CONCEPT ASSESSMENT FOR A 2-COLLINEAR-DoF STRUT PROTOTYPE WITH EMBEDDED ELECTROMAGNETIC SHUNT DAMPERS

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ABSTRACT

This paper presents the design challenges and preliminary tests of a 2-collinear-DoF strut prototype aimed at making this technology eventually available for future space missions. The strut is intended as a novel micro-vibration isolation system. Two separate, independent EMSDs are embedded within the strut to produce a two-level damping. This work demonstrates the feasibility of achieving a low cut-off frequency (below 10 Hz) and a remarkable decay rate of -80 dB/decade with a device that is smaller than previously-presented active struts (the proposed strut has an overall mass that is about 10% of the suspended mass) and does not require complex electronics to operate. The strut presented in this paper uses a small circuit board that consumes less than 0.1 Watts and is highly robust, which make this device extremely interesting for future space applications.

INTRODUCTION

Many space sensing and communication payloads such as astronomical telescopes and laser communication devices demand high pointing accuracy from the host spacecraft in order to perform their mission. Micro-vibrations are typically the main cause of limited platform stability and they are generated by several on-board subsystems and devices, such as reaction wheel assemblies, momentum wheel assemblies, control moment gyros and cryo-coolers. Developing a low-noise spacecraft bus to target a desired performance is usually one of the options to tackle this issue but it comes with a considerable increase in system cost. On the contrary, the use of isolation systems aimed at dynamically decoupling either the sensitive payload or the disturbance source from the satellite structure was proved to effectively counteract the micro-vibration issue without requiring expensive, time-consuming design and modification of the satellite bus.

Passive isolators are mostly preferred due to their constructive simplicity, compact size and reliability. Viscoelastic materials are often used by companies in the space sector given their low cost, reduced mass and good high frequency attenuation (final slope of -40 dB/dec) [1-2]. However, the impossibility to completely eliminate the resonance peak due to their limited loss factor and the strong dependency on the operating temperature and frequency ranges make the design of viscoelastic dampers quite complex.

Active or hybrid isolators could overcome some of these limitations, especially at low frequency. Active isolators are tuneable and capable of producing higher damping force that allow them to achieve more stringent conditions of isolation [3-4]. Nevertheless, active isolators (as well as hybrid systems because they are made of an active system in parallel with a passive one) use external actuators and sensors to provide control forces and feedback signals, and for this reason they require a significant amount of power to operate.

Differently from hybrid techniques, semi-active isolators might be considered mainly as passive methods with some components of the isolation system that can be actively modified. These isolators provide better mitigation performance than pure passive systems and are more reliable than pure active systems given the absence of actuators and the limited amount of power required to function. Among other techniques (e.g. use of smart materials like magnetorheological fluids), electromagnetic transducers connected to negative impedance circuits have been proved recently to function as semi-active components. In fact, although requiring input power, these systems can produce a considerable isolation performance without requiring any control algorithm [5].

This paper focuses on the development of a 2-collinear-DoF strut prototype with two embedded electromagnetic shunt dampers (EMSD) for micro-vibration isolation purposes. The main advantage of the proposed technology is the possibility to exploit the bipolarity of a single magnet to obtain two separate EMSDs [6]. A first design of the prototype is here presented followed by a preliminary analysis of the device. This work represents an initial step towards the full characterisation of the strut and the possibility to integrate six of them into a hexapod platform for 6-DoFs isolation purposes.

ANALYTICAL MODEL

A schematic of the system studied in this paper is shown in Fig. 1. This model consists of a magnet stack m_2 that is connected to a primary mass m_1 and to the ground via two separate springs with stiffness coefficients k_1 and k_2. The two masses can only move along their longitudinal axis, thus resulting in a 2-collinear-DoF system. By exploiting the bipolarity of a magnet, a two-level damping can be obtained using two separate electromagnets that are rigidly connected respectively to the suspended mass and to the ground. Each electromagnet is then connected to different shunt impedances thus forming two separate dampers ($EMSD_1$ and $EMSD_2$). By including negative resistance circuits in the shunts, it is possible to reduce the overall resistance of the EMSDs and increasing the current flowing in the electromagnets, thus producing higher damping force. The negative resistance converter circuit can be implemented using few electric components (one operational amplifier and three resistors, as shown in Fig. 2) and it requires little power to function because it does not need external sensors or control systems. These are main advantages with respect to active systems.

