Electromagnetic Shunt Damper for Spacecraft Micro-Vibration Mitigation

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Statement of Originality

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Alessandro Stabile
December 2017
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Abstract

The stringent stability requirements imposed by advanced, high-resolution payloads have produced an increased interest in the development of better-performing micro-vibration isolators. Several devices aimed at mitigating micro-vibrations have been studied and implemented, but their application is still far from being ideal due to the several drawbacks that they present, such as limited low-frequency attenuation for passive systems or high power consumption and reliability issues for active systems.

This research focuses on the modelling and testing of Electromagnetic Shunt Dampers (EMSD) characterised by the use of negative impedance converter circuits. An electromagnetic damper is a self-excited device that exploits the interaction between a moving magnetic field and a conductive material to provide a reaction force to the applied motion. An EMSD presents several advantages, but the high ratio of system mass over damping force produced has limited its application in space missions. The use of a negative resistance can considerably lower this ratio since it 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.

In this thesis, the development of a multiphysics, multi-parametric model of an EMSD is presented and accurately corroborated by an extensive test campaign. This damper can be classified as a semi-active damper since the negative resistance circuit does not require any control algorithm to operate. In terms of damping performance, this research demonstrates that an EMSD applied to a 1-DoF system is capable of behaving, throughout the whole temperature range of interest, like a 2nd-order mechanical filter in which the resonance peak is eliminated and the roll-off slope is -40 dB/dec. Additionally, the proposed EMSD is characterised by low required power, simplified electronics and small device mass that could allow it to be comfortably integrated on a satellite.

This study presents also a possible novel 2-collinear-DoF system design with embedded EMSDs. This isolator is capable of achieving a remarkable final decay rate of -80 dB/dec while completely eliminating the two resonance peaks due to the high attenuation performance of the dampers. Moreover, other aspects of the proposed 2-collinear-DoF system are investigated in order to assess not only the damping performance but also its features at system level. This work demonstrates that the fundamental advantages of this system can make it a viable, competitive alternative to other actively controlled struts.
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<td>AC</td>
<td>Alternate Current</td>
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<tr>
<td>DC</td>
<td>Direct Current</td>
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<tr>
<td>DoF</td>
<td>Degree of freedom</td>
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<td>ECD</td>
<td>Eddy-Current Damper</td>
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<td>EMSD</td>
<td>Electromagnetic shunted damper</td>
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<td>ESA</td>
<td>European Space Agency</td>
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<tr>
<td>FEA</td>
<td>Finite Element Analysis</td>
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<td>FEM</td>
<td>Finite Element Model</td>
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<td>GEO</td>
<td>Geostationary Earth Orbit</td>
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<td>GSD</td>
<td>Ground Sample Distance</td>
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<td>HAI</td>
<td>Hard Active Isolators</td>
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<td>JPL</td>
<td>Jet Propulsion Laboratory</td>
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<td>KT</td>
<td>Kistler Table</td>
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<td>LEO</td>
<td>low earth orbit</td>
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<td>MD</td>
<td>Magnetic damper</td>
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<td>MEO</td>
<td>medium Earth orbit</td>
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<td>MFS</td>
<td>Multiaxial Force Sensor</td>
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<td>MVIS</td>
<td>Miniature Vibration Isolation System</td>
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<td>NASA</td>
<td>National Aeronautics and Astronautics Administration</td>
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<td>PA</td>
<td>Piezoelectric Actuator</td>
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<td>PZT</td>
<td>Piezoelectric transducer</td>
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<td>Transfer function</td>
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<td>TMD</td>
<td>Tuned Mass Damper</td>
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Chapter 1

Introduction

1.1 Background

In the last few decades, the need for improving the platform stability of optical/imaging missions has resulted in an extensive investigation aimed at developing more accurate structural dynamic models and more effective damping systems to reduce disturbances on board satellites. The research on spacecraft micro-vibration and its control was mainly bolstered by the high-resolution requirements of the Hubble Space Telescope project in the early 1980s [1, 2]. Since then, it has gained an increasing interest both in the commercial and scientific world. These mechanical disturbances are usually in the magnitude order of micro-g accelerations which occur at frequencies ranging from a few Hz up to several hundred Hz.

The flutter produced by micro-vibration disturbances can significantly affect the pointing accuracy of a laser transmitter or the performance of a camera. An example is given in Figure 1.1 where the effect of micro-vibration is shown for two different types of detectors. In case of an area detector (typical of digital cameras where a 2-D matrix of pixels is recorded at once), micro-vibration would result in a blurred image with reduced sharpness, whereas in case of a push broom detector (where a 1-D array of pixels is recorded and the image of the ground is built up one line at a time), micro-vibration perturbations would result mostly in a distorted image.

The propagation of micro-vibrations throughout the satellite structure can be represented schematically as shown in Figure 1.2. Micro-vibrations are generated by several on-board subsystems and devices, such as reaction wheel assemblies, momentum wheel assemblies,
1.1 Background

**Figure 1.1:** Simulated images demonstrating effects of micro-vibration on different types of detector [3]

**Figure 1.2:** Schematic representation of the propagation of micro-vibrations
1.1 Background

control moment gyros, pointing systems and cryo-coolers. In particular, reaction wheels are seldom considered the most important source of micro-vibrations. Their disturbances come but significant disturbances can also be produced by other mechanisms such as thrusters and electronic components (motors, data storage devices and sensors). Micro-vibrations propagate through the satellite structure towards sensitive instruments and payloads, defined as receivers, thereby jeopardizing their correct functioning. The same satellite structure might amplify disturbances coming from the sources if these micro-vibrations excite the structure at its own natural frequencies.

The issue of the micro-vibration is addressed via four different approaches:

• Disturbance reduction at source level. Several studies on key mechanisms (mainly reaction wheels) aimed at better understanding the cause of micro-vibration generation have been carried out in the last two decades [4–6]. In particular, imperfections in the ball bearings have been identified as one of the main source of disturbances and for this reason passive damping methods are seldom included within the mechanisms to reduce those disturbances (e.g. in [7] a layer of viscoelastic material is placed between the outer race and the bearing support).

• Noise sensitivity reduction at payload level. Optical devices can be designed to be less sensitive to disturbances in certain frequency ranges. Image stabilisation can be performed either digitally (by manipulating the picture through a software algorithm [8]) or mechanically (by acting on the optical lenses [9, 10]).

• Alteration of the micro-vibration transmission path. Through extensive FEA, the prediction of the micro-vibration propagation in the spacecraft bus can be determined [11, 12]. By modifying the transmission path it is possible to reduce the disturbances reaching a specific area where a sensitive payload might have been placed [13, 14].

• Isolation of the sources or receivers from the satellite structure. Isolation systems prevent vibration disturbances from being transmitted to the satellite platform and vice-versa. They can be passive, active or a combination of the two, and several examples have been studied and developed for space applications [15].

These four methods are analysed in parallel by the academic community and space industry because of the unique advantages that each method offers, but it is also important to understand their limitations. For instance, optical image stabilisation is regularly included in modern smartphones to compensate for hand jitter, but its use in large cameras or telescopes (e.g. via deformable mirrors in adaptive optics) still requires several actuators and sensors which might result in a complex architecture. At source level, reducing the disturbances
by improving the manufacturing process of mechanisms has resulted in devices that are producing less micro-vibration disturbances. However, controlling and reducing the amount of imperfections is extremely expensive (since several iterations in the manufacturing process are needed) and it is still limited by the achievable machining tolerances, hence minor defects cannot be completely eliminated. On the other hand, reconfiguring a spacecraft bus in order to change the transmission path and target a desired performance might not only come with a considerable increase in system cost and design constraints but it is also incompatible with the current trend towards lighter and smaller structures. Finally, isolation systems can drastically improve the mitigation of micro-vibrations while alleviating some crucial design constraints in the overall spacecraft structure and mechanisms. Nevertheless, passive devices have limited performance and actively controlled systems come usually with considerable increase of mass and required power. For this reason, active isolation platforms are rarely employed in space missions.

Out of the four aforementioned methods, this work will focus on the isolation of micro-vibration disturbances with the aim of addressing fundamental aspects (i.e. attenuation performance, reliability, system mass and power consumption) that are limiting their use on board satellites.

1.2 Research Motivation

This project started with the identification of promising isolating technologies which, with the right improvement, would be able to significantly promote the mitigation of micro-vibration on board satellites. Passive isolators are usually the cheapest and lightest damping solutions and are widely used by space companies. However, they have some down sides that often limit their performance (i.e. viscoelastic materials are dependent on temperature and frequency operative ranges and this makes their design extremely complex). On the other hand, actively-controlled isolators can drastically improve the micro-vibration attenuation but their use in space applications is almost negligible due to excessive increase of costs, system mass and complexity. Therefore, there is an growing need both from the public and private sector to identify alternative isolating solutions that will allow the spacecraft platform to reach an unprecedented stability in order to meet more ambitious requirements (e.g. high resolution imagery from GEO orbit [16]).

In terms of optical payloads, a sub-metre image resolution is nowadays achievable in the lower range of LEO orbits (e.g. the 0.5-m-resolution image shown in Figure 1.3 was taken
1.2 Research Motivation

by WorldView-2, a satellite with an overall mass of 2800 kg that flies at an altitude of 772 km), but it still necessary to have better platform stability to obtain high resolution imagery from MEO or GEO orbits [16]. The common trend among companies providing imaging missions is to have smaller satellites ideally capable of acquiring images with resolution well below 1 m at higher altitudes, and this cannot be guaranteed with the current state of the art of isolating technologies.

Scientific missions that analyse the space surrounding the earth in order to better understand our universe often require ultra stable platforms to achieve their goals. An example is represented by the Euclid spacecraft, a survey mission of ESA aimed at investigating the nature of dark energy, dark matter and gravity [17] (see an artistic representation of the spacecraft in Figure 1.4). The pointing accuracy needed for this mission is extremely strict (relative pointing error of 75 milli-arcseconds over 700 sec and an absolute pointing error of 7.5 arcsec both with 99.7% confidence level) and during a preliminary design phase it was established that the noise coming from the reaction wheels would have compromised the success of the mission. For this reason, a cold-gas micro-propulsion system was added to be employed for fine attitude control while keeping the reaction wheels off so that the stability requirements would be met during science observations. Therefore, although solving the pointing issue, this system presents some drawbacks such as added pressurised tanks with 70 kg of Nitrogen, an intricate piping system and an overall cost attributed to it in the order of 10 M Euro. It is likely that if an enhanced isolation system had been ready to be used for a space mission back in 2012 when the Euclid mission was selected, then the use of reaction wheels could alone have been enough to meet the fine pointing requirements and save money and energy trying to come up with alternative solutions.

Isolation platforms capable of attenuating disturbances in all six degrees of freedom have been widely studied in the last three decades, but more research and a deeper analysis is still needed in order to make these systems viable solutions for long-lasting missions. Isolators not necessarily include a damping element to perform their tasks, but in case there is a need for isolating a broadband disturbance the use of a damper is crucial (more details can be found in the next chapter). Damping elements are used to attenuate isolators’ resonance peaks but they often reduce also the high frequency micro-vibration mitigation. Designing a damper capable of functioning only in restricted frequency ranges will be the challenge of the next generation of this devices and extensive research is still necessary to combine this performance feature with other aspects at system level (e.g. system mass and power consumption).
1.2 Research Motivation

During the first part of this work a wide variety of damping systems has been investigated with the main purpose of assessing advantages, disadvantages and most importantly the performance limits of each device. Among them, magnetic dampers exploiting the interaction between a magnetic field and a conductive material have drawn our attention. These devices present several advantages, such as:

- the contactless damping force that they produce,
- the great versatility that allows these systems to be used as passive or active dampers,
- the broad environmental conditions that they can sustain compared, for instance, to viscoelastic materials (since they are only made of metal parts).

Eddy-current dampers and electromagnetic dampers belong to the family of magnetic dampers. For more than twenty years, applications involving these kinds of dampers have been investigated, including magnetic braking systems (either applied to a sliding track [18, 19] or a rotating disk [20]), structural vibration suppression of beams and plates [21, 22], vibration isolation enhancement in levitation systems [23], and vehicle shock absorbers [24].

Nevertheless, the employment of magnetic dampers for space applications has been almost negligible. The main drawback is the limited damping coefficient that this technology can
achieve in the passive configuration [25, 26], and consequently the high ratio of system mass over damping force produced. Several studies have addressed this limitation with the employment of active electromagnets (instead of using permanent magnets) that are capable of increasing the damping performance through the control of the magnetic field [27, 28]. However, the substantial amount of input power required and the narrow operative frequency range still do not allow this configuration to be employed in a space mission.

On the other hand, the connection of a shunting circuit to the solenoid terminals of an electromagnetic damper can also produce a damping performance boost. By carefully selecting the electrical components of the circuit, it is possible to completely change the device frequency response and tune it to the satellite structural modes [29]. Recently, Electromagnetic Shunt Dampers (EMSD) that employ negative impedance circuits have been used successfully to mitigate vibrating cantilever beams and plates [30, 31]. These devices function as semi-active dampers because although they require power to operate (due to the use of op-amps to generate the negative impedance) they do not need any control algorithm or external sensor to perform their tasks. Still, the limitations related to dimensions, maximum input power and allowable temperature need to be addressed in order to make this damper suitable for use in space missions.
1.3 Aim and Research Objectives

The use of isolation systems is often considered an effective and efficient way to tackle the issue of micro-vibrations on board spacecraft [15]. However, it is still quite challenging to completely isolate noise sources from the spacecraft bus. For this reason, there is often a need to come up with alternative, more expensive solutions to overcome this issue (e.g. cold gas micro-propulsion system). The interest in improving the capability to effectively isolate either a noisy mechanism or a sensitive payload is especially driven by the companies that deal with satellites, as the current platform stability of satellites is not enough to guarantee sub-metre image resolutions at high altitudes. In this light, developing an innovative isolation system could be a game changer in the small-satellite market because it would allow higher image resolutions or better pointing accuracy at a more affordable cost. Hence, the EMSD was chosen as the technology that has the greatest potential to enable such a goal.

Therefore, the aim of this research is to advance the state of the art of the EMSDs with the employment of negative impedances. This advancement will lead to a better exploitation of the capabilities of this damper for its use in space applications, and will ultimately affirm this technology as a viable alternative with respect to the other well-established micro-vibration dampers.

In order to achieve this aim, the following research objectives needed to be addressed:

- Develop a multiphysics mathematical model of the EMSD capable of simultaneously evaluating its electro-mechanical and thermal behaviour;
- Investigate the properties of the negative resistance converter when connected to an electromagnetic damper;
- Determine through a trade-off what magnet configuration is capable of producing the highest ratio of damping force over system mass;
- Design a proof of concept that is able to reproduce accurately the presented analytical model;
- Verify the analytical predictions against test results for a 1-DoF system with an EMSD;
- Develop an analytical model for a novel 2-collinear-DoF system with embedded EMSDs and test its damping performance under laboratory testing conditions;
- Explore other aspects of the proposed isolator that go beyond the sole evaluation of the damping performance in order to address some of the limitations and issues that active
and hybrid isolators have encountered when used in space applications (e.g. system mass, power consumption or environmental conditions).

## 1.4 Novel Contributions

The space sector needs an isolation system that is capable of considerably improving the state of the art in the micro-vibration attenuation while being extremely reliable and cost effective. In order to achieve this, such a system will need to be characterised by mechanical simplicity and performance versatility along with contained costs and added mass and power. This study represents a step forward in this direction providing a full characterisation of a novel isolation system which takes into account not only the mitigation performance but also other important aspects at sub-system level.

More specifically, the envisaged novelties to the state of the art that have been made by this activity in pursuit of the objectives stated above are:

- The development of a multiphysics, multi-parametric model of an EMSD that has been corroborated accurately by an extensive test campaign. This model will help future research in the optimisation process of the whole device;

- The full characterisation of the negative resistance converter circuit that has simplified its use and has allowed for the isolator to increase considerably its damping performance without requiring heavy electronics or large amount of external power;

- The development of a novel 2-collinear-DoF isolator that combines the enhancement of the micro-vibration attenuation with the simplification of the system architecture and the absence of a control algorithm. The two DoFs have been obtained by exploiting the bipolarity of magnets which has allowed the inclusion of two separate EMSDs linked to the same magnet. This device has resulted extremely competitive with comparison to both passive isolators (due to the complete elimination of the resonance peak and the response versatility) and active isolators (since its linear behaviour is not determined by a control algorithm or large power supply).

The proposed isolation methodology could also find interesting applications in other fields with the possible advantage of having less constraints in volume and mass. For instance, in automotive there is a continuous interest in enhancing the passenger comfort and it could be achieved by better isolating the engine from the frame or by improving the seats damping system. Also in civil engineering, great attention is paid to the development of isolation
1.5 Structure of the Thesis

The overall structure of the thesis is organised into 7 chapters which are outlined in this section.

Chapter 2 contains the literature review related to the use of isolating systems aimed at mitigating micro-vibration disturbances on board satellites. In particular, great attention is given to the description of different damping methods that have been developed in the last forty years and that can be divided into four categories: passive, active, hybrid and semi-active dampers. The chapter concludes with a detailed analysis of several isolators that have been proposed for space applications with the goal of presenting their performance properties and limitations.

In Chapter 3 the analytical model of an EMSD is extensively investigated. First, the isolation performance of an EMSD applied to 1-DoF system is studied. It follows the analysis of two fundamental aspects of this damper: the choice of the magnetic configuration and the full characterisation of the negative resistance converter circuit. Finally, the development of a multiphysics approach that allows the system to be studied under a wide temperature range and consequently the parametric trade off aimed at meeting the performance requirements are presented.

In Chapter 4 the experimental validation of the analytical model reported in the previous chapter is described. In particular, the test rig is thoroughly investigated through FEA in order to give a complete overview of the expected results obtained with the test campaign.
and a clear distinction between the data related to the examined isolator and to the test rig itself.

Chapter 5 presents the novel 2-collinear-DoF system with two embedded EMSDs. A benchmark study with a 2-collinear-DoF system that uses two viscoelastic materials is first presented with the goal of creating a reference system that is then compared with the system proposed in this chapter. It follows the presentation of the analytical model and the parametric trade off that has brought to the determination of 6 parameters of the proposed isolator in order to meet the performance requirements. Finally, an extensive investigation of the modified test rig and the result data obtained through a second test campaign are reported.

Chapter 6 presents the analysis of other important aspects of the proposed isolator that goes beyond the sole consideration of the damping performance. First, a preliminary design of the prototype strut is presented, followed by an overview of the potential hexapod configuration made of six of the proposed struts and its comparison with other hexapods found in literature. The possibility to use a temperature control system, either active or passive, to guarantee the damping performance of the isolator for a wider temperature range is then reported. The chapter concludes with the description of the steering capability of the proposed system that can be actuated simultaneously and in parallel with respect to the advanced mitigation property of the isolator.

Chapter 7 draws the conclusions of this thesis and outlines the future work.
Chapter 2

Literature Review

In this chapter the relevant literature pertinent to the development and evaluation of micro-vibration mitigation tools is presented. First, the description of reaction wheels and their disturbance signature is presented. The difference between absorbers and isolators is then reported, with particular focus on the range of applications that these devices are usually applied to. It follows an in-depth look at the different damping methods that have been proposed and studied in the last four decades in the realm of micro-vibration mitigation on board spacecraft. Finally, the current limitations in the use of isolators for space missions are presented, and typical performance features and drawbacks of these devices are reported in order to give an complete overview of the state of the art in the mitigation of micro-vibrations.

2.1 Reaction Wheels

Among several subsystems and mechanisms, reaction wheels are commonly considered as the main source of micro-vibration disturbances on board satellites. Reaction wheel assemblies (i.e. cluster of two or more reaction wheels) are mainly used for spacecraft attitude control as they operate on the principal of conservation of angular momentum. By accelerating or decelerating about one axis, reaction wheels produce a torque on the spacecraft that makes it rotate in the opposite direction about the same axis, thus conserving the total angular momentum of the system. When no attitude control is needed, reaction wheels are usually kept spinning at a constant speed (i.e. they do not produce any torque on the spacecraft) in
order to avoid instabilities at speeds close to zero. However, such operational conditions bring to constant injection of micro-vibration disturbances into the spacecraft bus.

**Figure 2.1:** Example of reaction wheel. Most of the mass is concentrated on the rim so to have higher inertia.

**Figure 2.2:** The waterfall plot shows the disturbances produced by reaction wheels in the whole frequency range of interest for each operational wheel speed.

As shown in the waterfall plot of Figure 2.2, for each operational speed reaction wheels produce several disturbances that span throughout the whole frequency range of interest. In particular, these disturbances are mainly caused by two factors:
2.2 Micro-Vibration Absorbers and Isolators

- **Structural modes.** Depending on the configuration, reaction wheel can be described generally as a five DoFs system (the torsion about the spinning axis is neglected). This leads to five dominant structural modes which further reduce to three due to symmetry about the vertical axis. These modes can be either constant with wheel speed (e.g. axial and lateral modes) or be speed-dependent which generates gyroscopic effects that results in the split of the structural mode in two branches (e.g. the rocking mode splits into a precession whirl and nutation whirl);

- **Manufacturing or assembling imperfections.** The dominant disturbances are often generated by flywheel mass unbalance, bearing defects or motor irregularities. These harmonics present a linear behaviour with the wheel speed and their slope depends on geometrical parameters (e.g. number of balls, bearing diameter, contact angle, etc.).

### 2.2 Micro-Vibration Absorbers and Isolators

This section explains the fundamental difference between absorbers and isolators and how they relate with dampers. These concepts are often mistakenly interchanged in the field of vibration mitigation. However, their use can result in a system having a completely different behaviour. The choice of using one or the other depends on the specific application, and sometimes this choice is even dictated by structural constraints or requirements. For instance, if the undesired vibration involves an extensive panel with several critical devices on it then the use of absorbers will result to be the more practical solution. On the other hand, if the vibration disturbance is mainly produced by one device, then isolating it from the rest of the structure could be more effective.

#### 2.2.1 Absorbers

Absorbers use some sort of damping methods to remove energy from a vibrating structure by converting kinetic energy into another kind of energy (mostly dissipated via heat energy, but there is also an increasing interest in the development of energy harvesting devices [32–34]). Absorbers can be used against shock loading [24, 35–37] (e.g. for vehicle suspensions or for safely dislodge the umbilicals from the spacecraft before taking off from the launch pad) or against broadband signals applied to an extended structure [38–41]. However, when applying dynamic vibration absorbers to a continuous structure such as a plate or a beam, vibrations can be eliminated only at the absorber attachment point while amplification of the vibrations...
2.2 Micro-Vibration Absorbers and Isolators

may occur in other parts of the structural elements. In fact, a structure will most often vibrate as a superposition of several mode shapes and deciding where an absorber needs to be placed depends on the mode that is considered the most critical. Ideally, an absorber should be located in a spot that undergoes the maximum displacement during the targeted resonance frequency. However, the same spot might be a node (i.e. zero displacement) for a different resonance frequency (see Figure 2.3). Therefore, the position of the absorber needs to be accurately determined before hand and is responsible of the damping efficiency of the device [42].

Another important aspect of an absorber is the “tuning” of the damping element. Most dampers need to be tuned at the targeted frequency in order to be more effective. For example, the stiffness coefficient and the secondary mass in a tuned mass damper or the electric components in a piezoelectric shunt damper have to be chosen according to the resonance frequency of interest, and their choice exclude those devices to be effective at a different frequency. Self-tuning techniques have also been implemented to obtain frequency-independent gains of decentralised velocity feedback control loops and maximise their local absorbed power [43, 44]. Constant gain feedback loops are stable if the control-sensor pairs are dual and collocated. However, in a real case scenario the actuators and sensors approximate ideal force and velocity transducers only for a certain frequency band, and the presence of filters in feedback loop may introduce phase shift which can limit the control gain before instability.

Therefore, although having the advantage that they can be connected to a structure in a second moment during the design phase, absorbers are limited by their specific location and by the tuning that was chosen before hand. The use of absorbers might become a cumbersome method when several natural frequencies need to be attenuated.
2.2.2 Isolators

With the term isolation one refers to the attempt to prevent vibration disturbances from being transmitted through some load paths. In other words, an isolator reduces the ability of a system to react to an excitation. This is achieved by the use of flexible decoupling elements between the equipment and mounting surface. It allows the inertia of the isolated component to oppose and thereby reduce the vibratory motion transmitted to the support. Isolation works both ways: either a noise source can be decoupled from the rest of the structure or a sensitive device can be isolated from a noisy support (see Figure 2.4).

![Figure 2.4: Schematic representation of an isolation system. a) noise source is decoupled from the satellite platform. b) sensitive payload is decoupled from the noisy satellite platform](image)

An ideal isolator does not need damping to properly perform. In fact, it is just necessary to choose its main resonant frequency to be well below the primary forcing frequency of the isolated device. In this way, the intrinsic dynamic of the isolator does not interact with the supported device and it is capable of effectively attenuating the vibration disturbances.

