Design and verification of a negative resistance electromagnetic shunt damper for spacecraft micro-vibration

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Abstract

Active control techniques are often required to mitigate the micro-vibration environment existing on board spacecraft. However, reliability issues and high power consumption are major drawbacks of active isolation systems that have limited their use for space applications. In the present study, an electromagnetic shunt damper (EMSD) connected to a negative-resistance circuit is designed, modelled and analysed. The negative resistance produces an overall reduction of the circuit resistance that results in an increase of the induced current in the closed circuit and thus the damping performance. This damper can be classified as a semi-active damper since the shunt does not require any control algorithm to operate. Additionally, the proposed EMSD is characterised by low required power, simplified electronics and small device mass, allowing it to be comfortably integrated on a satellite. This work demonstrates, both analytically and experimentally, that this technology is capable of effectively isolating typical satellite micro-vibration sources over the whole temperature range of interest.

1. Introduction

The unprecedented endeavour to build the Hubble Space Telescope in the early 1970's and the desire to achieve the highest-ever imaging resolution bol-

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4 stered the research on the modelling and control of micro-vibration on board
5 spacecraft. Micro-vibrations are typically generated by several on-board sub-
6 systems and devices, such as reaction wheel assemblies, momentum wheel as-
7 semblies, control moment gyros, pointing systems and cryo-coolers. The in-
8 creasingly stringent stability requirements imposed by advanced, high-resolution
9 payloads have produced an increased interest in the development of better-
10 performing micro-vibration dampers. Several devices aimed at mitigating micro-
11 vibrations have been investigated and tested, and depending on the nature of
12 their functioning they can be classified as passive, active or semi-active dampers.

13 The passive dampers are mostly preferred due to their constructive simplic-
14 ity, relative low cost and reliability. Among this kind of damper, viscoelas-
15 tic materials are the cheapest and lightest damping solutions and are widely
16 used by companies in the space sector [1, 2, 3]. However, the limited loss fac-
17 tors achievable with these materials, along with the strong dependency on the
18 operating temperature and frequency ranges, make the design of viscoelastic
19 dampers quite complex. Other passive dampers studied in the last 25 years
20 include: Eddy-current dampers [4, 5, 6], D-struts [7, 8] and piezoelectric trans-
21 ducers shunted with passive electrical networks [9, 10]. Nevertheless, it is still a
22 challenge to employ pure passive isolation techniques to dampen low frequency
23 micro-vibrations onboard a spacecraft [11].

24 Active control techniques [12] and active dampers are sometimes necessary
25 in order to achieve more stringent conditions of isolation. Active dampers are
26 fully controllable and capable of producing higher-order vibration attenuation
27 with respect to passive dampers, but they also require a significant amount of
28 power to operate. Active isolation can be obtained by using external actuators
29 and sensors to provide control forces and feedback signals. Piezoelectric patches
30 have been used extensively in active vibration control applications [13, 14, 15],
31 and thanks to the development of the self-sensing technique these smart ma-
32 terials can be employed simultaneously as sensors and actuators [16, 17]. The
33 main advantages of this technique are the realization of a very robust, truly-
34 collocated control (given by the sensor and the actuator being placed in the

35 same position) as well as the reduction of system mass and complexity due to
 36 the elimination of independent sensors. However, variations of the piezoelectric
 37 properties produced by environmental and operating conditions would require
 38 a continual tuning of the circuit that, if not effectuated, could destabilise the
 39 closed-loop system. Also, electromagnetic transducers can be used either as
 40 velocity sensors or force actuators [18, 19], or simultaneously as both in the self-
 41 sensing technique [20, 21]. These transducers are characterised by large strokes,
 42 high bandwidth and low cost, but they may also suffer serious spillover (e.g.
 43 actuator saturation) and stability issues.

