Hindawi
Shock and Vibration
Volume 2019, Article ID 1250707, 16 pages
https://doi.org/10.1155/2019/1250707

Review Article

Review of Passive Electromagnetic Devices for Vibration Damping and Isolation

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Received 29 May 2019; Revised 1 August 2019; Accepted 4 August 2019; Published 20 August 2019

Academic Editor: Mahmoud Bayat

Passive electromagnetic devices for vibration damping and isolation are becoming a real alternative to traditional mechanical vibration and isolation methods. These types of devices present good damping capacity, lower cost, null power consumption, and higher reliability. In this work, a state-of-the-art review has been done highlighting advantages and drawbacks, application fields, and technology readiness level of most recent developments. In addition, a general introductory section relates presents key considerations that any engineer, electrical or mechanical, needs to know for a deep comprehension and correct design of these types of devices.

1. Introduction

Vibrations generate significant problems and issues in mechanical systems such as fatigue, fracture, and energy loss [1, 2]. Therefore, it is essential to damp the vibrations for overall performance and durability of machines and mechanisms. Performance of new magnetic materials in combination with optimization tools allows the development of efficient and tuneable vibration damping and/or isolation techniques. Moreover, electromagnetic damping and isolation can be clean and environmentally friendly since there is no need of using fluids (except in magnetorheological dampers). Thus, they can be applied in clean, harsh, and/or extreme temperature environments such as space, aerospace, electric vehicles, or microfabrication industries. As they provide damping through contactless magnetic forces, most friction and wear issues of conventional dampers also disappear, increasing their reliability. Nevertheless, there are still critical issues to be solved like design optimization, performance, cost, device ageing, reduction of external electromagnetic interferences, or frequency tuning.

In this paper, two different types of electromagnetic damping devices can be found: active and passive. Active devices are those devices that measure vibrations in real time and react accordingly under an active control system decision. Those devices present an outstanding customized performance. However, they require control systems, electric power systems, and sensors that increase the total complexity, price, and energy consumption while decreasing the reliability of the device. On the other hand, passive devices are designed and manufactured to respond in a certain manner against vibration without the need of active feedback and control. They are flexible and can be tuned in design or during assembly, but not during operation. These types of devices have a lower cost and higher reliability at the expense of lower performance under certain vibration variations. In this article, we focus on passive electromagnetic devices for vibration, damping, and isolation (PEDVDI) because of their larger applicability. Main

challenges of PEDVDI are to provide at least the same damping capacity in terms of damping coefficients and stiffness within the same mass, lifetime, and reliability of conventional mechanical dampers, and of course, with a competitive price.

Applications of electromagnetic devices require gathering different engineering disciplines. They are typically designed and manufactured by electrical engineers. Electrical engineers test the devices and provide performance of the device in some general known variables. However, the final applications of the devices which sometimes need specific performance values are typically developed and selected by mechanical engineers. It is thus important to link properly these two fields in order to obtain more efficient and optimized devices.

One of the main objectives of this article is to summarize and show general information of all types of passive electromagnetic devices applied in vibration damping and vibration isolation. PEDVDI have been categorized as follows: eddy current dampers (ECD), electromagnetic shunt dampers (EMSD), magnetic negative-stiffness dampers (MNSD), and passive magnetorheological dampers (PMRD). This provides a wide view of the existing technologies for a proper application selection. A second aim of this article is to collect and present recent and outstanding research articles on passive electromagnetic devices for vibration damping and isolation in civil and mechanical engineering and space applications. A comparison of the performance and application field is given, highlighting its main differences, pros, and cons.

In addition to the technology review part itself, a general introductory section relates main considerations and design key parameters that any engineer, electrical or mechanical, needs to know for a deep comprehension of any PEDVDI performance. Therefore, the article may be used as a design guide for specific applications. Moreover, as it describes the general design for each technology, it can be used as starting point for new designs.

This paper is organized as follows: Section 2 relates to general design considerations of PEDVDI, Section 3 describes the review of the different technologies explored: ECD in Section 3.1, Section 3.2 is the review for EMSD, and Section 3.3 shows the review of MNSD. The papers end with the review of PMRD described in section 3.4. Finally, general conclusions are listed in Section 4.

2. General Design Considerations of Electromagnetic Dampers and Isolators

When designing PEDVDI, three types of materials are mainly used: paramagnetic, soft ferromagnetic, and hard ferromagnetic. Paramagnetic materials are weakly affected by external static magnetic fields. Common paramagnetic materials used in designs are aluminium, copper, titanium, or polymers. However, when selecting paramagnetic materials, their electrical conductivity must be carefully considered, as we will describe later. Even if they are inert to static external magnetic fields, they can severely react to alternant magnetic fields if they are good conductors. If a paramagnetic material is inside a static magnetizing field H, its magnetic polarization M is negligible, acting as if they were air or vacuum.

On the contrary, ferromagnetic materials do react against external magnetic fields H. If an external magnetic field is applied to a ferromagnetic material, it gets magnetized, increasing significantly its volumetric magnetization M, and thus the total magnetic flux density B, i.e., more magnetic field accumulated within the same volume. Magnetic behaviour of the ferromagnetic materials is not linear, but it follows a hysteresis curve (Figure 1).

Based on the value of remanence B_R and coercivity H_C, we can determine whether the sample under study is a hard or a soft magnetic material. Those materials with large remanence and large coercive field are called hard magnetic materials because they are hard to demagnetize. Inversely, soft magnetic materials have very low remanence and low coercive field, and thus they are easily demagnetizable.

Soft magnetic materials have a thin hysteresis curve, so they are typically applied in applications where polarities change very often, such as in transformers and motor windings. Soft magnetic materials can sustain relatively small electrical losses. The hysteresis loop width tells much about the losses. Hard magnetic materials have a very wide hysteresis curve, which makes them practical in applications where they exert their magnetic field on soft magnetic materials. Their slope of demagnetization at the zero line is very shallow and does not steepen until it goes far to the left of the zero line. If hard magnetic materials changed polarity very often, the hysteresis losses would be huge. Hysteresis losses can be used in magnetic dampers as a mechanism to transform and dissipate kinetic energy, as shown in Section 3.4.

General values of electromagnetic properties for different engineering materials are listed in Table 1.

