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Review

A Review of Linear Compressor Vibration Isolation Methods

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Abstract: Linear compressors exhibit high compression efficiency and low noise characteristics, showcasing broad application prospects in various fields such as aerospace, medicine, household appliances, and more. However, due to the complexity of their structures and operation, the issue of vibration isolation in linear compressors has long been a research challenge within the industry. Addressing this challenge, this paper provides an overview of vibration isolation optimization methods for linear compressors. It delves into the discussion of different vibration sources in linear compressors and their respective measurement techniques. By integrating both single degree of freedom (SDOF) and multiple degree of freedom (MDOF) vibration isolation models, this paper describes both active and passive vibration isolation methods tailored to linear compressors. Furthermore, a feasible optimization approach is proposed. Finally, the paper offers insights into the developmental potential and feasibility of vibration energy recovery strategies.

Keywords: linear compressor; vibration isolation optimization; vibration isolation model; vibration energy recovery

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1. Introduction

Thanks to the invention of NdFeB permanent magnets in the 1980s, linear compressors have developed vigorously in the past decade. Compared to traditional compressors driven by rotating motors, they boast advantages such as a compact structure, an absence of friction loss due to crank-connecting rod mechanisms, low energy consumption, and high efficiency, and are suitable for air compressors, vehicle-mounted air conditioning compressors and Stirling refrigerators [1]. Notably, the overall efficiency of linear compressors is 20% higher than that of traditional compressors [2]. Due to their versatility, linear compressors demonstrate superior performance in compression efficiency across various fields, including aerospace, medicine, and household appliances. However, the vibration issues arising from their higher reciprocating motion speeds can affect the compressor's reliability and lifespan. Therefore, optimizing compressor vibrations while ensuring compression efficiency has become a research hotspot in the industry. Linear compressors can be applied in refrigeration systems across different spaces [3]. Initially, the research and development of linear compressors focused mainly on refrigeration systems in space or infrared fields, such as the use of linear Stirling cryocoolers in satellites [4]. As our understanding of linear compressors deepened, their application gradually expanded from the military to civilian domain starting in the 1990s, driving the development and research of household refrigerators [5]. Since 2000, LG Electronics in South Korea has been producing linear compressors for household refrigerators [6].

The types of linear compressors are categorized primarily based on the linear motors used for actuation, encompassing electromagnetic vibration compressors (i.e., reciprocating piston compressors driven by linear oscillating motors), linear electric compressors (i.e., compressors driven by composite secondary linear motors), and compressors driven by linear stepper motors [7]. Furthermore, linear compressors can be classified into moving-coil linear compressors, moving-iron linear compressors, and moving-magnet linear compressors, depending on the different moving components of the linear motors [8]. At present, research work aimed at optimizing the performance of linear compressors covers many aspects, including compressor component optimization technology, lubrication and sealing technology, and vibration reduction and noise suppression technology [9]. However, despite many efforts in these fields, the material characteristics and cost constraints are still the key factors affecting further development. For example, in the optimization of compressor components, the high cost of high-performance materials has become a bottleneck hindering their large-scale application. In terms of lubrication and sealing technology, it is also challenging to find efficient and economical solutions. Therefore, it is particularly critical to develop effective vibration and noise reduction technology. Researchers' efforts in this field mainly focus on structural stiffness optimization, vibration isolation measures, and the application of sound absorption materials and noise control technology. Through continuous improvement and in-depth research, it is expected the vibration and noise level of the linear compressor will be significantly reduced and that the operating stability of the equipment and the comfort of users will be improved, thus promoting its application in more fields [10,11]. The optimization technology and existing problems associated with linear compressors are shown in Figure 1.