44 Recently, the use of negative impedances was proved to considerably im-
 45 prove the damping performance when connected to piezoelectric transducers
 46 [22, 23] or electromagnetic transducers [24, 25, 26]. Negative impedances are
 47 used to cancel or reduce the inherent electrical properties of the transducers
 48 (e.g. piezoelectric patches are characterised by an inherent capacitance). In
 49 the case of electromagnetic transducers (characterised by an inductance and a
 50 resistance), the reduction of the overall resistance by means of the negative resis-
 51 tance increases the induced current flowing in the circuit and thus the damping
 52 performance.

53 This paper presents an electromagnetic shunt damper (EMSD) connected to
 54 a negative-resistance circuit, and demonstrates its vibration attenuation perfor-
 55 mance by applying the isolator to a single-degree-of-freedom (SDOF) system.
 56 Although the shunt circuit requires power to operate, this type of EMSD can be
 57 considered as a semi-active damper for two reasons: i) the negative-resistance
 58 circuit acts overall as a passive electrical component having a constant negative
 59 magnitude, without requiring any control algorithm; ii) the damper requires lit-
 60 tle external power to function because part of the energy is self-provided by the
 61 relative motion between the magnet and the coil (i.e. the damper would work
 62 as a passive damper if the coil terminals were short-circuited). The proposed
 63 damper is analysed not only in terms of damping performance, but also as a
 64 satellite subsystem that needs to withstand certain physical and environmen-
 65 tal criteria. Therefore, the mass and power budgets of the damper have been

66 taken into account, as well as stability conditions derived from considering an
67 operating temperature range from $-20\text{ }^{\circ}\text{C}$ to $+50\text{ }^{\circ}\text{C}$.

68 This work differs from previous articles on similar dampers [24, 26, 27] by
69 proving, both analytically and experimentally, that this technology is able to
70 effectively isolate typical sources of micro-vibration in space (in the region of
71 5 kg) without using any active control algorithm to adjust for changes in the
72 environmental conditions. In fact, the damper transmissibility shows a final de-
73 cay rate of -40 dB/dec , while drastically reducing the force amplification at the
74 resonance frequency throughout the whole temperature range of interest. There-
75 fore, the improved damping performance, along with the low power required to
76 operate it and an overall mass that is less than 4 % of the suspended mass, are
77 among the main features of the proposed EMSD, which make it highly compet-
78 itive when compared with other well-established micro-vibration dampers.

79 The rest of the paper is organised as follows. Section 2 presents the analytical
80 model of the proposed EMSD. Section 3 describes the experimental set up.
81 Section 4 reports the tests results and shows the correlation with the analytical
82 data. The conclusions are drawn in Section 5.

83 2. Analytical model

84 A schematic of the system studied in this paper is shown in figure 1. This
85 model consists of a mass suspended on a metal spring with stiffness coefficient
86 k and an EMSD. The damper consists of two electromagnets connected to-
87 gether and rigidly attached to a fixed boundary, a cylindrical magnetic assembly
88 bonded to the suspended mass and coaxially aligned with the electromagnets,
89 and a shunt impedance connected to the terminals of the coils. The closed
90 electric circuit formed by the electromagnets and the shunt can be equivalently
91 represented as a voltage source V_0 in series with an inductance L_e and two resis-
92 tances represented by the resistance of the electromagnet, R_e , and the equivalent
93 resistance of the shunt, R_s . It is noted that a SDOF system is not representative
94 of the actual behaviour of a reaction wheel, which produces disturbances in all

95 six degrees of freedom. However, this configuration is aimed at obtaining a first
 96 assessment of the damper performance that can then be compared with other
 97 well-established damping solutions [3, 18, 28] . In particular, special attention
 98 is given to the transfer function between the input force (i.e. the disturbance
 99 forces produced by the reaction wheels) and the force transmitted to the ground.
 100 In this study, the mass of the satellite where the reaction wheels are connected
 101 to has been assumed to be several orders of magnitude bigger than the reaction
 102 wheels, and hence the ground can be considered fixed in space.

