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Experimental Beam Structure With Magnetically Controlled Damping Blocks

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Abstract
This study considers the dynamic response of an experimental system combining two thin, aluminium beams with damping blocks attached. Damping blocks made of magnetorheological elastomer were placed at the tips of the beams and sequentially clamped using electromagnets to obtain energy dissipation. We analyse whether the magnetorheological elastomers can be effectively used in controlled damping systems with varying levels of stiffness and friction. The experimental results demonstrate that residual vibrations can be suppressed faster when the switching control is applied than if the electromagnetic actuator is turned on constantly. The effectiveness of the solution is discussed, based on the experimental results. The decay of vibration amplitude, damping and the system frequency is provided.

Keywords: vibration, damping, control, switching, friction, electromagnet.

1 Introduction

In order to obtain passive-adaptive or semi-actively controlled vibration abatement, smart materials like magnetorheological or electrorheological fluids or elastomers, electroactive polymers, piezoelectric patches can be applied, among others. These types of solution are in favour, since the motion of the system can achieve amplitude decay, instead of directly applying control forces, as is the case with active solutions.

The isotropic magnetorheological (MR) elastomer used in our study was composed by mixing carbonyl iron particles (permeability $\mu_p = 37$) with polymeric non-magnetic matrix (permeability $\mu_m = 1$). Due to the high volume fraction of the magnetic particles, reaching up to 40%, the magnetorheological elastomer reaches permeability of $\mu_m = 3 \div 4$ Schubert and Harrison (2016). In most of the studies on magnetorheological elastomers, vibration attenuation is achieved through modification of Kirchoff's modulus. In our study we use magnetorheological elastomers, but

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instead of changing the shear modulus, the friction mechanism is explored. The elastomeric blocks are placed at the tips of two, separated, adjacent aluminium beams. Both beams were equipped with electromagnets, which sequentially clamp and lock the magnetorheological elastomer blocks. Energy dissipation in obtained in an instantaneous sense by modulating the normal force at the friction interface. If simple rubber was used instead of MR elastomer it would not be possible to use electromagnets as actuators, since the non-magnetic rubber would make the solution ineffective.

One can imagine that there are particular moments when the electromagnets are switched that result in the most effective vibration abatement. In the paper Dyniewicz et al. (2015) the influence of the theoretically derived control was presented. The simple Kelvin-Voigt model of the material was used and the change in shear stiffness was assumed as a core dissipation mechanism. In further experiments, it was noticed that other deformation mechanisms and control strategies proved to be more highly efficiency. That is why the sliding of rubber surfaces was allowed and used as a principal feature of the present approach. The main part of the study is devoted to the experimental results, which were obtained for two different configurations of elastomeric damping blocks, used to damp the free vibrations of a double-beam system. The results demonstrate the efficiency of vibration attenuation obtained using magnetically controlled damping blocks. The discussion of the experimental results concerning the decay of the vibration amplitude, damping and the frequency of the experimental system is provided.

2 Switching times control for elastomer blocks

When using the switching times method, the particular system parameters are sequentially altered at predetermined moments to achieve the desired level of vibration damping. Typically, the actuator's operating regime is simplified to switching between two states of the device according to an on-off logic, providing fair damping efficiency. This type of control can be easily adapted to a control relay, where the actuator only assumes two states, i.e., off and on. In Mroz et al. (2010) a simply supported, double beam system with piezoelectric actuators was studied. Depending on the electric circuit conditions, adhesion or delamination of beams were performed and principal mode of vibrations was damped. Ostachowicz et al. Ostachowicz et al. (2007, 2008) proposed shape memory pseudoelastic actuators to suppress forced vibration of smart beams. Each of the actuators was activated only under compression, while at the same time the opposite actuator remained inactivated. Dyniewicz et al. Dyniewicz et al. (2015) were switching shear modulus of magnetorheological elastomer to damp fundamental mode of a sandwich beam. In Bajkowski et al. (2015) pneumatically operated airtight elastic pouch was filled with granules and used as a device with variable stiffness and energy dissipation.