However, it can happen that the main resonant frequency of an isolator might be close to (or above) the primary forcing frequency thus resulting in the amplification of the vibrations. In this case, adding a damping element would be required to reduce the amplitude of the resonant response, even though its introduction can negatively affect the high frequency attenuation efficiency (e.g. a viscous damper reduces the final roll-off slope of the transmission curve by 20 dB/dec).
2.3 Damping Methods Review

This section reviews the main micro-vibration dampers that have been studied and developed in the last decades. In particular, these devices are classified into four categories which represent the four possible functioning natures of each damper. The four categories are: passive, active, hybrid and semi-active dampers. The first three categories are shown in Figure 2.5 where different damping configurations are applied to a proof-mass system. A typical diagram of a semi-active damping system is instead shown in Figure 2.6. A damper is classified as passive if it does not require input power to operate. They are simple and generally low in cost but unable to adapt to changing needs. On the other hand, active dampers require power and are capable of producing a higher attenuation performance, but they usually come with increased complexity and mass. Hybrid dampers are characterised by a passive system and an active system working independently in parallel, whereas a semi-active damper is mainly a passive system in which some parameters can be actively controlled. It is noted that some technologies (e.g. electromagnetic transducers) are presented in more than one category because of their versatile nature (they are capable of producing a damping force either with or without an active control system). Particular attention is given to the characterisation of viscoelastic materials, since their damping performance is used as a comparison throughout this thesis.

![Schematic diagram of a proof-mass system with a passive, an active, and a hybrid damper](image)

**Figure 2.5:** Schematic diagram of a proof-mass system with a passive, an active, and a hybrid damper [45]
2.3 Damping Methods Review

2.3.1 Passive Dampers

Passive dampers are characterised by a pre-defined curve in units of force/velocity, and have a smaller range of damping forces if compared to semi-active and active systems. The main passive dampers are: viscoelastic materials, fluid viscous dampers (e.g. D-struts), particle dampers, tuned mass dampers, shunted piezoelectric transducers, eddy-current dampers and electromagnetic shunt dampers.

2.3.1.1 Viscoelastic Material

Viscoelastic materials have the capability to exhibit both time-independent elastic behaviour and time-dependent viscous behaviour when they undergo deformation. Therefore, they already function as isolators since they perform both tasks simultaneously. A convenient measure of damping is obtained by comparing the energy lost in a cycle with the total energy stored in the system during that cycle [46]. The loss factor $\eta$ is defined as

$$\eta = \frac{D}{2\pi W}$$

where $D$ denotes the energy dissipated per cycle (or the energy that must be supplied to the system to maintain steady-state conditions) and $W$ denotes the total (kinetic plus potential) energy associated with the vibration. The energy $D$ dissipated per cycle is equal to the
area enclosed in a typical hysteresis loop produced by viscoelastic isolators (see Figure 2.7) [47].

![Hysteresis Loop](image)

**Figure 2.7:** Example of elliptical hysteresis cycle in the \((\sigma \varepsilon)\) plane for the Kelvin-Voigt model [47].

**Analytical Models** Viscoelastic materials, such as amorphous polymers, semicrystalline polymers, and biopolymers, can be modelled in different ways depending on the properties that one wants to represent and simulate. The most-studied models are: Maxwell model, Kelvin-Voigt model, and the Standard Linear Solid Model [48] (respectively reported in Figure 2.8). The Maxwell model is able to predict the time-exponential stress decay when a viscoelastic material is put under a constant strain, whereas the Kelvin-Voigt model (also known as the Voigt model) is used to explain the creep behaviour of polymers. The Standard Linear Solid Model effectively combines the Maxwell Model and a Hookean spring in parallel to produce a more accurate material response than the Maxwell and Kelvin-Voigt models. However, the strength of model predictions decreases with increasing frequency, particularly for hysteresis, and it is rather difficult to determine its parameters.

**Elastomers** For the last 20 years, elastomers have found their way through the vast field of damping devices applied to space missions [50]. Nowadays, viscoelastic dampers are the cheapest and lightest damping solution that can be designed for a space application, and most space companies rely on them as the main damping system (see Figure 2.9). Elastomers, which belong to a wide family of polymers, can be very flexible, extremely elastic, gas-proof and thanks to an appropriate compound they can present a high damping capability. However, their properties strongly depend on the mechanical and thermal environments.
2.3 Damping Methods Review

Figure 2.8: Schematic representation of the most-studied models to simulate viscoelastic materials [49]: a) Maxwell model, b) Kelvin-Voigt model, and c) Standard Linear Solid Model

in which they will operate. In terms of the mechanical environment, the frequency seems to be the factor that most affects the damping behaviour of the elastomer [51, 52]: at low frequencies, the material is flexible and elastic (it does not present energy dissipation); when the frequency increases, a phase delay arises between the material response and the solicitation (hysteresis phenomenon); at higher frequencies, the tensions are permanent inside the material and it seems to become rigid. This behaviour cannot be predicted through the three aforementioned methods in which they describe the viscoelastic material as linear. Instead, the fractional derivative models [53] were proved to be able to accurately reproduce the frequency dependency of the Young’s modulus and loss factor. In particular, a recent paper [54] has shown that with an appropriate fitting of the experimental data it is possible to analytically simulate the behaviour of elastomers. Figure 2.10 shows the good correlation between the experimental data for Norsorex® 46925 and a 20th-order fractional derivative model fit. It is worth noting that this model, although accurately representing the frequency dependency of viscoelastic materials, is not capable of modelling the temperature effects on the materials behaviour (the tests shown in Figure 2.10 were performed at a constant temperature of 22 °C). In fact, the temperature also plays an important role in the evaluation of the damping properties [55, 56]: at very low temperatures, the elastomer is rigid and breakable, whereas at very high temperatures the material is purely elastic (no viscous
2.3 Damping Methods Review

properties. Moreover, the broadness of the temperature range at which the elastomer operates is one of the main sources of ageing, along with the effects of solar radiations and vacuum.

Finally, the amplitude of the input force is another source of nonlinearity for elastomers. Both dynamic stiffness and damping performance are highly affected by the load amplitude. For instance, in the work presented in [57] the stiffness increased approximately 100% when amplitude was reduced from 1 mm to 0.01 mm. The Kelvin-Voigt model is not capable of representing this amplitude dependency.

Therefore, it is still quite challenging to create an analytical model for viscoelastic materials that can combine the effects of frequency and temperature dependencies [58] along with other nonlinearities that characterise this kind of materials. An example of this modelling difficulty can be observed in Figure 2.11. In this case, a SMAC elastomer specimen’s experimental data provided by Surrey Satellite Technology Ltd (SSTL) is compared with the Kelvin-Voigt model and the fractional derivative model (the SMAC elastomer is different from Norsorex® 46925). The temperature of the specimen is considered constant.

The Kelvin-Voigt constitutive relationship transformed into the frequency domain is given by:

$$\sigma = (E_1 + i\omega E_2)\varepsilon$$  \hspace{1cm} (2.1)

where $E_1$ and $\omega E_2$ are respectively the real and imaginary part of the complex Young’s modulus [59]. The loss factor is defined as:

$$\eta = \frac{\omega E_2}{E_1}$$  \hspace{1cm} (2.2)

which reveals one of the critical failures of the Kelvin-Voigt scheme: a loss factor that is linear with frequency.

On the other hand, the fractional derivative models were proved to be efficient in describing the non-linear behaviour of real materials such as elastomers [53, 60]. The constitutive relationship of the fractional derivative model can be expressed starting from the conventional derivative model:

$$\sigma(t) + \sum_{j=1}^{N} \frac{1}{r_j} \frac{d^j}{dt^j} \sigma(t) = \sum_{j=0}^{N} \frac{E_j}{r_j} \frac{d^j}{dt^j} \varepsilon(t)$$  \hspace{1cm} (2.3)
and replacing the integer order derivatives with fractional ones, thus obtaining:

$$
\sigma(t) + \sum_{j=1}^{N} \frac{1}{r_j} \frac{d^{\beta_j}}{dt^{\beta_j}} \sigma(t) = \sum_{j=0}^{N} \frac{E_j}{r_j} \frac{d^{\alpha_j}}{dt^{\alpha_j}} \varepsilon(t)
$$

(2.4)

where $0 < \beta_j < 1$ and $0 < \alpha^k < 1$ are material constants as well as $E_j$ (modulus contribution) and $r_j$ (transition frequency). The fractional derivation, say the $\alpha$th order derivative of $\varepsilon(t)$, can be defined by the gamma function ($\Gamma$) as:

$$
\frac{d^{\alpha}}{dt^{\alpha}} \varepsilon(t) = \frac{1}{\Gamma(1-\alpha)} \frac{d}{dt} \int_0^t \frac{\varepsilon(\tau)}{(t-\tau)^{1-\alpha}} d\tau
$$

(2.5)

where the gamma function $\Gamma(1-\alpha)$ is given by:

$$
\Gamma(1-\alpha) = \int_0^\infty x^{-\alpha} e^{-x} dx
$$

(2.6)

The complex modulus of the model can be derived by transforming Eq. (2.4) into the frequency domain. The derivation is simplified by the following properties of the fractional derivative model:

$$
\Im[f^{(\alpha)}(t)] = (i\omega)^{\alpha} \Im[f(t)]
$$

(2.7)

where $\Im$ represents the Fourier transform. Therefore, by applying the Fourier transform to (2.4) and assuming the fractional order $\alpha = \beta = \phi$, the fractional derivative constitutive relationship in the frequency domain can be expressed as:

$$
\sigma = \left( E_s + \sum_{j=1}^{N} E_j \frac{(i\omega/r_j)^{\phi_j}}{1+(i\omega/r_j)^{\phi_j}} \right) \varepsilon
$$

(2.8)

where $E_s$ is the constant modulus at $\omega = 0$ and $0 < \phi_j \leq 1$ are material constants. Through an optimisation algorithm, it would be possible to evaluate the coefficients $E_j$, $r_j$ and $\phi_j$ to best fit the experimental data. An example of fitting process is shown in Figure 2.10. In that case, a 20-order fractional derivative model was fitted to represent the dynamic behaviour of the viscoelastic material Norsorex® 46925 at a temperature of 22 °C. In particular, the frequency dependency of such material is clearly visible through the increase of the Young modulus as the frequency increases.

The comparison of the two aforementioned methods applied to the SMAC materials used by SSTL is shown in Figure 2.11. The comparison not only shows the better accuracy of
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Figure 2.9: Isolation system used for reaction wheel assembly at SSTL.

the fractional derivative model over the Kelvin-Voigt one, but also highlights some intrinsic limitations in the use of viscoelastic materials that seem difficult to be overcome. For instance, the resonance peak cannot be completely eliminated (in this case it has an amplification of 14 dB at 56 Hz). This is due to a limited loss factor that becomes even smaller at low frequencies. Another issue can arise from the desire to lower the corner frequency in order to obtain a low pass filter. Reducing the stiffness of the viscoelastic element (e.g. by increasing the length of the element) can be challenging because it can introduce secondary modes within the viscoelastic specimen at the frequencies of interest and also cause buckling instability (see resonance peaks in Figure 2.11 that have appeared above 500 Hz). Finally, the effect of the elastomer’s frequency dependency can be seen in the reduction of the roll-off slope as the frequency increases. In fact, the slope associated with the experimental data starts with the expected $-40$ dB/dec and, as the specimen becomes stiffer at higher frequency, it gradually decreases.

Space Applications As stated above, viscoelastic isolators are typically considered by space companies as the first option in the design of the damping system of a spacecraft [61, 62]. Their damping performance over cost and added mass is superior to every other damping device. However, all the uncertainties and limitations in the application of elastomers usually force engineers to add other damping devices in the design of the suspension system in order to obtain higher force-transmission attenuation. This is the case, for example, of the satellite SSTL300-S1 0.75-1 m GSD (ground sample distance) Earth imaging 400 kg spacecraft, designed and manufactured by SSTL [63, 64]. A combination of different expedients (e.g. the employment of hybrid ceramic bearings) allowed the whole system to perform several orders of magnitude better than the original wheel over the entire frequency range.
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Figure 2.10: Viscoelastic materials: example of frequency dependency of Young’s modulus and loss factor at 22 °C for Norsorex® 46925 [54]

Figure 2.11: Comparison between the experimental data and two analytical models. The Kelvin-Voigt model and the fractional derivative model are here represented.

-40 dB/dec
However, the impossibility of eliminating the resonance peak and the unchangeable roll-off slope of $-40$ dB/dec typical of elastomers make it difficult to achieve an image resolution below 1 m GSD. Hence, an alternative solution needs to be studied in order to increase the platform stability and obtain a higher imaging resolution.

2.3.1.2 Fluid Viscous Damper

One of the first examples of fluid viscous damper for space applications is the D-Strut™ designed by Honeywell Inc. at the end of the 1980s [65, 66]. This damper was developed in collaboration with the US Air Force and Jet Propulsion Laboratory for use in the isolation of disturbances from a reaction wheel assembly. The viscous-damped D-Strut™ has been employed in several space applications, including the Hubble Space Telescope. A D-Strut™ is composed by a viscous fluid flowing through a controlled annulus between two hermetically sealed chambers (see Figure 2.12a). Mechanical energy can be dissipated in the shearing of the fluid as it is pumped through the annulus by the vibratory displacements. The advantage of this design is the possibility to change the stiffness of the secondary bellows which results in a three-parameter configuration characterised by improved isolation performance when compared to a conventional two-parameter configuration. As it can be observed from Figure 2.12c, the three-parameter isolator (modelled with one spring in parallel with a series connection of a dashpot and a second spring) is capable of considerably reducing the amplification factors at resonance while providing a much better isolation at high frequencies than that of conventional two-parameter isolators. The crucial role is played by the spring in series with the dashpot that reduces the stiffness of that branch at high frequency thus alleviating the effect of enhanced stiffness produced by the viscous fluid as the frequency increases. The main advantages of this technology are the very large dynamic range (no rubbing friction or hysteresis), high damping, low temperature sensitivity compared to viscoelastic materials, linear and predictable performance, and hermetically sealed fluid.

Honeywell proposed also a second viscous damper to be used in high specific-stiffness truss structures. For this device, the hydraulic fluid was replaced with a gas (compressible medium) [67]. The pneumatic D-Strut isolator is capable of produce the required stiffness while maintaining a good level of damping, but it has not been qualified yet for space applications.
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Figure 2.12: D-strut™: (a) Cross-section of the damper in the 1.5 Hz isolator struts; (b) schematic representation of the two-parameter and three-parameter configurations; (c) displacement transmissibility comparison between three-parameter and two-parameter isolators. In particular the dashed line with a loss factor $\eta$ of 0.5 is characterised by a roll-off slope of $-20$ dB/dec (typical of the two-parameter configuration) whereas the solid line with $\eta = 0.5$ representing the three-parameter configuration shows a slope of $-40$ dB/dec [66]
2.3 Damping Methods Review

2.3.1.3 Particle Damper

Particle Damping (PD) is a technique of providing damping with metal or ceramic particles or powders of small size ($\approx 0.05 - 0.5$ mm in diameter) embedded within small holes in a vibrating structure [68, 69]. Particle-to-wall and particle-to-particle collisions arise under the vibrating motion of the structure. In contrast to viscoelastic materials which dissipate the stored elastic energy, particle damping treatment focuses on dissipation of the kinetic energy. PD involves potential energy absorption and dissipation through momentum exchange between moving particles and vibrating walls, friction, impact restitution, and shear deformations. It is an attractive alternative in passive damping due to its conceptual simplicity, potential effectiveness over a broad frequency range, temperature and degradation insensitivity, and very low cost. Its main drawbacks are the low damping coefficient achievable and the difficulty to analytically model this damper. Attempts to use the discrete element method to simulate the PD has resulted in a good correlation with the experimental data, but the computational complexity limits the number of spheres that can be simulated simultaneously [70].

Recently, a new particle damper has been studied by Michon et al.[71]. The paper proposes the use of soft hollow particles (instead of the classical hard ones) to fill the cavities of honeycomb cells. In this case, instead of dissipating by friction and impact, the damping performance is produced by hysteretic damping (typical of viscoelastic materials).

2.3.1.4 Tuned Mass Damper

A Tuned Mass Damper (TMD) is a device consisting of a mass, a spring, and a damper that is attached to a structure in order to reduce the dynamic response of the structure [72, 73]. The damper is tuned to a particular structural frequency so that when the structure is excited at that frequency, the damper will resonate out of phase with the structural motion. Energy is dissipated by the damper’s inertia force acting on the structure. These dampers have been extensively used in the civil engineering field to attenuate earthquake vibrations on skyscrapers (e.g. on the 60-storey John Hancock Tower in Boston, and on the Citicorp Center in New York City [74]).

TMDs are extremely interesting when the damping-system designer is concerned only about one structural resonance, but they do not give any attenuation contribution with respect to other resonance modes. Multiple TMDs that are tuned to different modes and placed at various locations have also been studied. This configuration enhances the dampers’
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performance, with a vibration mitigation of 10-25% more than a conventional TMD [75]. However, it is noted that a significant auxiliary mass is necessary to effectively reduce the resonance peak, and this is a major drawback in their application to space missions.

2.3.1.5 Shunted Piezoelectric Transducer

Piezoelectric Materials (e.g. Quartz, Gallium Orthophosphate, and Lead Zirconate Titanate) are materials that produce an electric voltage when they undergo mechanical stress and they can equivalently be represented as capacitors due to their intrinsic high impedance [76, 77]. The piezoelectric process is also reversible, which means that they can change shape if an electric field is applied. Therefore, Piezoelectric Transducers (PZT) can be used either as sensors or as actuators. The employment of shunting circuits has allowed these devices to also operate as passive dampers. In fact, by connecting an electrical impedance to the terminals of a structurally bonded PZT it is possible to dissipate energy through resistive heating, so that it performs a damping action [78]. Shunting circuits are typically made of a simple resistor or a combination of a resistor and an inductor in series. The former configuration exhibits a frequency-dependent loss factor behaviour similar to viscoelastic materials, but results in it being much stiffer and more independent of temperature. The latter forms a resonant electrical circuit (i.e. RLC circuit) that can be tuned to a specific structural mode thereby increasing the attainable modal damping ratio (in an effect similar to a classical TMD).

In the last decade, several studies were aimed at developing multi-mode vibration damping methods using a single PZT [79]. Various configurations have been analysed (e.g. parallel RLC shunts tuned at different structural modes, or RL circuits with current-blocking LC circuits into each branch), and each one presents different pros and cons in regard to system response and stability. Hence, the best shunt configuration has to be chosen each time depending on the specific system requirements.

The main disadvantages of PZTs are the low damping ratio compared to viscoelastic materials and the necessity to use several PZTs to significantly reduce a system disturbance.

2.3.1.6 Eddy-Current Damper

Eddy-Current Dampers (ECD) belong to the family of magnetic dampers. They function through the induced currents that are generated when a conductive material is subjected to a time/spatial changing magnetic field (see Figure 2.13). The force produced by these devices
is proportional to the relative velocity between the conductive metal and the magnetic-field source, and so they act as viscous dampers (further details on the physics principles and the analytical model of magnetic dampers are given in chapter 3). This damper takes its name from the typical shape of little swirls, or “eddies”, which characterise the induced currents that flow in the conductive material.

![Figure 2.13: Schematic of conductive material passing through a magnetic field and the generation of eddy currents][80]

Among other advantages, the most interesting properties of an ECD are the contactless nature of its damping force and no performance degradation over time. Several studies have demonstrated that this kind of damper could be suitable for suppressing the vibration of beams, plates, and membranes, in which the required damping forces are relatively small [81–85].

Sodano et al. [80] studied the application of an ECD, both analytically and experimentally, for suppressing the vibrations of a cantilever beam. The configuration is shown in Figure 2.14a, and consists of a copper sheet at the tip of a cantilever beam and a single cylindrical permanent magnet that generates the magnetic field. The proposed eddy current damping mechanism showed the damping ratio increasing by up to 150 times compared with the intrinsic damping of the aluminium, and provided a sufficient damping force that quickly suppressed the beam’s vibration.

A different configuration of an ECD is characterised by a conductive tube where a cylindrical permanent magnet slides into it (see Figure 2.14b). This configuration has been widely studied [86–88], and it has been demonstrated that this ECD is capable of a higher damping coefficient compared with the copper-sheet configuration.

Rotating devices with embedded ECDs have been successfully used in several space applications such as solar array and antenna deployment [89, 90]. Nevertheless, their employment
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![Figure 2.14: Two examples of configuration for an ECD. a) cantilever beam in magnetic field generated by permanent magnet [80]; b) eddy current shock absorber [36]](image)

for different purposes (e.g. absorbing mechanism for vibrating membranes or panels) has encountered several difficulties, mainly related to the high ratio of mass introduced by the damper over damping force produced.

### 2.3.1.7 Electromagnetic Shunt Damper With Passive Components

Electromagnetic transducers function on the same principles of ECDs (they also belong to the family of magnetic dampers), but in these devices the solid conductive element is substituted by a solenoid. A passive electromagnetic damper can be obtained either by direct connection of the solenoid terminals or by attaching a shunting circuit at the two terminals. The latter option is usually preferred because an accurate choice of the shunting electrical components can considerably enhance the overall damping performance [91]. As already stated in section 2.3.1.5, different configurations of the shunting circuit have been studied for PZTs, and the same circuits can be applied to electromagnetic transducers. Dynamic analogy between an Electromagnetic Shunt Damper (EMSD) and a TMD has also been proved by Zhu et al. [92]. However, an EMSD presents some important advantages compared with a TMD: the EMSD does not involve any moving masses (this damper can imitate virtual mass using electronic capacitors, such that a substantial equivalent mass may be achieved by a small device) and it does not introduce any additional stiffness in the system due to the contactless nature of this device. Moreover, an EMSD is characterised by a higher specific damping coefficient with respect to a shunted PZT.

EMSDs can operate not only as mechanical dampers, but also as energy harvesters where the converted electrical energy is stored in capacitors or batteries [93–98]. In this way it
Figure 2.15: Schematic of an EMSD. The shunt is composed by an RLC circuit [92]

is possible to simultaneously reduce the vibration amplitude while accumulating electrical energy that can be used for secondary purposes. For instance, energy harvesters could be used on pedestrian bridges or train platforms where the vibrations produced by people’s walk could be used to power the street lights [99, 100].

2.3.2 Active Dampers

Active dampers are force generators that act on a structure to counteract a disturbance. They are fully controllable and capable of producing higher-order vibration attenuation with respect to passive dampers, but they also require a significant amount of power to operate, and might introduce instability issues [101, 102]. The main active dampers are: voice coil actuators, piezoelectric actuators and active magnetic dampers.

2.3.2.1 Voice Coil Actuator

Voice Coil Actuators (VCA) are direct drive, limited motion devices that use a permanent magnet and a coil winding to produce a force proportional to the current applied to the coil [42, 103] (see Figure 2.16). Originally used in radio loud speakers, VCAs have been extensively studied in the last two decades for applications where proportional or tight servo control was necessary. For instance, VCAs have been recently utilized to perform the optical image stabilization of digital cameras, in which they are used to compensate the hand jitter by acting on a lens or lens group, by modifying the optical path [104, 105].

The main advantages of the VCA are the linearity and the simplicity of the control, the significantly greater stroke when compared with PZT (typically in the millimetre range versus the micrometre range of PZT) and its contactless nature that makes it more preferable over
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hydraulic or pneumatic devices. Moreover, the versatile nature of electromagnetic transducers (they can be used as actuators or as sensors) allows the development of electromagnetic self-sensing actuators where the function of actuator and sensor are performed simultaneously by one voice coil [106, 107]. This technique involves the estimation of the system velocity from measurements of the transducer current and voltage. A feedback loop, driving the voice coil voltage, is constructed around the estimation to minimize the relative mechanical velocity between the coil and magnet. From a theoretical point of view, this control scheme is considered to be perfectly collocated [108], which improves the stability and robustness of the closed-loop system.

![Figure 2.16: Schematic of a VCA published in the NASA tech briefs [109]](image)

Recently, VCAs have been controlled to reproduce the behaviour of a cubic nonlinear viscous damper. This technique uses the approach of the output frequency response function [110], and it is based on the representation of the damping term as:

\[ F_d = C_1 \dot{x} + C_3 \dot{x}^3 \] (2.9)

where it can be seen that the damping force is also proportional to the cubic term of the velocity. By properly choosing the parameters of the damping term it is possible to reduce the amplification at resonance peak while alleviating the typical negative effect of linear viscous dampers at high frequency, thus allowing for better vibration control [111], see Figure 2.17. Although this technique could be implemented passively by damper manufacturers [112], the development of active control algorithms to reproduce the cubic nonlinear viscous damping has resulted in several advantages, such as the simplicity of its implementation and the effectiveness in reducing the vibration over the whole range of the system working frequencies [113]. The same technique, if used in shunted electromagnetic devices for energy-
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harvesting purposes, is capable of significantly increasing the harvested power compared to a linear energy harvester, therefore expanding the dynamic range of these devices [114].

Figure 2.17: The force transmissibility for a single DoF isolator subjected to force excitation with different linear damping coefficients and different cubic order nonlinear terms [115]. The solid line and the dashed line represent respectively systems with linear damping coefficient of 0.1 and 0.325. The effect of including nonlinear terms is instead represented by the stars and dots in the plot while the linear damping coefficient is kept to 0.1.

In space applications, VCAs are often considered as the first choice when an active control system is necessary [116]. However, the main drawbacks of these devices are the significant weight increase and the high external power required to operate, as well as the cooling issue since most power is dissipated in the coil within the actuator.

2.3.2.2 Piezoelectric Actuator

Piezoelectric Actuators (PA) have been successfully used in the closed loop control of a variety of active structures including beams, plates, and trusses [117–120]. These actuators exploit the piezoelectric materials’ property of straining when an electrical field is applied across them. Their analytical model has been extensively studied [121–123] and understood thus facilitating their use in damping solutions.