Figure 1: Schematic representation of the SDOF model, with the electrical schematic of the shunt circuit of the EMSD

103 2.1. State-space modelling

104 EMSDs function through the combination of two different physics phenom-
 105 ena that are described by the Faraday-Lenz law and the Lorentz force law. The
 106 Faraday-Lenz law asserts that a relative velocity between an electromagnet and
 107 a permanent magnet induces an electromotive force V_0 (i.e. electric voltage) at
 108 the terminals of the electromagnet. This induced voltage can be expressed with
 109 the equation:

$$V_0 = \oint (\vec{v} \times \vec{B}) \cdot d\vec{l} \quad (1)$$

110 where $\vec{v}$ is the magnet velocity with respect to the electromagnet, $\vec{B}$ is the
 111 magnetic field and $d\vec{l}$ is the infinitesimal length of the coil turns along which the
 112 integral is computed. Considering a cylindrical coordinate system ($\hat{r}$, $\hat{\phi}$, $\hat{z}$, with
 113 the z-axis along the magnet's longitudinal axis), and the only relative motion
 114 between the conductor and the magnet to be along the z-axis, the velocity, v ,
 115 and the magnetic field, B , can be written as:

$$\begin{aligned}\vec{v} &= 0 \hat{r} + 0 \hat{\phi} + v_z \hat{z} \\ \vec{B} &= B_r \hat{r} + B_\phi \hat{\phi} + B_z \hat{z}\end{aligned}\tag{2}$$

116 This results in an electromotive force equal to:

$$V_0 = \oint (v_z B_r \hat{\phi} - v_z B_\phi \hat{r}) \cdot r d\vec{\phi} = \oint v_z B_r r d\phi\tag{3}$$

117 where it can be seen that the magnetic field contributes to the induced voltage
 118 only with the radial component.

119 In the micro-vibration load case, it can be assumed that the magnetic field
 120 seen by the conductive material is constant in time, since the relative displace-
 121 ment is in the order of magnitude of tenths of a millimetre, and so its variations
 122 are almost negligible. Also, given the small cross section of the electromagnet,
 123 one can assume a linear trend of the magnetic field along the radial axis inside
 124 the coil. Therefore, Eq.(3) can be simplified as:

$$V_0 = 2\pi n_t r_{avg} \bar{B}_r v_z = K_d v_z\tag{4}$$

125 where n_t is the number of turns of the coil, r_{avg} is the average radius of the con-
 126 ductor, and $\bar{B}_r$ is the average radial component of the magnetic field through the
 127 coil cross section. K_d is defined as the electro-mechanical transducer coefficient.

128 Once the induced current is generated, it couples with the surrounding mag-
 129 netic field to produce the Lorentz force. This force is described by the equation:

$$\vec{F}_d = \oint I d\vec{l} \times \vec{B}\tag{5}$$

130 Adopting the same assumptions used for the electromotive force, and since
 131 $d\vec{l}$ has the only component along $\hat{\phi}$, the damping force F_d can be expressed as:

$$\vec{F}_d = -2\pi n_t r_{avg} \bar{B}_r I \hat{z} = -K_d I \hat{z}\tag{6}$$

132 These two laws considered together demonstrate that the force produced by
 133 a permanent magnet which moves close to a conductive material is proportional
 134 in magnitude and opposite in direction to their relative velocity. Therefore, this
 135 kind of force opposes the movement of the magnet and acts like a damping force.

136 The system shown in Fig. 1 can be modelled by including Eqs.(4) and (6)
 137 in a fully-coupled system of four equations:

$$\begin{cases} m\ddot{z} + kz = F_d + F_{in} \\ V_0 = K_d v_z \\ L_e \dot{I} + (R_e + R_s) I = V_0 \\ F_d = -K_d I \end{cases} \quad (7)$$

138 which respectively represent: the equation of motion (the input force is described
 139 by the term F_{in}), the Faraday-Lenz law of induction, Kirchhoff's voltage law
 140 associated with the electric circuit, and the Lorentz force.

