

Attenuation of relative oscillation by means of self-active composite elasto-magnetic attenuators

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Abstract – New experimental results on the capability of innovative elasto-magnetic composites to produce high attenuation of the relative oscillation between two bodies with different rigidities, as well as to elude the resonance effects, are reported. A simple, but very efficient, model of the innovative attenuator working principle, including the determinant role of magnetic force change with vibration, is proposed and its agreement with experimental features is explained. Peculiar potentialities for optimal application of the innovative composite elasto-magnetic attenuators are outlined.

I. INTRODUCTION

The field of vibration attenuation is in continuous development to find innovative and more effective solutions. This is due to its large impact in several and different fields: passenger comfort on trains, airplanes or vehicles (containment of oscillations, noise prevention, etc...) [1, 2]; civil buildings protection from earthquakes and from the resonance produced by periodic wind gusts or traffic solicitations [3, 4]; safety of workers in mechanical industry in presence of noise and vibration [5, 6]; vibration isolation of delicate apparatuses in laboratory of research or aerospace (anti-vibrating tables for microscopes, optical equipments, etc...) [7 - 9].

The methods and the devices generally used can be classified on the basis of the demand of adjunctive energy by means of external inputs. So we have passive attenuators (no external control required) [10], active attenuators (operating by an external input that activates and controls the functioning) [11], semi-active attenuators (passive attenuators in which it is possible to trigger an active component to reinforce the attenuation) [12] and adaptive attenuators (actively controlled attenuators which are able to adapt their functioning in dependence on different working conditions (frequency, power, etc...)) [13]. In the recent past, we proposed a sort of self-active attenuator in the sense that the devices is able

to self-produce change of internal magnetic forces as response to an external harmonic strain without necessity of input signals or energy from the extern (elasto-magnetic composite for vibration attenuation [14]). This kind of device results very effective in reducing the amplitude of both forced and spontaneous oscillations. At the same time, it favours the resonance interdiction by means of asymmetrical response of the magnetic forces during the compression and dilatation phases of the harmonic oscillation coming from the extern [15].

The aims of the present experimental investigation are: (i) to confirm the effectiveness of composite elasto-magnetic attenuators also in the case of horizontal relative oscillation between a vibrating mechanical system and a wall integral to earth (experimental condition very different from the case previously investigated concerning vertical oscillation of a mechanical system excited at the base or at the top [15]); (ii) to propose a simple model showing the inner work of the innovative attenuator.

Peculiar apparatuses were built up to perform proper experimental tests, as shown in the following section.

II. EXPERIMENTAL

In order to investigate the action of the attenuator in opposition to a horizontal relative oscillation of two bodies with different rigidity, the experimental apparatus, shown in figure 1, has been arranged. The shaker (S) has been governed by a signal generator and a power amplifier. It applies to the mechanical system (M) a horizontal harmonic force (variable in frequency but fixed in amplitude). The excited mechanical system (M) consists of a horizontal aluminium platform sustained by an accordion base with a stabilizing mass on the basis. On the opposite side of the point in which the shaker applies its force, two pillars are put face-to-face to contain the attenuation device (D): a pillar (P) is integral to the M platform and the other one (P') is integral to a panel in-built with a laboratory wall.

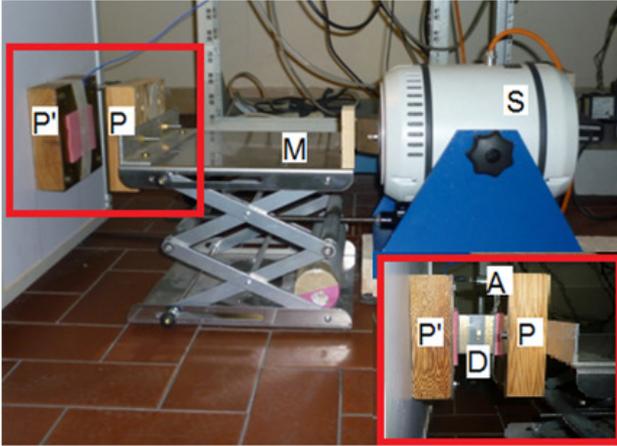


Fig. 1. Experimental apparatus used to perform test of relative vibration attenuation.

In the used configuration S+M simulate a vibrating body (e. g. a motor) and the composite elasto-magnetic attenuator (CEMA) allocated between P and P' has the objective to weaken its vibrations with respect another body rigidly connected to the earth.

The CEMA is constituted by two permanent magnets of NdFeB that are included into a silicone matrix with identical magnetic polarities “face-to-face” at a fixed distance, so to exert reciprocal repulsive forces (see Fig. 2); a soft iron nucleus lies between the magnets to reinforce the magnetic repulsion [15].