The damping performance of PAs can be considerably modified and enhanced by means of a proper feedback-loop design. Vasques et al. [124] reported the comparison between the classical control strategies (constant gain and amplitude velocity feedback), and the optimal control strategies (linear quadratic regulator and linear quadratic Gaussian controller). They concluded that the linear quadratic Gaussian presents a better control in bandwidth with a
2.3 Damping Methods Review

Figure 2.18: Piezoelectric actuators. a) flexible structure with a collocated pair of piezoelectric transducers; b) piezoelectric-based, self-sensing PA, generating an estimate of the mechanical strain [123]

lower input voltage, and more adaptability in design, but also the necessity of digital components (increased time delays in the control system, and instability issues), and the requirement of providing a precise model of the system to obtain an efficient optimisation.

Some of the limitations related to the feedback loop have been overcome with the development of the self-sensing technique [125, 126]. As already explained for VCAs, the actuator and sensor are combined into a single piezoelectric element called a self-sensing PA. Major advantages of this device are that actuator and sensor are coincident (resulting in a truly-collocated, very-robust control), and the elimination of possible closed loop control problems arising from the capacitive coupling between the sensor and the actuator elements (not an issue with the self-sensing PA because only a single element is used). A schematic representation of active feedback control and self-sensing technique can be seen in Figure 2.18.

These actuators share the same drawbacks of passive PZTs: their efficacy is strongly affected by their position within a structure. Therefore this method might result cumbersome if multiple structural modes associated with a large structure are targeted.
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2.3.2.3 Active Eddy-Current Damper

An ECD can be controlled actively if the permanent magnet is substituted by an electromagnetic coil to produce the required magnetic field (see Figure 2.19). By sensing the velocity of the vibrating structure and actively modifying the current flowing through the coil, a time-changing magnetic field is generated that induces eddy currents in the conductor and results in a magnetic force.

An active ECD was first tested on an aluminium beam by Sodano et al. [27, 28]. This damper was able to suppress each of the beam’s first five modes of vibration by more than 20 dB, and to have control on a larger bandwidth over previously used eddy current vibration control systems and dampers.

![Example of active ECD](image)

Figure 2.19: Example of active ECD [28]

Although this device is capable of changing the magnetic field strength without the need to increase the overall mass, the considerable amount of external power required by the electromagnet is one of the major drawbacks of this device. For this reason, such active damper could be beneficial mainly for the suppression of micro-vibrations on thin membranes where small damping forces are necessary.

2.3.3 Hybrid Dampers

Hybrid dampers have a passive system and an active system that are used simultaneously in order to combine the advantages of both dampers and possibly minimise their disadvantages. Hybrid dampers are usually developed with one part, either active or passive, as a master
system and the other part as a slave. Some hybrid dampers have been developed for micro-vibration mitigation on board spacecraft.

### 2.3.3.1 Hybrid Dampers with Electromagnetic Actuators

Electromagnetic actuators are often preferred in hybrid configurations due to the contactless nature of their force. In fact, the system stiffness is carried out by the passive component in parallel with the electromagnetic actuator which can considerably improve the damping performance at low frequency [127].

One of the first examples of this type of hybrid damper was the advanced D-Strut™ designed by Honeywell in the mid 1990s [128] that combined a traditional D-Strut™ with a VCA. This damper was capable of achieving the majority of the mitigation requirement passively. The active portion was designed to enhance the low-frequency isolation performance of the strut by modifying the passive system’s lower break frequency through apparent payload mass augmentation or by de-stiffening the passive system. In comparison with the passive D-Strut, this hybrid damper was capable of considerably lowering the resonance peak with the use of load cell feedback, thus increasing the disturbance attenuation in the range of up to 100 Hz. For greater frequencies the transfer functions of the passive and hybrid D-Struts overlap with a roll-off slope of $-40 \text{ db/dec}$ (i.e. this damper does not produce a further attenuation at high frequency). The main disadvantages of this device are related to the instability issues that might arise from the feedback loop, and the presence of nonlinear behaviour above 200 Hz.

Another example where an electromagnetic actuator was used in a hybrid damper was studied by Sodano in [129]. The proposed configuration was characterised by a shaker attached to a permanent magnet (see Figure 2.20a). By allowing the position of the magnet to change relative to the beam and thus allowing the net velocity between the two to be maximised, the damping force of this hybrid ECD can be significantly increased (vibration suppression could be increased by approximately 79% more than with only passive eddy current damping). However, the shaker used for the tests had an effective bandwidth limited to approximately 100 Hz, and so the attenuation of the beam modes beyond the first two has not been verified.

Hybrid dampers embedded in shock absorbers have also been widely studied. Ebrahimi et al. [130] proposed a configuration where eddy current damping effect was utilised as a source of the passive damping, whereas an externally-powered electromagnet functioned as active control (see Figure 2.20b). Ebrahimi demonstrated that the hybrid damper had more than 70%
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Figure 2.20: Two examples of hybrid magnetic dampers. a) Hybrid ECD [129]; b) hybrid electromagnetic damper [130]

less power consumption at resonance, while maintaining the active damping performance advantages. Asadi et al. [131] studied a shock absorber with a similar configuration where in this case the passive damping was performed by viscous fluid sealed inside the strut cylinder. The work showed that the damper was able to produce damping coefficients of 1300 and 0-238 Nsm$^{-1}$ through the viscous and electromagnetic components, respectively. Although, these shock absorbers with embedded hybrid dampers have shown good mitigation performance, their high ratio of device mass over provided damping performance makes these configurations still not suitable for space missions.

2.3.3.2 Hybrid Dampers with Piezoelectric Actuators

Several examples of hybrid dampers with PAs have been investigated and developed and some of them have also been implemented on satellites [132]. The use of PAs can introduce a considerable amount of stiffness in the system and this effect has pros and cons. Increasing the stiffness, especially in the case piezoelectric stack actuators, results in having a less complex isolation system since it is able to support itself during launch and so a lock-down system is not necessary. On the other hand, having a higher stiffness can compromise the low frequency micro-vibration attenuation.

Recently, Kamesh et al. [133] have combined folded continuous beams with piezoelectric actuators and sensors as shown in Figure 2.21. The passive isolation is performed by the folded continuous beam configuration which has a low dynamic stiffness. The piezoelectric materials can be used either as active actuators or sensors and are driven by an optimal
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control algorithm that is capable of effectively suppress disturbances coming from harmonic and impulse loading.

![Hybrid isolator consisting of folded continuous beams and piezoelectric actuators](image)

**Figure 2.21:** Hybrid isolator consisting of folded continuous beams and piezoelectric actuators

A similar configuration was also investigated by Zhou and Li [134]. They studied a low frequency flexible intelligent isolation platform to be used with reaction wheel and momentum wheel assemblies. The study showed that the passive isolation part is effective to isolate high frequency micro-vibration of momentum wheel assemblies, and the intelligent active part is effective to suppress low frequency disturbance of reaction wheel assemblies.

### 2.3.4 Semi-Active Dampers

Contrarily to hybrid dampers, semi-active dampers require less accessory devices (e.g. sensors, actuators, feedback control etc.) and also less power, which are the drawbacks that both hybrid and pure active dampers share. Semi-active dampers are considered an effective approach for micro-vibration mitigation because they are characterised by some system parameters (either the stiffness or the damping) that are tunable through active control techniques in the context of passive vibration attenuation [15]. Semi-active methods have the main advantage of operating in stable conditions and providing good mitigation performance even in case of a power supply failure. Smart materials have been extensively investigated to be used in semi-active configurations. For smart materials one refers to materials that have properties that react to changes in their environment. This means that one of their properties can be changed by an external condition, such as temperature, light, pressure or electricity. Examples of materials belonging to this category are electrorheological fluids,
magnetorheological fluids and piezoelectric materials. Electromagnetic transducers can also be used in semi-active techniques by connecting an active shunt circuit to the electromagnet terminals.

2.3.4.1 Electrorheological Fluids

Electrorheological fluids are catching much attention in the literature due to their capability of changing their damping features accordingly to the variation of the applied electric field [135, 136]. A simple mathematical model of this type of dampers was proposed in [137] which consisted of two springs, a variable Coulomb frictional element and a viscous damping element as shown in Figure 2.22. This model was proved to be accurate for both particle-dispersion type and liquid-crystal type of electrorheological fluids. However, the latter showed that the variable viscous damping element presents a greater variation range than the one of the variable Coulomb frictional element as the external applied voltage was changed, and this considerably differs from the behaviour of the particle-dispersion type.

![Figure 2.22: Electrorheological-fluid variable damper [137]. a) equivalent model; b) cross-section of the damper](image)

The main drawbacks of electrorheological fluids are the added complexity due to the need to seal the fluid inside the device and the significant amount of mass that is necessary in order to achieve a reasonable level of damping.
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2.3.4.2 Magnetorheological Fluids

Magnetorheological fluids present similar features to those of the electrorheological fluids. Magnetorheological fluids can have superior and stable properties in a wide temperature range (−40 to 150 °C) and the control of these fluids does not require a considerable amount of external power [138–140], and for these reasons they have been extensively studied and used in several micro-vibration mitigation systems. The analytical model and the basic configuration of the magnetorheological fluid dampers are essentially the same as those of the electrorheological fluid ones (see Figure 2.22) but in terms of micro-vibration attenuation the former presents a much better damping performance than the latter. Moreover, a combination of permanent magnets and electromagnets can be implemented on these dampers in order to optimise the passive damping, increase the reaction time and further reduce the power consumption [141].

On the other hand, magnetorheological fluid dampers present fundamental drawbacks that have limited their use in space structures, such as the considerable weight introduced with these dampers and their highly nonlinear nature that might complicate the control of their active part. In theory, this intrinsic nonlinearity could be exploited through a feedback control to reproduce a device that behaves like a cubic nonlinear viscous damper [142]. However, the implementation of such device and the evaluation of its full potential are still under investigation [143, 144].

2.3.4.3 Piezoelectric Materials as Semi-Active Controllers

Piezoelectric materials can operate as semi-active controllers if they are connected to switching shunt electrical circuits [145, 146]. This method differs from an active method because in this configuration the actuators are passive devices whose characteristics can be altered in real time. Mechanical vibration energy can be dissipated in the process of switching between the open-circuit (high stiffness) and resistive-circuit states (low stiffness). In other words, the energy is stored in the high-stiffness state and dissipated in the low-stiffness state with the whole recurring process that happens within each vibration cycle. Onoda et al. [147] presented an energy recycling technique with embedded piezoelectric transducers to implement semi-active vibration control of space truss structures. Although this technique offers good isolation performance and stability, many sensors and a huge calculation cost for vibration control are required, especially for complex structures, and the isolation performance at low frequency of these semi-active systems still need to be improved [15].

40
Another way to improve the damping performance of a piezoelectric material is by using a synthetic negative capacitance circuit. By reducing the inherent capacitance of PZT, this circuit is capable of controlling the elasticity of the piezo element and increasing the electro-mechanical coefficient $[148–150]$. By choosing an optimal value of the negative capacitance it is possible to considerably enhance the damping performance and also broaden the frequency range of application. Nevertheless, there is the risk of having instability at low frequency and so it is necessary to perform a trade-off of the electric parameters in order to find the right balance between attenuation performance and system stability $[151]$.

### 2.3.4.4 Electromagnetic Shunt Dampers with Negative Impedance

As shown throughout this chapter, electromagnetic transducers offer great versatility since they can be used either as passive or active dampers. However, the main drawback of these devices is the high ratio of system mass over provided damping performance. This ratio can be considerably reduced by adding a negative impedance in the shunt circuit. In fact, by using a negative resistance circuit it is possible to reduce the overall resistance of the electromagnet that results in an increase of the electric current and thus of the damping force $[152]$. Recently, negative impedance included in EMSDs were tested for absorber applications on vibrating beams $[30, 153, 154]$ and plates $[31, 155]$. By combining a negative inductance (to produce a frequency-independent damping force) and a negative resistance (to increase the magnitude of the force) in the shunt circuit these studies demonstrated the ability of the EMSD to considerably attenuate multi-mode vibrations on extended structures (see Figure 2.23).

Few examples of negative-resistance EMSDs used in isolators have been found in literature $[156, 157]$. These devices have shown to be able to reduce the isolator’s resonance peak, but their analysis is still partial and presents some limitations. For instance, the shunt circuits seem to be always implemented using a controller board (e.g. dSpace R1103) connected to an elaborated circuitry to obtain the desired value of the negative impedance (the electronic board from $[156]$ is reported in Figure 2.24). Moreover the evaluation of the power required by the shunt is not addressed in literature.

A detailed investigation on the use of this damper in a space application has not been conducted yet. In particular, a thorough assessment of the attenuation performance of this device in which thermal and energy aspects are also taken into account would be fundamental in the determination of the maximum potential of this device, and would finally affirm it as an advanced alternative with respect to the other well-known dampers.
2.4 Current Isolator Limitations

Although the previous section has shown several possible damping methods that have been developed in the last four decades, only few of them have actually been embedded in real isolators to be used in space applications. With the exception of viscoelastic materials (which are not bound to work uni-axially thus effectively behaving as a 6-DoF isolating system), isolators are usually presented as strut elements which allow for the longitudinal displacement while preventing or limiting the other five DoFs. Six of these struts are then connected to
form a compact hexapod platforms (or three bipods) in order to perform isolation in all six DoFs.

It is widely agreed in the field of micro-vibration mitigation that a combination of passive and active isolation (either in the hybrid or semi-active configuration) could be the best compromise to obtain the advantages of both sides [15]. Several examples of hybrid isolators have been studied and designed. This section provides an overview of the most interesting systems found in literature and highlights their features and limitations. These hybrid isolators can be divided in two different categories: Hard Active Isolators (HAI) and Soft Active Isolators (SAI) [158].

HAI use a stiff actuator in series with a stiffening element (e.g. spring or membrane). The Satellite Ultraquiet Isolation Technology Experiment (SUITE) was a piezoelectric-based technology that flew in 2001 on board of a PICOSat spacecraft [132]. Each strut was made of a flexure and a piezoelectric actuator that were spaced by a geophone motion sensor (see Figure 2.25a). This strut was characterised by a stroke of only 30µm peak to peak and a suspension frequencies located between 25 and 75 Hz (see Figure 2.25b). The SUITE was capable of performing narrowband vibration isolation with the reduction of disturbance to be at least 30 dB at the frequency of interest. However, the broadband vibration isolation was heavily affected by the high stiffness of the strut which resulted in an attenuation between 1 Hz and 100 Hz that did not exceed 10 dB.

Another example of HAI is represented by the Miniature Vibration Isolation System (MVIS) developed by the Air Force Research Laboratory and Honeywell Defense and Space Electronics [159, 160]. The isolator strut consisted of a D-Strut™ in series with a piezoelectric actuator, and two of them were connected as shown in Figure 2.26a to form a bipod. The active control of MVIS was able to further attenuate the disturbance at low frequency while the viscous damper guaranteed the classic −40 dB/decade roll-off slope (see Figure 2.26b). Despite having a larger stroke compared to the one of SUITE (76 µm peak to peak), this system presented also several drawbacks such as the need to perfectly seal the liquid inside the strut, the 9-kg electronic mass necessary to control the piezoelectric actuator (70% of the total isolator system mass and about 60% of the suspended payload mass), and the 3-W nominal power required by each actuator.

A larger stroke can instead be achieved with SAIs which are characterised by a soft actuator (typically a VCA) in parallel with a stiffening element. Although SAIs require the payload mass to be off-loaded during ground testing and satellite launching, they present lower corner frequency than HAIIs that result in a better broadband micro-vibration attenuation. A sub-hertz vibration isolator for a deep space optical communication transceiver was developed by
2.4 Current Isolator Limitations

**Figure 2.25**: SUITE strut [132]. a) strut assembly; b) strut force transmissibility in passive mode

**Figure 2.26**: MVIS bipod [159]. a) bipod assembly; b) bipod force transmissibility
2.4 Current Isolator Limitations

Figure 2.27: Sub-hertz vibration isolator for a deep space optical communication transceiver [161]. a) hexapod configuration; b) single strut force transmissibility

Figure 2.28: “Sky-hook” vibration isolator [162]. a) strut assembly; b) strut force transmissibility with and without active control
2.4 Current Isolator Limitations

**Figure 2.29:** Hybrid strut with viscous fluid in parallel with VCA [163]. a) strut assembly; b) strut force transmissibility with and without active control

**Figure 2.30:** Hybrid 2-collinear-DoF strut with magnetorheological fluid in parallel with VCA [164]. a) strut assembly; b) strut force transmissibility with and without active control
2.4 Current Isolator Limitations

Figure 2.31: For transmissibility for a dual stage passive vibration isolator

the JPL [161]. A soft stainless-steel membrane gave the low stiffness at the strut whereas the active control was able to reduce the disturbance transmission at the resonance frequency by almost 60 dB (see Figure 2.27b). A similar configuration was also developed by ULB [162]. The stiffness of the strut membrane was chosen so to produce an attenuation of at least 5 dB in the frequency band 5 – 400 Hz with a maximum attenuation of 40 dB near 100 Hz. The VCA was used through a “Sky-hook” feedback control strategy to considerably reduce the first resonance peak, as shown in Figure 2.28b. The main drawback of both configurations was the use of a single membrane which was characterised by low bending stiffness. For this reason, structural secondary modes were excited in the frequency range of interest (in [161] the strut presented undesired modes already from 20 Hz), thus compromising the high frequency micro-vibration mitigation.

Other examples of SAIs used viscous fluids to perform passive damping. A D-Strut in parallel with a VCA was recently tested for micro-vibration disturbances caused by reaction wheels [163]. The strut had a mass of 0.927 kg and was tuned for the isolation of a 5-kg suspended mass. This strut, although showing good correlation between the analytical model and the experimental results, did not present a significant advantage in using a high-viscosity fluid together with a VCA. As it can be observed from Figure 2.29b, the active control reduced the resonance peak by almost 13 dB, and for both configurations the maximum attenuation achievable is $-35$ dB at 75 Hz.

A 2-collinear-DoF strut made of a VCA in series with a magnetorheological fluid was instead studied in [164]. The smart material was actively controlled to tune the fluid viscosity to a
desired value, whereas the VCA was used for micro-positioning purposes. The strut presents interesting isolation performances both during the semi-active and active operating modes (see Figure 2.30b). However, this strut is characterised by an intermediate mass of 15 kg (corresponding to more than 90% of the suspended mass) and it requires around 24 W to control the magnetorheological damper at peak force.

All the limitations presented so far in this section are the main cause of the almost negligible use of hybrid struts in space missions. Although promising, these technologies are not ready yet to be included on board spacecraft that require high pointing stability. Therefore, passive dampers are still primarily used for micro-vibration mitigation purposes. In particular, vibration isolators utilizing viscoelastic materials have been extensively developed both for scientific missions (e.g. Advanced X-ray AstroPhysics Facility [165] and James Webb Space Telescope [166]) and for commercial space applications [167].

The isolation performance of viscoelastic materials has been chosen as benchmark throughout this report. Specifically, the force transmissibility of a passive isolator strut for optical interferometer missions [168] was picked due to the low corner frequency (see Figure 2.31) that is comparable with the targeted corner frequency of the devices proposed in this report.

### 2.5 Summary

This chapter described the main damping methods that have been studied and developed for micro-vibration mitigation. These devices were divided into four categories: passive, active, hybrid and semi-active dampers. Passive dampers are usually the cheapest and most reliable solution for micro-vibration attenuation, and for this reason they are often preferred for space applications. However, their limited damping performance is seldom not enough for high-resolution payloads. On the other hand, active dampers are capable of achieving a higher mitigation, especially at low frequency, but they are also characterised by significant amount of input power and the possibility of instability issues. Hybrid dampers are able to combine the advantages of passive and active dampers. Hybrid dampers have been widely developed to be embedded in real isolator applications as described in Section 2.4. However, the excessive weight of these devices along with their high power consumption and limited reliability have heavily contributed to their almost negligible use in space missions.

On the other hand, semi-active dampers could solve some of the problems shown by hybrid dampers, such as the need of several sensors or the risk of instability issues. Semi-active
dampers work mainly as passive devices with some parameters that can be actively modified. They are characterised by low power consumption, high reliability and performance versatility, and for these reasons they are considered the best candidates for future high-pointing space missions.

In the context of semi-active dampers, the EMSD with the use of a negative resistance converter in the shunt circuit was found to have the most interesting features and potential to become the next-generation damper to be employed in space. This device combines the advantages of the active control (i.e. amplification of the circuit current that produces a higher damping force) with the elimination of possible instabilities deriving from the control loop (i.e. the negative resistance does not require a control algorithm to operate). Examples of negative impedances used in conjunction with EMSDs have been found in literature and they have shown to be effective in improving the micro-vibration mitigation. However, their analysis is partial and needs further investigation. For instance, these dampers have been used mostly in absorbers applied either to beams or plates but their employment in isolators has been marginally studied. In terms of the negative resistance, this impedance is always obtained with complex circuitry and its power consumption has never been taken into account.

This thesis aims at investigating the advantages of using a negative resistance converter circuit. This simple circuit could considerably simplify the damper design, but a full characterisation still needs to be performed in order to assess the maximum potential of this device. A detailed analysis of the EMSD model in a space environment, addressing not only the complete evaluation of the device’s damping behaviour but also the determination of system mass and power consumption are the main goals of this work.
Chapter 3

Electromagnetic Shunt Damper

This chapter reports an in-depth analysis of an EMSD applied to a 1-DoF system. After an overview of the damper properties, the analytical model of the system under examination is reported. It follows an investigation on different magnet configurations in order to find the one with the best ratio of damping performance over system mass. The description of the negative resistance converter is then presented, followed by the multiphysics approach adopted throughout this thesis that has allowed the analysis of the EMSD in changing environmental conditions. Finally, the parametric trade-off of the damper aimed at meeting stability and performance requirements is reported.

3.1 Damper Properties

An EMSD is a device that produces a force every time there is an interaction between a time/spatial changing magnetic field and a conductive material. In particular, when a conductor is subjected to a variation of the surrounding magnetic field, induced currents are generated within the conductor in such a way that they produce their own magnetic field with the opposite polarity of the applied field, causing a resistive force. This repulsive force is proportional to the relative velocity between the conductive metal and the magnetic-field source. Since the currents are dissipated into heat within the conductor, energy is removed from the system, thus allowing the magnet and conductor to function like a viscous damper.

As already stated throughout chapter 2, an EMSD presents several advantages that are going to be thoroughly described in this section. First of all, an EMSD is extremely versatile since
it can be used either as passive, active or semi-active by just changing few components in the electric circuit. Secondly, this kind of damper is simple and mechanically robust, thus requiring little or no maintenance throughout its life. Its robustness is due to the use of only metallic materials, which avoids the occurrence of issues related to deterioration of seals, liquid leakages or outgassing. Thirdly, an EMSD produces a contactless damping force that not only facilitates its installation, and makes it preferable in applications where direct contact of an actuator could jeopardise the structural integrity (e.g. inflatable or deployable membranes), but also it avoids the introduction of extra stiffness, typical of other damping devices (e.g. D-Struts). The latter feature allows the enhancement of the damping properties of the system while keeping its natural frequencies and mode shapes unaltered. Finally, with regard to the environmental conditions, these dampers are relatively insensitive to temperature variations and have a linear behaviour throughout most operating conditions (which is one of the major limitations of viscoelastic materials).

However, in spite of these important advantages, the use of EMSDs for space applications has been almost negligible. The main reason is the strict dependence of the damping force on the geometrical and physical features of the permanent magnet and conductor, and consequently the high ratio of system mass over damping force produced (see Section 3.2). Several optimised designs have been proposed in the last decade, but the slightly-reduced damping coefficient density does still not justify their use in space. Palomera-Aria in his doctorate thesis [25] reached the conclusion that to obtain the same damping capacity of a fluid viscous damper, an electromagnetic transducer should use at least one-and-a-half times the volume of the fluid damper, and cost at least five times more.

3.2 Analytical Model

3.2.1 Physics Principles

An EMSD functions through the combination of two different physics phenomena that are described by the Faraday-Lenz law and the Lorentz force law. The Faraday-Lenz law asserts that a relative velocity between an electromagnet and a permanent magnet induces an electromotive force $E_0$ (i.e. electric voltage) at the terminals of the electromagnet (see Figure 3.1a). This induced voltage can be expressed with the equation:

$$E_0 = \oint (\vec{v} \times \vec{B}) \cdot d\vec{l}$$  \hspace{1cm} (3.1)
where $\vec{v}$ is the magnet velocity with respect to the electromagnet, $\vec{B}$ is the magnetic field and $d\vec{l}$ is the infinitesimal length of the coil turns along which the integral is computed.

![Diagram of Faraday-Lenz law and Lorentz force law](image)

**Figure 3.1**: Schematic representation of the Faraday-Lenz law and Lorentz force law. a) Faraday-Lenz law: an induced electromotive force $E_0$ is produced by the cross product of the relative velocity (between the magnet and the coil) and the magnetic field; b) Lorentz force law: the cross product between the induced current and the magnetic field produces a force that always opposes the direction of the velocity.