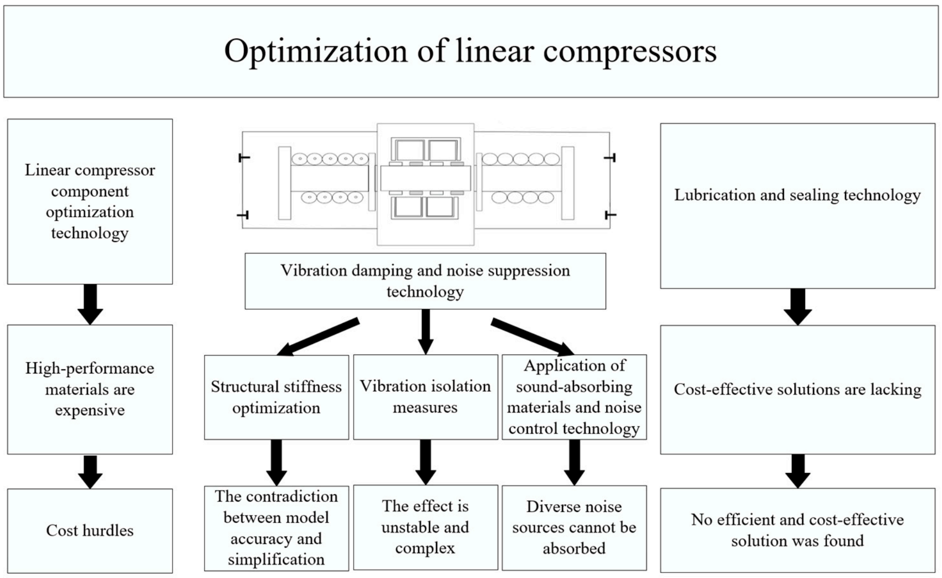


Figure 1. Optimization of linear compressors.

Addressing the aforementioned research challenges, this paper elaborates upon vibration control and optimization techniques for linear compressors. In the subsequent section, we introduce the vibration sources of linear compressors and their measurement methods. Section 2 discusses the research of single degree of freedom (DOF) and multi-DOF vibration isolation models for linear compressors, as well as active and passive vibration isolation. Section 3 explores effective optimization methods for vibration control in linear compressors. Lastly, Section 4 discusses the challenges and opportunities facing future research on vibration control in linear compressors.

2. Vibration Source Analysis and Measurement Methods

2.1. Vibration Source Analysis

Vibration and noise are two interrelated phenomena in linear compressors. Typically, radiated noise is influenced by vibrating surfaces, with the noise source and vibration source being the same [12]. The vibration sources in linear compressors are multifaceted, encompassing factors such as the electromagnetic drive, mechanical design, gas dynamics, and system damping. Considering these factors, the vibration sources of linear compressors can be classified into electromagnetic vibration, mechanical vibration, and spring resonance.

2.1.1. Electromagnetic Vibration

Linear compressors, particularly those utilizing electromagnetic vibration, primarily rely on the principles of electromagnetic force and mechanical resonance. In these systems, the electric motor resonates under the influence of springs and directly propels the piston in a reciprocal linear motion, classifying them as oscillating linear motors. Due to the excessively high oscillation frequency of the motor within the compressor, it can lead to excessive vibrations throughout the entire linear compressor. Therefore, the primary source of vibration in electromagnetic vibration compressors is electromagnetic in nature. The electromagnetic force (EMF), which serves as the main source of vibration for the linear motor housing, can be categorized into three main types: the force acting on the surface of the stator teeth (F_S), the force acting on the stator windings (F_W), and the magnetostrictive force (F_M) [13]. A schematic diagram of the structure of a linear compressor during operation is shown in Figure 2.

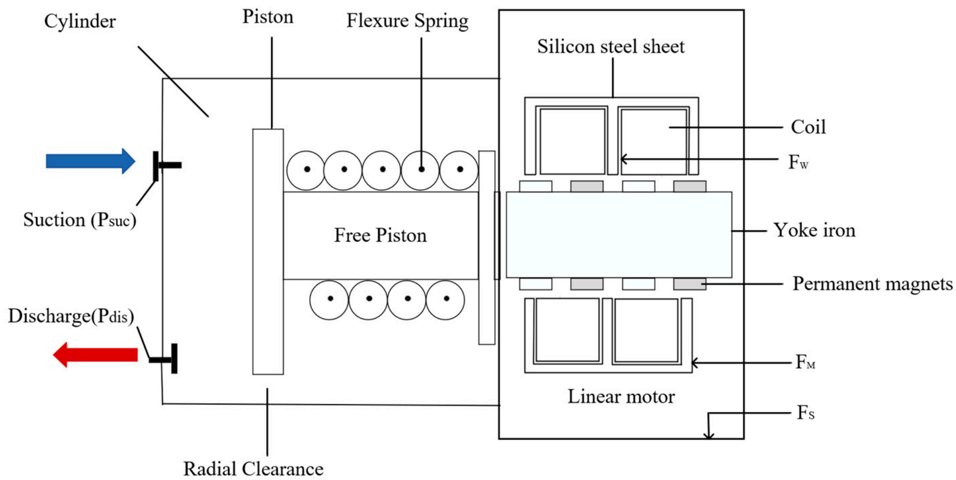


Figure 2. Linear compressor structure.