The vibration attenuation is active with respect oscillation that come parallel to the magnetic repulsive force. In static conditions the magnets placed at a distance of 20 mm between them repulse each other with a force of about 70 N (see Fig. 3), and the shaker works in order to apply to the mechanical system (M) a horizontal harmonic force with a fixed amplitude comparable (70 N).

The accelerometer (A), glued on pillar P, detects the horizontal acceleration of the mechanical system (proportional to the related displacement in the monitored point) and the output signal, after charge amplification, was visualised and elaborated by an oscilloscope.

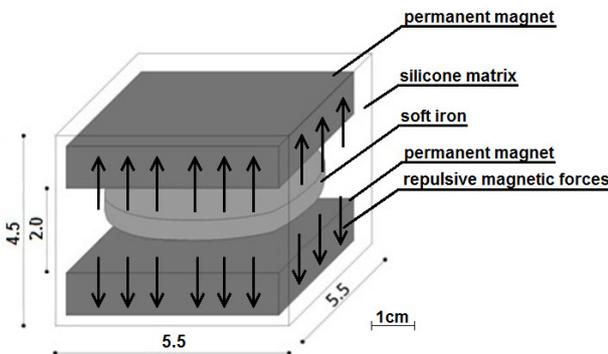


Fig. 2. Scheme of the used composite elastomagnetic attenuator (CEMA).

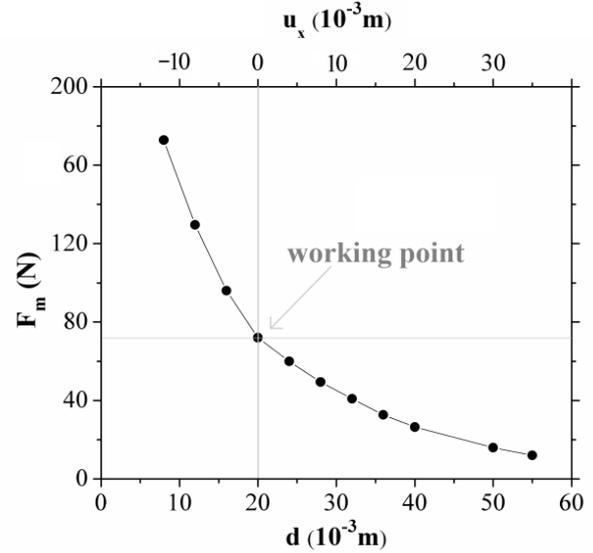


Fig. 3. Behaviour of the repulsive magnetic force modulus, F_m , as a function of the distance d (or relative displacement u_x) between the magnet face-to-face surfaces.

The oscillation measured by the accelerometer was detected in different experimental conditions: without any attenuator (see Fig. 4a); with a passive attenuator constituted by a silicone block (see Fig. 4b); with a passive attenuator constituted by a silicone block including brassy components (same size and position of the magnets in the CEMA) (see Fig. 4c); with a self-active CEMA (see Fig. 4d). In all cases, the shaker excitation was the same and the initial distance (not active shaker) between P and P' was such as to have a fixed pre-compression of about 2 mm when actuators are present.

III. RESULTS AND MODEL OF CEMA WORKING PRINCIPLE

The acceleration amplitude versus frequency of the harmonic applied excitation (at fixed amplitude of the force applied by the shaker (70 N)) is reported in figure 5, in the without attenuators condition.

In the low frequency range (< 100 Hz) four main picks are evidenced at 27.8 Hz, 40.7 Hz, 69.5 Hz, 105.0 Hz, corresponding to the resonance of the spontaneous vibration modes with the exciting shaker action.

In figure 6 it is possible to observe the progressive improvement introduced by more effective attenuation devices:

1st the elastic reaction of the silicone matrix, due to the periodic strain, slightly abates the picks amplitude of acceleration, but a negligible effect is evidenced on resonance frequency displacement;