Considering a cylindrical coordinate system ($\hat{r}$, $\hat{\phi}$, $\hat{z}$, with the $z$-axis along the magnet’s longitudinal axis), and the only relative motion between the conductor and the magnet to be along the $z$-axis, the velocity, $v$, and the magnetic field, $B$, can be written as:

$$\vec{v} = 0 \hat{r} + 0 \hat{\phi} + v_z \hat{z}$$
$$\vec{B} = B_r \hat{r} + B_\phi \hat{\phi} + B_z \hat{z}$$

This results in an electromotive force equal to:

$$E_0 = \int (v_z B_r \hat{\phi} - v_z B_\phi \hat{r}) \cdot r \, d\phi = \int v_z B_r \, r \, d\phi$$

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

In the micro-vibration load case, it can be assumed that the magnetic field seen by the conductive material is constant, since the relative displacement is in the order of magnitude of tenths of a millimetre, and so its variations are almost negligible. Also, given the small cross section of the electromagnet, one can assume a linear trend of the magnetic field along
the radial axis inside the coil. Therefore, Eq. (3.3) can be simplified as:

$$E_0 = 2\pi n_t r_{avg} \bar{B}_r v_z = K_d v_z \quad (3.4)$$

where $n_t$ is the number of turns of the coil, $r_{avg}$ is the average radius of the conductor, and $\bar{B}_r$ is the average radial component of the magnetic field through the coil cross section. $K_d$ is defined as the electro-mechanical transducer coefficient. This coefficient, determined by some intrinsic properties of the system, quantifies the coupling between the mechanical domain and the electric one, and it is obtained as:

$$K_d = 2\pi n_t r_{avg} \bar{B}_r \quad (3.5)$$

Once the induced current is generated, it couples with the surrounding magnetic field to produce the Lorentz force (see Figure 3.1b). This force is described by the equation:

$$\vec{F}_d = \int I d\vec{l} \times \vec{B} \quad (3.6)$$

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

$$\vec{F}_d = -2\pi n_t r_{avg} \bar{B}_r I \hat{z} = -K_d I \hat{z} \quad (3.7)$$

The combination of the Faraday-Lenz law and the Lorentz force law demonstrate that the force produced by a permanent magnet which moves close to a conductive material is proportional in magnitude and opposite in direction to their relative velocity. Therefore, this kind of force opposes the movement of the magnet and acts like a viscous damper, causing the vibration energy of the moving mass to dissipate through resistive/Joule heating generated in the conductive component.

### 3.2.2 Single DoF System

In order to demonstrate the validity of these assumptions and to assess the vibration attenuation performance of an EMSD, a single DoF system has been chosen [169]. A schematic of the system studied in this report is shown in Figure 3.2. This model consists of a mass suspended on a metal spring with stiffness coefficient $k$ and an EMSD. The closed electric
3.2 Analytical Model

circuit formed by the electromagnet and the shunt can be equivalently represented as a voltage source $E_0$ in series with an inductance $L_e$ and two resistances represented by the resistance of the electromagnet, $R_e$, and the equivalent resistance of the shunt, $R_s$. It is noted that a single DoF system is not representative of the actual behaviour of a real noise source (e.g. reaction wheel), which produces disturbances in all six degrees of freedom. However, this configuration is aimed at obtaining a first assessment of the damper performance that can then be compared with other well-established damping solutions [162, 168, 170]. In particular, special attention is given to the assessment of the force transmitted to the ground.

![Figure 3.2: Schematic representation of the single DoF model, with the electrical schematic of the shunt circuit of the EMSD](image)

In this study, the mass of the satellite where the reaction wheels are connected to has been assumed to be several orders of magnitude bigger than the reaction wheels, and hence the ground can be considered fixed in space.

The system shown in Figure 3.2 can be modelled by including Eqs. (3.4) and (3.7) in a fully-coupled system of four equations:

$$
\begin{align*}
    m\ddot{z} + kz &= F_d + F_{in} \\
    E_0 &= K_d v_z \\
    L_e I + (R_e + R_s) I &= E_0 \\
    F_d &= -K_d I
\end{align*}
$$

(3.8)

which respectively represent: the equation of motion (the input force is described by the term $F_{in}$), the Faraday-Lenz law of induction, Kirchhoff’s voltage law associated with the electric circuit, and the Lorentz force. An equivalence between the damping force produced by an EMSD and the one generated by a viscous damper can be made at the resonance frequency in terms of equivalent damping coefficient. In fact, the inductance can be neglected.
at low frequency and by combining the Kirchhoff’s voltage law with the Faraday-Lenz law of induction, the current can be written as:

\[ I = \frac{K_d}{R_e + R_s} \cdot v_z \]  

(3.9)

and by substituting Eq. (3.9) in the equation of motion, the damping force is equal to:

\[ F_d = \frac{K_d^2}{R_e + R_s} \cdot v_z \]  

(3.10)

which corresponds to a damping coefficient at the resonance frequency defined as:

\[ c_m = \frac{K_d^2}{R_e + R_s} \]  

(3.11)

### 3.2.3 State Space Modelling

The assumptions of micro-vibration load case (i.e. displacements in the order of tenths of a millimetre) and steady-state conditions allow this system to be studied as a linear, time-invariant model. Therefore, Eq. (3.8) can be linearised and rearranged via a state-space representation. The use of state space modelling considerably simplifies the analysis of the system in the frequency domain into which it can be converted by taking the Laplace transform. The state-space representation of a linear, time-invariant-coefficient system is written in the following form:

\[
\dot{x} = Ax + Bu \quad y = Cx + Du
\]  

(3.12)

where \( x \) and \( A \) are respectively the state vector and the state matrix, \( u \) and \( B \) are the input vector and input matrix, \( y \) and \( C \) are the output vector and the output matrix, and \( D \) is the feedthrough matrix (different to zero in case the system has a feedforward loop).

In the system under examination, the state vector consists of three state variables: mass displacement (\( z \)), mass velocity (\( \dot{z} = v_z \)), and circuit current (\( I \)). Rearranging the four equations shown in Eq.(3.8), the state space model is obtained as

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

(3.13)
The output vector $F_{out}$ represents the force transmitted to the satellite structure. This single-input-single-output (SISO) system can be converted to the frequency domain, and solved for the force transfer function (TF). The TF (often indicated in literature with the function $H(s)$) is a mathematical representation of the relation between the input and output of a linear, time-invariant system. Based on Eq. (3.12), the TF is obtained as

$$H(s) = \frac{y(s)}{u(s)} = C(sI - A)^{-1}B + D = \frac{P(s)}{Q(s)}$$

where the solutions of the two polynomials $P(s)$ and $Q(s)$ are respectively the zeros and poles of the TF. In particular, the poles are a characteristic property of a system (whereas the zeros can change by changing the matrix $C$, and so by considering a different output), and they must have a strictly-negative real part in order to make the system stable. Moreover, in a typical Bode plot, each pole produces a TF slope reduction ($-20 \text{ dB/dec}$ or $-40 \text{ dB/dec}$ depending on whether the pole is real or complex), while each zero increases the TF slope ($+20 \text{ dB/dec}$ or $+40 \text{ dB/dec}$ depending on whether the zero is real or complex).

In order to better understand how different poles and zeros affect the overall behaviour of a TF, an example with three different system configurations is shown in Figure 3.3. In the case of a simple mass-spring system, the TF would be only characterised by a couple of complex-conjugate poles, which determine the resonance peak and the roll-off slopes of $-40 \text{ dB/dec}$. Adding to this system a viscous damper would produce a reduction of the resonance peak but also the introduction of a real zero that brings the decay rate to $-20 \text{ dB/dec}$. The EMSD proposed in this report has the capability of considerably reducing the resonance peak while introducing an extra real pole that causes the roll-off slope to be again $-40 \text{ dB/dec}$. This additional pole is due to the presence of high inductance in the circuit with respect to the overall resistance. In fact, in a typical Eddy-current damper (where the inductance is neglected [36, 80, 171]) the device would behave as a perfect viscous damper and the TF would show the characteristic roll-off slope of $-20 \text{ dB/dec}$. However, the inclusion of the inductance produces the third differential equation in the state space model (as shown in Eq. (3.13)) and this results in the system being characterised by three poles (a complex conjugate pair from the equation of motion, and a real one from the electric circuit equation).
3.3 Magnet Configuration

Figure 3.3: Comparison between TFs associated with three different 1-DoF systems: mass-spring system characterised by one couple of complex-conjugate poles (blued line), mass-spring-dashpot system characterised by one additional real zero (green line) and mass-spring-EMSD system that introduces an extra real pole thus restating the final decay rate to $-40$ dB/dec (red line).

In particular, the frequency associated with this third pole depends on the characteristic conductor ratio [172, 173] defined as:

$$f_{\text{extrapole}} = \frac{R}{2\pi L}$$  \hspace{1cm} (3.16)

For instance, the mass-spring-EMSD system shown in Figure 3.3 is characterised by a resistance of $0.35\,\Omega$ (chosen so that the amplification at the resonance frequency was the same as the case with the viscous damper) and an inductance of $0.22\,\text{mH}$. These parameters resulted in the extra pole to be at $250\,\text{Hz}$. Therefore, it is clear that having a low ratio of resistance over inductance can shift this pole at lower frequencies, thus improving the overall damping performance at high frequencies.

3.3 Magnet Configuration

The damper performance can be improved, besides other means, through the increasing of the electro-mechanical transducer coefficient, $K_d$. The EMSDs found in literature are often
3.3 Magnet Configuration

characterised by low values of $K_d$ (e.g. the system proposed in [156] has a coefficient equal to 1.38 N/A). During this research, an extensive trade-off between various magnet configurations was conducted with the software COMSOL Multiphysics. All the configurations use Neodymium-Iron-Boron magnets with grade N35 characterised by a residual induction of 1.2 T. The residual induction is enough to define the magnetisation of a magnet in COMSOL and its sign determines the direction of the polarisation. For vacuum and ferromagnetic materials, COMSOL-built-in properties have been used. In order to compare these different designs a specific parameter was used and it was defined as the ratio of the coefficient $K_d$ to the mass of the magnetic assembly. The cross section of the electromagnet and its number of turns were fixed so to facilitate the comparison of the different configurations. In particular, the cross section was chosen to be 1 x 0.5 cm and by using a 0.25-mm-radius, enamel-coated copper wire the number of turns resulted to be approximately 136.

![Figure 3.4](image1)

**Figure 3.4:** One-ring-magnet configuration. (a) 3D view; (b) 2D-axisymmetric view; (c) magnetic flux density

![Figure 3.5](image2)

**Figure 3.5:** Two-ring-magnet configuration. (a) 3D view; (b) 2D-axisymmetric view; (c) magnetic flux density
The first configurations that have been investigated involved the use of ring magnets with radial magnetisation. This choice was dictated by the fact that only the radial component of the magnetic field $B_r$ affects the damper performance, hence a radial magnetisation seemed to be the most logical option. A ring with a cross section of $1 \times 1$ cm and radius of 1.5 cm was initially studied. The electromagnet was placed outside the ring magnet with a clearance gap of 1 mm (see 2D-axisymmetric configuration and 3D model in Figure 3.4). It is important to note that the magnetic field lines follow the shortest, easiest path to go from the north pole to the south one (i.e. in case the magnet is close to ferromagnetic materials, the high magnetic permeability of these materials constitutes a preferable path for magnetic field). For the one-ring configuration the flux lines exiting the external face of the ring turn almost immediately toward the south pole thus resulting in low radial magnetic field within the electromagnet (phenomenon that can be observed in Figure 3.4c where the cross section of the coil is coloured in dark blue). In order to increase the radial magnetic field and “force” the flux lines to cross the electromagnet, an additional ring magnet with same cross section as the previous one was added facing the outer part of the coil, see Figure 3.5. As it can be observed from Tab. 3.1, the use of the second ring magnet increases the radial magnetic field within the electromagnet as expected thus producing a $K_d$ that is almost three times more than the one of the single ring configuration. On the other hand, the 2-ring configuration presents a considerably increased mass that results in a specific electro-mechanical transducer coefficient that is less than half of the other configuration. Moreover, it is worthwhile to mention that obtaining a true radially-magnetised ring magnet is expensive and only few companies in the world claim that are able to achieve such magnetisation. Instead, it is more common to obtain an approximation of radial ring magnetisation made of several arc segments where the magnetic orientation is along a straight axis (see Figure 3.6). Therefore...
the use of ring magnets has been discarded in this research due to the increase in mass, costs and manufacturing complexity.

Figure 3.7: Single-cylindrical-magnet configuration. (a) 2D-axisymmetric view; (b) magnetic flux density

Figure 3.8: Single-cylindrical-magnet configuration plus iron yoke. (a) 2D-axisymmetric view; (b) magnetic flux density

The second part of the trade-off involved the use of cylindrical magnets that are longitudinally magnetised. Cylinders are preferable with comparison to other shapes (e.g. cubes) due to
3.3 Magnet Configuration

the perfect axial symmetry of their magnetic fields. First, a single cylindrical magnet characterised by 3 cm of diameter and 2.5 cm of height was analysed (see Figure 3.7). The dimensions were chosen so that its mass was identical to the one of the single ring magnet (about 135 g). As reported in Tab. 3.1, this configuration not only produces a slightly greater radial magnetic component compared to the single-ring-magnet configuration, but also allows the use of a second coil at the south pole of the magnet (i.e. due to the bipolarity of magnets, the two poles present the same magnetic field in magnitude but with opposite direction). In this way the coefficient $K_d$ can be considerably enhanced since the two electromagnets connected together have a total of 272 turns. Another configuration analysed was characterised by the same cylindrical magnet studied before with the addition of two iron yokes at the two magnet’s ends (see Figure 3.8). As already mentioned above, the iron yokes (i.e. ferromagnetic material) create a preferential path for the magnetic field that produces a slight increase in the radial component of the magnetic field seen by the electromagnets. However, this increase does not outweigh the added mass of the yokes thus resulting in a lower ratio of $K_d$ over mass when compared to the one of the single cylindrical magnet.

![Figure 3.9: Magnet stack with iron yokes. (a) 2D-axisymmetric view; (b) magnetic flux density](image)

The last configuration studied was formed by three cylindrical magnets arranged with opposing polarity and alternated with two iron yokes (see Figure 3.9). This configuration foresees the use of a non-magnetic, M4 screw to keep the magnets in place. The dimensions of the whole magnet stack (iron yokes included) were chosen in order to have a mass comparable to the one of the single magnet configuration. By forcing two magnets with opposite polarity to face each other, it is possible to considerably increase the radial component of the magnetic

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3.4 Negative Resistance

Field in the gap region between the magnets. This effect can even be bolstered by placing an iron yoke in that gap (see Figure 3.9b). From Tab. 3.1, it can be observed that this configuration produces a higher $B_r$ than the one of a single magnet that ultimately results in the highest ratio of $K_d$ over mass among all the configurations studied. Moreover, the last configuration presents a further advantage since the two free ends of the magnet stack can be potentially exploited by using two other electromagnets. For these reasons, the magnet assembly shown in Figure 3.9 was chosen to be used throughout this report. In particular, for the EMSD shown in Figure 3.2 the two electromagnets are connected together (but with opposite orientation to take into account the different direction of the magnetic field coming out of the iron yokes) and they are rigidly attached to a fixed boundary. The magnetic stack is instead bonded to the suspended mass and coaxially aligned with the electromagnets.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>$B_r$ (T)</th>
<th>$K_d$ (N/A)</th>
<th>mass (kg)</th>
<th>$K_d$/mass ($N/A$ kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 ring magnet (Figure 3.4)</td>
<td>0.14</td>
<td>2.81</td>
<td>0.14</td>
<td>19.93</td>
</tr>
<tr>
<td>2 ring magnets (Figure 3.5)</td>
<td>0.34</td>
<td>6.82</td>
<td>0.74</td>
<td>9.17</td>
</tr>
<tr>
<td>1 cylindrical magnet (Figure 3.7)</td>
<td>0.20</td>
<td>6.45</td>
<td>0.135</td>
<td>47.78</td>
</tr>
<tr>
<td>1 cylindrical magnet with yokes (Figure 3.8)</td>
<td>0.23</td>
<td>7.41</td>
<td>0.23</td>
<td>32.01</td>
</tr>
<tr>
<td>magnet stack with yokes (Figure 3.9)</td>
<td>0.28</td>
<td>6.7</td>
<td>0.135</td>
<td>49.63</td>
</tr>
</tbody>
</table>

Table 3.1: trade-off of magnetic assembly

3.4 Negative Resistance

As already shown in Section 3.2.3, the reduction of the ratio $R/L$ in the electric circuit results in the shift of the extra pole toward lower frequencies. One way to do this is by increasing the inductance in the circuit by means of passive inductors. Although this approach has some benefits, it has also some intrinsic limitations that cannot be overcome. First of all, there is limited availability of high-value components that can be found off-the-shelf. Secondly, these passive components have inherent resistances that, besides increasing the ratio $R/L$, negatively affect the response at the resonance peak. In fact, by adding resistance to the electric circuit the dynamic response of the system at the resonance frequency is worsened due to a reduction of the current flowing in the circuit and consequently reduced Lorentz force (see Eq. (3.7)). An effective way to tackle the issue of inherent properties of passive
3.4 Negative Resistance

electric components (e.g. parasitic resistance) or to reduce their effects is to include a negative impedance in the electric circuit. Negative impedances are externally-powered circuits that are capable of simulating passive components characterised by negative magnitudes. Their discovery has allowed engineers to explore new possible applications where small magnitudes of circuit features were needed but not achievable before then. Over the last decade, the use of negative impedances connected to transducers for the purpose of structural damping has been widely investigated and the benefits of this technology are thoroughly reported in literature (see Section 2.3.4.4).

The main focus for the 1-DoF system presented in this chapter is to obtain an isolator that resembles a 2nd-order filter characterised by almost no amplification at the resonance frequency and roll-off slope of $-40$ dB/dec. This can be achieved with the sole use of a negative resistance that has the dual benefit of reducing the overall resistance and decreasing the ratio of $R/L$. As shown in Section 2.3.4.4, several examples aimed at demonstrating these beneficial effects on the damping performance of EMSDs have been found in literature, mainly in absorbers but also in isolators. However, the limitations posed, for instance, by the employment of complex circuitry and the lack of an evaluation of power consumption need to be addressed before this technology can be used in a space application.

In this study, the negative resistance is created by utilising an analogue circuit called negative impedance converter [174]. This extremely simple circuit consists of a single operational amplifier (op-amp) and three resistors that are connected as shown in Figure 3.10. The equivalent resistance of the shunt $R_s$ is:

$$R_s = -R_x \left( \frac{R_z}{R_y} \right)$$  \hspace{1cm} (3.17)

By considering the resistances $R_y$ and $R_z$ to be equal to each other, the shunt produces a negative resistance $R_s = -R_x$. Moreover, in order to minimise the current loss from the op-amp output pin to the ground, the resistances $R_y$ and $R_z$ are chosen to be at least three order of magnitude greater than $R_x$, so that the current flowing in that branch is in the order of micro amperes. From a theoretical point of view, this circuit behaves linearly throughout the whole frequency range and tends to be stable as long as the resistance of the electromagnet, $R_e$, is greater in magnitude than the shunt resistance, $R_s$ (this aspect will be taken into account in the next section).

As mentioned before, the negative resistance converter is an active circuit that requires external power to function. The supplied power required by the shunt circuit can be expressed
3.5 Multiphysics Approach

Figure 3.10: Electrical schematic of the negative resistance impedance converter connected to the electromagnet

\[ P = (V^+ - V^-) \cdot (I_q + I_o) \]  \hspace{1cm} (3.18)

where \( V^+ - V^- \) is the total supply voltage, \( I_q \) is the quiescent current consumed by the op-amp (i.e. idle current drawn from the power supply), and \( I_o \) is the op-amp output current. An op-amp is ideally characterised by having zero current flowing into the input pins, which results in the current flowing through \( R_y \) being the same as the one through \( R_z \). Therefore, \( I_o \) can be approximated to the current flowing in the resistor \( R_x \), because by selecting \( R_y \) and \( R_z \) to be three orders of magnitude greater than \( R_x \) the current \( I_{R_yR_z} \) can be neglected. Since the signal is sinusoidal, the rms value of the power can be obtained as:

\[ P_{avg} = (V^+ - V^-) \cdot (I_q + I_{rms}) \]  \hspace{1cm} (3.19)

where \( I_{rms} \) is the rms value of the op-amp output current.

3.5 Multiphysics Approach

The damping performance and the stability of a system within the operating conditions are two crucial aspects of every isolation device. Space missions typically undergo wide temperature ranges that are characteristic of each subsystem, and depending on the type of missions these ranges can also span within temperatures that are always below 0 °C (e.g. the ESA’s Euclid is a high precision survey mission that requires the payload to operate between
3.5 Multiphysics Approach

$-170 \, ^\circ C$ and $-120 \, ^\circ C$ in order to achieve high thermal stability [175]). It is quite common to find in literature the characterisation of an isolation device without the temperature at which it is analysed or tested being specified, assuming that it has a linear behaviour throughout the whole temperature range of interest. However, this assumption is often proved not to be accurate. For instance, passive dampers although being inherently stable are often characterised by a strong dependency on the temperature and frequency range of operation (e.g. silicone fluids used in viscous dampers, specifically developed to be thermally stable, exhibit a viscosity drop of 50\% over a temperature range of $+20 \, ^\circ C$ to $+50$, as it can be seen in Figure 3.11). On the other hand, active dampers can present stability margins that do not cover the full range of environmental conditions.

![Figure 3.11: Temperature characteristics of the damping constant for silicone fluids [176]](image)

There is an increasing interest in the development of damping solutions that guarantee good isolation performances in a wide temperature range, and for this reason it is necessary that dampers undergo an extended analysis that takes into account also the thermal domain. An EMSD (made uniquely of metal parts such as Nd-Fe-B magnets and copper) has the advantage that only few parameters are notably affected by the temperature change. In particular, the copper resistivity of the coils increases linearly with respect to the temperature with a thermal coefficient of $0.00386 \, ^\circ C^{-1}$, whereas the residual induction of the Nd-Fe-B magnet (which affects the electro-mechanical coefficient, $K_d$) is characterised by a linear temperature coefficient of $-0.0012 \, ^\circ C^{-1}$. The different sign of these two coefficients means that at low temperature the coils would have a reduced magnitude of resistance and the magnet would present a higher residual induction, whereas they will behave in an opposite fashion at high temperature. Therefore, an EMSD works better at low temperatures (due to
3.6 Parametric Trade-off

the increase of \( K_d \) even though there is the risk of the system to become unstable if the coil resistance \( R_e \) is smaller in magnitude than the shunt resistance \( R_s \) (i.e. the total resistance \( R_e + R_s \) is negative).

In terms of the electric circuitry, the three resistors \( R_x \), \( R_y \) and \( R_z \) that form the negative resistance converter can be chosen among space-qualified, off-the-shelf components that have tolerances down to 0.005\% and temperature coefficients of \( 1 \cdot 10^{-6} \, ^\circ C^{-1} \). Hence these resistors can be considered constant over a wide temperature range when compared with the electromagnet resistance, \( R_e \).

All these effects have been included in a multiphysics model of the system that has been implemented on COMSOL Multiphysics. This model integrates the thermal domain with the electromagnetic and mechanical domains in order to assess how the damping performance of an EMSD changes within the temperature range of interest. Through this model, it has been possible to demonstrate that the small amount of energy dissipated by the coil in the form of Joule heating can be completely removed from the system if the coil is thermally coupled with a thermal conductive material (radiation would be less effective given the small surface of the coil and convection is not possible in space due to the absence of atmosphere).

Therefore, by removing the thermal power produced within the electromagnet the system is capable of maintaining the same temperature of the surrounding environment thus allowing it to be studied in steady-state conditions. In fact, the magnitudes of the coil resistance, \( R_e \), and the average radial component of the magnetic field, \( \vec{B}_r \) used in Eq.(3.13) are initially determined by the environmental temperature and are assumed to remain constant throughout the analysis. This assumption is corroborated by the fact that in a typical micro-vibration load case only a small amount of thermal power needs to be dissipated in the process of attenuating micro disturbances. The magnitudes of \( R_e \) and \( \vec{B}_r \) at +15 \( ^\circ C \) were taken as reference values, and a thermal analysis was performed in COMSOL to evaluate their variations within the temperature range of interest.

3.6 Parametric Trade-off

Having the analytical model fully defined, the next step has been the determination of the case study parameters. The suspended mass has been chosen to be 5 kg and corresponds approximately to the mass of two 100SP-O reaction wheels manufactured by SSTL. As already mention, the analysis of a 1-DoF system is a simplification of the actual behaviour of a reaction wheel, which produces disturbances in all six degrees of freedom. The operating
temperature range for this type of reaction wheel is from $-20^\circ C$ to $+50^\circ C$ [63], and this is
going to be the reference temperature range throughout this report.

The system under examination presents several parameters that need to be chosen and their
values can considerably affect the damping performance of the device. The full optimisation
of the whole system would be the ideal option in order to meet all the requirements of
micro-vibration attenuation but also mass and supplied power. Although such optimisation
has been initially attempted, it was immediately clear that integrating the magnetic domain
with the mechanical, electromagnetic and electrical domains would have been a tedious,
time-consuming process given the large amount of parameters involved and their non linear
relations with each other. For this initial phase it was decided that a rigorous system
optimisation was beyond the scope of this project and a reasonable trade-off would have
been sufficient.

The trade-off covering the whole temperature range of interest was chosen for some param-
eters after having fixed some other damper features. In particular, the dimensions of the
magnetic stack was set to be 58 mm in height and 20 mm in diameter, whereas the coil
cross section was chosen to be 11 x 5 mm, with 1 mm clearance gap between the magnet
and the coil. In this way, some parameters were pre-determined (e.g. average radius, $r_{\text{avg}}$, and
average radial magnetic field within the coil, $\bar{B}_r$) leaving only a relative small amount
of parameters that needed to be determined through the trade-off. These parameters were:
spring stiffness $k$, electric properties of the shunt circuit, wire diameter of the copper coil
that affect the electromagnet resistance $R_e$ and also the inductance $L_e$ through the number of
turns.