141 Through the assumptions of micro-vibration and steady-state conditions,
 142 the system shown in Eq.(7) can be linearised and rearranged via a state-space
 143 representation. The use of state space modelling considerably simplifies the
 144 analysis of the system in the frequency domain into which it can be converted
 145 by taking the Laplace transform. The state-space model can be written as

$$\begin{bmatrix} \dot{z} \\ \ddot{z} \\ \dot{I} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -k/m & 0 & -K_d/m \\ 0 & K_d/L & -(R_e + R_s)/L \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ I \end{bmatrix} + \begin{bmatrix} 0 \\ 1/m \\ 0 \end{bmatrix} F_{in} \quad (8)$$

146

$$Y = \begin{bmatrix} k & 0 & K_d \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ I \end{bmatrix} + \begin{bmatrix} 0 \end{bmatrix} F_{in} \quad (9)$$

147 where the state vector consists of three state variables: mass displacement (z),
 148 mass velocity ($\dot{z} = v_z$), and circuit current (I). The output vector Y represents
 149 the force transmitted to the satellite structure. This single-input-single-output
 150 (SISO) system can be converted to the frequency domain, and solved for the

151 force transfer function (TF). The TF considered in this case is the relation
 152 between the input force F_{in} and the force transmitted to the ground. The
 153 TF is characterised by poles and zeros that describe the system's dynamic be-
 154 haviour in the frequency domain. In the case of a mass-spring system, the TF
 155 is only characterised by a couple of complex-conjugate poles, which determine
 156 the resonance peak and the roll-off slopes of -40 dB/dec. Adding to this sys-
 157 tem a magnetic damper with almost negligible inductance (e.g an Eddy-current
 158 damper as shown in Ref. [29]) produces a reduction of the resonance peak but
 159 also the introduction of a real zero that brings the decay rate to -20 dB/dec.
 160 The EMSD proposed in this paper presents a relatively high resistance-over-
 161 inductance ratio that introduces a real pole at low frequency, thus causing the
 162 roll-off slope to be -40 dB/dec, while having a considerable reduction of the
 163 resonance peak.

164 2.2. Magnet configuration

165 The damper performance can be improved, besides other means, through
 166 the increasing of the electro-mechanical transducer coefficient, K_d . As shown in
 167 Eq.(4), K_d depends on the radial component of the magnetic field. A trade off
 168 between various configurations of the magnetic assembly in order to increase $\bar{B}_r$
 169 was conducted through the software COMSOL Multiphysics. Three different
 170 configurations were analysed: a single cylindrical magnet (see Fig. 2a), a single
 171 cylindrical magnet with two iron yokes at the ends (see Fig. 2b) and three
 172 cylindrical magnets arranged with opposing polarity and alternated with two
 173 iron yokes (see Fig. 2c). The last configuration uses a non-magnetic, M4 screw
 174 to keep the magnets in place. The cross-section of the two electromagnets was
 175 kept constant throughout the analysis, and the diameter of the coils was changed
 176 in order to maintain a safe gap of 1 mm between the magnet and the coils. A
 177 parameter defined as the ratio of the coefficient K_d to the mass of the magnetic
 178 assembly was used to compare these configurations. Tab. 1 shows the main
 179 parameters for each configuration. The magnetic assembly shown in Fig. 2c and
 180 2f was ultimately chosen not only for the better K_d -over-mass parameter, but

181 also because the two free ends of the magnet stack can be potentially exploited
 182 by using two other electromagnets.

Figure 2: Comparison between three different configurations of the magnetic assembly. The analyses were carried out through COMSOL Multiphysics using a 2D-axisymmetric formulation. The top row shows the disposition of the magnets and yokes, and the magnetic polarization, whereas the bottom row shows the magnetic flux density. (a,d) single cylindrical magnet; (b,e) single cylindrical magnet with two iron yokes at the ends; (c,f) three cylindrical magnets arranged with opposing polarity and alternated with two iron yokes.