In mechanical engineering applications, we used magnetic materials to exert forces between themselves to provide an output torque like in motors or damping forces as in PEDVDI. Magnetic forces between two magnetic elements depend on the strength and orientation of the magnetic field that element 1 applies on element 2 and on the strength and direction of the magnetization of element 2 [3–6]. The force that element 1 exerts on 2 depends on the gradient of its volumetric magnetization M and on the magnetic field generated by the element 2:

$$\overrightarrow{F_{12}} = \nabla\left(\overrightarrow{m_2}\cdot\overrightarrow{B_{\text{applied by 1}}}\right) = \nabla\left(\overrightarrow{M_2}\cdot \text{Vol}_2\cdot\mu_0\cdot\overrightarrow{H_{\text{applied by 1}}}\right). \tag{1}$$

Thus, the larger is its magnetization the larger will be the forces acting on it. Materials with very large magnetization values suffer larger forces under the same external magnetic field. Inside up, materials with very large magnetization generate larger external magnetic fields. Therefore, in first term, the larger is the magnetization of the elements, the larger will be the forces of/on the device. Moreover, the magnetic field strength that a certain element generates in its surroundings is inversely proportional to the cube of distance. Hence, it is very important to approximate magnetic elements as much as possible in order to increase magnetic

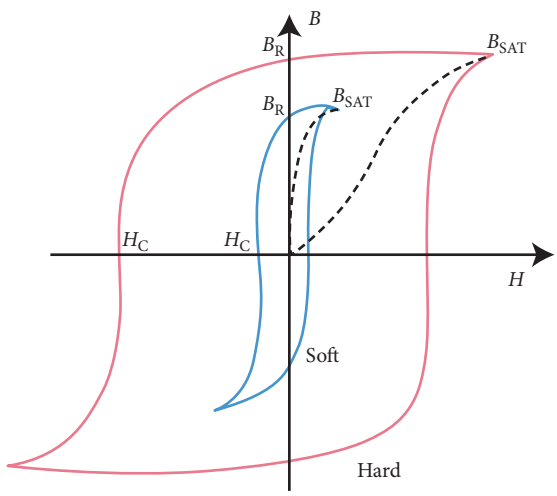

Figure 1: Diagram of B–H hysteresis curve for a hard and a soft ferromagnetic material.

Table 1: Magnetic properties of different engineering materials.

Material type	Coercivity, bHc (kA/m)	Remanence, Br (T)	Relative permeability	Saturation, bsat (T)
Ferrite magnet	150–250	0.35–0.4	1.2–1.9	0.35–0.4
Alnico magnet	30–151	0.7–1.11	1.2–1.9	0.7–1.2
Neodymium-iron magnet	675–1090	0.87–1.5	1.05	0.87–1.5
Samarium-cobalt magnet	493–790	0.83–1.15	1.05	3000
Iron	0.006–0.080	0.01–0.06	150–2000	1.6–2.15
Nickel	0.056–2.01	0.01–0.15	100–600	0.4–0.6
Cobalt	0.80–71.62	0.02–0.2	70–250	0.7–0.9
Co-Fe alloy (49% Co)	0.07–0.09	0.1–0.3	8000–15000	2.2–2.4
Ni-Fe alloy (75% Ni)	0.001–0.002	0.05–0.2	60000–250000	0.6–0.8
Ni-Fe alloy (50% Ni)	0.005–0.010	0.03–0.15	7000–100000	1.5
Electrical steel	0.032–0.072	0.5–0.8	3000–4000	1.6–2
Stainless steel 304/316	1–3	0.001–0.002	1.003–1.012	0.025–0.03
Stainless steel 430	0.5	0.8	750	1.4
Copper OFCH	—	—	0.99	—
Aluminium 7075	—	—	1.004	—
Titanium grade 5	—	—	1.0001	—
PTFE Teflon	—	—	1	—
Polyimide	—	—	1	—
Nylon	—	—	1	—

forces. Air gaps between moving elements must be always minimized.

There are two main methods to achieve large magnetizations inside a certain volume: nonpermanent magnetization of soft ferromagnetic materials and permanent magnetization of hard ferromagnetic materials. Nonpermanent magnetization is generally obtained through magnetizing fields generated by currents circulating through coils or windings in the device. This option is not affected by other phenomena provided that the current is maintained, so it is a reliable method. Its major issue is that it requires a continuous power consumption and current control.

Permanent magnetization of hard ferromagnetic materials has a main advantage which is that it does not require continuous currents flowing since magnetization is done once in factory. However, other problems must be considered. For example, as the magnetization is permanent, forces between magnets will always appear even in nondesired orientations, so it is important to analyse the magnetic forces at each of the motion positions. Permanent magnetizations usually have lower flux density on the materials in comparison with fully saturated soft ferromagnetic materials. Permanent magnets, if once fully magnetized, retains magnetism permanently, but its intensity does not remain constant and normally decreases gradually with elapse of time. This change is known as permanent magnet ageing, which is caused either naturally or by external disturbance. The external disturbance mentioned before can be classified into the following four types according to characteristics: magnetic circuit reluctance change, external field application, mechanical shock, and temperature change [7].

For extreme applications like in space or in cryogenic environments, other aspects also affect significantly to

permanent magnetizations. Radiation, as found in space applications, can permanently demagnetize magnets [8]. Therefore, special cares as correct selection of materials, even if they have lower remanence or radiation shielding, must be considered. Temperature also plays a main role when analysing permanent magnetization. For all permanent magnets, increasing temperature means to lose their magnetization. Some materials like SmCo may resist higher temperatures than others, but high temperatures always imply permanent magnetization issues. In contrast, lower temperatures usually increase permanent magnetizations for most of the materials [9, 10]. Nevertheless, at lower temperatures, permanent magnets get more brittle, so applications with very large magnetic forces may need wall reinforcement of magnetic pieces.

As stated, magnetic forces are generated by applying external magnetic fields to magnetized volumes. This surely implies that volumetric magnetization changes, according to the material hysteresis curve. When having permanent magnets in a device, they should be checked for possible demagnetisation caused by external magnetic fields. Normally, only a simple check is needed at the end of the analysis. The check is made accordingly: to find the highest possible temperature inside a magnet and to find the lowest possible field value inside a magnet given by a finite element method (FEM) model. This lowest parametric model values should be treated as average values inside a magnet. Then, by using the BH-curve of the used magnet material, check that the point of the lowest field value is above the knee point (point where magnetizations changes are significant and irreversible) [11].

Another main point to analyse in electromagnetic devices is the eddy current generation and its associated issues and/or benefits. When magnets move through the inner conductor, the moving magnetic field induces an eddy current in the conductor. The flow of electrons in the conductor immediately creates an opposing magnetic field, generating Lorentz forces, which result in damping of the magnet motion and produces heat inside the conductor. The amount of energy transferred to the conductor in the form of heat is equal to change in kinetic energy lost by the magnets.

Power loss due to eddy currents in a conductive sheet per unit of mass can be calculated as

$$P = \frac{\pi^2 \sigma B_p^2 d^2}{6D} \cdot f^2, \tag{2}$$

where P is the power lost per unit mass (W/kg), B_p is the moving magnetic field peak (T), d is the sheet thickness (m), σ is the electric conductivity of the conductive sheet (S/m), D is the conductive sheet density (kg/m^3), and f is the oscillation frequency (Hz), or variation, of the applied magnetic field. Thus, power losses depend directly on material conductivity, quadratically with the applied magnetic field on the conductive element and on geometrical dimensions. The component of the magnetic field that affects for the eddy current generation is the one perpendicular to the sheet. This must be considered when designing towards eddy current motion damping.