The requirements that were initially set for the trade-off can be divided into two subcategor-
ies: stability and performance requirements. In terms of stability, as already stated above, the
EMSD could become unstable at the lowest temperature range limit (where $R_e$ reaches its
minimum value and it could result in an overall resistance being negative). In order to prevent
this, a minimum value of 0.1Ω was imposed on the total resistance $R_e + R_s$ at a temperature
of $-20^\circ C$. In regard to the attenuation performance, it is important to notice that different
requirements would be necessary whether it is intended to isolate either a sensitive payload
or a noise source. A payload needs to be isolated from every disturbance signal (even at low
frequency), so ideally there should be no amplification at all within the whole frequency
range. On the other hand, a noise source tends to be more problematic at higher frequencies
which means that the attenuation at low frequency could be relaxed in order to gain higher
attenuation at high frequency. In fact, by increasing the inductance it is possible to move
the extra real pole toward lower frequencies (from Eq. (3.16)) up to a point that it will start
affecting the amplification at the resonance peak. For this reason, the isolation of a noise source allows the extra pole to be further ‘pushed’ toward lower frequencies while relaxing the strict requirements on the resonance amplification.

Since this project is focused on the isolation of reaction wheels from the satellite structure, the latter approach was adopted. In particular, the performance requirements that were established a priori were: the amplification at the resonance frequency to not be greater than 6dB throughout the whole temperature range, the corner frequency (i.e. the point in a TF where the output is half the value (−3dB) of the input) to be equal to 10 Hz, and to have a final slope of −40 dB/dec starting from at least 100 Hz.

The trade-off of the system parameters was performed to meet both the stability and performance requirements within the temperature range of interest. It was carried out by analysing the system at three different temperatures: −20 °C, +50 °C and the range mid point, +15 °C. The final choice of the parameter set that has met the predefined requirements is reported in Table 3.2.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass, m (kg)</td>
<td>5</td>
</tr>
<tr>
<td>Spring Stiffness, k (N/m)</td>
<td>4600</td>
</tr>
<tr>
<td>Coil Inductance, L_e (mH)</td>
<td>1.3</td>
</tr>
<tr>
<td>Shunt Resistance, R_s (Ω)</td>
<td>−1.5</td>
</tr>
<tr>
<td>Number of turns, n_t</td>
<td>270</td>
</tr>
<tr>
<td>Wire diameter, d_w (mm)</td>
<td>0.5</td>
</tr>
<tr>
<td>Coil Resistance, R_e (Ω)</td>
<td>1.60</td>
</tr>
<tr>
<td>E-M Transducer Coefficient, K_d (N/A)</td>
<td>6.98</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temp.</th>
<th>−20 °C</th>
<th>15 °C</th>
<th>50 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coefficient, K_e (N/A)</td>
<td>6.98</td>
<td>6.7</td>
<td>6.42</td>
</tr>
</tbody>
</table>

Table 3.2: Final choice of the parameter set obtained through a trade-off

A comparison of the force TFs, obtained from Eq. (3.12), between the system without damping (i.e. the shunt circuit is disconnected and the electromagnet is in open loop conditions) and the system with the EMSD is shown in Figure 3.12. 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 $-40\text{dB/dec}$ that is typical of a mass-spring system.

**Figure 3.12:** Comparison of the force TFs between the system without EMSD and the system with EMSD at three different temperatures

![Figure 3.12](image)

**Figure 3.13:** Force TF for a three parameter isolator analysed in [177]. (a) Schematic model of the isolator; (b) Force TFs associated with different values of the damping coefficient

![Figure 3.13](image)
3.6 Parametric Trade-off

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. This behaviour can also be seen in Figures 3.14 and 3.15. In particular, Figure 3.14 shows the evolution of the three eigenvalues starting from the mid-range temperature, 15 °C (marked with green squares). As the temperature decreases, the pair of complex conjugate poles tend to the real axis, and at the temperature of −1 °C ($R_{\text{tot}} = 0.24\Omega$) the three poles become all real and negative (purple squares in the plot). This condition is maintained only until the temperature reaches −1.5 °C, after that a pair of complex conjugate poles reappears but this time their real part is greater than the real pole (i.e. the complex poles have swapped with respect to the real one). As the temperature continues to decrease, the real part of the three poles tends to zero, and at a temperature of −34 °C the three poles cross the imaginary axis and the system becomes unstable. Therefore, given the set minimum temperature to be −20 °C, there is still a margin of 14 °C before reaching instability.

From a physical point of view, the relationship between the eigenvalues and the overall circuit resistance (i.e. system temperature) has the following explanation. As stated in Section 3.2.3, 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 [37]), 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. A similar behaviour was also shown in [177] where a three parameter isolator was analysed. From Figure 3.13, it can be observed how increasing the damping coefficient above an optimal value produces the negative effect of shifting the resonance frequency and increasing the amplification of the peak.

In the system under examination, the resistance of 0.1 Ω reached at the temperature of −20 °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.

![Eigenvalues Vs Temperature](image)

**Figure 3.15:** 3D representation of the variation of the system eigenvalues with respect to the overall resistance (i.e. system temperature).
3.7 Summary

This chapter presented a thorough analysis of an EMSD with the use of a negative impedance converter. In particular, several aspects of this technology were studied. An analytical model of the EMSD applied to a 1-DoF system was first presented. The system was studied both in the time domain and in the frequency domain (by taking the Laplace transform) and the advantageous characteristics of its TF (i.e. elimination of the resonance peak and final roll-off slope of $-40$ dB/dec) were outlined in comparison to mass-spring and mass-spring-dashpot systems. The configuration of the magnet was also investigated through a trade-off and it was concluded that the stack of magnets with opposite polarity alternated with iron yokes gave the highest ratio of electro-mechanical transducer coefficient over mass of the magnetic assembly. Another fundamental aspect of this chapter was the characterisation of the negative resistance converter circuit in which the benefits in terms of damping performance and power consumption were reported. The chapter concluded with the analysis of the damper performance at different environmental conditions which was the starting point for the parametric trade-off. In particular, the main parameters that were investigated were the spring stiffness, the magnitude of the negative resistance and the diameter of the copper wire (which affected also the number of turns of the coil) and a final configuration of these parameters that allowed for the 1-DoF system to meet all the predefined requirements was found.
Chapter 4

Model Validation

This chapter describes the experimental set-up and the test results that have validated the model presented in the previous chapter. First, an overview of the test rig which includes also a finite element analysis of the whole assembly is reported. It follows a brief description of two key aspects of the EMSD manufacturing such as the realisation of the electromagnet and the choice of the pcb components. Finally, several TFs obtained experimentally are compared with the numerical data in order to show the good correlation between test results and analytical model.

4.1 Experimental Set-up Design

A laboratory test rig was built in house and its fully-assembled configuration is shown in Figure 4.4a. It was bolted on a SSTL-customised multi-axial dynamometric table. This device consists of a very rigid plate that is supported by four three-axial load cells (manufactured by Kistler) in order to retrieve the forces and moments in all six DoFs. The test rig was designed to satisfy the requirement of the mass having only a single DoF, while trying to avoid the test rig’s vibration modes interfering with the data acquisition. The mass of 5 kg was obtained through the combination of a dummy mass formed by three steel plates bolted together (4.85 kg) and the mass of the magnetic assembly (150 g), see Figure 4.1. It is noted that the magnetic assembly and the two electromagnets weighted 190 g overall, which represents less than 4% of the total suspended mass. In order to simulate microgravity, the dummy mass was offloaded using bungee cords so that the system had a sub-hertz resonance frequency. A suspended mini-shaker has then been connected to the centre of the dummy
mass to reproduce the vertical disturbance of a reaction wheel. A stinger rod (see Figure 4.2) was used to convey a force since its axial rigidity is few orders of magnitude bigger than the bending stiffness. Therefore, even in case of not perfect centring of the mini-shaker with respect to the dummy mass, the stinger rod would considerably reduce the bending moments generated by such misalignment while still transferring the desired vertical force.

![Experimental test to verify the axial stiffness of the flexures. Low-stiffness bungee cords were used to simulate microgravity. (a) dummy mass made of three steel plates bolted together and suspended with large flexures; (b) magnetic assembly suspended with narrow flexures and bolted to an aluminium element that was used to rigidly connect the magnet to the dummy mass.](image)

**Figure 4.1:** Experimental test to verify the axial stiffness of the flexures. Low-stiffness bungee cords were used to simulate microgravity. (a) dummy mass made of three steel plates bolted together and suspended with large flexures; (b) magnetic assembly suspended with narrow flexures and bolted to an aluminium element that was used to rigidly connect the magnet to the dummy mass.

![Mini-shaker connected to the dummy mass through a steel stinger](image)

**Figure 4.2:** Mini-shaker connected to the dummy mass through a steel stinger
Steel flexures were used to guarantee that the mass could only move in the vertical direction, whereas displacements/rotations along the other five DoFs were minimised. A fixed-fixed configuration (see Figure 4.3) was chosen in order to support both the dummy mass and the magnet stack. The four flexures arranged in this configuration are characterised by low stiffness in the orthogonal direction (i.e. along the z-axis according to Figure 4.3) and high stiffness in the other two directions, along with high bending and torsional stiffness. In particular, the overall orthogonal stiffness obtained as the sum of the four flexures considered individually can be written as:

\[ k_z = 4 \cdot \frac{12 EI_x}{L^3} = 4 \cdot \frac{E wt^3}{L^3} \]  

(4.1)

where \( E \) is the Young modulus, \( I_x \) is the moment of inertia with respect to the x-axis, \( L \) and \( t \) are respectively the length and the thickness of the flexures whereas \( w \) is the width of the flexures measured along the x-axis. The stiffness along the y-axis is equivalent to:

\[ k_y = 4 \cdot \frac{Et w}{L} \]  

(4.2)

which can be found directly from the Hooke’s law. With regard to the stiffness along the x-axis \( k_x \), the flexure cannot be considered as a beam any more because the width is comparable with the length. Therefore, in this case the transverse shear stress becomes dominant over normal stress and the flexure results extremely stiff along the x direction.

**Figure 4.3:** Fixed-fixed flexure configuration chosen to support both the dummy mass and the magnet stack. This configuration guarantees low stiffness in the orthogonal direction and high stiffness in the other five DoFs.
4.1 Experimental Set-up Design

By choosing the thickness to be a couple of order of magnitude smaller than the width and the length it is possible to obtain a vertical stiffness $k_z$ that is at least $10^7$ times smaller than $k_x$ and $10^5$ times smaller than $k_y$. Moreover, since the flexures directly connect the two masses (dummy mass and magnet stack) to the ground their vertical stiffness also contributes to the total stiffness of the 1-DoF system (i.e. the stiffness $k$ shown in Eq. (3.8)). Therefore, a trade-off was necessary to establish the flexures’ dimensions in order to obtain the desired vertical stiffness while having local secondary modes at the highest frequency as possible. The trade-off was limited by the dimensions of the Kistler table. Two different sets of dimensions were assigned to the top flexures (supporting the dummy mass) and to the bottom flexures (supporting the magnet stack) and they are reported in Tab. 4.1. These two configurations, made of four flexures each, produced a vertical stiffness of about 760 N/m (for the top flexures) and 500 N/m (for the bottom flexures). Since they were in parallel, the eight flexures altogether provide a stiffness of about 1260 N/m.

These calculations were confirmed through a Nastran analysis after having built the model of the whole test rig on Patran (see Figure 4.4b). This model was made of 38920 hexahedral elements and 4216 quadrilateral shell elements (representing the flexures). The connection between the magnet stack and the dummy mass was obtained by linking two rigid elements (RBE2 Multi point constraints) through a spring connector (CBUSH) characterised by a vertical stiffness of $10^7$ N/m to simulate the rigid connection between the two masses. Another RBE2 Multi point constraint was used at the bottom of the magnet stack and connected to ground with a CBUSH characterised this time by a stiffness of 3340 N/m (derived from the desired stiffness of 4600 N/m reported in Tab. 3.2 minus the stiffness produced by the flexures).

<table>
<thead>
<tr>
<th>Property</th>
<th>Top Flexures</th>
<th>Bottom Flexures</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness, t (mm)</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Length, L (mm)</td>
<td>48</td>
<td>37</td>
</tr>
<tr>
<td>Width, w (mm)</td>
<td>100</td>
<td>30</td>
</tr>
</tbody>
</table>

*Table 4.1: Flexure trade-off*

The finite element analysis conducted with Nastran showed that the test rig behaved as a single DoF up to 150 Hz, where secondary modes in the flexures were excited, affecting the force TF (see Figure 4.5). Ideally, these secondary modes were desired to be above 300 Hz, but the dimensions of the Kistler table constrained the design of the flexures. It is also noted that the disturbances produced by the flexure secondary modes are conveyed to the ground.
4.1 Experimental Set-up Design

Figure 4.4: Experimental test rig for 1-DoF set-up. (a) test rig mounted on the Kistler table; (b) Patran model of the test rig

Figure 4.5: Force TF of the 1-DoF system without EMSD obtained through FEM (Nastran).
through the walls of the test rig. The damping introduced with the EMSD is not capable of damping those disturbances as it only acts on the forces conveyed through the centre of the test rig. Therefore, these modes that are visible both on the system without and with EMSD from 150 Hz onward are due to the configuration of test rig rather than the damper itself.

4.2 EMSD Manufacturing

The design and realisation of the EMSD required an appropriate selection of the circuit components in order to reproduce the same conditions simulated in the analytical model. The clearance of 1 mm between the magnet and the coil (as already shown in Section 3.3) was chosen as safety margin to prevent the two elements to get in contact in case the device would be tested for the launch vibration environment and lateral or tilting motions would occur. The choice of using self-bonding magnet wire allowed to fulfill the 1-mm-gap requirement. This type of wire is coated with an additional layer of adhesive polymer that is activated by heat or solvents. Once activated the adhesive bonds adjacent turns of wire together, forming a compact self-supporting coil. In this way, the inner spindle can be removed, and any extra spacing between the magnet and the coil is eliminated.

![Figure 4.6: Electromagnet made of self-bonding magnet wire](image)

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. 4.2). The three resistors have been picked among surface mount components characterised by a tolerance of 0.5%. The resistors $R_y$ and $R_z$ were chosen to be 1 kΩ whereas $R_x$ was equal to 1.5 Ω so to obtain the desired magnitude of the negative resistance.

![Printed circuit board used to reproduce the negative resistance converter](image)

**Figure 4.7:** Printed circuit board used to reproduce the negative resistance converter

<table>
<thead>
<tr>
<th>Property</th>
<th>Typ</th>
<th>Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Noise Voltage ($nV/\sqrt{Hz}$)</td>
<td>3.8</td>
<td></td>
</tr>
<tr>
<td>Quiescent Current (mA)</td>
<td>3.7</td>
<td></td>
</tr>
<tr>
<td>Output Current (mA)</td>
<td>50</td>
<td>80a</td>
</tr>
<tr>
<td>Supply Voltage (V)</td>
<td>±5</td>
<td>±6.3</td>
</tr>
</tbody>
</table>

* Value measured experimentally.

**Table 4.2:** Main features of the LT1722 op-amp

### 4.3 Experimental Results

The Kistler table is characterised by a 3-cm-thick aluminium plate that gives high bending and torsional stiffness to the whole supporting system (see Figure 4.8). The Kistler table was clamped on a granite block (with dimension of $40 \times 40 \times 10$ cm with a mass of approximately 60 kg) which was suspended on a pneumatic isolation system. This pneumatic suspension device was necessary in order to act as a low pass filter which was capable of reducing the mechanical noise coming from the surrounding laboratory environment. The purpose of the granite block was to increase the overall suspended mass and so decrease the first resonance peak. Nevertheless, the isolation system 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 sinusoidal input force sweeping from 15 Hz to 350 Hz (in the plots that will be shown later in this section the data from 15 Hz to 20 Hz, recorded in both tests, are taken from the latter sine sweep). The input force of the first sine sweep was set to be 0.5 N so as to prevent the flexures from operating in the nonlinear region due to relatively large displacements (e.g. in the order of millimetres). Due to the difficulty to move the whole test set-up inside a thermal chamber, this test campaign was performed at only one temperature (same condition for all the tests reported in this thesis). The lab temperature was approximately 18 °C.

![Image of Kistler table clamped on a granite block. The 3-cm-thick aluminium plate is supported by four three-axial load cells.](image)

**Figure 4.8: Kistler table clamped on a granite block. The 3-cm-thick aluminium plate is supported by four three-axial load cells**

The comparison between the experimental results and the analytical data (after tweaking the parameter set accordingly to the lab temperature) is shown in Figure 4.9. Through the observation of the test results for the system with the shunt circuit disconnected (see Figure 4.9a), it has been able to assess the test rig’s mechanical damping that was initially omitted from the analytical model. This damping naturally occurs in a mechanical assembly and can be produced by several factors (e.g. micro-friction between adjacent components, air resistance, hysteresis damping in the steel flexures and Eddy-current damping due to the relative motion between the magnet and other surrounding conductive materials, like aluminium). By adding some viscous damping in Eq. (3.8) (about 6% of the critical damping), it can be observed that good correlation between the analytical data and the experiments has been achieved. The behaviour of the system above 180 Hz is characterised...
by some resonance peaks that, as expected through the Nastran analysis, are attributable to the secondary vibration modes of the flexures. Good correlation can also be observed for the system with the EMSD (see Figure 4.9b). As already shown in Figure 3.12, 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. (3.14) as follows:

$$
Y = \begin{bmatrix} Y_v \\ Y_c \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \dot{z} \\ \dot{I} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \end{bmatrix} F_{in} \tag{4.3}
$$

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 (an example of the measured voltage signal is shown in Figure 4.10). The comparison of these analytically-obtained TFs with the experimental data is shown respectively in Figures 4.11 and 4.12. 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. (3.19) the average power required by the shunt at 5.60 Hz is obtained as:

$$
P_{avg} = (V^+ - V^-) \cdot (I_q + I_{rms}) = 10V \cdot (3.8mA + 0.707 \cdot 70mA) = 0.53W \tag{4.4}
$$

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. [132] the single strut requires a minimum of approximately 15 W to operate).
Figure 4.9: 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.
4.3 Experimental Results

Figure 4.10: Voltage measured across the electromagnet through an oscilloscope probe during the sine sweep test.

Figure 4.11: Transfer function between the input force and the mass velocity. The system considered is without EMSD.
4.4 Summary

This chapter showed the importance of fully analyse and design a test rig fit for purpose. In fact, although a 1-DoF system should be easy to test, its implementation required several iterations and try-and-error attempts to reach the final rig design. By knowing before hand via FEA the modal behaviour of the test rig under sine sweep load case, it allowed a better understanding of the test data. In particular, this analysis helped to confidently state that the disturbances showed in the TFs were not generated from some nonlinearity in the EMSD but they were due to the test rig itself. Important was also the determination through experiments findings of the mechanical damping which was originally omitted from the model. Overall, this chapter showed the good correlation between the analytical model and the test results across all measured parameters.

Figure 4.12: Transfer function between the input force and the induced current in the coil. The system considered is with EMSD.
Chapter 5

2-Collinear-DoF System

This chapter presents the development of a 2-collinear-DoF system aimed at further enhancing the isolation performance of the proposed damping method. Chapter 3 has shown that an EMSD installed in a 1-DoF system produces a roll-off slope of $-40$ dB/dec, which is characteristic of a mass-spring system, and is capable of completely eliminating the resonance peak (contrarily to the behaviour of viscoelastic materials). However, such attenuation is usually not sufficient in case a higher payload image resolution is required. The use of a 2-collinear-DoF system has been found in literature as one possible solution to produce a higher isolation performance. Contrarily to a TMD where the auxiliary mass is placed on top of the primary mass, this 2-DoF configuration has the auxiliary mass inserted between the primary mass and the base structure. The use of the second mass has the main advantage to introduce extra poles in the TF with the result of considerably increasing the final roll-off slope. In order for this configuration to be successful it is necessary to have two separate dampers acting between the two masses and between the secondary mass and the ground. The difficulty to actually achieve this has resulted in one of the major drawbacks in development of this type of isolators.

In this chapter, a benchmark example of a 2-collinear-DoF system using viscoelastic materials is first reported. Then the concept that has allowed the development of the 2-collinear-DoF isolator with two embedded EMSDs is presented. It follows the analytical model and the parametric trade-off that has brought to a configuration of the isolator capable of meeting the performance and system requirements. The chapter ends with the experimental validation of the analytical model and the performance comparison between the proposed 2-collinear-DoF isolator and other damping solutions found in literature.
5.1 Benchmark Study

In order to fully understand the advantages of the proposed 2-collinear-DoF isolator, a benchmark study is first reported. The schematic model can be seen in Figure 5.1. It consists of a 2-DoF system where two viscoelastic elements (with stiffness $k_1^*$ and $k_2^*$ respectively) are placed between the two masses and between the secondary mass and the ground. Usually viscoelastic materials are employed to sustain shear forces, but for the purpose of this benchmark it is assumed that they only counteract longitudinal forces (i.e. they work as compression elements).

![Figure 5.1: Schematic representation of a 2-DoF system configuration with two pieces of viscoelastic material ($k_1^*$ and $k_2^*$) placed between the two masses and between the secondary mass and the ground.](image)

As already reported in Section 2.3.1.1, viscoelastic materials are characterised by several nonlinearities (e.g. frequency and temperature dependencies) that are quite difficult to be investigated simultaneously. A method that has shown to give a good outcome in the description of the viscoelastic behaviour (at least in terms of frequency dependency) is the fractional derivative model [53, 54]. In particular, by defining the complex Young’s modulus $E^*$ as (see Eq. (2.8)):

$$ E^* = E_0 + \sum_{j=1}^{N} E_j \frac{(i\omega/r_j)^\theta_j}{1 + (i\omega/r_j)^\theta_j} $$

(5.1)

the viscoelastic material can be converted in a compression spring characterised by a complex stiffness $k^*$ expressed as:

$$ k^* = E^* \cdot \frac{A}{L} $$

(5.2)
where \( A \) and \( L \) are respectively the area and length of the elastomer specimen.

The 2-collinear-DoF system shown in Figure 5.1 can then be represented via the two following equations of motion:

\[
\begin{align*}
    m_1 \ddot{z}_1 + k_1^* \cdot (z_1 - z_2) &= F_{in} \\
    m_2 \ddot{z}_2 + k_1^* \cdot (z_2 - z_1) + k_2^* z_2 &= 0
\end{align*}
\] (5.3)

where the primary mass \( m_1 \) was fixed and equal to 5 kg.

The TF between the input force and the force transmitted to the ground presents two pairs of complex-conjugate poles that are characterised by limited loss factor at low frequency and increasing loss factor at higher frequencies. This system depends on three factors \((k_1^*, k_2^* \text{ and } m_2)\) and their effects on the TF were separately investigated (the temperature dependency of the viscoelastic material was omitted by considering all this simulations at a fixed temperature).

The variation of the longitudinal stiffness of the viscoelastic material can be obtained by changing the geometric features of the elastomer element. In fact, by considering only the constant modulus \( E_0 \), the static stiffness of each viscoelastic piece \((k_1 \text{ and } k_2)\) is a function of the area \( A \) and the length \( L \) of the elastomer specimen. However, it is worth noting that excessively reducing the longitudinal stiffness can bring to instability of the element (e.g. buckling load is inversely proportional to the square of the length \( L \)). In this benchmark study the instability was not considered. Figure 5.2 shows that reducing the stiffness \( k_2 \) by more than three times (while keeping the other two parameters constant) produced little improvement at high frequency whereas the first peak slightly shifted toward lower frequency. The same effect was observed with the variation of \( k_1 \). A more substantial improvement at high frequency was seen when the stiffness of both elastomer elements were varied simultaneously (see Figure 5.3). On the other hand, this simultaneous variation results also in a more-amplified first resonance peak.

Completely different was instead the behaviour of the system when only the secondary mass \( m_2 \) was modified. Figure 5.4 shows that the increase of \( m_2 \) considerably changed the frequency of the second peak while leaving the first peak basically unaltered. However, the beneficial effect on high frequency attenuation obtained by increasing the mass is clearly visible when \( m_2 \) is still a small fraction of the primary mass (less than 25\%), but becomes almost negligible for larger values of \( m_2 \). To conclude this benchmark study, the system analysed in this section gave a good idea of the performance limitations of a 2-collinear-DoF system with two elastomer elements. In particular, it was clear that the amplification at the first resonance peak could not be reduced (due to limited loss factor) and, in order to have
5.1 Benchmark Study

Figure 5.2: Force TF of the benchmark study varying the static longitudinal stiffness of the top viscoelastic piece ($k_2$) while keeping all the other parameters constant.

Figure 5.3: Force TF of the benchmark study varying simultaneously the static longitudinal stiffness of both viscoelastic pieces ($k_1 = k_2$) while keeping all the other parameters constant.
5.1 Benchmark Study

**Figure 5.4:** Force TF of the benchmark study varying the secondary mass ($m_2$) while keeping all the other parameters constant.

**Figure 5.5:** Final choice of the system parameters for the benchmark study. This force TF is characterised by an attenuation of $-40$ dB at 100 Hz and a final roll-off slope that starts at $-80$ dB/dec and slowly decreases due to the typical increase of the viscoelastic material stiffness with the frequency.
5.2 Concept

A better mitigation at high frequency while guaranteeing the system stability, the variation of the secondary mass resulted the most beneficial. However, to have a more consistent comparison with the isolator proposed in this thesis, \( m_2 \) was limited to 0.2 kg (similar to the mass of the magnetic stack) and the static stiffness \( k_1 \) and \( k_2 \) were chosen so that the TF showed an attenuation of \(-40\) dB at 100 Hz (see Figure 5.5). As already mentioned in Section 2.3.1.1, viscoelastic materials are characterised by a stiffness that increases with the frequency and this explains the roll-off slope after the second resonance that starts by being \(-80\) dB/dec and gradually decreases as the frequency increases.