Friction interfaces are rarely discussed when considering the switching times control. However, the damping performance of the passive friction system could be improved with moderate effort by the semi-active control of the normal force applied to the friction damper Lane et al. (1992); Dupont et al. (1997). Applications of adjustable friction dampers may be found in the fields of seismic protection or building upgrading. Plenty of publications deal with the friction phenomena in the terms of the dynamics of turbine blades in aircraft engines and power plants. The authors in Bhaskararao and Jangid (2006) looked into the performance of buildings joined with friction dampers. Li and Reinhorn in Li and Reinhorn (1995) evaluated friction dampers for bridges, with the friction force adjustable through appropriate torque of the pressure bolts. In Stammers and Sireteanu (1998) numerical study of a semi-active dry friction damper operated with sequential logic was presented. The successful implementation of this strategy has showed a significant reduction in the forces transmitted to the foundation by a machine subjected to a rotating imbalanced force or acceleration in a car riding on a bumpy road. Switching the friction in vibration damping was explored earlier, but the use of MR elastomers and electromagnets for this operation has not yet been discussed.

3 Experimental system

The investigated structure consists of two adjacent, thin beams made of aluminium of Young modulus 69 GPa. Each of the beam is 720 mm long, and has rectangular cross-section 40×1.0 mm. The gap between these parallel beams is 20 mm. The top end of each beam is fixed, while the bottom end is free. To establish initial clamping force and keep the vibrating beams at close distance to achieve effective operation of electromagnets, some auxiliary small springs ($k = 25$ N/m)were placed at 1/3 and 2/3 of the beam's length (Fig. 1). The parameters of the system are collected in Table 1.

The frictional damping blocks were placed at the beam's tips. There were fabricated from a custom, isotropic, magnetorheological elastomer of 3560 kg/m^3 density. The natural rubber SVR 3L was used as a polymeric matrix with sulfuric cross linking additive and plasticizer. The fraction of BASF carbonyl iron content was 40% in volume concentration. Particles were uniformly mixed into the matrix, and the elastomer was cured at 145°C for 20 minutes. Two configurations of the damping blocks were considered for the experiments (Fig. 1).

In the **Case A**, two identical damping blocks were bonded to these planes of the beams which were facing each other. When the beams were clamped together with electromagnets, the friction occurred between the unbounded surfaces of the blocks, rubbing each other. The friction plane lies in the vertical symmetry plane of the two beams system. The damping members were 40 mm long, and had rectangular cross section 40×10 mm. The total mass of the elastomer blocks in the Case A was 115 g. The kinetic friction coefficient for MR elastomer block on MR elastomer was 0.6.

In the **Case B**, a single damping block was placed inside an elastic envelope

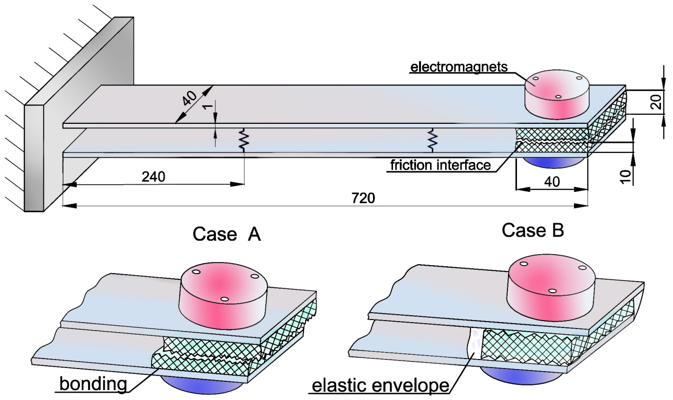


Fig. 1. Dimensions (in mm) of the experimental system, with two configurations of the elastomer: two bounded blocks - Case A and single unbounded block - Case B.

sealed at the beams' tips. The special thin envelope made of PVC of thickness 0.5 mm was preventing the unbounded damping member from slipping or falling out of the slot during vibrations. The damping block in the Case B was 40 mm long, and had rectangular cross section 40×20 mm, so its thickness was twice the size of block in the Case A. The mass of the block was 115 g. In this case, when the beams are being clamped by the electromagnets, two friction planes can be distinguished - the unbounded elastomer rubs on surfaces of the left and right beam. The kinetic friction coefficient for elastomer on aluminium is 0.4. This configuration is less practical than the Case A.