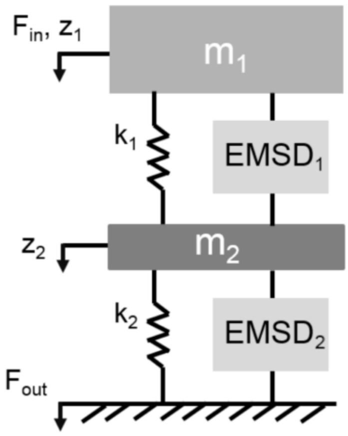


Figure 1. Schematic representation of the 2-collinear-DoF model

The analytical model was presented in [6] and for simplicity it was decided not to include it in this paper. However, it is important to note that by fixing the geometrical features of the magnetic stack, there were still six parameters that could be tweaked to adapt the dynamic response of the system to meet pre-determined requirements. The six parameters were the stiffness k_1 and k_2 and the electric properties of the two EMSDs (resistance and inductance). The attenuation performance of the proposed isolator was defined by three goals that needed to be met in the force transfer function between the input force, F_{in}, and the force transmitted to the ground, F_{out}. They were:

- Maximum amplification below 6 dB.

– Corner frequency at 10 Hz or below.

– At least −40 dB at 100 Hz.

Along with these goals, it was crucial to guarantee the stability of the system for the whole temperature range of operation (from -20 °C to +50°C which is a typical temperature range for a reaction wheel). The only parameters that have been assumed temperature dependent were the electromagnet resistance (the copper resistivity has a thermal coefficient of 0.00386 $°C^{-1}$) and the magnetic field (the residual induction of the Nd–Fe–B magnets is characterized by a thermal coefficient of -0.0012 $°C^{-1}$). The other parameters were considered temperature independent (e.g. the electric components for the negative resistance circuits can be chosen among space-qualified parts that have tolerances down to 0.005% and temperature coefficients of $1 \cdot 10^{-6}$ $°C^{-1}$). A trade-off was carried out and the final set of parameter values produced a system capable of meeting all the requirements in the whole temperature range of interest.

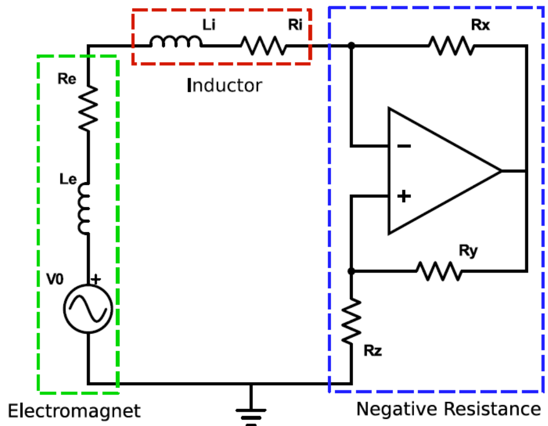


Figure 2. Schematic of the EMSD electric circuit used for the 2-collinear-DoF system. The shunt circuit includes the negative resistance converter and an iron-core inductor

Fig. 3 shows the comparison of the force transfer function between the system when the EMSDs are switched off and the one with the EMSDs working nominally. It can be observed how the temperature only slightly affects the dynamic response of the system around the two resonance frequencies. Nevertheless, these variations do not compromise the dynamic response of the system and the final slope of -80 dB/dec is preserved. Fig. 4 reports the transfer function of the proposed isolator compared with the transfer function of a 1-DoF system with EMSD [5] and the one of a passive isolator that uses viscoelastic materials [1]. The performance advantages of the proposed strut are clearly visible not only at low frequency (with the elimination of the resonance peak that is characteristic of viscoelastic materials) but also at high frequency with a remarkable final roll-off slope.