Equation (3) is valid only under the so-called quasistatic conditions, where magnet motion frequency is not fast enough to generate the skin effect, i.e., the electromagnetic wave fully penetrates into the material. In relation to very fast-changing fields, the magnetic field does not penetrate completely into the material. However, increased frequency of the same field value will always increase eddy currents, even with nonuniform field penetration. The skin depth or penetration depth, δ, is defined as the depth where the current density is just $1/e$ (about 37%) with respect to the value at the surface. Penetration depth for a good conductor can be calculated from the following equation [12]:

$$\delta = \frac{1}{\sqrt{\pi f \mu \sigma}}, \tag{3}$$

where δ is the penetration depth (m), f is the frequency (Hz), μ is the material magnetic permeability (H/m), and σ is the material electrical conductivity (S/m). Penetration depths for different materials are plotted on Figure 2. As design criteria, penetration depth at a certain frequency must be in the same order than the characteristic geometric value of the conductive elements. This allows to maximize eddy current generation and so the damping forces.

Eddy current generation can be linked with mechanical damping. In a viscous damper, mechanical power losses can be expressed as

$$P = F_D \cdot v, \tag{4}$$

where F_D is the damping force and v is the moving mass speed. By linking eddy current power losses and mechanical power losses, we can state that

$$\frac{\pi^2 \sigma B_p^2 d^2}{6D} \cdot f^2 = F_D \cdot v. \tag{5}$$

An oscillatory linear motion frequency is directly proportional to linear speed amplitude as $v = A \cdot 2 \cdot \pi \cdot f$, where A is the displacement amplitude. Thus, we can determine that the damping force-speed ratio is a constant c depending on the eddy current electromagnetic behaviour as

$$c_{\text{ed}} = \frac{F_D}{v} = \frac{\sigma B_p^2 d^2}{24 D \cdot A^2}. \tag{6}$$

Therefore, in order to maximize damping coefficient, several parameters must be optimized. If the magnetic field applied is larger, damping coefficient will increase quadratically. As already stated, maximizing a generated magnetic field can be done by selecting a material with large magnetization and by reducing distances between magnetic field generator magnet and conductive element. By reducing motion amplitude, damping coefficient can also be larger. Opposite considerations must be taken when trying not to damp motion but to allow it smoothly.

Magnetic forces are volumetric forces that depend on the magnetization and magnetic field directions and orientations. Some designs may require forces in radial, tangent or longitudinal directions in order to damp or to transmit forces. However, for most of the cases, only one direction of the forces is required while other two directions must be constraint or locked. If they are not locked, undesired motions can appear. Earnshaw's theorem states that a

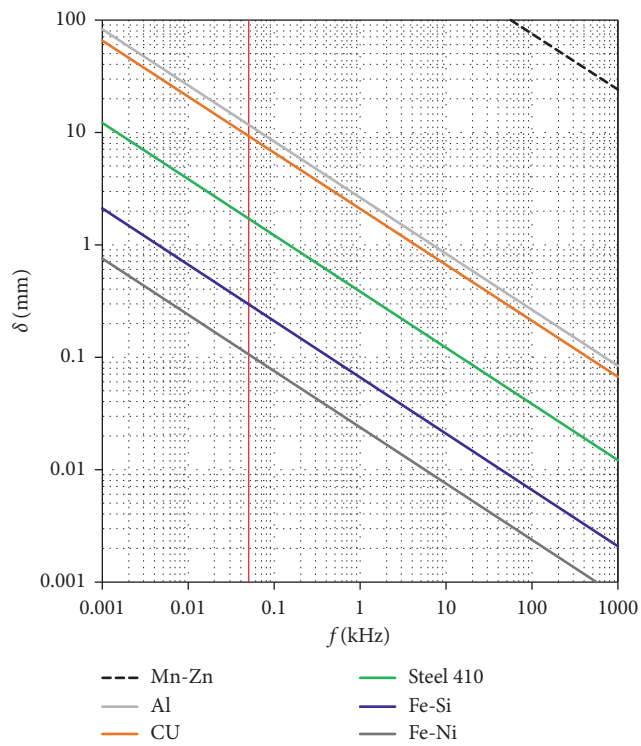

Figure 2: Skin depth vs. frequency for some materials at room temperature; red vertical line denotes 50 Hz frequency [13].

collection of permanent magnets cannot be maintained in a stable stationary equilibrium configuration by themselves, thus mechanical constraints must be included in any device. These mechanical constrains or kinematic pairs generate undesired frictions. Therefore, in order to minimize the loads on these kinematic pairs, symmetrical magnetic configurations and balanced assemblies are highly recommended. This means to include pairs of magnets instead of odd number of magnets with counter directions and symmetrical magnetic mass configurations.

We can summarize some key considerations when designing electromagnetic dampers and isolators as

(i) The larger the magnetization of the elements, the larger will be the forces

(ii) Air gaps between moving elements must be always minimized

(iii) Permanent magnets must be analysed in all their motion positions

(iv) Remanence of permanent magnets is affected by mechanical shocks and external fields

(v) Temperature is also a critical aspect for permanent magnet remanence

(vi) Demagnetization of permanent magnet pair has to be analysed and prevented

(vii) Eddy current damping depends quadratically on applied magnetic field

(viii) Penetration depth at a certain frequency must be analysed in eddy current damping

(ix) Magnetic assemblies have to be symmetric and magnetically balanced, if possible

(x) Simple and standard magnet shapes like cylinders, ring, or blocks on prototypes must be selected.

Along next sections, we will overview different types of PEDVDI analysing their characteristics and constructive properties and also their application fields and performance.

3. Passive Electromagnetic Technologies for Vibration Damping and Isolation

3.1. Eddy Current Dampers. Eddy current dampers (ECDs) are based on the interaction between a nonmagnetic conductive material and a time varying magnetic field in their relative motion. Eddy currents are generated either by movement of the conductive material through a stationary magnet or by strength or position change of the magnetic field source. This induces a magnetic field with opposite polarity to the applied field and a repulsive electromotive force (EMF) which is dependent on the applied field change rate, as shown in Section 2. Due to the conductive material internal resistance, induced currents are dissipated into heat and the energy transformed from the system is removed [14].

General design of an ECD is depicted in Figure 3. It consists of a set of permanent magnets, typically made of NdFeB or SmCo because of their large magnetic quality, which are aligned in front of conductive elements made in aluminium or copper (preferred as its conductivity is the largest). Design and device optimization are currently done through numerical simulations on the magnetic field distribution and eddy current, generally FEM based.