183 2.3. Negative resistance

184 The use of negative resistance in the shunt (R_s) produces a reduction of the
 185 overall circuit resistance that results in an increase of the electric current, I ,
 186 and hence of the damping force, F_d . Several examples of the use of negative
 187 impedances have been found in literature [26, 27], but these circuits are always
 188 implemented using a controller board (e.g. dSpace R1103) connected to an
 189 elaborated circuitry to obtain the desired value of the impedance (the electronic

Property	Value		
	Config. 1	Config. 2	Config. 3
$\bar{B}_r$ (T)	0.20	0.23	0.28
K_d (N/A)	6.45	7.41	6.7
K_d/mass ($\frac{\text{N/A}}{\text{Kg}}$)	47.78	32.21	49.63

Table 1: trade-off of magnetic assembly

board from Ref. [26] is reported in Fig. 3). Moreover the evaluation of the power required by the shunt has never been taken into account. In this study, the negative resistance is created by utilising an analogue circuit called negative impedance converter [30]. It consists of a single operational amplifier (op-amp) and three resistors that are connected as shown in Fig. 4. The equivalent resistance of the shunt is,

$$R_s = -R_1 \left(\frac{R_3}{R_2} \right) \quad (10)$$

By considering the resistances R_2 and R_3 to be equal to each other, the shunt produces a negative resistance $R_s = -R_1$. In terms of stability, this circuit tends to be stable as long as the resistance of the electromagnet, R_e , is greater in magnitude than the shunt resistance, R_1 (this aspect will be taken into account in the next paragraph). The supplied power required by the shunt circuit can be expressed as

$$P = (V^+ - V^-) \cdot (I_q + I_o) \quad (11)$$

where $V^+ - V^-$ is the total supply voltage, I_q is the quiescent current consumed by the op-amp, and I_o is the op-amp output current. I_o can be approximated to the current flowing in the resistor R_1 , because the choice of selecting R_2 and R_3 to be three orders of magnitude greater than R_1 produces a current, $I_{R_2 R_3}$, that can be neglected. Since the signal is sinusoidal, the rms value of the power can be obtained as:

$$P_{avg} = (V^+ - V^-) \cdot (I_q + I_{rms}) \quad (12)$$

208 where I_{rms} is the rms value of the op-amp output current.

Figure 3: Block diagram and circuit board of the negative resistance shunted impedance presented in Ref. [26]

Figure 4: Electrical schematic of the negative resistance impedance converter connected to the electromagnet

209 2.4. Parametric trade-off and expected results

210 The environmental conditions at which a satellite operates can usually affect
 211 the performance of dampers (e.g. viscoelastic materials are strongly dependent
 212 on the surrounding temperature). The suspended mass has been chosen to be 5
 213 kg and corresponds approximately to the mass of two 100SP-O reaction wheels
 214 used by Surrey Satellite Technology Ltd (SSTL). The operating temperature
 215 range for this type of reaction wheel is from -20°C to $+50^{\circ}\text{C}$ [1]. Therefore,
 216 there is an increasing interest in the development of damping solutions that
 217 guarantee good isolation performances in this temperature range. An EMSD is
 218 made uniquely of metal parts (Nd-Fe-B magnets and copper) whose character-
 219 istics with respect to temperature variations are well established. In particular,
 220 the resistivity of the copper increases linearly with respect to the temperature
 221 with a thermal coefficient of $0.00386^{\circ}\text{C}^{-1}$. Also, the residual induction of the
 222 Nd-Fe-B magnet is characterised by a linear temperature coefficient of -0.0012
 223 $^{\circ}\text{C}^{-1}$. These two effects have been taken into account for the assessment of the
 224 damper performance.

225 Regarding the negative resistance circuit, the three resistors R_1 , R_2 and
 226 R_3 can be chosen among space-qualified, off-the-shelf components that have
 227 tolerances down to 0.005% and temperature coefficients of $1 \cdot 10^{-6}^{\circ}\text{C}^{-1}$. Hence
 228 these resistors can be considered constant over a wide temperature range when
 229 compared with the electromagnet resistance, R_e . As already stated above, the
 230 EMSD would become unstable if the total resistance ($R_e + R_s$) is negative,
 231 and this could happen at the lowest temperature range limit (where R_e reaches
 232 its minimum value). In order to prevent this, a minimum value of 0.1Ω was
 233 imposed on the total resistance at a temperature of -20°C . A trade-off of the
 234 damper parameters was performed to meet the stability requirement and also
 235 to guarantee an amplification at the resonance frequency to not be greater than
 236 6dB throughout the whole temperature range. The corner frequency (i.e. the
 237 point in the TF where the output is half the value (-3dB) of the input) was
 238 set to be equal to 10 Hz for this test case. The trade off was carried out by
 239 analysing the system at three different temperatures: -20°C , $+50^{\circ}\text{C}$ and the

range mid point, +15 °C.