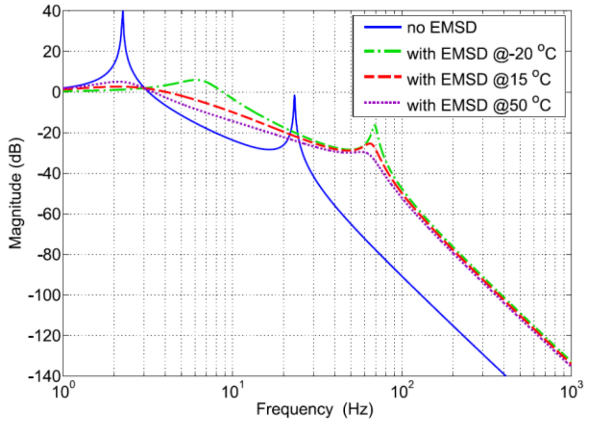


Figure 3. Comparison of the analytical solution of the force TFs between the system without EMSD (i.e. electromagnets in open-circuit conditions) and the system with EMSD at three different temperatures.

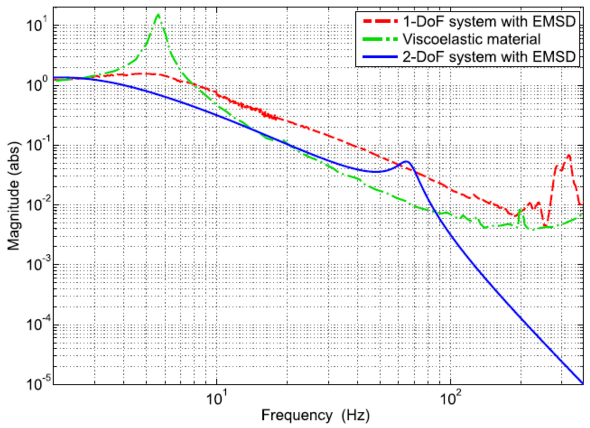


Figure 4. Comparison of the force transfer functions between three different damping systems: 1-DoF system with EMSD presented in [5], viscoelastic-material passive isolator presented in [1], and the 2-collinear-DoF system proposed in this paper

STRUT CONCEPT DESIGN

A previous work demonstrated the proof of concept of the isolator using a cumbersome test rig with an overall weight of more than 25 kg [6]. The ability to scale down the proposed technology into something small and compact would be fundamental to make it suitable for a space mission. A concept design of the strut was first attempted. This device would behave as described by the analytical model only as long as the two DoFs are along the same longitudinal axis while all the other displacements and rotations are prevented. To obtain such a configuration, thin stainless-steel flexures with circular shape (resulting in an isotropic behaviour) were thoroughly investigated. The final choice can be seen in Fig. 5. This flexure has a diameter of 86mm and a thickness of 0.15mm.

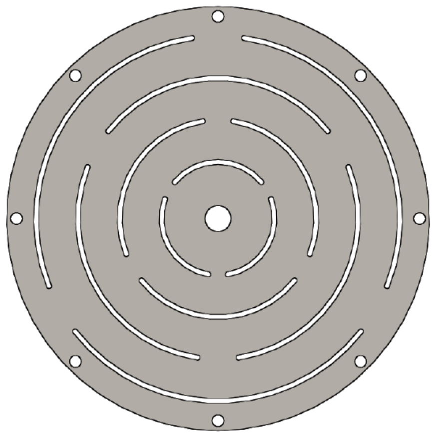


Figure 5. Flexure design finally chosen for the strut

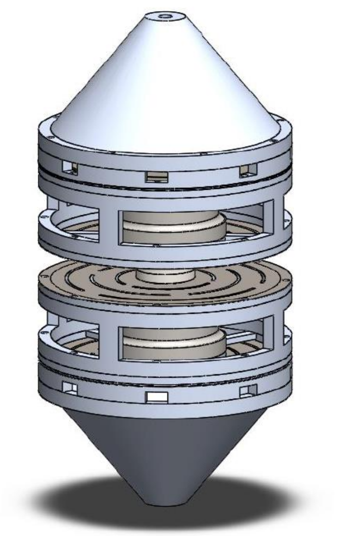


Figure 6. Preliminary design of the 2-collinear-DoF strut prototype.