There are three main points to consider during design phase in order to enhance the damping coefficient of an ECD. First is to properly orient permanent magnets poles in respect to the conductive elements. Magnetic field vector components must be perpendicular to the conductive plane as much as possible because eddy currents are generated by those components. However, there are cases where magnet polarization capacity prevents an optimal magnets layout, for example, radial magnetizations could be an optimal choice for cylindrical devices, but radial polarized magnets are not as strong as axial polarized ones yet. Second point is to maximize the magnetic field variation, i.e., to maximize the peak-to-peak value of the applied magnetic field. This maximization can be done by combining reduced air gaps, long displacement variations, large magnetic quality of permanent magnets, and large magnet sizes. An alternative to large magnets could be to assemble more magnets but with reduced sizes. In this way, eddy currents generated per magnet will be smaller but multiplied by the number of magnets. From a certain point, increasing the size of the magnets does not increase the applied magnetic field; therefore, there is an optimized size wherein it is worthier to

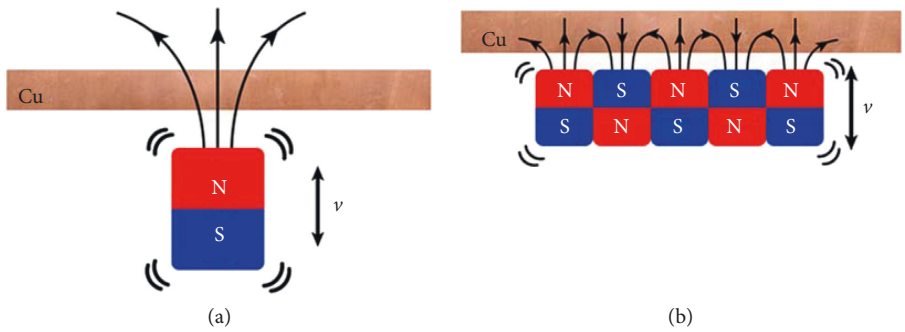

Figure 3: ECD common configurations: (a) Single magnet against conductive element. (b) Multipole magnets against conductive element.

add other magnets with alternative polarizations instead of increasing the own magnet size.

Third method, and most relevant, to enhance damping coefficient is to increase cinematically the motion of the permanent magnets towards faster, i.e., larger variation frequency, magnet motions. We consider this method as the most relevant because there are multiple mechanical options to enhance or multiply magnet's motion towards a maximization of the variation of the magnetic field. A common method is to couple a mechanical speed multiplication stage between vibration source and magnets frame. Mechanical speed multiplication selection will depend on the vibration source type of displacement. It has been typically done by using track-pinion elements or lever arms [15] for linear vibration oscillations and planetary speed multipliers for rotational vibrations. This last type of devices is profusely commercialized in aerospace applications [16, 17]. These commercial elements operate at temperature ranging from −40 to + 70°C, having a significant variation in damping depending on the operational temperature. The variation of damping coefficient with respect to temperature is typically −0.5%/°C. The main disadvantage of using mechanical multipliers is that this part may need maintenance, lubrication, and of course contact, limiting those advantages provided by the eddy current dampers. In addition, mechanical issues like large hysteretic forces or gear backlash prevent their use in low-amplitude vibration damping applications like for microvibrations [18, 19]. To solve those issues, an outstanding and unique eddy current magnetic damper with mechanical multiplication has been proposed and successfully tested [20]. This commercialized device includes an innovative multiplication stage made through linear magnetic gearing instead of mechanical that prevents almost all issues appearing in conventional mechanical multiplication stages while providing excellent results both at room and high temperature environments, with a damping coefficient of 35000 Ns/m for a 19 kg device [21]. This eddy current damper has one of the best specific damping coefficients ever demonstrated, 1842 Ns/m·kg, making it very adequate for cars or aircrafts. Damper tested in [20] has an operational temperature range from −40 to + 250°C with a very low complexity of its moving parts. The major con is that magnetic parts generates magnetic contamination in its surroundings.

There are multiple research articles related to ECDs. Ahn [22] presents a design procedure of an ECD for a linear motor motion stage. This device overcomes the disadvantages of the spring type mechanism such as resonance and assembly difficulties due to the spring. However, the design is simple and constraint to the specific linear motor stage. In [23], the eddy current damping is applied in a passive tuned mass damper using a Halbach array of magnets against a copper plate. They demonstrated that plate thickness severely affects the damping coefficient passing from 25 Ns/m for a 4 mm plate thickness to more than 35 Ns/m if the plate is 20 mm for a constant speed. Berardengo et al. [24] presented a new type of adaptive tuned mass damper based on shape memory alloys and eddy current damping. The former element is used to adapt the Eigen frequency of the device, while the latter to tune the damping. Again damping coefficient is highly affected by geometrical parameters and layouts. This can be an advantage during the design process, since it gives flexibility to the designers but it can lead to nondesired performance if some geometric values are modified during assembly or operation.

Besides the studies conducted in the previous references, other applications like in civil engineering, rotors applications, precision instrumentation, robotics, or automotive can also be found. For example, Jo et al. [25] proposed to include ECD in an air-bearing precision stage to improve the vibration isolation characteristics. A Halbach magnet array was devised to increase the density of the magnetic flux of the ECD because a stronger magnetic field generates a greater damping force. In this case, vibrations lower than 100 Hz are damped; however, vibrations above are not damped efficiently. This is explained because ECD damping coefficient decreases with frequency as also found in previous references. ECD can also be found working independently or in combination with tuned mass dampers (TMD) [26, 27] or magnetorheological dampers [28]. In any case, references [22–28] are far from a product-oriented design and they remained just as interesting proof-of-concept prototypes.

The main commercial application field for eddy current dampers is aerospace mechanism where cleanness and reliability are critical requirements. Since it is rather difficult to implement maintenance and the operational environment is severe in aerospace, the damping device should be advanced in long function fatigue life, high reliability, and good applicability in vacuum and heat transformation conditions. Eddy current dampers with instinct natures, such as

noncontact, nonleakage, and easy implementation, become a candidate to suppress vibrations of in the aerospace application system. There are different constructions and designs for ECD depending on the manufacturer. While the design and "temperature factors" may vary from manufacturer to manufacturer, the basic principle of using a high-speed magnetic damper and a gearbox to increase the damping rate and torque capacity is almost universal [29, 30]. Previous studies report rotational eddy current dampers demonstrating damping coefficients ranging between 24 and 1000 Nms/rad. The specific damping coefficient of these devices ranges between 1000 and 2000 Ns/m·kg. Damping coefficient varies with frequency, decaying significantly for greater than 50 Hz frequencies. Therefore, they are adequate for low-frequency damping but not very performant above those frequencies. They both use a single magnet against a copper plate and both they calculate equivalent viscous damping coefficient from hysteresis force-displacement curve.

Other product-oriented applications can also be found. For high-resolution and precision instruments, such as scanning tunnelling microscopy (STM) and atomic force microscopes (AFM), the effective isolation of environmental vibrations plays a key role. Different types of eddy current dampers have been presented and analysed in the literature. A comparison of one- and two-stage spring-suspended systems with magnetic eddy current damping showed acceptable vibration isolation levels [31]. The advantage of using eddy current dampers is avoiding the use of grease-lubricated elements near the probe, which may damage the instrumentation. Most of the recent STM and AFM use magnetic eddy current damping for low-frequency vibration isolation [32, 33]. Typical values of damping coefficient for eddy current dampers applied in instrumentations vary from 0.25 Ns/m to 5 Ns/m. For scientific on-ground instrumentation, specific damping coefficient is not relevant. ECD in [33] was used in the pressure ranges between 10^{-7} Pa and 10^{-9} Pa, showing a significant vibration displacement attenuation going from 50 nm at 16 Hz in the input to just 50 pm at 16 Hz after using ECD. Major pitfall of ECDs in STM and AFM is the magnetic contamination that ECDs may induce in the system and in the samples.