In both cases two island-pole 24 VDC operated electromagnets were attached to the outer surfaces of the beams (Fig. 1). The total mass of both electromagnets was 370 g, so they lowered vibration frequency of the system. To rout the magnetic field through the elastomer, a part of the aluminium surface was removed from underneath the electromagnets' poles. The magnets were in direct contact with the surface of the elastomer, so the magnetic field tends to flow through the elastomer. The magnets were polarized to provide highest value of magnetic field flux inside the elastomer blocks, and keep attraction of them.

By simple shunting switch of the circuit, the beams and damping blocks were being clamped by the electromagnets. The actuators were not capable of being positioned in intermediate configurations (between clamped or loose state), since electromagnets force varied roughly between minimum and maximum value. The average force clamping the elastomer blocks were different for each case considered. The exact value was hard to measure, as it depends on the size of the air gap between the magnets, and relative position of the magnets' poles. For the Case A there is only one air gap between the blocks, while for the Case B two air gaps are present (between the block and the surface of left and right beam).

Table 1. Parameters of the system for Case A and B.

	Case A	Case B
Average kinetic friction coeff.	0.6	0.4
Minimum clamping force [N]	0.5	0.5
Maximum clamping force [N]	5.0	4.0

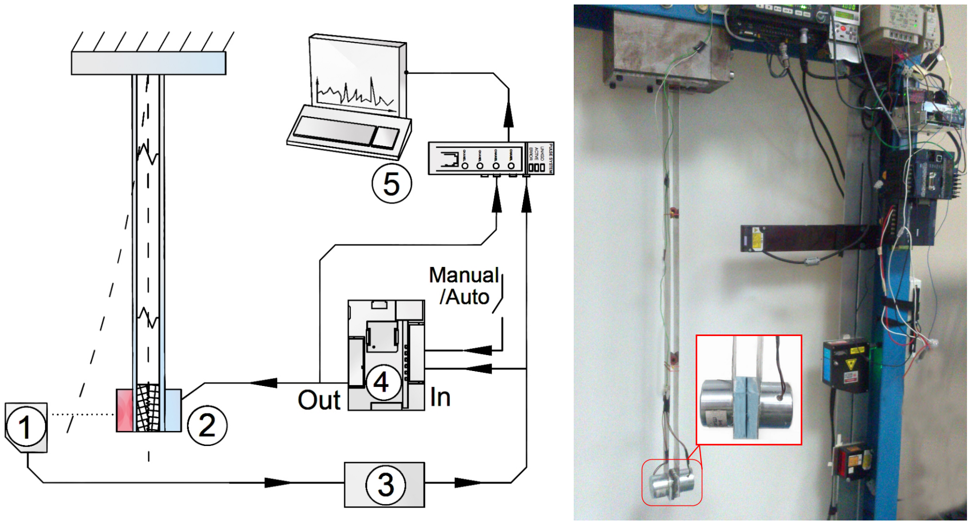


Fig. 2. Scheme of the experimental setup and photo of the test rig: 1–laser displacement sensor, 2—electromagnets, 3—signal amplifier, 4-–programmable logic controller, 5—data acquisition system.

4 Experimental results

The purpose of the research was to determine if the concept of using frictional pads clamped with the electromagnets is effective in vibration attenuation. The beams were tightly fixed in a vertical cantilever orientation, which is later called as vertical-neutral position. The main elements of the measurement equipment included high resolution, fast response laser displacement sensors with amplifiers. The laser amplifiers featured the function of inaccuracy compensation of the transverse displacement measurement up to 15 degrees of inclination angle. The lasers measured transverse displacements at the beams tips, where the maximum displacement amplitude occurred for the first mode of vibrations. The switching control of the electromagnets was performed by the programmable logic controller connected to the electromechanical relay outputs. The measurements were recorded with a 16-bit data acquisition card (Fig. 2).

For the purpose of free-vibration tests, both beams were initially deflect according to the first modal shape, and released from the fixed position. Some initial strategy for switching the electromagnets was established. At this stage of early mathematical modelling, the moments of switching were roughly estimated on the basis of reviewed literature and our previous experience on switched-stiffness strategies Dyniewicz et al. (2015); Pisarski et al. (2016); Bajkowski et al. (2016).