1. The forces acting on the surface of stator teeth

Some vibrations are generated by the forces acting on the stator teeth. These forces are perpendicular to the surface of the iron, and in the motor, there are both tangential (axial) and radial forces [14]. Both tangential and radial forces acting on the stator teeth can cause vibrations in the internal motor of the linear compressor, which in turn leads to vibrations throughout the compressor. For linear compressors, the internal motor, under the action of electromagnetic forces and resonant springs, causes the piston to perform a reciprocating linear motion. Compared to tangential forces, radial forces lead to more significant vibrations in linear compressors.

2. The forces acting on the stator windings

Sang-Ho Lee et al. [15] proposed a method to improve the resonance frequency of the motor stator by changing the connection thickness of the stator and the thickness of the yoke, thus reducing the vibration. The resonance frequency of the motor stator is related to the total structural stiffness and the overall mass, with the relationship expressed as follows:

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \tag{1}$$

where f represents the resonance frequency of the motor stator in the linear compressor; K is the total stiffness of the stator system; and M is the overall mass of the stator.

3. Magnetostrictive force

Magnetostrictive force (MF) is generated in ferromagnetic materials when they are subjected to an external magnetic field. Severe saturation of the stator due to the influence of rapidly changing magnetic fields may lead to a significantly larger MF. Considering the linear elastic behavior, the total strain (ε) is the sum of the elastic strain (ε_{e}) and the strain (ε_{m}) caused by the MF. The total strain (ε) is related to the overall vibration of the linear compressor. Within the range of elastic deformation, the larger the total strain (ε), the greater the vibration amplitude. Mininger et al. [16] developed a strain model based on magnetostrictive force, linking the magnetostrictive strain (ε_{m}) to the magnetic induction intensity (B). Therefore, based on Hooke's Law, the magnetostrictive effect is considered in the mechanical behavior, and its strain formula is as follows:

$$\sigma(B,\varepsilon) = C * (\varepsilon - \varepsilon_{\mathrm{m}}(B)) \tag{2}$$

$$\varepsilon_{\mathrm{m}}(B) = \sum_{n=0}^{N} \beta_n B^{2(n+1)} \tag{3}$$

where $\sigma(B,\varepsilon)$ is the stress depending on the magnetic induction intensity B and the overall strain ε; $\varepsilon_{\mathrm{m}}(B)$ is the magnetostrictive strain (m denotes magnetostriction); and C is the Young's modulus. β_n is the coefficient of the model given by Equation (3).

The easiest model is quadratic ($N = 0$), valid only in the case of a low frequency. Higher orders of the model allow the consideration of the saturation of the ferromagnetic materials. For the current application, the order of the model is 10 ($N = 4$), and the coefficients β_n of the magnetostriction strain model are deduced from experimental curves, with classical Fe-Si materials.

Therefore, the vibration response of the motor housing under instantaneous excitation cannot be ignored, and studying the vibration characteristics of the motor housing under the influence of electromagnetic fields is crucial for researching the vibration of linear compressors [17,18].

2.1.2. Mechanical Vibration

The air valve is one of the most crucial components in the development of compressors. The valve of a linear compressor is installed on top of the moving piston, and it opens when the piston moves. The valve bends under the action of a pressure difference and inertial forces, colliding with the piston and discharge valve, thus generating vibrations. If the piston collides compressor body for a long time or many times, the compressor body will be damaged, meaning that the compressor will not be able to work properly or even stop running. The air valve directly affects the working efficiency and reliability of the linear compressor, and it is also one of the root causes of vibration [19].