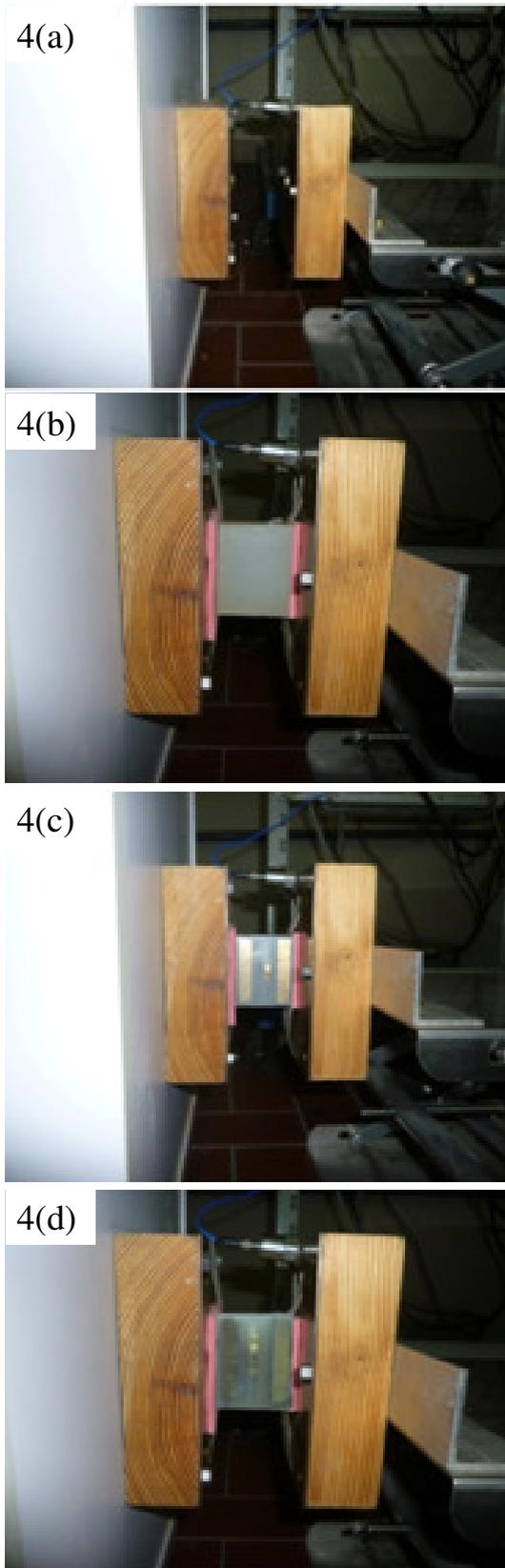


Fig. 4. Actuation conditions in the different experimental tests.

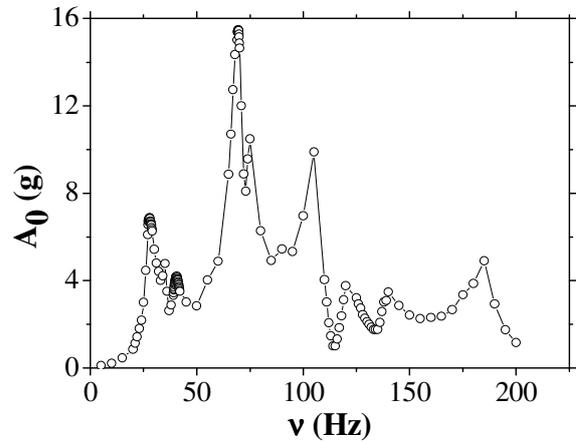


Fig. 5. Acceleration amplitude A_0 vs frequency ν in free vibration condition.

2nd when rigid components are added into the silicone matrix, the increases of the energy consumption, due to internal friction between rigid bodies and soft elastic silicone, produces a significant decrement of the acceleration pick amplitude;

3rd the introduction of the repulsive magnetic forces, which obstacle contraction but facilitate dilatation, demonstrates its potentiality in reducing main pick amplitude of about an ulterior 50% in comparison with an equivalent composite attenuator with soft and rigid alternate components in absence of magnetic forces. Moreover, it appears evident the resonance control at low frequency (< 100 Hz) where prominent picks disappear.

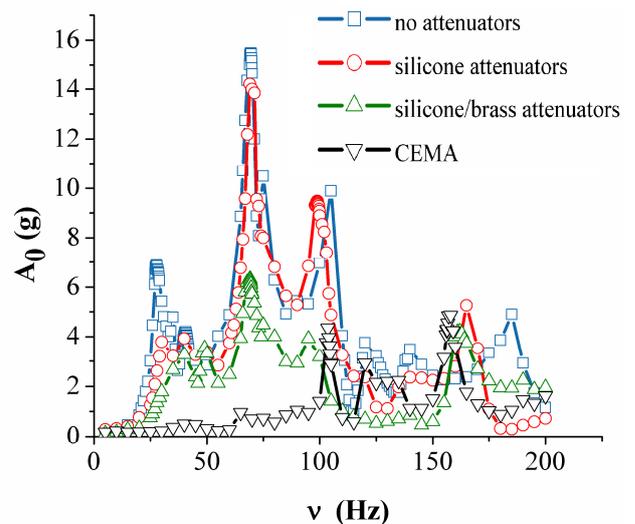


Fig. 6. Attenuation improvement due to progressive introduction of silicone passive action (\circ), brass-silicone internal friction (Δ) and magnetic repulsive forces (∇).