Moreover, active electromagnetic dampers used in automotive vehicle suspension systems have also drawn so much attention in recent years, due to the developments in power electronics, permanent magnet materials, and microelectronic systems. One of the main drawbacks of these electromagnetic dampers is that they are not fail-safe in case of power failure. A passive damping element can make the active electromagnetic dampers fail-safe. ECD has the potential to be used in electromagnetic dampers, providing passive damping for a fail-safe hybrid electromagnetic damper. ECD for automotive applications has demonstrated a damping coefficient 1880 Ns/m for a weight of 3.25 kg [34], which leads to a specific damping density of 578 Ns/m·kg. However, a comparison of the ECD presented in [34] with the off-the-shelf passive dampers reveals that the size and cost of the ECD are higher than those of passive oil dampers. Moreover, the ECD cost is more than twice of a commercial passive damper, due to the high cost of rare earth magnets. This is why it is essential to optimize the selection of shapes and sizes also for decreasing cost and not only to increase performance, as recommended in Section 2.

On the contrary, ECD has not been widely used for civil engineering applications because its performance remains rather limited due to its low density of energy dissipation. ECD can offer advantages in building vibration damping compared with other damping devices, such as friction damping and viscous fluid damping. A notable advantage is that the eddy current dampers may operate in outdoor under severe temperature conditions. Additionally, there is no fluid inside the damper and the damping generation is independent of friction, potentially increasing eddy current damper longevity and lowering maintenance requirements [35, 36]. In those studies, it was demonstrated that the linear damping assumption in the analytical model is only valid for a limited range of low velocity and this velocity. It is important, thus, to determine the aimed frequency range when designing ECD, because for high frequencies, they are not so performant. However, for large-scale massive structures, the required damping will be of several orders of magnitude larger than that for eddy current dampers applied in mechanisms. It is therefore more practical and economical to apply eddy current dampers as a damping element for a resonant-type absorber or tuned mass damper. The auxiliary mass weights of a TMD are just a small fraction (commonly 0.5–2%) of the controlled modal mass of the primary structure, and the damping required to mitigate the vibration of the auxiliary mass of a TMD is greatly reduced. A ECD optimized for TMD has demonstrated damping coefficients of 321.34 Ns/m, with a mass of 2 kg.

Last but not least, EDC can be found in manufacturing applications [37], specifically in robotic milling. This type of machining has become a new choice for machining of large complex structure parts. However, due to its serial structure, the industrial robot has several limitations such as low stiffness that causes a low accuracy in the machining due to chatter vibrations. In order to mitigate these chatter vibrations, a novel ECD has been designed for vibration suppression in the robotic milling process [38]. The ECD proposed is a multipole set of magnets against a cooper plate oriented to damp two vibrations directions. The damping coefficient measured in this element is 165.6 Ns/m with 0.6 kg mass, which leads to a specific damping coefficient of 276 Ns/m·kg. The results showed that the peaks of the tool tip FRFs caused by the milling tool modes were damped by 22.1% and 12.4% respectively, in the vertical axis, which increase the precision of the milling process and increase the reliability of the tool.

As a conclusion, ECDs can increase the damping property of the structures they are attached to in a broader frequency range over the classic tuned mass dampers. Also, they are clean and temperature resistant and they are not quite sensitive to the change of structural modal frequencies, thus having good robustness. Moreover, ECDs are passive dampers and do not require complex control laws, and therefore they are easy to be implemented. These advantages

make ECDs a good choice for vibration attenuation in mechanical systems.

3.2. Electromagnetic Shunt Dampers. An electromagnetic shunt damper (EMSD) is essentially an electromagnetic motor/generator that is connected to a shunt circuit (Figure 4), in which the electromagnetic motor converts mechanical oscillation into electrical energy, whereas the shunt circuit design controls the characteristic behaviour of EMSDs [39, 40]. The main features of EMSDs are as follows: easy design, passive control, energy harvesting, and motion multiplication.

Analogy between mechanical and electrical systems allows flexible EMSD designs by adjusting the external electric shunt circuit, which is generally compact in size and allows an easy element replacement. When EMSDs are combined with gear components, as in [41], EMSDs can transform linear motion to rotary motion and provide a great damper force with a small size/weight. In addition, unlike conventional dampers that dissipate kinetic energy into heat, EMSD converts kinetic to electrical energy through the electromechanical coupling effect, where electrical energy can be potentially harvested and reused for other functions if necessary. However, the optimal performance of EMSD is inevitably constrained by the inherent resistance of motor coils and circuit elements (such as inductors and capacitors) in practical applications.

There are two main practical advantages of this passive damper system. First is that the vibration energy is not merely dissipated but it can be reutilized. In [42], researchers demonstrated that by using EMSDs, vibrations were attenuated while energy is transferred to an electric circuit for its use. Secondly, vibration energy transferred to the damper can be transported easily through wires. This permits to locate the dissipation in other places far from the vibration origin. For example, in [43], an EMSD showed that it is capable of isolating the first-order and the third-order vibrations (larger than 80 Hz) far away from the vibration source. In applications where on-place thermal generation is a critical issue, like cryogenic or space applications, having the possibility to select the most adequate dissipation location is an interesting feature. On the other hand, EMSDs present several limitations; for example, the presence of inherent resistance considerably caps the maximum damping and causes the divergence of the damper performance from the design. Moreover, power generation performance of a EMSD working as an energy harvester is limited by resonance excitation as shown in [44]; therefore, other methods as multigenerator methods or multiresonance modes have to be applied. Marinkovic and Koser [45] showed wide bandwidth energy harvested from vibrations by using multigenerator method; however, its damping capacity at frequencies lower than 20 Hz is much reduced, acting as a bandwidth pass filter. Multiresonance mode method was used in [46] showing a device with 2 degrees of freedom that has two resonant peaks which may be tuned independently maintaining fairly uniform power output over a frequency range.

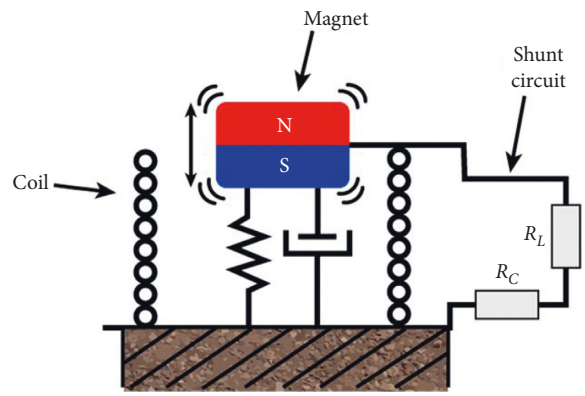

Figure 4: EMSD common configuration: a permanent magnet oscillates along a coil generating electricity from the vibration damping. This electricity is then reoriented to a load resistance to be dissipated externally or to be used or accumulated for other applications.

Main application fields for EMSDs are microvibration [47], low-frequency energy harvesting [48–52], mostly for microelectronics applications, and regenerative shock absorbers mostly for vehicle applications [53–57]. Other applications like human motion energy harvesting or combined with tuned mass absorbers [58–60] can be also found. In any case, all the developments found are limited to the laboratory demonstrator or applications and far from its extensive commercialization.