During the analysis it was assumed that the energy dissipated by the coil through Joule heating was removed from it (only through conduction due to the absence of atmosphere in space) so that the electromagnet did not change its temperature with respect to the surrounding environment. This assumption is corroborated by the fact that only a small amount of thermal power needs to be dissipated in a typical micro-vibration load case (see Section 3). Also, this hypothesis allows to study the system in steady-state conditions using Eq.(8), where the magnitudes of the coil resistance, R_e , and the average radial component of the magnetic field, $\bar{B}_r$, are determined by the initial environmental temperature and are constant throughout the analysis. The magnitudes of R_e and $\bar{B}_r$ at +15 °C were taken as reference values, and a thermal analysis was performed in COMSOL Multiphysics to evaluate their variations within the temperature range of interest. The final choice of the parameter set that has met the predefined requirements is reported in Tab. 2.

Property	Value		
Mass, m (kg)	5		
Spring Stiffness, k (N/m)	4600		
Coil Inductance, L_e (mH)	1.3		
Shunt Resistance, R_s (Ω)	-1.5		
	<i>Temp.</i>	-20 °C	15 °C
		50 °C	
Coil Resistance, R_e (Ω)	1.60	1.85	2.10
E-M Transducer Coefficient, K_d (N/A)	6.98	6.7	6.42

Table 2: Final choice of the parameter set obtained through a trade-off

A comparison of the force TFs between the system without damping and

the system with the EMSD is shown in Fig. 5. It can be seen that the use of the EMSD produces the expected reduction of the resonance peak, and presents an amplification that is always below 6dB in all three cases studied. The corner frequency is equal to 10 Hz for the case at 15 °C, but it moves slightly within the temperature range, going from 8 Hz to 20 Hz. However, at high frequency the three curves almost overlap, showing the final slope of -40dB/dec that is typical of a mass-spring system.

The shift of the corner frequency at low temperatures is due to the swap of the complex-conjugate poles with the real pole introduced with the proposed damper. As stated in Section 2.1, reducing the overall resistance produces an increase of the damping ratio associated with the complex-conjugate poles and moves the real pole toward lower frequencies. However, when the overall resistance decreases beyond a certain value and tends to zero the electric circuit becomes less effective in dissipating energy through Joule heating, which corresponds to the reduction of the damping performance. Also, the increased current produces a stronger magnetic field with opposite polarity with respect to the permanent magnet. The repulsive force produced by these two opposite magnetic fields is directly proportional to the displacement (i.e. it behaves like a spring [31]), which results in the increase of the system stiffness. Therefore, reducing beyond a certain value the overall resistance produces eventually the reduction of the damping force and the appearance of the spring-like behaviour. In the system under examination, the resistance of $0.1\ \Omega$ reached at the temperature of $-20\ ^\circ\text{C}$ causes the swap of the poles. However, this phenomenon does not compromise the EMSD damping performance since the maximum amplification is kept below 6dB.