When the linear compressor piston approaches the stroke limit position, due to the change in the direction of movement, the piston speed and acceleration will be drastically changed, thus generating a large vibration. And in the linear compressor reciprocating the movement process, the piston will be displaced, making it difficult to accurately monitor the displacement of the piston in real time, which increases the possibility of vibration caused by contact and collision between the valve plate and the compressor body. Jian S et al. [20] found that a dual-piston system can significantly reduce piston deviation. Therefore, a dual-piston linear compressor can minimize its own vibrations to a certain extent [21]. To better reduce piston deviation and minimize the vibrations of the linear compressor, a dual-cylinder opposed-piston configuration is adopted in the installation of the dual-piston system on the linear compressor. The structure of the dual-cylinder linear compressor is shown in Figure 3.

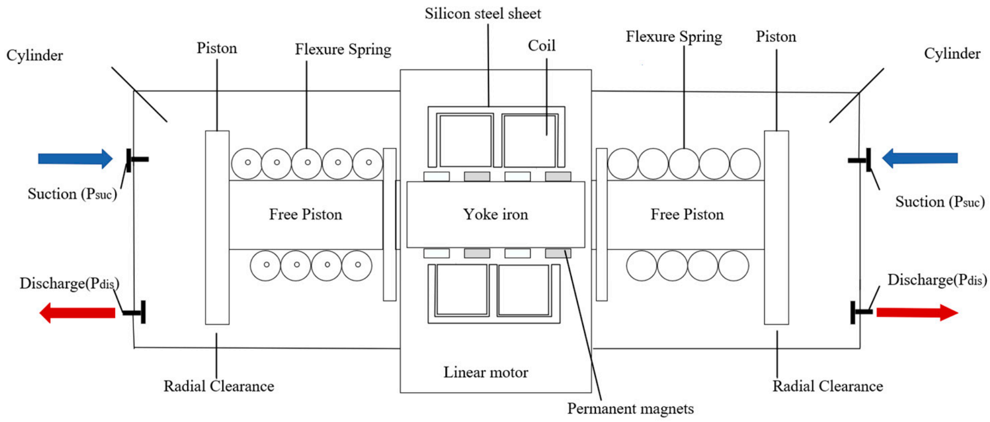


Figure 3. Dual-cylinder linear compressor structure.

2.1.3. Spring Resonance

In a linear compressor, the internal motor is driven by electromagnetic force and resonant springs to push the piston in a reciprocating linear motion. During operation, the piston produces resonance to minimize the drive current and resistance loss. Resonance is determined by the moving mass and spring stiffness, which consists of two components during the compression process: the mechanical spring and the "gas spring" effect [22]. The "gas spring" effect refers to the change in the gas spring within the cylinder due to changes in the suction/discharge pressure. The compressor, due to its structural characteristics, generates airflow pulsations. The periodic changes in cylinder pressure and velocity caused by intermittent intake and exhaust can lead to gas column resonance and pipe vibration. Since the "gas spring" effect is nonlinear, the stiffness of the mechanical spring must be carefully considered. While producing resonance, it is essential to minimize the transmission of vibrations to the compressor body.

2.2. Vibration Testing Methods and Evaluation Indicators

To better reduce the vibration of linear compressors while ensuring compression efficiency, it is necessary to conduct vibration tests on them. Vibration testing primarily involves monitoring the operating state of linear compressors. This is achieved by continuously monitoring the vibration levels of critical components (such as motors, pistons, and housings) using vibration sensors (e.g., accelerometers, velocity sensors) under the actual operating conditions of the linear compressor. The most common testing method involves installing vibration sensors on the linear compressor to measure both axial and radial displacements and accelerations separately. Displacement and acceleration are the most commonly used parameters for quantitatively assessing vibrations [23]. Rijpma et al. [24] measured the vibrations of a linear compressor using accelerometers and preamplifiers placed on the compressor, achieving millimeter-level accuracy and high sensitivity. Olivieri E et al. [25] established a vibration measurement system comprising a high-sensitivity PCB393B04 vibration accelerometer, a PCB-480E09 signal conditioner, and a 16-bit National Instruments DAQ-6218 data acquisition module.

