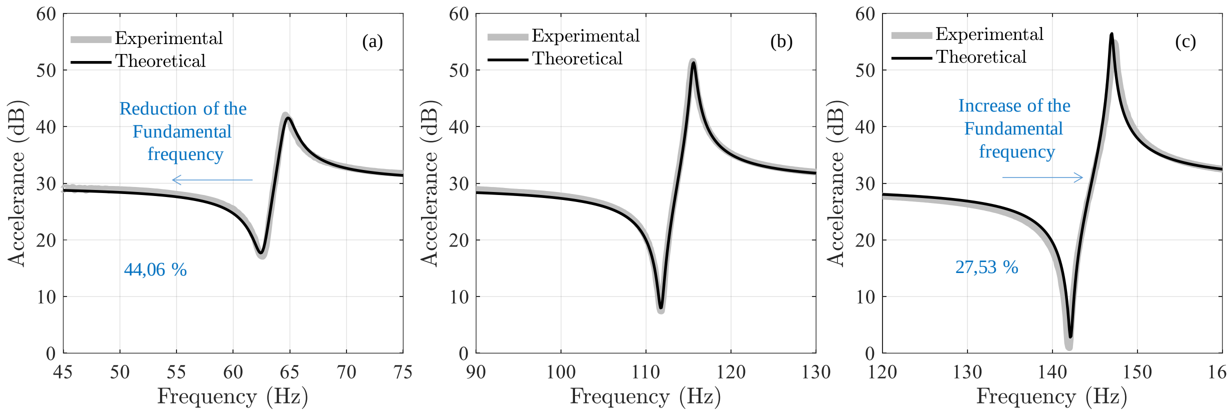

Figure 4. Comparison between the experimental point accelerance and the point accelerance of the two-degree-of-freedom model obtained using equation (1): (a) repelling configuration (k=340 Nm^{-1}, η = 0.018), (b) without magnet (k=1085 Nm^{-1}, η = 0.005) and (c) attracting configuration (k=1755 Nm^{-1}, η = 0.003). (dB ref. 1 m/s²/N).

A noticeable shift in the fundamental frequency of the neutralizer is observed, decreasing from 115.5 Hz to 65.6 Hz, marking a 44.1% reduction in frequency when the magnet polarity is to repel, while there is an increase from 115.5 Hz to 147.3 Hz, marking a 27.5% increase in frequency when the magnet polarity is to attract.

A major issue is that the dynamical behaviour of a nonlinear structure strongly depends on the magnitude of the displacement response. Thus, the identified model was tested with different levels of excitation, to illustrate the effects of nonlinearity for different magnet configurations. In this work only the amplitude of 0.008 mm base discplacement is shown because the nonlinearity is more evident, these responses are plotted in Figures 5(a) and (b). A sampling frequency of 500 Hz was used, and the signal was divided into two parts: 'up' and 'down'. The sweep rate was set to 1 Hz/min, so that the excitation could be considered to be quasi steady-state, and the behavior of the nonlinear systems could be seen. The displacement amplitude was kept constant throughout the sweep using the control capabilities of the SPEKTRA VCS 403 system.

Depending on the magnet configuration at position (P1), the axial force exerted on the beam can be compressive (repelling configuration) or tensile (attracting configuration). It can be observed in Figure 5(a) that the effect of the compressive axial force applied by the magnet on the beam shifts the resonant peak to lower frequencies in the transmissibility, as the compressive force decreases the bending stiffness of the neutralizer. The opposite occurs when there is a tensile axial force, which shifts the resonant peak to higher frequencies due to the increase in the bending stiffness of the neutralizer, as shown in Figure 5(b). Additionally, the effect of nonlinearity on the transmissibility can be seen as the bend the resonant peak to the right when the magnetic force is compressive (repelling configuration - Figure 5(a)), similar to the behavior of a base-excited hardening Duffing oscillator[10], and bend the resonant peak to the left when the magnetic force is tensile (attracting configuration - Figure 5(b)).

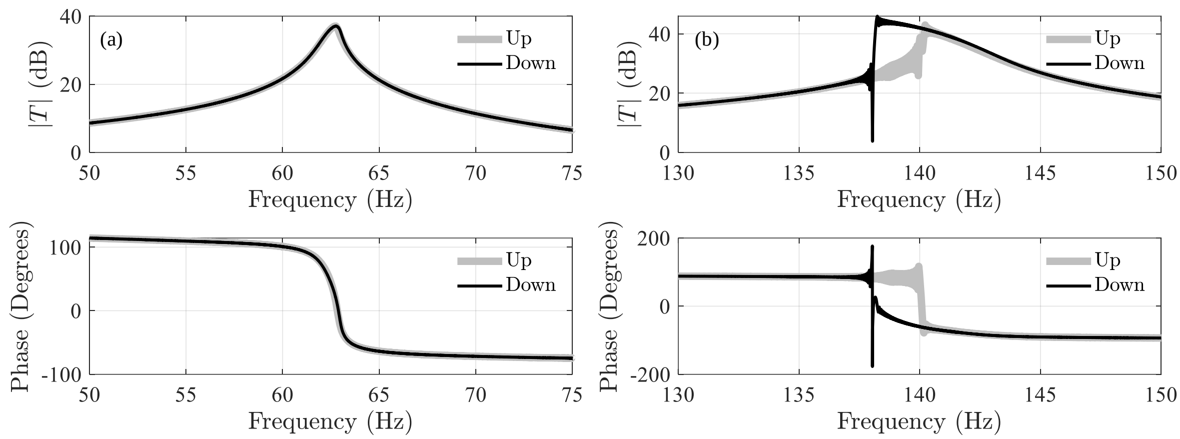

Figure 5. Transmissibility and phase: (a) repelling configuration at position P1 with a frequency range of 50–75 Hz, and (b) attracting configuration at position P1 with a frequency range of 130–150 Hz. Both cases used linear swept sine excitation with a displacement amplitude of 0.008 mm and a sweep rate of 1 Hz/min (dB ref. 1 m/s²/N).

4. DISCUSSION AND CONCLUSIONS

This paper has introduced a vibration neutralizer design, where the fundamental frequency can be significantly reduced by using a permanent magnet (0.4 g) without substantial mass addition. The device was characterized through experimental tests, and a magnet configuration was tested, demonstrating the feasibility of the design. To achieve the same effect of reducing the fundamental frequency, it would be necessary to increase the mass of the beam tip by about 277%, while for a similar effect on increasing the fundamental frequency by approximately 27%, the mass of the beam tip should be reduced by about 40%. This demonstrates that the use of magnetic force, although it adds superficial mass, has a significant effect on increasing and decreasing the system's fundamental frequency simply by changing the magnetic polarity. Additional effects on the system include the nonlinear behavior of the neutralizer, which presents an interesting possibility for studying nonlinearity in dynamic systems. Future work will explore other configurations of permanent magnets, aiming to optimize the system's performance.

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