The idea was to distribute friction over time, and thus, modify global stiffness of the system to achieve fast vibration abatement. The control aims at converting the strain energy of a vibrating system into kinetic energy and alter equilibrium of the system, as a consequence of clamping beams through the friction pads. Additionally, some part of energy is dissipated by rubbing of the pads.

Some initial test were performed for switching friction each time when the system reaches point of maximum deflection (maximum left or right swing, $x \to \max$ and $\dot{x} \to 0$) or minimum deflection (no swing, $x \to 0$ and $\dot{x} \to \max$) measured from the vertical-neutral position. This operation is equal to switching parameters of the system, when kinetic energy of the system transfers to maximum potential energy and when potential energy transfers to maximum kinetic energy (see Fig. 3).

In the first phase when the beams start approaching vertical-neutral position, electromagnets are deactivated for short time to extract strain energy, and then rapidly activated when $x = 0$, $\dot{x} = \max$ to establish equilibrium position. The swinging beams pass the vertical-neutral position, but the imposed friction constraints counteract the beam movement. When the clamped beams reach maximum deflection ($x = \max$, $\dot{x} = 0$), the friction pads are quickly unclamped to extract the stored strain energy. The magnets are then again rapidly activated to establish another equilibrium. The return movement of the beams is counteracted by the clamped frictional pads. When the beams reach vertical-neutral position, the strain energy is once again released. Another quick activation of electromagnets establishes new equilibrium and the cycle starts from the first phase, but some part of the energy is already removed.

During the experiments it turned out that the combination of thin beams and heavy, high holding force electromagnets resulted in some additional, unwanted, vibrations when the sequence of rapid on-off-on switching was performed. The full sequence of switchings was simplified to half-cycles, as presented in Fig. 3, which allowed us to avoid uncontrolled vibrations introduced by electromagnets. The experimentally validated control is less efficient then the theoretically described, but is easier to adapt for the considered slender beams.

4.1 Case A - two blocks, one friction plane

Fig. 4a compares the transverse motion of one of the beams' end point for inactivated electromagnets, with results obtained for both electromagnets turned permanently on and for the switching control. The vibration damping increases considerably when electromagnets are turned constantly on, since beams are clamped and additional normal force provides higher friction force. When the switching of the magnets

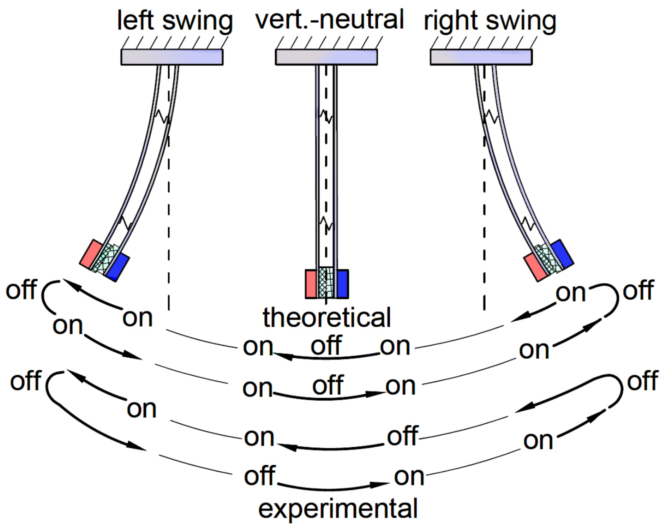


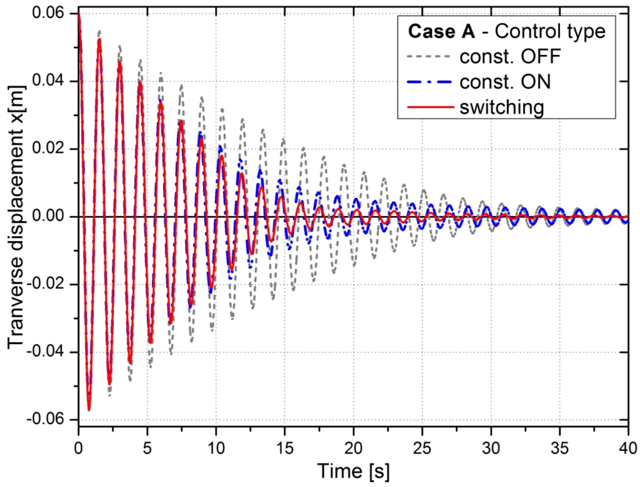
Fig. 3. Control sequence for shunting switch of the electromagnets.