3. Vibration Isolation Methods

3.1. Vibration Isolation Model

3.1.1. Single Degree of Freedom (SDOF) Model

When it comes to rapidly assessing the fundamental characteristics of an overall linear compressor system, the Single Degree of Freedom (SDOF) model can be employed. The SDOF model represents a simplified holistic system model, which treats the entire system as a single mass point and only considers vibrations along a single direction. This model is well suited to investigating the basic properties and dynamic responses of the overall system, such as the natural frequency, damping characteristics, and vibration amplitude. The SDOF model is generally simpler, facilitating easier calculation and analysis, making it ideal for rapid evaluation and preliminary design stages. The SDOF vibration model of a linear compressor is illustrated in Figure 4.

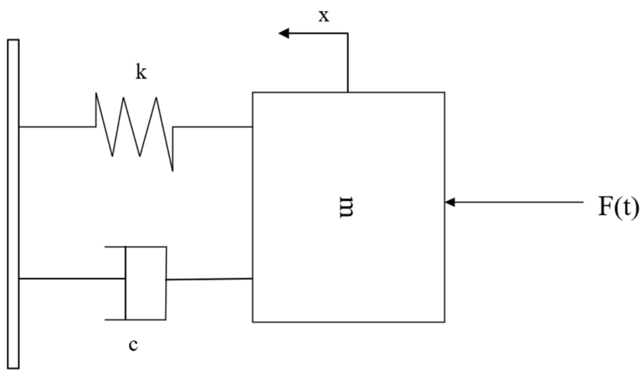


Figure 4. Single Degree of Freedom (SDOF) vibration model.

The equation of motion for a SDOF vibration system is typically established based on Newton's Second Law of Motion, and it takes the following form:

$$m\ddot{x} + c\dot{x} + kx = F(t) \tag{4}$$

where x is the displacement of the system, which is the deviation of the mass block from its equilibrium position. $\ddot{x}$ and $\dot{x}$ are, respectively, the second derivative (acceleration) and the first derivative (velocity) of the displacement with respect to time. c is the damping coefficient. k is the spring stiffness coefficient. $F(t)$ is the external force, which can be a function of time and used to describe the external excitation.

For a linear compressor, the moving mass (m) in the single degree of freedom (SDOF) model represents the piston and its attached components. The stiffness (k) in the SDOF model, which arises from the elastic restoration of the system under compression or expansion forces, encompasses the stiffness of the compressor's resonant spring and the elastic modulus of the gas within the cylinder, among other factors. Additionally, the damping coefficient (c) is usually related to viscous force. This coefficient is related to the resistance caused by friction and other resistance when the piston moves in the cylinder.

3.1.2. Multi-Degree of Freedom (MDOF) Model

When a more detailed and accurate analysis of the vibrational behavior of the entire linear compressor system is required, a multi-degree of freedom (MDOF) model becomes more suitable. The MDOF model considers the vibrations of multiple key components (such as springs, dampers, control arms, etc.) within the overall system and takes into account their interactions. This model can more accurately predict the dynamic response, resonant frequencies, modal coupling, and other characteristics of the entire system. The MDOF model typically requires more complex mathematical modeling and computation, making it suitable for in-depth studies of the vibrational characteristics and optimization design of the overall system. According to the linear compressor studied in 2018 [26], it is simplified into a dynamic model with two degrees of freedom (TDOF), as shown in Figure 5.

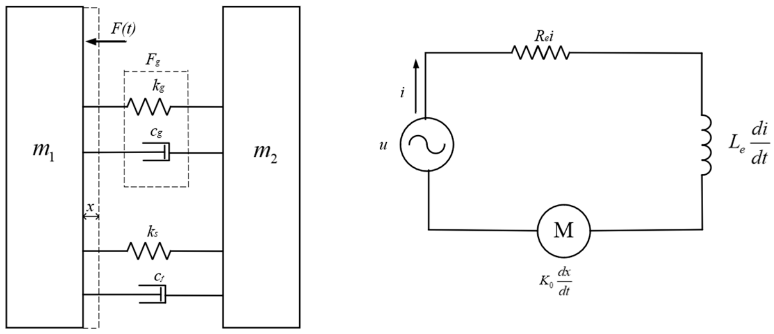


Figure 5. TDOF vibration models.