It is important to note that the nonlinearities related to temperature and displacement amplitude were not considered in this study. Therefore, it is evident that this system although using the simplest, cheapest damping system (i.e. viscoelastic materials) would require complex modelling to represent its behaviour at every possible environmental and operating conditions.

5.2 Concept

The idea behind the concept of the 2-collinear-DoF system raised with the realisation that a magnet presents two poles that can be used independently \([178, 179]\). In fact, the bipolarity of magnets can be exploited to create a two-level damping (i.e. by using two separate EMSDs) that considerably enhances the isolation performance without the need to increase the secondary mass. This goal has been achieved by analysing two key aspects of this system such as the magnetic assembly and the shunt circuit.

The magnetic assembly strongly affects the damping force \( F_d \) through the electro-mechanical transducer coefficient \( K_d \) which depends on the radial component of the magnetic field, as shown in Eq. (3.7). The configuration considered for this study is the same as the one reported in the Section 3.3 which was obtained through a trade-off conducted via COMSOL Multiphysics. The chosen magnetic assembly (see Figure 5.6) is characterised by two central regions where the radial component of the magnetic field is considerably enhanced due to the magnets facing each other with opposite polarity. This configuration presents also two extra poles at the stack’s extremities that can be exploited by two other coils. In the 2-collinear-DoF system analysed in this chapter, each electromagnet is made of two coils connected together, as it can be observed from Figure 5.6a. These coils need to have opposite winding orientation (i.e. one is winded clockwise and the other anticlockwise) in order to take into account the different direction of the magnetic field at the two magnetic poles.
Figure 5.6: Magnetic assembly analysis carried out through COMSOL Multiphysics using a 2D-axisymmetric formulation. (a) disposition of the magnets and yokes with relative magnetic polarization; (b) magnetic flux density.

Figure 5.7: 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.

The micro-vibration isolation of the proposed isolator can also be improved by modifying the electrical features of the shunt circuit. As already stated in Section 3.4, reducing the overall resistance has a beneficial effect on the amplification at the resonance peaks, whereas reducing the ratio $R/L$ produces the shift of the extra pole toward lower frequencies which results in a better high frequency isolation performance. By using a negative resistance converter in the shunt circuit it is possible to achieve both tasks. This simple circuit (made of
one op-amp and three resistors) produces an equivalent resistance equal to:

\[ R_s = -R_x \left( \frac{R_z}{R_y} \right). \]  \hspace{1cm} (5.4)

which simplifies to \( R_s = -R_x \) if \( R_y \) and \( R_z \) are considered equal to each other.

Although reducing the overall resistance has the effect of enhancing the damping performance by increasing the electric current, the magnitude of the negative resistance is limited by two factors. First of all, it can result in instability if the positive resistance becomes smaller than the negative one. This condition can occur due to the typical temperature fluctuations of a space mission (this aspect will be taken into account in Section 5.4). Secondly, reducing the overall resistance beyond a minimum value has the effect of increasing the amplification at the resonance peak and the overall system stiffness (see Figures 3.12 and 3.13). Therefore, to further reduce the ratio \( R/L \) and so enhance the isolation properties of an EMSD it is necessary to include an inductor in the shunt circuit (see Figure 5.7). In particular, the inductor should be characterised by having a high ratio of inductance over internal resistance, and for this reason iron-core inductors are usually preferable (they can present values of inductance in the order of mH and inherent resistances in the order of tenths of a Ohm).
5.3 Analytical Model

The analytical model presented in this chapter is based on the system of equations presented in Section 3.2. In particular, through the assumptions of constant magnetic field seen by the conductive material (due to the relative displacement being in the order of magnitude of tenths of a millimetre for a micro-vibration load case), linear trend of the magnetic field along the radial axis inside the small cross section of the electromagnet, and relative motion between the conductor and the magnet to be only along the z-axis, the equations related to the Faraday-Lenz law and Lorentz force law can be simplified as:

\[ E_0 = K_d v_z \]  \hspace{1cm} (5.5)

\[ \vec{F}_d = -K_d \vec{I} \hat{z} \]  \hspace{1cm} (5.6)

where the electro-mechanical coefficient \( K_d \) is defined as:

\[ K_d = \frac{2 \pi n_t r_{avg} \bar{B}_r}{5.7} \]

These two equations can be integrated with the equation of motion and the Kirchhoff’s voltage law to fully model the behaviour of an EMSD. The system showed in Figure 5.9 presents a two-level damping made of two EMSDs and it can be represented via a fully-coupled system made of eight equations. The system can be written as:

\[ \text{Equations of motion and Kirchhoff’s voltage law} \]

Figure 5.9: Schematic representation of the 2-collinear-DoF model
5.3 Analytical Model

\[
\begin{align*}
\begin{cases}
m_1 \ddot{z}_1 + k_1 (z_1 - z_2) = F_1 + F_{in} \\
m_2 \ddot{z}_2 + k_1 (z_2 - z_1) + k_2 \ddot{z}_2 = F_2 - F_1 \\
V_1 = K_d (\dot{z}_1 - \dot{z}_2) \\
V_2 = K_d \dot{z}_2 \\
L_1 \dot{I}_1 + R_1 I_1 = E_1 \\
L_2 \dot{I}_2 + R_2 I_2 = E_2 \\
F_1 = -K_d I_1 \\
F_2 = -K_d I_2
\end{cases}
\end{align*}
\]

where \( E_1 \) (induced voltage), \( F_1 \) (Lorentz force), \( L_1 \) and \( R_1 \) are the features related to EMSD\(_1\), whereas \( E_2, F_2, L_2 \) and \( R_2 \) are related to EMSD\(_2\).

Through the assumptions of micro-vibrations (i.e. magnetic field seen by each point in space is assumed practically constant due to small displacements), this system can be linearised via a state-space representation. The state vector consists of six state variables: displacements \((z_1, z_2)\), velocities \( (\dot{z}_1, \dot{z}_2)\), and circuit currents \((I_1, I_2)\). The state space model can be written as:

\[
\begin{bmatrix}
\dot{z}_1 \\
\dot{z}_2 \\
\ddot{z}_1 \\
\ddot{z}_2 \\
\dot{I}_1 \\
\dot{I}_2
\end{bmatrix}
= \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
K_e & 0 & 0 & 0 & 0 & 0 \\
0 & K_e & 0 & 0 & 0 & 0 \\
-C_L & 0 & 0 & 0 & 0 & 0 \\
0 & -C_L & 0 & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
z_1 \\
z_2 \\
\dot{z}_1 \\
\dot{z}_2 \\
I_1 \\
I_2
\end{bmatrix}
+ \begin{bmatrix}
0 \\
0 \\
1/m_1 \\
0 \\
0 \\
0
\end{bmatrix} F_{in}
\]

where \( 0_{2,2} \) and \( \mathcal{I}_2 \) are respectively the 2x2 null and identity matrices. \( \mathcal{K} \) and \( C_m \) represent the stiffness and damping matrices in the equation of motion and are defined as:

\[
\mathcal{K} = \begin{bmatrix}
-k_1/m_1 & k_1/m_1 \\
k_1/m_2 & -(k_1 + k_2)/m_2
\end{bmatrix}
\]

\[
C_m = \begin{bmatrix}
-K_d/m_1 & 0 \\
K_d/m_2 & -K_d/m_2
\end{bmatrix}
\]

whereas \( C_L \) and \( \mathcal{R} \) are representative of the induced voltages and the resistances that are associated with the two electric circuits and are defined as:

\[
C_L = \begin{bmatrix}
K_d/L_1 & -K_d/L_1 \\
0 & K_d/L_2
\end{bmatrix}
\]
5.3 Analytical Model

\[ \mathcal{R} = \begin{bmatrix} -R_1/L_1 & 0 \\ 0 & -R_2/L_2 \end{bmatrix} \]  \hspace{1cm} (5.13)

In order to assess the damping performance, it is necessary to evaluate the TF between the input force \( F_{in} \) and the force transmitted to the satellite structure. The latter can be obtained by setting the output vector as:

\[ Y = \begin{bmatrix} 0 & k_2 & 0 & 0 & K_d \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ \dot{z}_1 \\ \dot{z}_2 \\ I_1 \\ I_2 \end{bmatrix} + \begin{bmatrix} 0 \end{bmatrix} F_{in} \]  \hspace{1cm} (5.14)

and consequently the force TF can be evaluated after converting this single-input-single-output system to the frequency domain.

\[ \text{Figure 5.10: Comparison between TFs associated with three different 2-DoF systems: mass-spring system characterised by two couples of complex-conjugate poles (solid line), mass-spring-dashpot system characterised by two additional real zeros (dashed line) and mass-spring-EMSD system that introduces two extra real poles thus restating the final decay rate to } -80 \text{ dB/dec (dotted line)} \]

In order to better understand how the introduction of two EMSDs affects the system response, an example with the TFs associated with three different system configurations is shown in Figure 5.10. A typical TF for a two-masses-two-springs system is characterised by two
resonance peaks (i.e. two couples of complex-conjugate poles) and a final roll-off slope of $-80$ dB/dec (solid line). Adding viscous dampers in parallel to the springs (dashed line) would help reducing the force amplification at the resonance peaks, but it would also reduce the high-frequency attenuation due to the introduction of two real zeros in the TF that brings the slope to $-40$ dB/dec (i.e. each real zero increases the decay rate by $+20$ dB/dec). The two-level damping made of two separate EMSDs that is proposed in this thesis is capable of combining the advantages of viscous dampers (i.e. considerable reduction of the resonance peaks by decreasing the overall circuit resistance) and of 2-DoF mass-spring systems, since the relatively low resistance-over-inductance ratio introduces a real pole per EMSD that restores the final roll-off slope to $-80$ dB/dec (dotted line).

### 5.4 Parametric Trade-off

The proposed isolator was designed to provide good micro-vibration mitigation within the operating temperature range without requiring any active control. Similarly to what reported in Section 3.6, the suspended mass $m_1$ was chosen to be 5 kg which corresponds approximately to the mass of two 100SP-O reaction wheels used by SSTL. Reaction wheels work usually within a temperature range that spans from $-20\, ^\circ\mathrm{C}$ to $+50\, ^\circ\mathrm{C}$ [63, 64]. A parametric trade-off has been conducted in order to meet the following goals throughout the whole temperature range of interest:

- 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 all the operating conditions. An EMSD would become unstable if the negative resistance is greater in magnitude than the positive one ($R_e + R_l$) and this could happen at the lowest temperature range limit. In fact, at such temperature the electromagnet and the inductor reach their minimum values, whereas the three space-qualified, surface-mount resistors forming the negative resistance circuit can be considered constant over a wide temperature range (tolerances down to $0.005\%$ and temperature coefficients of $1 \cdot 10^{-6} \, ^\circ\mathrm{C}^{-1}$). In order to prevent the system from becoming unstable, the total resistance at the temperature of $-20\, ^\circ\mathrm{C}$ was set to a minimum value $0.1 \, \Omega$. 

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Another important aspect that needs to be taken into account is the understanding of the role of the inductance in terms of system dynamic response. By increasing the inductance it is possible to produce a similar effect that can be achieved by increasing the secondary mass $m_2$ (having assumed that all the other parameters, e.g. $\tilde{B}_r$ and suspended mass $m_1$, remain constant). Figure 5.11 shows for example that changing the inductances $L_1$ and $L_2$ from 1 mH to 5 mH (red curve) would generate a high-frequency attenuation that is comparable with the system where the mass $m_2$ is set to 3.5 kg (purple curve). For this reason, in this study the secondary mass $m_2$ was kept at 0.18 kg and was not included in the parametric trade-off.

![Figure 5.11: Effect of changing the inductance and the secondary mass on the system damping performance](image)

The trade-off was carried out by analysing the system at three different temperatures: $-20$ °C, $+50$ °C and the range mid point, $+15$ °C. The analysis included few assumptions that simplified the integration between the mechanical, electromagnetic and thermal domains. First of all, it was assumed that all the components of the electric circuit have the same temperature of the surrounding environment (i.e. the small amount of thermal energy dissipated by the coil in a typical micro-vibration load case was removed from it). In this way it was possible to study the system in steady state conditions, where all the temperature-sensitive parameters are determined by the initial environmental temperature and are constant throughout the analysis. Secondly, it was assumed that the two springs were identical.
(\(k_1 = k_2\)), as well as the electrical properties of the two electromagnetic circuits (\(R_1 = R_2\) and \(L_1 = L_2\)). The final choice of the parameter set is reported in Tab. 5.1.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suspended mass, (m_1) (kg)</td>
<td>5</td>
</tr>
<tr>
<td>Magnet mass, (m_2) (kg)</td>
<td>0.18</td>
</tr>
<tr>
<td>Spring Stiffness, (k_1 = k_2) (N/m)</td>
<td>2000</td>
</tr>
<tr>
<td>Coil Inductance, (L_1 = L_2) (mH)</td>
<td>8</td>
</tr>
<tr>
<td>Shunt Resistance, (R_s) ((\Omega))</td>
<td>−3.0</td>
</tr>
<tr>
<td>Inductor Resistance, (R_i) ((\Omega))</td>
<td>0.35, 0.40, 0.45</td>
</tr>
<tr>
<td>Coil Resistance, (R_e) ((\Omega))</td>
<td>2.85, 3.25, 3.65</td>
</tr>
<tr>
<td>E-M Transducer Coefficient, (K_d) (N/A)</td>
<td>11.28, 10.79, 10.29</td>
</tr>
</tbody>
</table>

**Table 5.1:** Final choice of the parameter set for the 2-collinear-DoF system obtained through a trade-off

**Figure 5.12:** 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.

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5.4 Parametric Trade-off

![Graph](image)

**Figure 5.13:** Comparison between the TFs related to the two circuits with respect to the force TF. From the plot, it can be seen how the top circuit becomes more relevant at higher frequencies.

![Graph](image)

**Figure 5.14:** 2D representation of the variation of the system eigenvalues with respect to the overall resistance (i.e. system temperature). The trend of the variations of the six eigenvalues can be seen going from the green squares to the red ones.
A comparison of the force TFs between the system without damping (i.e. the shunt circuits were disconnected and the electromagnets were in open-circuit conditions) and the system with the two EMSDs is shown in Figure 5.12. It can be observed that the use of the EMSDs produced the desired reduction of the two resonance peaks and the final roll-off slope of $-80$ dB/dec. The maximum amplification was kept below 6 dB in all three cases studied, and the corner frequency, although moving slightly within the temperature range, did not exceed 10 Hz. Also, the force attenuation at 100 Hz was greater than 40 dB. Therefore, the three aforementioned goals were all met in the trade-off analysis. It is noted that the shift of the first resonance peak within the temperature range of interest is in agreement to what already reported in Section 3.6.

Another interesting aspect is the contribution of the two circuit to the overall isolation performance. Figure 5.13 shows the comparison between the TFs related to the two circuits (through the two currents, $I_1$ and $I_2$) with respect to the force TF for the case at 15 °C. From the plot, it can be seen that the two circuits contribute equally at low frequency, but as the frequency increases, their behaviour starts diverging. In particular, although both decreasing after the second peak, the current in the top coil has a roll-off slope of $-40$ dB/dec whereas the current in the bottom coil has the same slope as the force TF ($-80$ dB/dec). The
5.5 Experimental Validation

5.5.1 Test Setup

The laboratory test rig used for this experiment was obtained by modifying the rig used for the 1-DoF test set-up. In particular, one of the main differences was the substitution of the rigid link between the dummy mass and the magnet stack with a metal spring in order to decouple the two masses, as shown in Figure 5.16. Another difference was the decision to use PTFE instead of aluminium for the coil supports so that any undesired damping force coming from eddy currents in the supports was eliminated (the white parts in Figure 5.16b are made of PTFE). In fact, although this effect was negligible compared to the damping force produced by the EMSDs, this 2-collinear-DoF system was characterised by a second resonant mode that involved the movement of only the magnet stack. Therefore, given the small mass of the magnetic assembly even this little damping force resulted in the impossibility to observe the relative peak in the TF (i.e. the second resonant peak). By using a non-conductive material as PTFE it was possible to remove undesired damping effects that were not included in the model and just analyse the behaviour of the EMSD.

This test rig (shown in Figure 5.17a) represented a physical proof of concept of the proposed isolator and, similarly to what explained for the 1-DoF test rig, it required steel flexures in order to guarantee that the two masses had only a vertical displacement while trying to minimise displacements/rotations along the other five DoFs. Although their use was necessary
to meet the requirements for a 2-collinear-DoF system, the inclusion of the flexures produced a slight change in the analytical model of the test rig. In this experimental configuration the primary and secondary masses were further connected to the ground through the flexures that can be represented with vertical springs called respectively $k_{bf}$ and $k_{sf}$ (see Figure 5.17c). The system model reported in Eq. (5.9) can be adapted to the actual configuration by solely changing the matrix $\mathcal{K}$ as:

$$\mathcal{K} = \begin{bmatrix}
-(k_1 + 2 \cdot k_{bf})/m_1 & k_1/m_1 \\
 k_1/m_2 & -(k_1 + k_2 + 2 \cdot k_{fs})/m_2
\end{bmatrix}. \tag{5.15}$$

where $k_{bf}$ and $k_{sf}$ are considered twice because each of them represents only a pair of the four flexures supporting either the dummy mass or the magnet stack.

Figure 5.16: Main differences between the 1-DoF and 2-DoF test rig. a) 1-DoF system with rigid connection between the dummy mass and the magnet stack and coil support made of aluminium. b) second spring to connect dummy mass and magnet and the coil support is now made of PTFE.

A model of the test rig was evaluated through a FEM analysis on Nastran. In comparison to the model reported in Section 4.1, the spring connector (CBUSH) placed between the dummy mass and the magnet stack was modified from having infinite stiffness (i.e. $10^7$ N/m) to the desired value of stiffness reported in Tab. 3.2 (2000 N/m). This analysis showed that the extra load paths introduced with the flexures and transmitted to the KT considerably modified the force TF altering the expected final roll-off slope of $-80$ dB/dec, as it can be observed from Figure 5.18. Moreover, disturbances coming from the flexures’ secondary modes were recorded by the KT from about 200 Hz, thus affecting the force TF.
5.5 Experimental Validation

Figure 5.17: Experimental test rig. In particular, $K_{bf}$ and $K_{sf}$ are the equivalent stiffnesses of the flexures connected respectively to the top mass $m_1$ and the magnet mass $m_2$, $k_1$ and $k_2$ represent the two springs that connect $m_2$ respectively to $m_1$ and to the ground, MFS is the multiaxial sensor placed on top of the KT. (a) test rig mounted on the KT; (b) exploded view of the CAD of the test rig; (c) Schematic representation of the test rig
5.5 Experimental Validation

In order to tackle both issues, a Multiaxial Force Sensor (MFS) was inserted on top of the KT and underneath the support that holds the bottom electromagnet and the spring $k_2$. The force TF between the input force and the force recorded on the MFS has been evaluated with Nastran (see Figure 5.18). It is characterised by the elimination of the high-frequency modes associated with the flexures and by almost entirely overlapping with the TF of the theoretical 2-collinear-DoF isolator described in Eq. (5.9). Therefore, with the inclusion of the MFS in the test rig it has been possible to obtain an almost accurate assessment of the isolation performance of the proposed isolator. The main differences between the two curves can be observed at low frequency where the TF associated with the MFS starts below 0 dB because part of the low-frequency force is conveyed to the ground through the rig’s walls. Also, the first resonance frequency slightly shifts toward higher frequencies due to the extra stiffness $k_{bf}$ and $k_{sf}$ produced by the flexures. It is noted that a small amount of viscous damping (less than 6 % of the critical damping) has been added to both Nastran and Matlab models in order to reflect the inherent mechanical damping that was observed in previous studies on a similar version of this test rig [169]. This damping naturally occurs in a mechanical assembly and can be produced by several factors such as micro-friction between adjacent components, air resistance and hysteresis damping in the steel flexures.

![Figure 5.18: Comparison of the TFs between the analytical solution of the ideal isolator (evaluated on Matlab) and the FEM results obtained through Nastran and relative to the KT and the MFS](image)

Another fundamental aspect of the proposed isolator is the low system mass compared to the suspended mass. The magnetic assembly weighs 180 g which represents less than 4% of the
total suspended mass. The dummy mass of 5 kg is obtained through the combination of steel plates bolted together and aluminium/plastic harnesses. In order to simulate microgravity, the primary mass was offloaded using elastic cords characterised by a sub-hertz resonance frequency.

Two different tests were performed on this test rig. First, a sine sweep was applied to the dummy mass through a suspended mini-shaker that was connected to its centre. The scope of this test was to verify the system behaviour at a wide frequency range. In fact, the mini-shaker is able to introduce a sufficiently high input force throughout the frequency range of interest that allows the output signal to mostly overcome the background noise and reach a maximum attenuation level of about -80 dB. The second test involved the use of a real reaction wheel provided by SSTL. A spinning reaction wheel produces a broadband disturbance due to microscopic defects that are present inside the device (e.g. flywheel unbalance or cage disturbance). This test aimed at demonstrating the capability of the EMSD to respond to a broadband input signal while maintaining its linear behaviour.
5.5 Experimental Validation

5.5.2 Noise source: mini-shaker

The KT is placed on top of an isolation system (a suspension system that acts as a low pass filter) characterised by an 8-Hz resonance mode that required the tests to be divided into two different sine sweeps: a low-frequency one with the KT locked to the ground, and a high-frequency one with the isolation system in operation. The mini-shaker was controlled through a force feedback loop to generate first a 0.1-N-amplitude sinusoidal input force sweeping from 1 Hz to 20 Hz, and then a sinusoidal input force with amplitude linearly increasing from 1 N at 10 Hz to 5 N at 400 Hz. The input force of the first sine sweep was set to be 0.1 N so as to prevent the flexures from operating in the nonlinear region due to relatively large displacements (e.g. in the order of millimetres). The lab temperature was approximately 20 °C and the parametric set in the analytical model was tuned to account for this temperature. The comparison between the experimental results and the analytical data is shown in Figure 5.20. The background noise represents the environmental disturbances measured by MFS and it was recorded at the beginning of the test campaign. From Figure 5.20a the comparison between the force TFs associated with the KT and MFS for a system without EMSD can be observed. In particular, the first two resonance modes are clearly visible, and above 30 Hz it is noted that the force TF associated with the MFS shows the expected final slope of −80 dB/dec up until it reaches the background noise level. Good correlation can also be seen for the system with EMSD (see Figure 5.20b). As already shown in Figure 5.12, the TF is characterised by the elimination of the first resonance peak, cut-off frequency below 10 Hz and (for the MFS case) roll-off slope of −80 dB/dec and attenuation greater than 40 dB at 100 Hz. The coherence between analytical data and experimental results of Figure 5.20 is 0.92 on average from 1 Hz to 150 Hz. For higher frequencies, the coherence reduces due to the background noise but also secondary modes not included in the analytical model.

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. (5.14) as follows:
5.5 Experimental Validation

\[ Y = \begin{bmatrix} Y_{v1} \\ Y_{v2} \\ Y_{I1} \\ Y_{I2} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ \dot{z}_1 \\ \dot{z}_2 \\ I_1 \\ I_2 \end{bmatrix} + \begin{bmatrix} 0 \end{bmatrix} F_{in} \quad (5.16) \]

where \( Y_{v1} \) and \( Y_{v2} \) represent respectively the velocity of the primary mass and the magnetic assembly while \( Y_{I1} \) and \( Y_{I2} \) are the induced currents in the top and bottom electromagnets. The velocities were obtained after integrating the acceleration signals measured through accelerometers placed on top of the dummy mass and on the sides of the magnetic assembly. The currents were evaluated using an oscilloscope probe connected to the ends of each electromagnet. The comparison of these analytically-obtained TFs with the experimental data is shown respectively in Figures 5.21 and 5.22. The use of different kind of sensors that has allowed to analyse different aspects of the damping system and the good correlation that can be observed among all the presented TFs are further confirmations of the accuracy of the analytical model and also the correct functioning of the negative-resistance circuits.

In terms of power required by the two EMSDs, by observing Figure 5.22 it can be observed that the frequency corresponding to the maximum current drawn by the electromagnets was about 4 Hz. In particular, the magnitude of the two currents \( I_1 \) and \( I_2 \) was approximately \(-23\) dB with respect to the input force. Therefore, if an input force of 0.5 N was considered (equal to the input force used for the 1-DoF test campaign reported in Chapter 4), then the current flowing in each electromagnet would be 35 mA (obtained as \(-23\) dB of 0.5 N), and using Eq. (3.19) the average power required by the single EMSD at 4 Hz would be:

\[ P_{avg} = (V^+ - V^-) \cdot (I_q + I_{rms}) = 10V \cdot (3.8mA + 0.707 \cdot 35mA) = 0.28W \quad (5.17) \]

which would correspond to an overall average power required by the two EMSDs (0.56 W) that is comparable to the required power for the 1-DoF system reported in Eq. (4.4) (0.53 W). This result is extremely interesting because it shows that the 2-collinear-DoF system still requires a marginal amount of power to operate when compared with active isolation methods. It is important to note that this outcome was made possible by the use of the two extra coils to exploit the magnetic field produced at the magnet stack’s ends and so by increasing the coefficient \( K_d \) due to the increase of coil turns.
5.5 Experimental Validation

Figure 5.20: Comparison between the test results and the analytical model for the 2-collinear-DoF system. The solid lines represent the test results, whereas the dashed lines are relative to the analytical model. The background noise is also included. (a) system without EMSD; (b) system with EMSD.
5.5 Experimental Validation

**Figure 5.21:** TF between the input force and the mass velocity. The system considered is without EMSDs. (a) primary mass $m_1$; (b) magnet mass $m_2$

**Figure 5.22:** TF between the input force and the induced current in the electromagnets. The system considered is with EMSDs. (a) top electromagnet; (b) bottom electromagnet
5.5 Experimental Validation

5.5.3 Noise source: 10SP reaction wheel

The reaction wheel used for this test belongs to the 10SP series made by SSTL. This reaction wheel is smaller than a typical wheel from the 100SP series but it was still chosen due to availability issues at the time of the tests. The 10SP reaction wheel has a mass of approximately 1 kg (against 2.5 kg of 100SP reaction wheel) and in order to compensate for the added mass one of the plates forming the dummy mass was changed from being in steel to being in aluminium. The overall supported mass \( m_1 \) was about 5.3 kg. Four force transducers were placed between the reaction wheel and the dummy mass (see Figure 5.19) and the total vertical input force generated by the wheel was measured by summing the signals of the four transducers. The rotational speed of the wheel was set to 1500 rpm. As shown in Figure 5.23a, the input force ranges from 0.1 mN at low frequency to about 5 mN at high frequency. The two input forces (for the case with and without EMSD) should perfectly overlap since the wheel speed was the same. However, the small discrepancy that can be observed between the two curves might be due to speed errors introduced by the controller and time of recording.