is performed (according to experimental sequence in Fig. 3), even much better performance is noted. Fig. 4b presents the fast Fourier transform vibration spectra of the signal. The resonance peaks are decreased when additional friction is present, but second mode peak can be noted. When controlled switching is performed, the amplitude peaks are the lowest, and the second peak is flattened. The frequency of the system is barely changed, because the global stiffness is varied sequentially, and turns back to initial value when the electromagnets are deactivated. The dissipation factor *DF* was determined via the half-peak bandwidth at the resonance peaks:

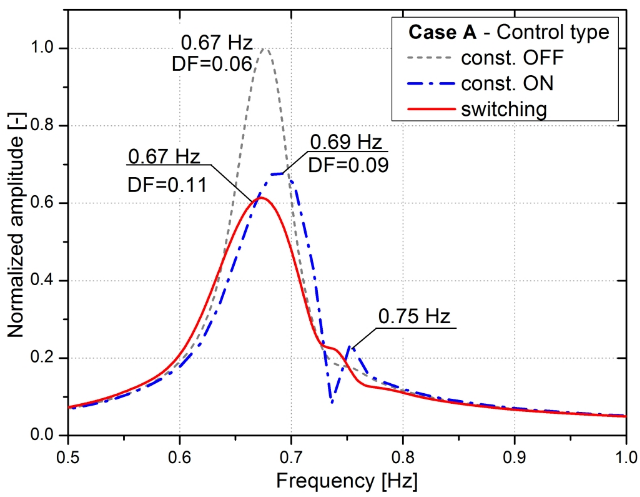
$$DF = \frac{\Delta f}{f_r}, \tag{1}$$

where f_r is the resonant frequency, and Δf is the bandwidth over which the power of vibration is greater than half the power at the resonant frequency. In other terms the factor expresses the ratio of the energy dissipated per cycle by damping processes related to the energy stored. The highest damping is obtained when the electromagnets are switched per control law, which is illustrated by flattening of the resonance peaks. The dissipation factor for turned off electromagnets is 0.06. It is increased to 0.09 for constant activation of the electromagnets. When the control is applied, the damping is increased by 67% compared with the deactivated magnets, and the dissipation factor reaches 0.11.

The analysis of the experimentally obtained state-space trajectory (Fig. 5) shows that the proper switching of parameters allows the system to jump to the trajectory providing faster convergence to equilibrium (continuous line) than the trajectory for the system with fixed stiffness values (grey dashed line).



a)



b)

Fig. 4. Transverse displacement of the beam's tip -a), and frequency of vibrations -b) for Case A.