The model of a two-cylinder linear compressor can be represented as follows:

$$(m_1 + m_2)\ddot{x} + c_f \dot{x} + k_s x + F_g = F(t) \tag{5}$$

$$R_e i + L_e \frac{di}{dt} + K_0 \dot{x} = u \tag{6}$$

$$F(t) = K_0 i \tag{7}$$

$$k_g = \frac{1}{\pi x} \int_0^{2\pi} F_g \cos\theta d\theta \tag{8}$$

$$c_g = \frac{1}{\pi 2\pi f x} \int_0^{2\pi} F_g \cos\theta d\theta \tag{9}$$

where m_1 and m_2 are the equivalent mass of the left–right moving magnet assembly. x represents the displacement of the system, which is the deviation of the mass block from its equilibrium position. $\ddot{x}$ and $\dot{x}$ are, respectively, the second derivative (acceleration) and the first derivative (velocity) of the displacement with respect to time. c_f is the viscous damping coefficient. k_s is the support spring stiffness. F_g is the gas force. $F(t)$ is the electromagnetic force output by the motor inside the compressor, which can be a function of time and is used to describe external excitation. i is the current. R_e is the equivalent resistance. L_e is the equivalent inductance. u is the voltage. k_g is the equivalent stiffness of the gas force. c_g is the equivalent damping coefficient of the gas force. θ is the angular displacement. f is the working frequency.

3.2. Passive Vibration Isolation

Passive vibration isolation technology refers to the use of passive techniques to suppress vibrations. It is a means of reducing or isolating the transmission of vibrations through mechanical components such as springs, dampers, or other elastic materials, without relying on external energy sources. A passive vibration isolation system harnesses the properties of physical elements (like springs and dampers) to absorb or divert vibrational energy. When external vibrations act on the system, these elements undergo deformation, thereby consuming the vibrational energy and reducing the amplitude of the vibrations transmitted to the protected equipment. For linear compressors, implementing vibration isolation at the source primarily involves installing vibration isolators between the vibration source and the supporting structure. Common passive vibration isolation techniques primarily include isolation through spring isolators, rubber isolators, magnetorheological fluid isolators, and composite damping isolators (such as those utilizing air, asphalt, or composite damping materials).

3.2.1. Spring Isolators

Spring isolators typically consist of elastic elements and damping elements. The rational design of these two components enables the isolators to support heavy loads statically while exhibiting low dynamic stiffness [27].

Cam–roller–spring mechanism spring isolators stand out due to their low-cost production advantages, and at the same time, show an excellent adjustment performance and good lateral vibration isolation ability, which effectively meets the vibration control needs in a variety of application scenarios [28,29]. In transportation, these isolators can reduce vibrations during vehicle driving, improve ride comfort, and protect vehicle structures from damage caused by long-term vibrations. In the energy field, such as in wind power generation and hydropower generation, cam–roller–spring mechanism spring isolators can reduce the wear and failure of equipment due to vibration, and improve the operation efficiency and reliability of equipment. Zhou [30] designed a quasi-zero stiffness (QZS) vibration isolator with cam–roller–spring mechanisms (CRSMs), excelling in the 3–10 Hz range over linear systems. Yao [31] introduced a high-static–low-dynamic stiffness isolator (HSLDS) with adjustable cam, lowering the isolation frequency. Nevertheless, this proposed vibration isolation mechanism runs the risk of disengaging the rollers from the cams in the case of high vibration.

Negative-stiffness spring isolators are known for their excellent dynamic response, high static load carrying capacity and good tunability, which give them significant advantages in a wide range of vibration control applications [32–34]. Based on the above advantages of negative-stiffness springs, Wu [35] developed a magnetic spring with negative stiffness (MS-NS) isolator for heavy loads, combining positive and negative stiffness springs and reducing the natural frequency from 13.43 Hz to 6.07 Hz at 1.35 kg. However, these negative stiffness springs have the following drawbacks: it is difficult to adjust the stiffness, and the stability of negative-stiffness springs is poor after long-term use. Yan [36] crafted an anti-spring isolator from blade springs, achieving a <3.6 Hz isolation frequency under a 21 kg load, showcasing a superior low-frequency performance. Sugahara Y [37] achieved control over the damping characteristics of air springs by manipulating the variable orifice, effectively reducing the vertical vibrations on the vehicle body floor. This confirmed that controlling the damping of air springs can mitigate vibrations.