3. Experimental set-up

3.1. Mechanical test rig

A laboratory test rig was built in house. It was designed to satisfy the requirement of the mass having only a single DoF, while trying to avoid the

Figure 5: Comparison of the force TFs between the system without EMSD and the system with EMSD at three different temperatures

285 test rig's vibration modes interfering with the data acquisition. The mass of 5
 286 kg was obtained through the combination of a dummy mass formed by three
 287 steel plates bolted together (4.85 kg) and the mass of the magnetic assembly
 288 (150 g). It is noted that the magnetic assembly and the two electromagnets
 289 weighted 190 g overall , which represents less than 4% of the total suspended
 290 mass. In order to simulate microgravity, the dummy mass was offloaded using
 291 bungee cords characterised by a sub-hertz resonance frequency. A suspended
 292 mini-shaker has then been connected to the centre of the dummy mass so that
 293 the vertical disturbance of a reaction wheel was reproduced. Steel flexures
 294 were used to guarantee that the mass could only move in the vertical direction,
 295 whereas displacements/rotations along the other five DoFs were minimised. The
 296 fully-assembled test rig bolted on a Kistler table is shown in Fig 6. A model
 297 of the test rig was built on Patran and analysed with Nastran (see Fig. 7).
 298 This finite element analysis showed that the test rig behaves as a SDOF up to
 299 150 Hz, where secondary modes in the flexures are excited, affecting the force
 300 TF. Ideally, these secondary modes were desired to be above 300 Hz, but the

301 dimensions of the Kistler table constrained the design of the flexures.

Figure 6: Experimental test rig. (a) test rig mounted on the Kistler table; (b) Patran model of the test rig

302 3.2. EMSD manufacturing

303 The design and realisation of the EMSD required an appropriate selection of
 304 the circuit components in order to reproduce the same conditions simulated in
 305 the analytical model. The clearance of 1 mm between the magnet and the coil
 306 (as already shown in Section 2.2) was chosen as safety margin to prevent the
 307 two elements to get in contact in case the device will be tested for the launch
 308 vibration environment. The choice of using self-bonding magnet wire allowed to
 309 fulfill the 1-mm-gap requirement. This type of wire is coated with an additional
 310 layer of adhesive polymer that is activated by heat or solvents. Once activated
 311 the adhesive bonds adjacent turns of wire together, forming a compact self-
 312 supporting coil. In this way, the inner spindle can be removed, and any extra
 313 spacing between the magnet and the coil is eliminated.

Figure 7: Force transfer function of the system without EMSD obtained through FEM (Nas-tran).

Another fundamental step was the realization of the negative-resistance electric circuit. Based on the analysis made with the freely-available LTSpice program, several op-amps have been investigated to verify that they performed as expected. Some of the required features include low quiescent current (which drives the minimum supplied power for the op-amp to operate), low input noise and the output current to be at least 50 mA. The LT1722 op-amp was finally selected as it met all the desired requirements (see Tab. 3).

4. Experimental results and discussions

The present study verifies the analytical model of Eq.(7) with experiments. The test rig was mounted on a Kistler table. The Kistler table isolation system (a suspension system lays underneath the table acting as a low pass filter) presented an 8-Hz resonance mode that required the tests to be divided into two different sine sweeps: a low-frequency one without the suspension system engaged, and a high-frequency one with the isolation system. The mini-shaker was controlled through a force feedback loop to generate first a 0.5-N-amplitude sinusoidal input force sweeping from 2 Hz to 20 Hz, and then a 1-N-amplitude

Property	Typ	Max
Input Noise Voltage ($nV/\sqrt{Hz}$)	3.8	
Quiescent Current (mA)	3.7	
Output Current (mA)	50	80 ^a
Supply Voltage (V)	± 5	± 6.3

^a Value measured experimentally.

Table 3: Main features of the LT1722 op-amp

330 sinusoidal input force sweeping from 15 Hz to 350 Hz. The input force of the
 331 first sine sweep was set to be 0.5 N so as to prevent the flexures from operating
 332 in the nonlinear region due to relatively large displacements (e.g. in the order of
 333 millimetres). The lab temperature was approximately 18 °C. The comparison
 334 between the experimental results and the analytical data (after tweaking the
 335 parameter set accordingly to the lab temperature) is shown in Fig. 8. Through
 336 the observation of the test results for the system with the shunt circuit discon-
 337 nected (see Fig. 8a), it has been able to asses the test rig’s mechanical damping
 338 that was initially omitted from the analytical model. This damping naturally
 339 occurs in a mechanical assembly and can be produced by several factors (e.g.
 340 micro-friction between adjacent components, air resistance, hysteresis damping
 341 in the steel flexures and Eddy-current damping due to the relative motion be-
 342 tween the magnet and other surrounding conductive materials, like aluminium).
 343 By adding a small amount of viscous damping in Eq. (7) (about 6% of the criti-
 344 cal damping), it can be observed that good correlation between the analytical
 345 data and the experiments has been achieved. The behaviour of the system above
 346 180 Hz is characterised by some resonance peaks that, as expected through the
 347 Nastran analysis, are attributable to the secondary vibration modes of the flex-
 348 ures. Good correlation can also be observed for the system with the EMSD (see