Recent studies for EMSDs applied as microelectronics energy harvesting are divided into three groups according to their objectives and approaches. The main point is to reduce the harvester resonant frequency in order to collect low-frequency vibration energy from the environment. Also, broadening current technology harvester bandwidth to increase the utilization of the random vibration energy is sought [61]. It is impossible to make a perfect harvester with low resonant frequency, wide frequency band, and good output performances at the same time.

In [62], power harvesting is simply achieved from relative oscillation between a permanent magnet allowed to move freely inside a tube-carrying electrical coil with two-end stoppers and directly connected to the vibration source. The proposed harvester with free/impact motion shows a nonresonant behaviour in which the output power continuously increases with the input frequency and/or amplitude. In addition, the allowable free motion permits significant power harvesting at low frequencies. Hence, proposed harvester is well suited for applications involved in variable large amplitude-low-frequency vibrations such as human-powered devices. Another example of nonresonant magnetomechanical low-frequency vibration energy harvester can be found in [63]. In this article, energy harvester converts vibrations into electric charge using a guided levitated magnet oscillating inside a multiturn coil that is fixed around energy harvesters exterior. In this case, the fabricated energy harvester is hand-held and the prototype generates a normalized power density of approximately $0.133\,\mathrm{mW/cm^3\,g^2}$ at 15.5 Hz.

Regenerative shock absorbers are based on electromagnetic rotary or linear motors connected to a shunt resistance operating as generators. Motors are linked to the vibration source directly or through a mechanical multiplication stage as for eddy current dampers. Three drive modes of the regenerative shock absorber systems can be found: direct drive mode, indirect drive mode, and hybrid drive mode [64]. The direct drive system has attracted substantial amount of interests due to its compact design and simple manufacturing. The direct drive mode directly connects vibration source to magnet stator to generate electricity. Direct drive regenerative shock absorbers provide damping coefficient between 1500 and 2000 Ns/m.

Microvibration on board a spacecraft is an important issue that affects payloads requiring high pointing accuracy. Although isolators have been extensively studied and implemented to tackle this issue, their application is far from being ideal due to the several drawbacks that they have, such as limited low-frequency attenuation for passive systems or high power consumption and reliability issues for active systems. In [65], a novel 2-collinear-DoF strut with embedded electromagnetic shunt dampers (EMSD) is modeled and analysed and the concept is physically tested. The combination of high-inductance components and negative-resistance circuits is used in the two shunt circuits to improve EMSD microvibration mitigation and to achieve an overall strut damping performance that is characterised by the elimination of the resonance peaks. EMSD operates without requiring any control algorithm and can be comfortably integrated on a satellite due to the low power required, the simplified electronics, and the small mass.

EMSDs demonstrate a unique feature when handling the damped vibration energy because it can be transported and/or stored as electrical energy. This makes EMSDs a very interesting type of dampers for microelectronics and vehicle applications. However, their wide implementation in industry has been limited by their low specific damping capacity as well as by the fact the performance is highly dependent on the vibration frequency.

3.3. Magnetic Negative-Stiffness Dampers. Negative-stiffness elements have been identified as unique mechanisms for enhancing acoustical and vibrational damping. Examples of negative-stiffness mechanisms include mechanical systems with negative spring constants and materials with negative moduli [66–69]. Negative-stiffness elements contribute to damping behaviour because they tend to assist rather than resist deformation as a result of internally stored energy [70]. Negative-stiffness isolators employ a unique and completely new mechanical concept in low-frequency vibration isolation. Vertical-motion isolation is provided by a stiff spring that supports a weight load, combined with a negative-stiffness structure. The net vertical stiffness is made very low without affecting the static load-supporting capability of the spring. Reducing the net vertical stiffness implies that resonant frequency is highly reduced since resonant frequency is proportional to the root square of the stiffness.

Generally, negative-stiffness spring is made by two bars hinged at the centre, supported at their outer ends on pivots, and loaded in compression by compressive forces [69, 71–73]. Both bars are brought to almost buckling operation point. The equilibrium positions of the buckled beam correspond to local minimum and maximum of the strain energy curve. Since the beam stiffness corresponds to the spatial derivative of its strain energy, the buckled beam exhibits negative stiffness over a certain interval [74].

Permanent magnets are an easy and reliable way of obtaining negative-stiffness springs. By using magnets poles, it is possible to tune operation range where the stiffness becomes negative [75]. A common setup is to locate magnets in unstable equilibrium point by facing equal poles, north against north and south against south, as shown in Figure 5(a). In this way, magnet repulsion will act with negative stiffness for displacements above and below the preload equilibrium point, compensating coil spring positive stiffness of the and thus, minimizing effective stiffness in the operation point (Figure 5(b)). By minimizing dynamic stiffness, the resonance frequency is lower and therefore, vibration damping capacity in higher frequencies is enhanced.

The main advantage of negative-stiffness dampers is that the resonance frequency is lower; therefore, it is very suitable not only for low-frequency vibration damping but also to enhance damping in higher frequencies significantly. By using magnetic negative springs, the system can be more reliable and long-lasting since one of the main problems of mechanical negative springs is the fatigue of the structures. Magnets will not suffer from fatigue or permanent plastic deformation; therefore, creating negative springs through magnetic forces is worthwhile. Application fields for magnetic negative-stiffness dampers (MNSD) are the same as those for negative-stiffness dampers made with structural elements: precision manufacturing, optical and scientific instrumentation, vehicles seat, and rotary machinery.

Among the most interesting developments, we found a magnetic vibration isolator with the feature of high-static-low-dynamic stiffness which is developed in [76, 77]. The device was constructed by combining a magnetic negative-stiffness spring with a spiral flexure spring for static load support. The magnetic spring comprised three magnetic rings configured in attraction, and it was used to reduce the resonant frequency of the isolator. Experimental results demonstrated that the magnetic negative-stiffness spring can reduce the resonant frequency in more than a half while it can expand the isolation frequency band.

Two novel designs of negative-stiffness dampers based on magnetism were designed, optimized, manufactured, and tested in [78, 79]. The two designs from those studies can efficiently integrate negative stiffness and eddy current damping in a simple and compact design. The proof-of-concept experiments were conducted through the scaled prototypes cyclic loading on a vibration machine. Nonlinearity in negative stiffness was observed in both configurations. Nonlinear problems in the vibratory system are hard to solve analytically, and many efforts have been