A CAD model of the preliminary design of the strut can be seen in Fig. 6. Each strut is made of two identical halves with a magnetic stack in between. The stack is only connected to each half of the strut through four of the aforementioned flexures. The main structure was initially thought of being made of aluminium (light grey parts), but a graphite reinforced plastic was finally chosen for the first prototype. This material is characterised by remarkable mechanical properties and the possibility to rapid prototyping the parts via additive manufacturing. The coil holders (dark grey parts) were instead made of magnetic steel because they could provide a double benefit: increasing the radial magnetic field seen by the coils (i.e. enhancing the electromechanical coupling) and at the same time shielding the external space from the strong magnetic field produced by the permanent magnets [6]. The overall mass of the strut is about 500 g.

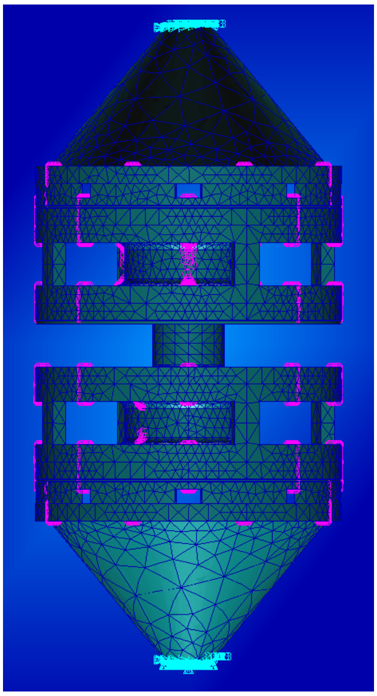
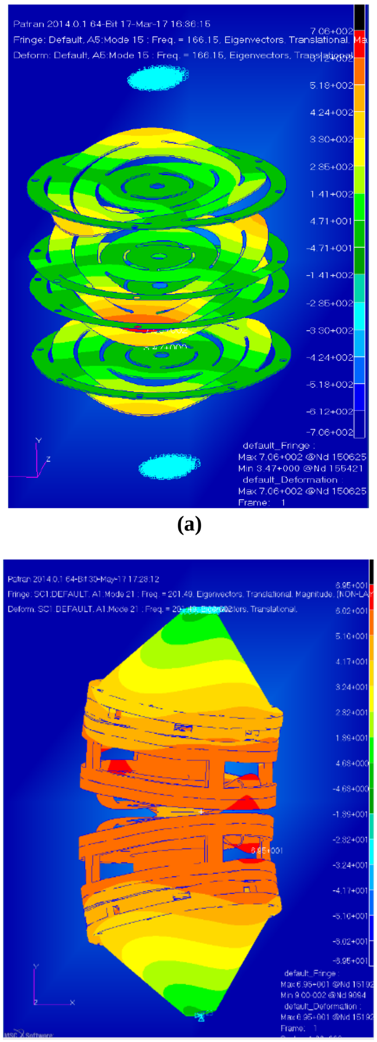


Figure 7. Patran model of the 2-collinear-DoF strut prototype

A finite element analysis of the strut was performed in Patran/Nastran. The extremity of the bottom cone was pinned (all displacements prevented) whereas the top cone's end had the vertical displacement allowed along with all the three rotations (see Fig. 7). These are the boundary conditions that the strut would more likely have if it were integrated in a hexapod platform.