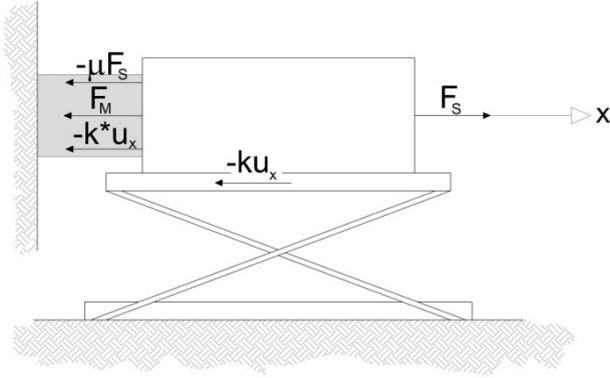


Fig. 7. Scheme of the mechanical system and the applied horizontal force due to oscillation.

In a simplified model the Newton equation for the centre of mass of the mechanical system, schematized in figure 7, is given by the following equation:

$$m (d^2 u_x / dt^2) = F_0 \sin(2\pi \nu t) - K u_x - K^* u_x + (\delta F_m / \delta u_x) u_x - \mu F_0 \sin(2\pi \nu t) \quad (1)$$

where m is the mass of the mechanical system; $F_s = F_0 \sin(2\pi \nu t)$ the shaker exciting force; $F_K = -K u_x$ and $F_E = -K^* u_x$ the mechanical system and CEMA elastic reaction, respectively, where K and K^* are the system and CEMA rigidity, respectively, and u_x is the horizontal displacement component of the mass centre; $F_M = (\delta F_m / \delta u_x) u_x$ the magnetic force change due to u_x ; $F_v = -\mu F_0 \sin(2\pi \nu t)$ a dissipative force due to the CEMA internal friction between soft silicone and rigid magnets, where μ is proportional the friction coefficient. We didn't consider the presence of a viscous force because the used silicone has shown an elastic behaviour in the range of the used strains and frequencies (practically linear and reversible stress vs strain characteristic), as well as the external vertical forces because they equilibrate each other and the mechanical system is subjected only to the scalar horizontal component of the external forces.

Taking into account the general solution of equation (1) of the type $u_x = u_0 \sin(2\pi \nu t)$, by substitution it is easy to find for the displacement amplitude $U_0 = |u_0|$ the following equation:

$$U_0 = (1 - \mu) F_0 / \left| [K + K^* - (\delta F_m / \delta u_x) - (2\pi \nu)^2 m] \right| \quad (2)$$

Considering that a horizontal translation of the mechanical system occurs, it is acceptable the assumption that the theorized amplitude of the mass centre displacement can be considered very close to that one detected by the accelerometer. Therefore, in addition to the expected direct proportionality to F_0 and the obvious U_0 decrement with the increase of the mechanical system + CEMA rigidity, equation (2) is completely consistent

with the other effects which have been experimentally evidenced:

1st the oscillation amplitude decreases if the internal friction is increased by means of rigid and soft component presence (μ effect);

2nd higher is the decrement of magnetic force versus the displacement (that is equal to the F_m derivative with the distance between the magnets in Fig. 3) higher is the U_0 decrease and the insertion of magnetic forces improves the vibration abatement;

3rd the magnetic force presence produces an increase of the frequency for which the denominator goes to zero, in other words increases the resonance frequency.

Even though $(\delta F_m / \delta u_x)$ was considered constant for the deduction of equation (2), one can understand another effect, namely the asymmetric action of $(\delta F_m / \delta u_x)$: in effect it is possible show in the investigated case that $|\delta F_m / \delta u_x|$ is slightly higher for negative values of u_x than for positive ones with the consequence that in an oscillation the dilatation phase is facilitated in comparison to contraction, so producing a not harmonic response of the attenuator which contrasts resonance (it compromises the frequency purity).

IV. CONCLUSIVE REMARKS

Novel experimental tests, expanding CEMA application to horizontal relative oscillations, have confirmed the competitive performance of composite elasto-magnetic attenuators both for vibration amplitude abatement and resonance effects limitation. A schematic model has been developed in order to clarify the statement that the magnetic force attenuation mechanism is self-activated by the displacement itself and, moreover, it has not a symmetrical effect in the expansion and in the contraction phase of a vibration, so preventing resonance by breakage of the harmonic behaviour. In particular, it is very incisive to describe the separated, but cooperative, actions of silicone matrix elasticity, internal friction between rigid and soft component, and especially of repulsive magnetic forces on working performance of CEMA. On the basis of these experimental results, the CEMA represents an efficient and not expensive alternative device to standard adaptive, active and semi-active vibration attenuation systems. Furthermore, CEMA appears easy-fitting to different practical situations due its adaptability in shape and size.

V. ACKNOWLEDGMENTS

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