Fig. 8b). As already shown in Fig. 5, the TF shows a maximum amplification of 4.12 dB, a cut-off frequency at about 10 Hz, and a roll-off slope of -40 dB/dec. For this case, it can be seen that the additional mechanical damping that was previously added has almost a negligible effect on the damper performance.

The analytical model was further evaluated through the analysis of two other TFs. The relations between the input force and the mass velocity (for the case without the EMSD) and the induced current (for the system with the EMSD) can be obtained by modifying the output vector in Eq. (9) as follows:

$$Y = \begin{bmatrix} Y_v \\ Y_c \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} z \\ \dot{z} \\ I \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \end{bmatrix} F_{in} \quad (13)$$

These two outputs are measured respectively through accelerometers placed on top of the dummy mass and an oscilloscope probe connected to the coil's ends. The comparison of these analytically-obtained TFs with the experimental data is shown respectively in Figs. 9 and 10. The good correlation that can be observed for both TFs, along with the use of different kinds of sensors, are further confirmations of the accuracy of the analytical model and also of the correct functioning of the shunt circuit.

The maximum power required by the shunt (0.53 W) was registered at 5.60 Hz, where the highest current was drawn from the power supply. In particular, the current flowing in the electromagnets had an amplitude of 70 mA, and using Eq. (12) the average power required by the shunt at 5.60 Hz is obtained as:

$$P_{avg} = (V^+ - V^-) \cdot (I_q + I_{rms}) = 10V \cdot (3.8mA + 0.707 \cdot 70mA) = 0.53W \quad (14)$$

This value of the average power of this semi-active system is considerably lower when compared with active isolation methods, where actuators and sensors need to be driven (e.g. in Ref. [14] the single strut requires a minimum of 15 W to operate).

(a)

(b)

Figure 8: Comparison between the test results and the analytical model. The dashed line does not include the extra mechanical damping, whereas the dotted line does include it. (a) system without EMSD; (b) system with EMSD.

Figure 9: Transfer function between the input force and the mass velocity. The system considered is without EMSD.

Figure 10: Transfer function between the input force and the induced current in the coil. The system considered is with EMSD.

372 5. Conclusions

373 In the present work, a negative-resistance EMSD has been studied both an-
374 alytically and experimentally. The proposed isolator uses a negative-resistance
375 analogue circuit to reduce the inherent resistance of the electromagnetic coils,
376 and as a result, the damping of the system can be increased. A 5-kg, SDOF
377 system has been employed to demonstrate the performance of the proposed
378 damper. The governing system of equations for this system has been established,
379 and the relative state-space model has been reported. The electro-mechanical
380 transducer coefficient has been analysed through the software COMSOL Multi-
381 physics and its value has been increased by changing the disposition of magnets
382 and iron yokes. A parametric trade-off has been carried out in order to meet
383 the requirements of maximum amplification at the resonance peak and low cut-
384 off frequency within the operating temperature range of -20°C to $+50^{\circ}\text{C}$. The
385 test rig has been built in-house, and the results from a test campaign have cor-
386 roborated the accuracy of the analytical model. This paper has demonstrated
387 that this technology is capable of isolating micro-vibration sources (e.g. reac-
388 tion wheel assemblies) without requiring an active control algorithm. The low
389 required power, the simplified electronics and the small device mass make this
390 EMSD extremely interesting for future space applications.

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