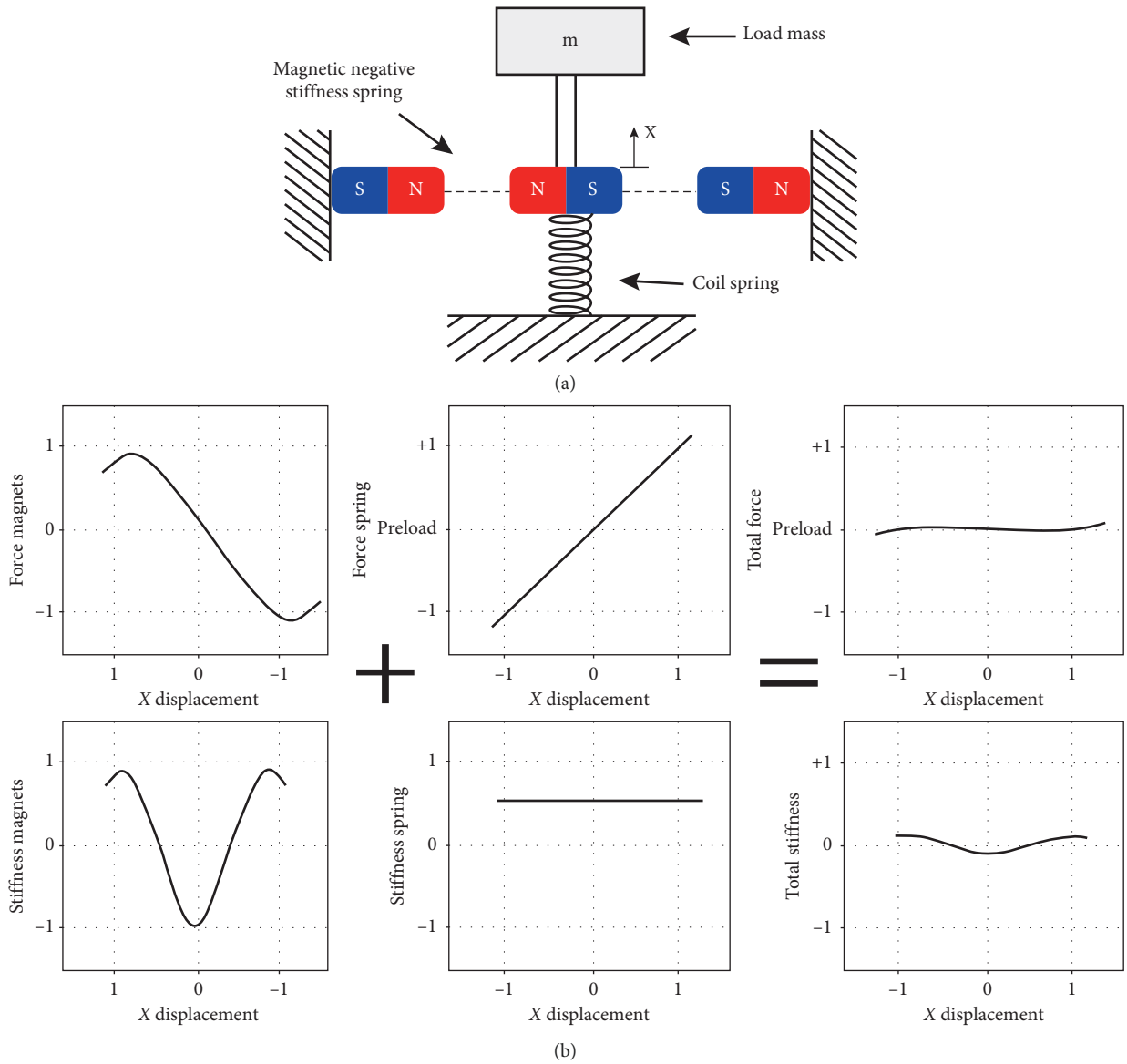

Figure 5: (a) Magnetic negative-stiffness spring configuration with coil spring for static forces. (b) Normalized forces and stiffness against normalized displacements for magnetic spring, coil spring, and its addition.

devoted to its solution [80–82]. Designs in [78, 79] exhibited the hardening and softening patterns of negative stiffness, respectively, with increasing displacement. Compared with the existing designs of negative-stiffness systems, unique features of the proposed designs include symmetrical negative-stiffness behaviour; integrated damping characteristic; and a compact design that can be installed in any direction. Proposed designs have a great potential to replace semiactive or active dampers in diverse vibration suppression or isolation applications.

To suppress rotor systems vibration, a vibration absorber combining together negative stiffness and positive stiffness is proposed in [83]. Firstly, the magnetic negative-stiffness producing mechanism using ring-type permanent magnets is presented and negative-stiffness characteristics are analysed. Then, absorber-rotor system principles and nonlinear dynamic characteristics are studied numerically. Experiments are carried out to verify the numerical conclusions. The results show that the proposed vibration absorber is effective in order to suppress the rotor system vibration, the negative-stiffness nonlinearity affects the vibration suppression effect, and the negative stiffness can broaden the effective vibration control frequency range of the absorber.

A special application of magnetic negative-stiffness dampers has been found to isolate low-frequency seismic noise from the ground in the fields such as gravitational wave detecting [84, 85]. The ultra-low-frequency isolator is made by a magnetic spring composed of a pair of ring magnets in parallel with the conventional pendulum. The magnetic spring can produce magnetic torque to cancel the gravitational torque of the pendulum and hence to reduce

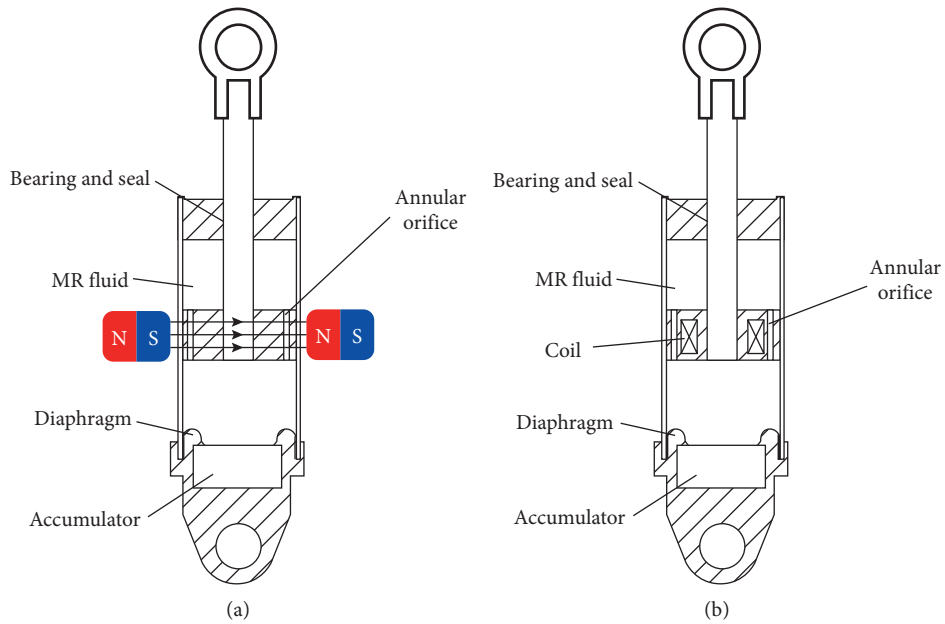

Figure 6: Permanent magnet-based magnetorheological damper (a) compared with active magnetorheological damper (b).

the resonance frequency. In this case, resonance frequency can be up to ten times lower, hence multiplying low-frequency vibration damping capacity.

MNSD can significantly decrease the resonance frequency while keeping the same static load capacity, therefore enhancing vibration damping capacity in higher frequencies.