A modal analysis was carried out in the frequency range from 1 to 500 Hz. Apart from the first two modes that were expected because of the two collinear DoFs, the analysis showed several secondary modes due to the flexures starting from 160 Hz onward. An example of one of these modes can be seen in Fig. 8a. Given the small masses of the flexures, these local modes should have a little impact on the strut transmissibility. However, expedients to reduce such modes are under investigation. The finite element analysis determined also the presence of other two structural modes in the frequency range of interest: a bending mode at about 201 Hz (see Fig. 8b) and a torsional mode at about 317 Hz. The latter mode would be less likely to be excited, whereas the former one needs particular attention because it would be complicated to obtain zero misalignment of the input force along the vertical axis, and this could excite the bending mode.



(a)

(b)

Figure 8. FEA of the proposed strut conducted with Nastran. Some critical modes are reported in this figure. a) one of the local modes due to the thin flexures starting from 160 Hz. b) bending mode of the whole strut happening at about 201 Hz

For this preliminary test phase, it was decided not to use pin joints. Their design would require further investigation because they are crucial elements also for the future design of the hexapod platform. Instead, a fixed-free configuration is going to be used as a first characterisation of the strut. This configuration, apart from being easily implementable, allows for the elimination of the torsional mode in the range of interest while the bending mode is still around 200Hz.

PRELIMINARY TESTS

The problem of suspending the primary mass with flexures was the creation of a direct load path from the mass m_1 to the ground (partly bypassing the secondary mass) which resulted in the limitations explained in [6]. However, eliminating the flexures and using only low-stiffness bungee cords to suspend a mass of 5 kg is practically extremely challenging. The application of the dynamic stiffness method could help overcome this issue.

The system under examination can be separated into two subsystems, as shown in Fig. 9, where the subsystem 2 represents the strut. This method would allow for the strut to be tested without the suspended mass and the dummy mass could be added analytically afterwards to retrieve the overall transfer function as explained in this section.

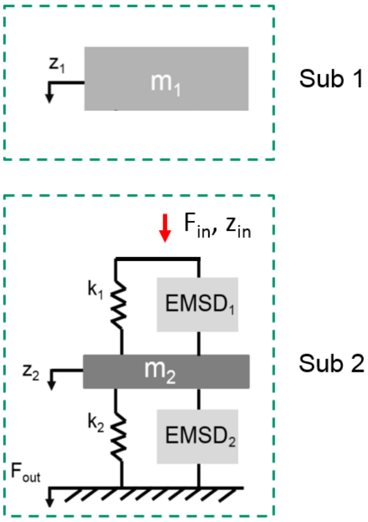


Figure 9. Separation of the 2-collinear-DoF system into two subsystems

The force transfer function for the subsystem 2 (TF_{sub2}) can be obtained using two force transducers to measure both the input force at the top of the strut and the force transmitted to the ground, thus resulting in:

$$TF_{sub2} = \frac{F_{out}}{F_{in}} \tag{1}$$

The interface interaction between the subsystem 1 and 2 can be represented via the dynamic stiffness (see Fig. 10). This stiffness (DS_1) is defined as the ratio between the force and the displacement in the interface point. In this case, DS_1 is equal to:

$$DS_1 = \frac{F_{in}}{z_{in}}. \tag{2}$$

Therefore, by adding an accelerometer on top of the strut and double integrating the measured acceleration to obtain the displacement, one can determine the dynamic stiffness in the frequency domain. The equation of motion of the subsystem 1 becomes:

$$m_1\ddot{z_1} + DS_1 z_1 = F_1 \tag{3}$$

And after using the Laplace transform it is possible to compute the transfer function as:

$$TF_{sub1} = \frac{F_{int}}{F_1} = \frac{DS_1 z_1}{F_1} = \frac{DS_1}{ms^2 + DS_1} \tag{4}$$

In which the force F_{int} represents the intermediate force exchanged between the subsystem 1 and 2.

Having the two transfer functions, the overall transfer function can easily be obtained as:

$$TF = TF_{sub1} \cdot TF_{sub2} = \frac{DS_1}{ms^2 + DS_1} \cdot \frac{F_{out}}{F_{in}} \tag{5}$$

Where the only unknown is the mass of the suspended mass. Therefore, this method allows the strut to be tested for different values of the primary mass.