The force TFs between the vertical disturbance generated by the reaction wheel and the vertical force measured via the MFS are reported in Figure 5.23b. It is noted that the disturbance force generated by the reaction wheel is at least a couple of orders of magnitude smaller than the input force applied through the mini-shaker. For this reason, it can be observed that the two TFs in Figure 5.23b are characterised by a maximum attenuation of only 50 dB before hitting the background noise.

Another important aspect that needs further explanation is the behaviour of the two TFs at low frequency. From a theoretical point of view, the system without EMSDs should show a first-resonance peak characterised by higher amplification with respect to the case with the dampers ON. It is believed that the behaviour shown in Figure 5.23b could be produced by the fact that a reaction wheel generates disturbances in all 6 DoFs (and the vertical one is often the smallest) that could have introduced other displacements and rotations, thus preventing the test rig to work nominally. In other words, although the flexures should have prevented all other displacements and rotations, the high lateral forces generated by the reaction wheel could have triggered undesired motions that affected the test results.

Nevertheless, both curves show overall a good agreement with the results reported in the previous section thus demonstrating that the negative-resistance circuit is capable of working with broadband signals without showing any nonlinear behaviour.
Figure 5.23: Experimental results for the test with the reaction wheel. (a) Input force produced by the reaction wheel along the $z$ axis and measured by summing the signal of the four force transducers placed between the reaction wheel and the dummy mass; (b) TF between the input force and the vertical force measured with the MFS
5.6 Isolation Performance Comparison

In this section, the 2-collinear-DoF isolator proposed in this thesis is compared to other damping solutions found in literature. In terms of damping performance, Figure 5.24 shows the force TFs of four different systems: 1-DoF system using a viscoelastic material presented in [168], 1-DoF system with EMSD presented in [169], 2-collinear-DoF benchmark system presented in Section 5.1, and the 2-collinear-DoF system with embedded EMSDs (at the temperature of 15 °C). At low frequency, the proposed isolator is capable of eliminating the resonance peak that is characteristic of a viscoelastic-material isolator, whereas at high frequency it presents a final slope that is 40 dB/dec greater than the one showed by the 1-DoF system with EMSD. Also in comparison to the benchmark study, the proposed isolator maintains the final roll-off slope of −80 dB/dec (before hitting the background noise) which confirms the absence of nonlinearities, contrarily to what has been demonstrated for viscoelastic materials in terms of stiffness.

![Figure 5.24: Comparison of the force TFs between four different damping systems: 1-DoF system using viscoelastic material presented in [168], 1-DoF system with EMSD presented in [169], 2-collinear-DoF benchmark system presented in Section 5.1, and the 2-collinear-DoF system proposed in this thesis.](image)

With an eye on Section 2.4 where some of the most advanced strut isolators were presented, it is clear that obtaining a disturbance attenuation of at least 40 dB for frequencies beyond 100 Hz is still quite challenging, especially if combined with considerable reduction of the
5.6 Isolation Performance Comparison

first resonance peak, but this is instead achievable with the isolator proposed in this thesis. Furthermore, the 2-collinear-DoF isolator with embedded EMSDs offers also other important advantages in comparison to the hybrid isolators presented in Section 2.4. The control methods used in the active part of these devices are typically characterised by complex, cumbersome electronics that considerably affects the overall mass of the isolation system (e.g. the hexapod presented in [160] has an electronics mass of 9 kg which corresponds to 72% of the overall hexapod mass) and require a significant amount of power to drive actuators and sensors (e.g. the single strut in [132] needs a minimum of 15 W to operate). Contrarily, the proposed 2-collinear-DoF isolator uses small circuit boards composed by few electric parts that require less than 0.7 W to produce the high damping performance reported in this report.

Another fundamental feature of the proposed isolator is the great versatility of the system response that can be considerably modified by solely changing the properties of the shunt circuits. Figure 5.25 shows the device TF as a function of the inductance and total resistance. Without the need to intervene on the isolator design, the variation of the inductance $L_i$ or the negative resistance $R_s$ produce the shift of the second peak thus adjusting for the attenuation performance required for a specific space mission. Also, this versatility makes the strut able to adapt its isolation performance to different masses that might be placed on top of the hexapod, thus allowing for a wide range of device masses to be isolated.

![Figure 5.25: Different configurations of the shunt circuit components showing the variation of the force TF with respect to inductance and resistance](image)

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5.7 Summary

This chapter presented the novel 2-collinear-DoF isolator made with two embedded EMSDs. A benchmark study of a 2-DoF system with viscoelastic materials was first presented in order to have a reference system for the following analysis. In particular, the use of elastomers in a 2-stage system produced a TF characterised by the amplification of the first resonance peak by more than 20 dB and a final roll-off slope that started at $-80$ dB/dec and decreased due to the nonlinearity of the material stiffness. It followed the description of the novel concept for a 2-collinear-DoF isolator that exploited the bipolarity of a magnet to create a 2-level damping system. The analytical model was reported and a parametric trade-off aimed at meeting the mitigation requirements was performed. The final choice of the isolator parameters resulted in a TF that was characterised, within the whole temperature range of interest, by the elimination of the first resonance peak, the corner frequency set below 10 Hz, an attenuation of more than 40 dB from 100 Hz and a final slope of $-80$ dB/dec. The proposed isolator was verified through an experimental campaign after having analysed the test rig and slightly modified the analytical model.
This chapter presents other aspects of the proposed 2-collinear-DoF isolator that were taken into account during this research. The concept design of the strut is first presented. In this section, particular focus is given to two key aspects of the design such as the realisation of the spring system and the magnetic shielding. It follows an overview of five main hexapod platforms that have been found in literature and they are compared with the concept design of a hexapod made of six of the aforementioned struts. An analysis of the temperature control is then reported and two methods (one active and one passive) to reduce the temperature dependency of the isolator are proposed. Finally, the capability of the 2-collinear-DoF isolator to produce a static force while isolating micro-vibration disturbances is presented.

6.1 Strut Concept Design

The previous chapter has demonstrated that a 2-collinear-DoF system can easily be modelled analytically and through an extensive test campaign the performance of the proof of concept was also corroborated experimentally. However, scaling down from the cumbersome test rig (overall weight more than 25 kg) to a compact device that guarantees the right stiffness while being light enough for a space application proved to be extremely challenging.

The first concern was on the realisation of the two springs shown in the schematic model reported in Figure 5.9. This device would work as modelled only as long as the two DoFs are along the same longitudinal axis while all the other DoFs are prevented. In order to obtain such a configuration, thin steel flexures with circular shape (resulting in an isotropic behaviour) were thoroughly investigated. Figure 6.1 shows six different configurations that
have been studied. For comparison purposes, the diameter was fixed and the out-of-plane stiffness (i.e. axial stiffness) was set to be $430 \pm 5\% \text{ N/m}$. The outer edge was considered clamped. Tab. 6.1 reports the main results of this trade-off analysis in which radial, bending and torsional stiffness are also included (description of the analysis run in Nastran is reported in Appendix A). The configuration $f$ from Figure 6.1 was finally chosen because it presented good values both of the bending and torsional stiffness. Moreover, this configuration was less affected by the nonlinearity of the stiffness compared for instance with the first three configurations. In fact, by applying a 0.2-mm, out-of-plane displacement at the centre of the flexures $a$ and $f$, the former presented an axial stiffness of $778 \text{ N/m} \ (+79\%)$ whereas the latter showed an axial stiffness of $462 \text{ N/m} \ (only \ +7\%)$. This property will allow the strut for more flexibility since the requirement of perfect alignment of all the flexures will be less strict. The bending stiffness, although small for the single flexure, can be considerably increased by having multiple flexures connected at different points along the magnet stack. In this configuration and by considering small rotations, the overall bending stiffness would be mainly driven by the radial stiffness as shown in Figure 6.2.

![Figure 6.1](image)

**Figure 6.1**: Different configurations of flexures that have been studied

Another aspect that was taken into account was the shielding of the magnetic field. Dealing with such strong magnets could result in a not-negligible surrounding magnetic field that could negatively affect the neighbouring electronic devices. For this reason, the cylindrical
### Table 6.1: Mechanical properties of six different flexure configurations

<table>
<thead>
<tr>
<th>Flexure ID</th>
<th>Thickness (mm)</th>
<th>Axial Stiffness (N/m)</th>
<th>Radial Stiffness (Nm·rad)</th>
<th>Bending Stiffness (Nm·rad)</th>
<th>Torsional Stiffness (Nm·rad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>0.17</td>
<td>434</td>
<td>6.89 \cdot 10^6</td>
<td>0.110</td>
<td>58.3</td>
</tr>
<tr>
<td>b</td>
<td>0.16</td>
<td>427</td>
<td>6.53 \cdot 10^6</td>
<td>0.105</td>
<td>71.9</td>
</tr>
<tr>
<td>c</td>
<td>0.15</td>
<td>443</td>
<td>6.80 \cdot 10^6</td>
<td>0.097</td>
<td>82.6</td>
</tr>
<tr>
<td>d</td>
<td>0.14</td>
<td>440</td>
<td>4.55 \cdot 10^6</td>
<td>0.115</td>
<td>87.7</td>
</tr>
<tr>
<td>e</td>
<td>0.12</td>
<td>430</td>
<td>4.20 \cdot 10^6</td>
<td>0.090</td>
<td>805</td>
</tr>
<tr>
<td>f</td>
<td>0.15</td>
<td>432</td>
<td>4.23 \cdot 10^6</td>
<td>0.125</td>
<td>1055</td>
</tr>
</tbody>
</table>

**Figure 6.2:** Comparison between two configurations in which a single or a double flexure support a mass. a) The single flexure counteracts a bending moment through its bending stiffness; b) for small rotations it can be assumed that the two flexures react to a bending moment via radial forces, thus making the radial stiffness the dominant factor.
6.1 Strut Concept Design

Figure 6.3: Magnetic flux density of the magnetic stack with austenitic steel cases

case supporting the coils was chosen to be made in austenitic steel (i.e. it has good magnetic properties, contrarily to a martensitic steel that is basically not magnetic). Figure 6.3 shows the magnetic flux density of the magnetic stack surrounded by two steel cases. The introduction of the steel case, although slightly increasing the strut mass, has the two advantages that it increases the radial magnetic field in the gap where the coil is (around 25\% more) while almost nullifying the magnetic field outside the case.

The prototype design of the strut can be observed in Figure 6.4. It is made of two identical halves that are connected to the magnetic stack through the steel flexures. All the supporting parts of the struts are in aluminium so to guarantee the right stiffness and limit the strut mass as much as possible. The proposed design is characterised by an overall mass of about 550 g. A modal analysis of the strut in the frequency range from 1 Hz to 500 Hz was performed on Patran/Nastran. The two ends were pinned (i.e. only the three translations were prevented). Apart from the expected mode due to the suspended magnet mass between the top and bottom flexures, the analysis showed several local modes at about 170 Hz (see Figure). This modes are due to the thin flexures and so little mass actually participates to these modes. However, further assessment will be necessary to determine the effect of these local modes to the overall performance of the strut.
Figure 6.4: Preliminary design of the 2-collinear-DoF strut prototype. a) isometric view; b) section view

Figure 6.5: Patran model of the 2-collinear-DoF strut prototype
6.2 Hexapod Configuration

When a subsystem (either a noise source or a sensitive payload) needs to be isolated in all six DoFs, one solution that is often adopted is to place such subsystem on a hexapod.

The hexapod platform is a mechanism system consisting of six linear actuators with variable length. It was first proposed in the mid 50s by V.E. Gough [180] even if it was named upon D. Stewart who used this configuration in 1965 to design a 6-DoF flight simulator for pilot training purposes [181]. The main characteristic of this platform is the cubic configuration which can be obtained by cutting a cube with two parallel planes at 45° as illustrated in Figure 6.7. This configuration offers interesting features such as the orthogonality of adjacent legs (resulting in a minimum cross-coupling among the individual legs and maximum uniformity of control effort in all directions) and platform symmetry (all the links are identical in the nominal configuration).

Hexapods can be divided into soft and hard Stewart platform depending on whether they use soft or stiff actuators. This section focuses on five hexapods that have been chosen among.
many others because they have either flown in real space missions or they are commercially available. The first two hexapods presented in this section were developed by Honeywell in collaboration with US Air Force research laboratory. They both included a D-Strut as the passive element but differed for the active part: the Vibration Isolation and Suppression System (VISS) [182] used a voice coil actuator whereas the Miniature Vibration Isolation System (MVIS) [159] used a piezoelectric stack actuator. The third hexapod that has been considered is the Satellite Ultraquiet Isolation Technologies Experiment (SUITE) [183] that was developed by the US Air Force research laboratory and CSA engineering. Each of the SUITE’s struts were made of a viscoelastic material in series with a piezoelectric stack actuator. Finally, the last two hexapod that have been reviewed are commercially available and are respectively manufactured by Physik Instruments [184] and CSA engineering [185]. It is worth noting that also the soft hexapod developed by the Université libre de Bruxelles [186] (see Figure 6.13) was initially considered along with the aforementioned five hexapods due to the interesting isolation performance obtained through parabolic flights. However, the lack of information about the system overall mass and the required external power did not allow the analysis of this hexapod at system level and therefore it was discarded.

The main features of the five hexapods under examination are reported in Table 6.2. As it can be observed, the overall system mass would become relevant in the comparative analysis of these systems if it was put in relation to the suspended mass that they can isolate. For instance, the overall masses of MVIS (including the electronics), SUITE and H-840 are almost the same, but these hexapods have ranges of suspended masses that are quite different. This causes the ratio of the overall mass (hexapod mass HM plus the electronic mass EM) over the suspended mass SM to vary considerably, swaying from 203 % for SUITE to 28 % for MVIS. Unfortunately, it was not possible to find in literature the overall mass of VISS but only the mass of the single strut. However, according to Ref. [159] MVIS struts have 91 % less mass than the struts proposed for VISS, and so one could extrapolate the VISS overall mass to be somewhere between 20 and 35 kg, which would correspond to a ratio of system mass over suspended mass between 86 and 150 %. Another common limitation of current hexapods is the need of a considerable amount of external power which is not always available on board a spacecraft. The analysed hexapods require power that ranges from 18 W to 60 W. Finally, some hexapods are also characterised by a high suspension frequency (in the region between 25 and 90 Hz) that heavily affects the low frequency attenuation and can reduce the high frequency mitigation (if compared for example with a hexapod with low suspension frequency).
All the drawbacks and limitations presented in Table 6.2 have been considered during the design phase of the hexapod made of six of the 2-collinear-DoF struts presented in the previous section. A CAD model of the hexapod prototype can be seen in Figure 6.14. Tab. 6.3 reports the main features of the proposed hexapod. In particular, the mass that this hexapod would be capable of isolating is fairly flexible (between 1 and 60 kg) and this is possible due to the great versatility that the EMSD offers. This flexibility of the suspended mass, combined with an overall mass of the hexapod that has been foreseen to be around
5 kg, would give a ratio that is 8% at its minimum. In terms of power, this system would require little power (expected less than 5 W to isolate two 100SP-o reaction wheels) due to the small amount of electronic components that are necessary for the negative resistance circuit.

![Hexapod Configuration](image)

**Figure 6.9:** Hexapod drawing of the Miniature Vibration Isolation System (MVIS) [159]

![Hexapod Configuration](image)

**Figure 6.10:** Hexapod drawing of the Satellite Ultraquiet Isolation Technologies Experiment (SUITE) [183]

Although being at an early stage, the first design of the proposed hexapod presents several advantages that would make it highly competitive with other hexapods that have been developed and commercially sold. It is clear that this isolation device has great potential not only in terms of performance but also at system level. For this reason, further investigation should be carried out in order to fully understand the pros and cons of the proposed system.
6.2 Hexapod Configuration

Figure 6.11: H-840 6-Axis Hexapod manufactured by Physik Instruments [184]

Figure 6.12: HX-V100 isolation Hexapod manufactured by CSAS Engineering [185]
6.2 Hexapod Configuration

Figure 6.13: Drawing of the soft hexapod developed by the Université libre de Bruxelles [186]

Figure 6.14: CAD model of the proposed hexapod

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<th>Research Projects</th>
<th>Commercially Available</th>
<th>Hexapod Configuration</th>
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<td>Vibration Isolation and Suppression System (VISS)</td>
<td>H-840 6-Axis Hexapod</td>
<td>HX-V100 Isolation Hexapod</td>
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<td></td>
<td>VISS</td>
<td>VISS</td>
</tr>
<tr>
<td>Satellite Ultraquiet Isolation Technologies Experiment (SUITE)</td>
<td>VISS</td>
<td>VISS</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Type of isolation</th>
<th>Hexapod mass (HM)</th>
<th>Electronic mass (EM)</th>
<th>Suspended mass (SM)</th>
<th>Best ratio (HM+EM)/SM</th>
<th>Attenuation at 100 Hz</th>
<th>System Power</th>
<th>Additional comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>D-Strut + voice coil actuator</td>
<td>1.13 kg (mass of single strut)</td>
<td>N/A</td>
<td>23 kg</td>
<td>29% (w/o including strut supporting structure)</td>
<td>−14 dB</td>
<td>21 W</td>
<td></td>
</tr>
<tr>
<td>D-Strut + piezo stack actuators</td>
<td>3.5 kg</td>
<td>9 kg</td>
<td>1−45 kg</td>
<td>28%</td>
<td>−20 dB</td>
<td>18−42 W</td>
<td></td>
</tr>
<tr>
<td>Viscoel. material + piezo stack actuators</td>
<td>12.6 kg</td>
<td>N/A</td>
<td>6.2 kg</td>
<td>20%</td>
<td>−9 dB</td>
<td>20−30 W</td>
<td>main purpose is positioning rather than isolating</td>
</tr>
<tr>
<td>DC motor + piezo ceramic actuator</td>
<td>12 kg</td>
<td>N/A</td>
<td>10−30 kg</td>
<td>40% (w/o including electronic mass)</td>
<td>N/A</td>
<td>N/A</td>
<td>suspension: 25−75 Hz</td>
</tr>
</tbody>
</table>

| | | | | | | | |

Table 6.2: hexapods
6.3 Temperature Control

The stability and performance requirements presented throughout this report have always taken into account the temperature range at which the isolator is going to operate, and both for the 1-DoF and 2-DoF systems it was demonstrated that this technology was capable of meeting those requirements without any sort of external control. However, there might be applications where a broader temperature range is needed or where it is requested that the isolator performance would be almost insensitive to any temperature variation.

This section proposes two methods (one active and one passive) that could be included in the shunt circuit to considerably reduce the temperature dependency of the proposed technology.

<table>
<thead>
<tr>
<th>Type of isolation</th>
<th>EMSD + soft spring</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hexapod mass (HM)</td>
<td>~ 5 kg</td>
</tr>
<tr>
<td>Electronic mass (EM)</td>
<td>&lt; 0.5 kg</td>
</tr>
<tr>
<td>Suspended mass (SM)</td>
<td>1 – 60 kg</td>
</tr>
<tr>
<td>Best ratio (HM+EM)/SM</td>
<td>8%</td>
</tr>
<tr>
<td>Attenuation at 100 Hz</td>
<td>&gt; –40 dB</td>
</tr>
<tr>
<td>System power</td>
<td>&lt; 5 W</td>
</tr>
</tbody>
</table>

Table 6.3: System features of the proposed hexapod made of six struts with embedded EMSDs

Figure 6.15: Active and passive method to perform a temperature control on the EMSD. a) The junction gate field-effect transistor (JFET); b) Nickel thin film linear thermistors
Both methods are based on the assumption that the trend of the variation of the copper resistivity with temperature is well known beforehand.

The first proposed methodology involves the use of an active control to modify the magnitude of the shunt resistance. One device that is capable of behaving as an almost pure ohmic resistor, under certain operating conditions, is the junction field-effect transistor (JFET) [187]. This three-terminal component can operate as a voltage-controlled resistor where the resistance between two of the terminals is controlled by a voltage potential applied to the third. The control algorithm will use thermal sensors attached to the electromagnets to determine their temperature and to assess their resistance based on the preloaded copper resistivity trend. A JFET might show some degree of nonlinearity as the input voltage approaches the device cut-off voltage. However, through a feedback neutralisation technique it is possible to reduce the signal distortion to less than 0.5% within a temperature range that can span from $-55^\circ$C to $+125^\circ$C (see Figure 6.16).

![Figure 6.16: Signal distortion produced by a junction field-effect transistor [187]. If the gate-source voltage $V_{GS}$ is increased, the controlled resistance will also increase. a) without feedback neutralisation technique; b) with feedback neutralisation technique](image)

The second proposed methodology deals with the use of thermistors. A thermistor sensor is a temperature-sensing element composed of sintered semiconductor material which exhibits a large change in resistance proportional to a small change in temperature. Thermistors can have both negative and positive temperature coefficients which means that the resistance of this component respectively decreases or increases as the temperature rises. In particular, Nickel thin film linear thermistors [188] have been considered for this application because their resistivity shows a temperature dependency that has a linear trend with a positive...
6.3 Temperature Control

The slope (see Figure 6.17). Moreover, they present a temperature coefficient that is similar to the one of the copper (the thermistor has a temperature coefficient at 25 °C of 4100 ppm/K, corresponding to 0.0041°C⁻¹, whereas the one for the copper is 0.00386°C⁻¹) that would result in the electromagnet and the thermistor to have the same resistance variation in percentage if they are exposed to the same temperature. For this reason, it is fundamental to thermally couple the copper coil and the thermistor, and given the advantage of the small dimensions of the considered thermistors (these components are surface mounted), the latter could be glued with thermal paste to the electromagnet. The thermistor would replace $R_x$ in Figure 5.7.

One of the drawbacks of these thermistors is the high value of inherent resistance. In fact, although Nickel thin film thermistors were found to be characterised by the smallest amount of resistance (100 Ω at 25 °C) compared to other commercially available thermistors, this value is almost a couple of order of magnitude greater than a typical value that is required in a shunt circuit (for instance, in Chapter 5 the shunt resistance was set to $-3 \Omega$). One possible way to tackle this issue is by connecting multiple thermistors in parallel. This would cause the resistance to be one $n$-th of the original 100 Ω (where $n$ is the number of thermistors connected in parallel) and it would have also the advantage of averaging the coil temperature since it is measured in different points, thus resulting in a more accurate compensation of the electromagnet resistance variation.

If the parallelisation of thermistors was still not enough (due for example to the large number of components that are necessary to obtain the desired value of shunt resistance), the exploitation of the peculiar features of the negative impedance converter could be used to further reduce the shunt resistance. So far in this report, the negative resistance was obtained by tuning the value of $R_x$ in the equation:

$$R_s = -R_x \left( \frac{R_z}{R_y} \right).$$  \hspace{1cm} (6.1)

while considering $R_z$ and $R_y$ equal to each other. However, $R_s$ can be considerably modified by choosing different values for $R_y$ and $R_z$. For instance, if four of the aforementioned thermistors were placed in parallel, the overall resistance would be 25 Ω, and by considering a target resistance of 3 Ω, the shunt resistance would be 8.33 times bigger than the one requested. In this case, $R_z$ and $R_y$ could be chosen among commercially-available components to be respectively 8.2 kΩ and 1 kΩ, thus producing a final shunt resistance of approximately 3.05 Ω. Therefore, the proposed passive methodology is capable of producing the required shunt resistance with a temperature coefficient of resistance that is comparable with the
6.4 Static Compensation and Steering Capability

one of the copper coil. If the thermistors are properly coupled with the electromagnets, the shunt resistance should be able to follow the variation of the coil resistance and produce an almost-temperature-independent performance behaviour.