3.4. Permanent Magnet-Based Magnetorheological Dampers. Similar to passive hydraulic dampers, a magnetorheological (MR) damper consists of a fluid that moves between different chambers via small orifices in the piston, converting "shock" energy into heat. However, in an MR damper, an electrical circuit is introduced in the piston assembly. As electrical current is supplied to the damper, a coil inside the piston creates a magnetic field and instantaneously changes the properties of the MR fluid in the piston annular orifice. Consequently, the dampers resistance can be continuously changed in real time by modulating the electrical current to the damper. Magnetorheological dampers are typically active elements since they need current circulating through the coils in order to generate the magnetic field [86–90]. This goes against passive devices benefits promoted in this article, so in this section, we will only focus on a passive special type of MR damper which is the permanent magnet-based MR dampers (PMRD).

Electromagnetic coils to generate the magnetic field intensity from the input current activate most of the proposed or developed MR dampers. So, to obtain the desired damping force, various control-related devices such as current amplifiers, signal converters, and signal processors are absolutely required. Furthermore, coil and wire modules of the current make the damper structure more complicated and difficult to assemble. In order to improve the commercial feasibility, self-powered MR dampers have been developed featuring energy harvesting by piston movement. However, they are not convincing powerless options. In this sense, in PMRD, damping force is tuned by a permanent magnet instead of electromagnetic coil circuits typically used to drive MR dampers [91]. PMRD consists of a ferromagnetic piston that is magnetically activated through a permanent magnet external motion (Figure 6). Damping force variation of PMRD is realized by the magnetization area or magnet flux dispersion variation, not by the input current magnitude. Thus, the PMRD input variable is totally different from the conventional MR damper.

The major advantage of PMRD is the specific damping coefficient magnitude since they can reach values as high as oil viscous dampers while keeping a tuning capacity of their performance. In comparison, with active MR dampers, the reduced power consumption is the second main benefit when PMRD was chosen for certain applications. The main drawback for this kind of devices is that most of them need magnetorheological fluid for its operation, eliminating all the benefits associated to oil-free devices, for example, cleanness, maintenance free, and reliability. In fact, PMRD are hybrid devices between oil dampers and PEDVDI. The fields of application for this type of device are mainly automotive elements like dampers [92], clutches [93, 94], or brakes [95] and also prosthetic mechanical systems. However, most of studies found in the literature are still laboratory prototypes, and as far as we are concerned, there is no commercialized device based on this technology.

A novel type of tunable magnetorheological PMRD damper based only on the location of a permanent magnet incorporated into the piston was designed, built, and tested in [96]. It was observed that the damping force reached up to 390 N according to the piston location. The maximum damping coefficient is 21428 Ns/m for an estimated mass of 1 kg. To reduce the nonlinearity of on the magnet location, a modified structure of the sidebar was utilized in the piston. It

Table 2: Summary and main conclusions of PEDVDI review.

Type	Fields of application	Advantages	Drawbacks
ECD	Aerospace, civil engineering, precision instrumentation, robotics, and automotive	Contactless damping, best specific damping coefficient, cleanness, temperature resistant	Magnetic contamination and heat generation
EMSD	Microvibration and low-frequency energy harvesting	Dissipation transportable and energy harvesting	Performance restricted to resonance frequency
MNSD	Precision manufacturing, optical and scientific instrumentation, vehicles seat, and rotary machinery	Reduced resonance frequency, enhancement of damping at higher frequencies, and easy manufactory, reliability	No direct damping capacity
PMRD	Automotive elements like dampers, clutches, or brakes	High specific damping coefficient	Oil contamination
Type	**Characteristic parameters**	**Development level**	
ECD	Damping coefficient from 1 to 35000 Ns/m; specific damping coefficient between 1000 and 2000 Ns/m·kg	Commercialized products	
EMSD	Damping coefficient from 1 to 10000 Ns/m; energy harvesting density from 0.1 to 10 mW/cm^3	Laboratory demonstrator	
MNSD	Reduced resonance frequency up to ten times	Laboratory demonstrator	
PMRD	Damping coefficient from 1 to 50000 Ns/m; specific damping coefficient between 10000 and 20000 Ns/m·kg	Laboratory demonstrator	

was experimentally shown that a linear variation of the damping force can be obtained based on the sidebars curved shape. It was also reported in a previous study on this damper that its response time is relatively slow compared with a conventional MR damper using an electromagnetic coil [97]. It should be remarked that the damping force can decrease a lot, perhaps due to the slow response time of the magnet when the excitation frequency is increased.

Sato [98] presents a power-saving magnetizing device for magnetorheological fluids. This device encompasses a permanent magnet for magnetizing the device, instead of an electromagnet that consumes electric power. The permanent magnet applies a magnetic field to the device through a specially designed magnetic yoke. The field intensity can be controlled by moving the magnet. When the magnetic field is controlled by a permanent magnet, thrust that attracts the magnet into the yoke normally acts on the magnet and consumes the power holding and moving the magnet.

As stated, PMRD has also been oriented for its applications in prosthetics. The development presented in [99] is a special damper mechanism proposed to make a prosthetic leg which can derive from on mode to off by using permanent magnet only. The mechanism design is undertaken, and the damping force is analysed in order to validate the effectiveness of the proposed damper system for the patient's motion without the control device. The system can provide up to 1500 Ns/m when it is "on" while the "off" damping coefficient is not larger than 55 Ns/m, demonstrating a very high variation capacity given by the permanent-based actuation.

PRMDs are hybrid devices combining the high specific damping capacity of oil dampers with the electromagnetic damping modulation capacity. The main drawback for this type of devices is that they need magnetorheological fluid for its operation, eliminating all the benefits associated to oil-free devices, for example, cleanness, maintenance free, and reliability.

4. Conclusion

Passive electromagnetic devices for vibration damping and isolation are becoming a real alternative to traditional mechanical vibration and isolation methods. Passive devices are designed and manufactured to respond in a certain manner against vibration without the need of active feedback and control. These types of devices present good damping capacity, lower cost, null power consumption, and higher reliability.

We have presented e a general description of all types of passive electromagnetic devices applied in vibration damping and vibration isolation. Those devices have been categorized as follows: eddy current damper (ECD), electromagnetic shunt damper (EMSD), magnetic negative-stiffness damper (MNSD), and passive magnetorheological damper (PMRD). We have analysed their topologies and pros and cons in the fields of applications, and we have given some characteristic parameter values. All this information has been summarized in Table 2 (advantages and drawbacks are described comparatively with other types of PEDVDI).

In addition to the technology review part itself, a general introductory section relates main considerations and design key parameters that any engineer, electrical or mechanical, needs to know for a deep comprehension of any PEDVDI performance.

Therefore, the article may be also used as a design guide for specific applications. Moreover, as it describes the general design for each technology, it can be used as a starting point for new designs. A practical list of key considerations when designing electromagnetic dampers and isolators is given in the general design considerations of electromagnetic dampers and isolator section.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

The authors would like to recognize the work of Alba Martínez Pérez in the preparation of figures.

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