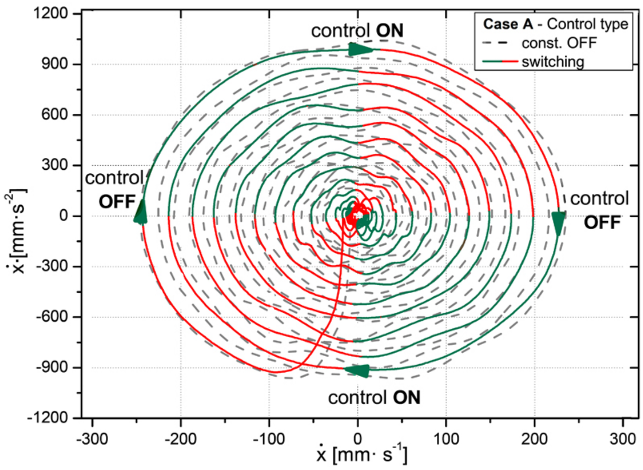


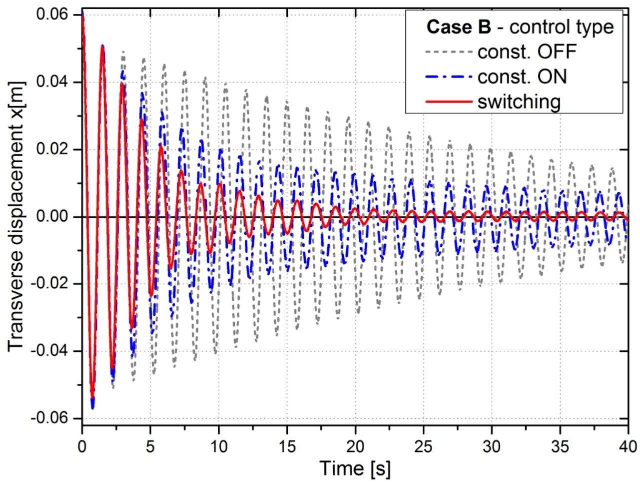
Fig. 5. State-space bounding orbit for Case A obtained with no control, and switching control.

4.2 Case B - one block, two friction planes

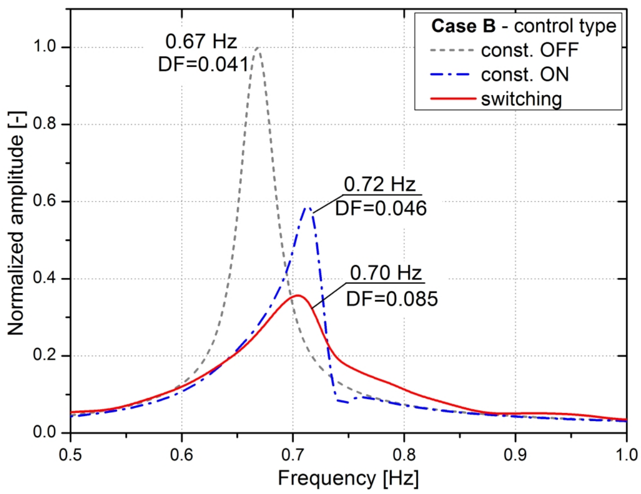
Figure 6a compares the dynamic response results obtained for the Case B, where the single damping element is secured by a thin elastic envelope. Since in this configuration there are two frictional planes (but both are elastomer-aluminium type), the system should provide better damping capacity than in the Case A. The free-vibration plots clearly show that the damping is the highest when the switching times technique is performed, and the vibrations are damped in a shorter time then in the Case A. One can note, that the envelopes of the signal in Fig. 4a exhibited high influence of dry friction. For the Case B, other forms of dissipation like slippage and viscousity were present. The frequency shift is noted for the permanent activation of electromagnets (Fig. 6b), as well as for the switching times control, due to higher stiffness of the clamped system than in the Case A. The calculated dissipation factor in the absence of magnetic field for the aluminium beams is 0.041. It increases to 0.046 for the permanently activated electromagnets. The highest dissipation factor of 0.087 is noted, when the sequential control is performed.

5 Conclusions

The free vibrations of two adjacent beams damped with MR elastomer pads were studied under experimental conditions. The elastomer pads were used to provide additional friction to the system while being clamped by electromagnets. The increased friction causes the increased stiffness of the system, which is sequentially varied. As a result, a non-typical semi-active vibration damping element was obtained. The residual vibrations were suppressed faster when the switching control was applied than



a)



b)

Fig. 6. Transverse displacement of the beam's tip a), and frequency of vibrations b) obtained for the Case B.

when the electromagnetic actuator was turned constantly off or constantly on. Two configurations of the blocks were considered. The configuration of Case A turned out to be practical, but less efficient than that of Case B. Case B showed better damping performance but was rather impractical from an engineering point of view. The experimental study illustrates that magnetorheological elastomers perform well and are useful, as friction pads clamped at specific moments of the system operation.

If surfaces are properly treated, it could result in higher friction forces and thus better performance. A mathematical model of the system's behaviour needs to be developed to determine the optimal placement of the damping blocks and the optimal control strategy. The particular case of free vibrations of a cantilever beam was considered, but the principle could be applied to more complex systems in order, to counteract the time-varying excitation.

6 Acknowledgements

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