Figure 6.17: Ratio of variation of resistance with comparison to the one at 25 °C for a Nickel thin film linear thermistor [188]

6.4 Static Compensation & Steering Capability

An EMSD, based on the same principle of how voice coils work, could be used as an actuator if external power is actively injected in the system. In fact, if DC voltage is applied to the terminals of the coil, the overall effect is the production of a static force along the longitudinal axis that can stretch or shorten the isolator (either with a tension or compression force). This section is going to demonstrate that an EMSD could function simultaneously as actuator and isolator without these two features interfering with each other. This property could be extremely beneficial both at the noise source side and at the sensitive payload side. In particular, if a noise source placed on top of the isolator produced a static force (e.g. for attitude control purposes) that caused the strut to lose its neutral alignment, this static force could be counteracted by feeding direct current (with opposite direction) to the two coils. As it can be observed from Figure 6.18a, the top coil would produce a force that opposes the input force whereas the bottom coil would generate a force that has the double function to restate the magnet in its neutral position while transmitting the original input force to the main structure. On the other hand, if a suspended payload needed to be steered by $\Delta z_1$, the
forces generated within the coils could produce the whole strut to stretch or shorten (see Figure 6.18b). It is worth noting that $\Delta z_1$ is equal to twice $\Delta z_2$, which means that each spring is only compressed or stretched by half of the desired final displacement (i.e. each coil would need to produce a force equal to $k_1 \cdot \Delta z_2 = k_2 \cdot \Delta z_2$).

The simultaneous steering-isolating property mentioned above can be achieved by exploiting one of the features of an op-amp: an op-amp is characterised by infinite input impedance which means that the input pins draw no current. Therefore, by placing a voltage source between the inverting input $V_{-}$ and $V_{node}$ (see Figure 6.19) it is possible to change the input voltage without affecting the behaviour of the negative resistance circuit. In other words, if the voltage source $V_{con}$ (used to actively control the isolator’s length) were connected directly to $V_{-}$, the inherent resistance of the source $R_{con}$ would not affect the choice of $R_x$ since there is no current drawn in the input pins, whereas it would need to be taken into account if $V_{con}$ were placed in series with the electromagnet.

Moreover, the position of the DC voltage source shown in Figure 6.19 has the other advantage that it would require half of the voltage compared to the case with the voltage source in series with the coil to obtain the same current offset (see Appendix B for the analytical
6.4 Static Compensation and Steering Capability

demonstration). Simulations to verify this behaviour were carried out in LTspice software. Figure 6.20a shows the current flowing inside the electromagnet when $V_0$ was set to have an AC voltage of 1 mV of amplitude which simulates the induced voltage produced by the relative movement between the magnet and the coil. In this case $V_{con}$ was set to 0 V and $R_{con}$ to 50 Ω. Instead, the effect of adding a DC voltage source at the inverting pin is shown in Figure 6.20b. The source $V_{con}$ was set to 10 mV and as a result the current signal shifted by 40 mA while keeping the original sinusoidal wave form with an amplitude of 2 mA.

Therefore, the simplicity of this technique (which could be implemented already on a voice coil actuator) might result in a game changer because by combining in one device these two fundamental aspects (steering capability and high isolation performance) several missions will be able to achieve unprecedented stability requirements with a system that is far less cumbersome and power demanding than the devices that are currently used.

![Figure 6.19: Electrical schematic of the EMSD that can function simultaneously as an isolator and an actuator](image)

**Figure 6.19:** Electrical schematic of the EMSD that can function simultaneously as an isolator and an actuator
6.5 Summary

This chapter dealt with other important aspects of the proposed isolator at sub-system level in order to address some of the issues that have limited the use of isolation systems in space missions. It started with the presentation of the strut concept design aimed at obtaining a device capable of accurately reproducing the analytical model presented in Chapter 5 while being characterised by a relatively small amount of device mass and input power required. This was addressed via an extensive analysis on steel flexures that have the crucial role of

Figure 6.20: Simulation of the effect of a DC voltage source connected in the shunt circuit using LTspice software. (a) current produced by the induced voltage in the electromagnet; (b) current maintains the original sinusoidal wave form but presents an offset of 40 mA due to the DC voltage source inserted in the shunt.
allowing a vertical displacement while preventing movements in all the other directions. The problem that could arise from using powerful magnets was also investigated (e.g. magnetic cleanliness outside the strut) and it was drastically reduced by using mild steel cases as magnetic shielding. The single strut was then included in a hexapod configuration capable of isolating a device in all six DoFs. Five hexapods found in literature were investigated and they were compared with the target goals of the proposed hexapod. This comparison showed that the isolator presented in this thesis is highly competitive not only in terms of attenuation performance but also at system level. The third section of this chapter dealt with the possibility to reduce the temperature dependency of the isolator. In particular, an active and a passive methods were suggested with the goal of controlling the negative resistance in order to follow the variation of the coil resistance. The chapter concluded with the description of the capability of the EMSD to function simultaneously as an actuator and an isolator. This feature is extremely interesting because it could allow, for instance, the steering of a payload for fine pointing purposes while still performing the high attenuation performance reported throughout this thesis.
Chapter 7

Conclusions and Future Work

This chapter summarises the completed work and draws conclusions from the research presented. Areas in which further investigation is needed are noted including design considerations and recommendations based on the research findings.

7.1 Thesis Summary

Micro-vibrations represent a growing topic of interest in spacecraft structural dynamics, and consequently a significant amount of research is being undertaken to deal with the various issues that affect satellites with high stability requirements. Among the different research areas involving micro-vibrations, this thesis has been focussing on their attenuation through an isolation system. The aim of this work was to investigate and develop a better-performing, versatile, low-impact isolation system which could drastically improve the mitigation of micro-vibrations while alleviating some crucial design constraints in the overall spacecraft structure and mechanisms.

The literature review of Chapter 2, after differentiating between absorbers and isolators, described the main damping methods that have been studied and developed for micro-vibration mitigation in the last three decades. These methods can be divided in four categories: passive, active, hybrid and semi-active dampers. Passive dampers are capable of operating without requiring input power, thus making them extremely reliable and preferable for space applications. However, they are characterised by a pre-defined, relatively-small damping force that is usually not enough for high-resolution payloads. Higher mitigation can be achieved with active dampers through a controlled force that actively counteracts
7.1 Thesis Summary

a disturbance. The main limitations of these devices are represented by the significant amount of input energy required to operate them and the possible occurrence of control instabilities. The advantages of passive and active dampers can be combined through a hybrid configuration. Although several hybrid dampers have been used in real isolator devices, their application is still far from being ideal due to the several drawbacks that they present, such as the large input power they still require (due to the active part) or the excessive mass they introduce. Therefore, these technologies are not yet ready to be included on board spacecraft that require high pointing stability.

Semi-active dampers are drawing increasing interest from the academic community and space industry for their potential to solve some of the problems shown by hybrid dampers, such as the need of several sensors or the risk of instability issues. Smart materials (e.g. electrorheological or magnetorheological materials) are mostly used as semi-active dampers because they work mainly as passive devices with some parameters that can be actively modified. Also electromagnetic transducers could be used as semi-active dampers if a negative impedance is included in the shunt circuit. In particular, the use of a negative resistance can considerably improve the damping performance of an EMSD since it produces an overall reduction of the circuit resistance that results in an increase of the induced current in the closed circuit. However, the analysis on these devices found in literature was still partial and further investigation was needed in order for this technology to be able to compete with other well-established damping systems. For instance, considerations on the power consumption or its behaviour over a wide temperature range have not been encountered in the literature.

The development of a multiphysics, multi-parametric model of an EMSD with the use of a negative resistance converter circuit was reported in Chapter 3. In particular, this damper was used in conjunction with a spring to reproduce a 1-DoF system. Several aspects of this system were investigated. First, the analysis in the frequency domain highlighted the fundamental role played by the ratio of resistance over inductance. By decreasing this ratio, it was possible to move the extra pole toward lower frequencies and improve the high frequency attenuation. Secondly, a trade-off on the magnetic configuration was carried out with the goal of increasing the radial component of the magnetic field and thus the electro-mechanical transducer coefficient $K_d$. The configuration that was ultimately chosen was characterised by a coefficient $K_d$ which was considerably larger than the ones found in literature while presenting a relatively small mass. The chapter continued with the description of the negative resistance converter circuit, which was made of one op-amp and three resistors and was capable of creating a passive-like resistor with constant negative magnitude. The
chapter concluded with the analysis of the damper performance at different environmental conditions and a parametric trade-off that allowed for the 1-DoF system to meet the predefined requirements, such as the maximum amplification below 6 dB, the corner frequency less than 10 Hz and a final slope of $-40$ dB/dec, all within a temperature range from $-20 \, ^\circ C$ to $+50 \, ^\circ C$.

The analytical model was corroborated by an extensive test campaign whose results were presented in Chapter 4. Great attention was given to the design and analysis of the test rig. In particular, it was fundamental to guarantee that the system presented a single DoF while preventing the rig itself to contaminate the test results. A finite element analysis (Nastran) of the full test set-up helped to better understand the test results and differentiate between the forces transmitted to the ground through the isolator and the disturbances coming from the test rig (e.g. the secondary modes of the flexures). The comparison between analytical and experimental data demonstrated the accuracy of the mathematical model as well as the linearity of the negative resistance converter circuit.

A novel 2-collinear-DoF isolator system that used a single magnet to produce a two-level damping with two separate EMSDs was the main focus of Chapter 5. An initial evaluation of a benchmark study was presented for performance comparison purposes. The chosen system was made of two masses alternated with two viscoelastic parts, and the fractional derivative model was adopted to analyse the elastomers. This benchmark study demonstrated that the increase of the secondary mass was the most influential parameter that allowed for the second resonance to be moved toward lower frequencies. The 2-collinear-DoF system proposed in this thesis offered instead the possibility to achieve a better performance by keeping the secondary mass low and tuning the other six parameters. Through a parametric trade-off, the isolator’s TF resulted to be characterised, within the whole temperature range of interest, by the elimination of the first resonance peak, the corner frequency set below 10 Hz, an attenuation of more than 40 dB from 100 Hz and a final slope of $-80$ dB/dec. The accuracy of these mathematical predictions was corroborated by a test campaign in which not only a mini-shaker but also a real reaction wheel were used to generate the input disturbance.

The evaluation of the attenuation performance is usually the main aspect that is analysed when a novel isolator is reported in literature. However, in order for an isolation system to be considered for a space application, it is necessary to address also other aspects at system level. This was the main goal of Chapter 6. A concept design of the strut was first presented. This analysis was fundamental because it gave a preliminary assessment on the possibility to scale down the cumbersome test rig into something suitable for a space mission. The crucial investigation on the steel flexures helped to choose their final design that was able
to guarantee the desired longitudinal stiffness while presenting a considerable amount of bending stiffness and a high value of torsional stiffness. The problem of magnetic cleanliness was also tackled by using coil-support cylinders made of magnetic steel. Following the design of the single strut, a preliminary analysis of the hexapod platform made of six of the aforementioned struts was presented and its main features (e.g. mass and power consumption) were compared with other five hexapods that were found in the literature. Though still at an early stage, the proposed hexapod presented several advantages that would make it highly competitive with other hexapods that were developed and commercially sold. The third section of the chapter dealt with the possibility of reducing the temperature dependency of the proposed technology. An active method and a passive one were presented as a basic overview on the opportunity to thermally control an EMSD. Finally, the description of the capability of the EMSD to function simultaneously as an actuator and an isolator was reported. This double functionality could be achieved by connecting a DC source to the inverting input pin of the op-amp and without affecting the magnitude of the negative resistance. Therefore, by combining these two effects in one device, the proposed isolator could allow for higher stability requirements to be achieved while having a system that introduces less mass and requires less power than the currently used technology.

7.2 Main Achievements

The work of this thesis has been published in the two journal papers listed in the introduction as well as been presented in two international conferences:

- SPIE 9431, Active and Passive Smart Structures and Integrated Systems, 2015 [189]
- 14th European Conference on Spacecraft Structures, Materials and Environmental Testing (ECSSMET), 2016 [178]

Three novel contributions to the state of the art have been made in this research as presented initially in Chapter 1:

1- Multiphysics, multi-parametric model of an EMSD

This thesis presented a compact analytical model of an EMSD connected to a negative resistance converter circuit which simplified the analysis both in the time domain and in the frequency domain. The accuracy of this model was strongly supported by the experimental results. By analysing the temperature dependency of each parameter of the system, it was
7.2 Main Achievements

possible to study the effectiveness of an EMSD in a wide temperature range. Moreover, such analysis helped to better understand the role of the resistance and the inductance in terms of dynamic response of the system. Their ratio was responsible for determining the frequency of the extra pole introduced by the EMSD. The increase of the inductance produced the shift of the pole toward lower frequencies. The same effect could be achieved by reducing the total resistance. However, contrary to what has been reported for absorbers (that is, reducing the resistance toward zero produces a monotonic increase of the attenuation [31]), this thesis showed that for an isolator there was an optimal value of resistance, below which it negatively affected the damping performance (i.e. the damping force decreases). Ideally, having a total resistance equal to zero would result in a system with zero damping (because there would be no resistance to dissipate the energy) and increased stiffness due to the spring-like force generated by the interaction between the coil and the magnet.

2- Characterisation of the negative resistance converter circuit

Few examples of the negative resistance converter circuit were found in the literature and those which were found only applied to absorbers [152]. This research demonstrated the simplicity of implementing the negative resistance converter circuit as well as its linear behaviour when used in isolators. An assessment of the power consumption was also reported. This circuit was characterised by low power consumption not only because part of the power was self induced by the vibration, but also because the resistance of the coil and shunt circuit were deliberately chosen to be as small as possible while still being able to produce the desired damping level and meet the attenuation requirements. The magnitude of those resistances drives the amount of power required by the circuit to operate. Also remarkable was the potential for the EMSD to function simultaneously as an actuator and an isolator. Because this was a direct exploitation of the characteristic of the op-amp to have infinite input resistance, this feature could be considered an intrinsic property of the negative resistance converter circuit.

3- Novel 2-collinear-DoF isolator with embedded EMSDs

The novelty of the proposed 2-collinear-DoF isolator resided in the ability to create a two-level damping system with the use of a single magnet. This configuration allowed for the secondary mass to be kept at less than 4% of the main suspended mass, especially due to the role played by the inductance. Through a parametric analysis it was demonstrated that having a large inductance could be beneficial not only in terms of performance but also at system
level since it would not be necessary to increase the intermediate mass (as shown instead for the benchmark study). Overall, the proposed system simplified considerably the isolator architecture (the complexity to produce two level of damping in a 2-collinear-DoF system while preventing all the other displacements/rotations is the typical drawback of this type of system found in the literature) and also offered enhanced versatility due to the possibility to modify several parameters. In terms of damping performance, the proposed isolator met all the attenuation requirements in the temperature range of interest and it showed the remarkable roll-off slope of $-80$ dB/dec starting already before 100 Hz. Although the characterisation of the strut is preliminary, this device could potentially compete with both passive isolators (due to the complete elimination of the resonance peak and the response versatility) and active isolators (because its linear behaviour is not determined by a control algorithm or large power supply).

7.3 Future Work

The research presented in this thesis is meant to be a starting point in the establishment and development of this technology as competitive alternative of active and hybrid isolation systems. With this work the proof of concept of the proposed isolator was verified both analytically and experimentally. In order for this technology to continue its path towards a flight-ready status there are some aspects that need further investigation and/or implementation.

- Development of an optimisation algorithm capable of evaluating all the system parameters given certain micro-vibration attenuation requirements or design constraints. Such algorithm, which needs to control also the geometric features of the magnetic stack, will allow for this isolator to be scaled down or up without losing its main advantages. The possible advantage of having an asymmetric configuration in which the two circuits have different parameters should also be investigated

- Verification of the magnetic cleanliness. The magnetic field produced by strong magnets (as Nd-Fe-B magnets adopted in this thesis) could heavily affect the electronic equipment surrounding the isolator, and for a space mission that might mean interfering with some payload functionalities. Further investigation is needed to assess the effectiveness of the proposed magnetic shield or to improve it in case it is not enough

- Evaluation of the electronic circuit in space environment. It is widely known among electronic engineers that the space environment can be extremely harsh for electronic components (especially due to unshielded radiation coming from the sun). Although
the negative resistance converter used in this study was proven to work as predicted, more analyses in thermal and vacuum environment need to be performed as well as failure analysis

- Manufacturing and assembling of the strut. The strut will perform as modelled in this thesis as long as all the DoFs but the longitudinal displacements are prevented. The design proposed in Section 6.1 is the result of a preliminary assessment, but an extensive test campaign in parallel with FEA are necessary in order to prove that this technology can be a viable alternative to the current state of the art. The first characterisation of a strut prototype is currently under development (see Appendix B). Once enough confidence will be gained on the single strut, the next step will be the realisation of a fully-functional 6-DoF hexapod platform

7.4 Conclusion

The contribution given in this thesis is a first step in the development of the proposed technology, and significant improvement is still necessary to fully establish this device as future isolator for high-resolution space missions. Nevertheless, the combination of novelty, simplicity and high performance that were addressed in this research have resulted in a considerable boost of this technology, and the impressive potentiality outlined throughout the thesis has enhanced its visibility to the point of gaining the attention of the private sector as well as the European Space Agency.
References


References


References


References


References


Appendix A

Flexure Non-Linear Analysis

The computation of Tab. 6.1 was obtained through a non-linear analysis on Nastran (solver 106). Four different subcases were performed, each with a different load case. The boundary conditions were fixed all around the flexures and free inside the flexure. The load was applied at the centre of the flexure. The four load cases were:

- Axial force;
- Lateral force;
- Bending moment;
- Torsional moment.

By applying a small load and observing the displacement/rotation of the central node of the flexures it was possible to retrieve the four stiffnesses reported in Tab. 6.1. Th Nastran code to run such analysis is reported below.

```
$ NASTRAN input file created by the Patran 2014.0.1
$ translator on April 20, 2016 at 18:02:21.
$ Direct Text Input for Nastran System Cell Section
$ Nonlinear Static Analysis, Database
SOL 106
CEND
TITLE = MSC.NASTRAN JOB CREATED ON 18-APR-16 AT 12:11:00
ECHO = NONE
SUBCASE 1
   TITLE=MSC.NASTRAN JOB CREATED ON 18-APR-16 AT 12:11:00
```

156
SUBTITLE=AXIAL_FORCE
NLPARM = 1
SPC = 2
LOAD = 9
DISPLACEMENT(SORT1,REAL)=ALL
OLOAD(SORT1,REAL)=ALL

SUBCASE 2
TITLE=MSC.NASTRAN JOB CREATED ON 18−APR−16 AT 12:11:00
SUBTITLE=LATERAL_FORCE
NLPARM = 2
SPC = 2
LOAD = 11
DISPLACEMENT(SORT1,REAL)=ALL
OLOAD(SORT1,REAL)=ALL

SUBCASE 3
TITLE=MSC.NASTRAN JOB CREATED ON 18−APR−16 AT 12:11:00
SUBTITLE=BENDING_MOMENT
NLPARM = 3
SPC = 2
LOAD = 13
DISPLACEMENT(SORT1,REAL)=ALL
SPCFORCES(SORT1,REAL)=ALL
STRESS(SORT1,REAL,VONMISES,BILIN)=ALL

SUBCASE 4
TITLE=MSC.NASTRAN JOB CREATED ON 18−APR−16 AT 12:11:00
SUBTITLE=TORSIONAL_MOMENT
NLPARM = 4
SPC = 2
LOAD = 15
DISPLACEMENT(SORT1,REAL)=ALL
SPCFORCES(SORT1,REAL)=ALL
STRESS(SORT1,REAL,VONMISES,BILIN)=ALL

BEGIN BULK
PARAM POST 0
PARAM AUTOSPC NO
PARAM WTMASS 1.
PARAM LGDISP 1
PARAM PRTMAXIM YES

NLPARM 1 10 AUTO 5 25 PW NO
1.0 7 .01
NLPARM 2 10 AUTO 5 25 PW NO
1.0 7 .01
NLPARM 3 10 AUTO 5 25 PW NO
1.0 7 .01
NLPARM 4 10 AUTO 5 25 PW NO
1.0 7 .01

$ Loads for Load Case : SUBCASE1.SC2
LOAD 9 1. 1. 1

$ Loads for Load Case : SUBCASE2.SC2
LOAD 11 1. 1. 3

$ Loads for Load Case : SUBCASE3.SC3
LOAD 13 1. 1. 5

$ Loads for Load Case : SUBCASE4.SC4
LOAD 15 1. 1. 7

$ Nodal Forces of Load Set : force.5
FORCE 1 20000 0 0.10000 0.1 0

$ Nodal Forces of Load Set : force.7
FORCE 3 20000 0 0.10000 1. 0 0

$ Nodal Forces of Load Set : moment.1
MOMENT 5 20000 0 0 0.1 1. 0 0

$ Nodal Forces of Load Set : moment.3
MOMENT 7 20000 0 0.1 0. 1. 0

$ Referenced Coordinate Frames
INCLUDE 'flexure_a.bdf'
ENDDATA e990bccc
Appendix B

Steering Capability Analytical Demonstration

This appendix demonstrates that placing a DC voltage source between the inverting pin \( V_- \) and \( V_{node} \) would not only preserve the attenuation performance of the negative resistance circuit, but also require half voltage that it would be needed if the voltage source were in series with the electromagnet. To begin this demonstration it is important to understand the features of the negative resistance circuit.

![Electrical schematic of the EMSD with a DC voltage source in series with the electromagnet](image)

**Figure B.1:** Electrical schematic of the EMSD with a DC voltage source in series with the electromagnet
Figure B.2: Electrical schematic of the EMSD with a DC voltage source connected at the inverting pin of the op-amp

Looking at Figure B.1, the current flowing through the resistance $R_y$ is equal to the one flowing through $R_z$, therefore:

\[
\frac{V_{out} - V_+}{R_y} = \frac{V_+}{R_z}
\]  

(B.1)

and if the resistance $R_y$ and $R_z$ were considered equal to each other, then it would result as:

\[
V_{out} = 2 \cdot V_+ = 2 \cdot V_-
\]

(B.2)

because one of the main features of an op-amp is that the input pins have always the same voltage.

In order to make this comparison meaningful, it has been assumed that the induced voltage $V_0$ is equal to zero and that the overall resistance, obtained as the sum of the positive and negative resistance, is the same for both configurations (i.e. for the configuration shown in Figure B.1 the resistance $R_x$ is increased by the magnitude of $R_{con}$ to produce the same behaviour of the other configuration).
For the case with the DC voltage source in series, the current flowing through $R_e + R_{con}$ is equal to the one flowing through $R_x$, resulting as:

$$\frac{V_{con} - V_{node}}{R_e + R_{con}} = \frac{V_{node} - V_{out}}{R_x}$$  \hfill (B.3)

and remembering Eq. (B.2) and that $V_{node}$ is equal to $V_-$, Eq. (B.3) can be simplified as:

$$\frac{V_{con} - V_-}{R_e + R_{con}} = -\frac{V_-}{R_x}$$  \hfill (B.4)

$$V_- (R_e + R_{con} - R_x) = -V_{con} R_x$$  \hfill (B.5)

$$V_- = -V_{con} \frac{R_x}{R_e + R_{con} - R_x}$$  \hfill (B.6)

$$V_- = -V_{con} \frac{R_x}{R_{tot}}$$  \hfill (B.7)

where $R_{tot}$ has been defined as $R_e + R_{con} - R_x$.

The contribution of $V_{con}$ can be determined by computing the current flowing through $R_x$, as:

$$I_x = \frac{V_{node} - V_{out}}{R_x}$$

$$I_x = -\frac{V_-}{R_x}$$

$$I_x = V_{con} \cdot \frac{1}{R_{tot}}$$  \hfill (B.8)

which shows that the current is directly proportional to the controlled DC voltage with a coefficient of $1/R_{tot}$.

When the DC voltage source is directly connected to the inverting input pin of an op-amp (see Figure B.2), half of the voltage is needed to generate the same current shown in Eq. (B.8). This can be demonstrated by evaluating the current balance as shown in Eq. (B.3). This time the voltage $V_{node}$ is equal to:

$$V_{node} = V_- - V_{con}$$  \hfill (B.9)
whereas for $V_{out}$ Eq. (B.2) is still valid. Therefore:

$$-\frac{V_{node}}{R_e} = \frac{V_{node} - V_{out}}{R_x} - \frac{V_- + V_{con}}{R_e} = - \frac{V_- + V_{con}}{R_x}$$  \quad (B.10)

$$(-V_- + V_{con}) R_x = -(V_- - V_{con}) R_e$$  \quad (B.11)

$$V_- = -V_{con} \frac{R_e + R_x}{R_e - R_x}$$  \quad (B.12)

$$V_- = -V_{con} \frac{R_e + R_x}{R_{tot}}$$  \quad (B.13)

where in this case $R_{tot}$ is defined as $R_e - R_x$ and is the same in magnitude as the one in Eq. (B.7).

As done before, the contribution of $V_{con}$ can be determined by computing the current flowing through $R_x$, as:

$$I_x = \frac{V_{node} - V_{out}}{R_x}$$

$$I_x = -\frac{V_- + V_{con}}{R_x}$$

$$I_x = -\frac{-V_{con} \left( \frac{R_e + R_x}{R_e - R_x} \right) + V_{con}}{R_x}$$

$$I_x = V_{con} \cdot \frac{2}{R_{tot}}$$  \quad (B.14)

Therefore, by comparing Eq. (B.8) with Eq. (B.14) it can be observed that by connecting the DC voltage source to the inverting input pin only half voltage is necessary to produce the same current as in the first configuration. This occurs because the other half of the power is introduced by the op-amp itself.
Appendix C

Strut Prototype

The strut prototype is under development and its first characterisation is foreseen to be completed by December 2017. Here are reported some of the stages of the strut assembly.

*Figure C.1: Single flexure mounted on two Graphite reinforced plastic rings*
**Figure C.2:** Intermediate stage during the assembly of one of the strut’s halves

**Figure C.3:** Completion of the two strut’s halves
Figure C.4: Support in aluminium that is used to guide the two halves together preventing the two magnets to crash against each other.

Figure C.5: Fully-assembled strut. The cones at the two ends are going to be used for the test campaign.