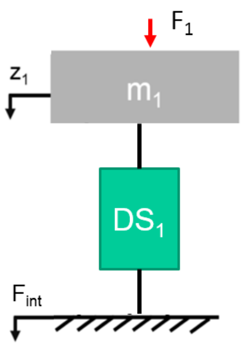


Figure 10. Representation of the subsystem 1 that is connected to the equivalent dynamic stiffness produced by the subsystem 2

Before proceeding with the strut characterisation through an experimental campaign using a multi-axial dynamometric table to record the transmitted force to the ground, the single flexures were tested with a compression/tension machine. In particular, three flexures were tested separately, both in the top and bottom configuration (i.e. the flexures were flipped between tests). Each one was individually clamped between two rings made of graphite reinforced plastic as shown in Fig. 11. The assembly was then simply supported on a table where the compression/tension machine could apply the desired force through a narrow rod pressing at the centre of the flexure. The test results have been compared with the simulation results obtained via a nonlinear static analysis in Nastran. Fig. 12 shows that in the linear region the test results confirmed the predictions from the analytical data. However, as the displacement increased, a larger discrepancy could be observed between tests and simulation, and the real specimens resulted to have smaller stiffness than predicted. This behaviour could be explained by the fact that the graphite reinforce plastic support is not infinitely rigid and micro displacements could have occurred. This condition would then differ from the boundary conditions defined in the Nastran model in which all the displacements and rotations at the edge of the flexure were prevented. Nonetheless, for the first characterisation of the strut the displacements are going to be in the order of few tenths of a millimetre and the flexures show to maintain a linear behaviour in that region.

Figure 11. Support used to measure the axial stiffness through a compression/tension machine

The evaluation of the stiffness together with the assessment of the electric coefficients (inductance and overall resistance for both EMSDs) are the parameters needed in the analytical model to predict the strut performance. The two electric circuits have been manufactured and connected to the electromagnets. The features of the two shunt circuits have been accurately measured with a multimetre and their dynamic behaviours (i.e. amplification of the current flowing in the circuit) have been verified using a function generator to reproduce the sinusoidal induced voltage within the electromagnets. With the determination of all these parameters, the analytical model is now calibrated with respect to the test rig and an experimental campaign will be soon conducted to corroborate such predictions.

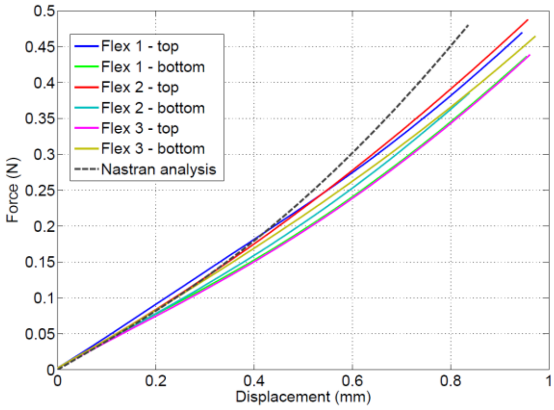


Figure 12. Comparison of the force/displacement behaviour between 3 different flexures (top and bottom side of each one) and the analytical data obtained with nonlinear analysis in Nastran

CONCLUSIONS

This paper presented some of the design challenges and preliminary tests of a 2-collinear-DoF strut prototype with two embedded EMSDs. Previous research demonstrated that the proof of concept of the strut was capable of achieving a remarkable performance compared with other well-established isolation systems, but a cumbersome test rig was used to verify such behaviour. A concept design of the strut was here proposed. The device is mainly made of graphite reinforced plastic and it has an overall mass of about 500g corresponding to approximately 10% of the suspended mass. A modal analysis conducted in Nastran showed that the strut was characterised by a couple of undesired structural modes and several secondary modes produced by the flexures. Expedients to eliminate or reduce these modes will be the main focus of future research. The paper concluded with the description of the dynamic stiffness method that could considerably simplify the test rig (no need to suspend the primary mass on top of the strut), and the evaluation of the flexures stiffness through a compression/tension machine which is an important parameter that will help in the correlation between the analytical model and future test results.

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