To better apply spring isolators to vibration isolation for linear compressors, spring-lever isolators provide excellent vibration isolation and reduce the transmission of harmful vibrations. Their design helps to reduce static distortion, maintain system stability, and achieve high energy recovery, contributing to energy conservation and emission reduction. The most important thing is to use eddy current damping (ECD) technology to further improve the stability and performance of the vibration isolation system [38,39]. Lee [40] proposed a novel linear compressor structure. This involved connecting a set of springs and mass blocks in series behind the resonant spring of the linear compressor, functioning as a dynamic vibration absorber. But the vibration isolator proposed in this study still has problems, such as a high noise level, and needs further improvement.

Yan [41] designed a novel quasi-zero-stiffness vibration isolation (L-QZS-VI) system utilizing the lever principle and incorporating an eddy current damper (ECD) to enhance the vibration isolation performance. Through experimental validation, the experimental peak transmissibility and corresponding frequency of the linear vibration isolator were 16.86 and 1, respectively, while those of the QZS-VI were 14.19 and 0.78, respectively. This indicates that QZS-VI has a better vibration isolation performance. However, this new type of vibration isolator is large and heavy, and is not suitable for some cases where the size of the vibration isolator is strictly required. Nguyen [42] designed an air spring vibration isolator system (ASVIS) based on a negative-stiffness structure (NSS). Through experimental data, they found that compared to traditional air spring isolators, the ASVIS with NSS reduced seat displacement by 77.16%. The variable hole air suspension diagram is shown in Figure 6.

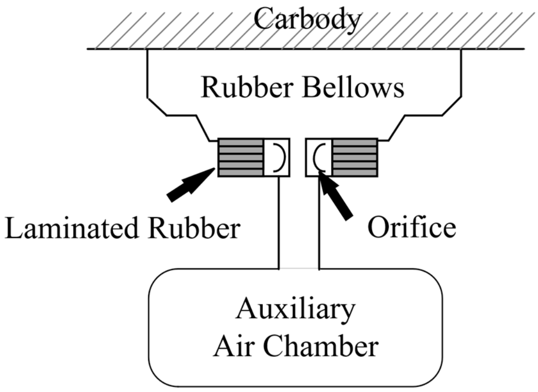


Figure 6. Variable hole air suspension diagram.

3.2.2. Rubber Vibration Isolators

Rubber materials possess immense elastic resilience and high energy dissipation capabilities, making them highly suitable for vibration control [43]. Nitrile-fluoroelastomer isolators show significant advantages, such as their radial stiffness being able to be flexibly adjusted according to actual needs, making these isolators suitable for complex vibration environments such as high-frequency, wide, and dynamically varying loads. In addition, their strong environmental adaptability allows them to maintain a stable performance even in extreme temperatures, humidity and chemical media environments [44,45]. What is particularly outstanding is their excellent mechanical properties, which provide a solid guarantee of vibration and noise reduction. Li et al. [46] designed a novel rubber vibration isolator based on a blend of chlorobutyl rubber (CIIR) and natural rubber (NR). The resonant frequency of this isolator increases with an increasing torque, while both the amplitude and transmissibility decrease. The minimum transmissibility achieved is in the order of 10^{-6}. The annular metal rubber isolator is known for its unique structural design, which is not only exquisite, but also enables the adaptive adjustment of frequency; it can automatically adjust its own performance according to different vibration frequencies to achieve the best vibration isolation effect. In addition, it has the ability to provide broadband vibration isolation, which provides effective vibration isolation over a wide frequency range. Wang Lei et al. [47] conducted research on the radial stiffness calculation and parameter optimization of vehicle annular rubber shock absorbers based on the Mooney–Rivlin model for rubber materials. The experimental data showed that nitrile rubber shock absorbers exhibit a higher radial stiffness in environments ranging from 20 °C to 70 °C.

Additionally, Nahal [48] developed a floating rubber–concrete isolation panel system utilizing high-damping rubber (HDR) to isolate the horizontal and vertical vibrations caused by mechanical or other load sources. The experimental data revealed that, compared to conventional structures, the lateral drift of 1-story, 2-story, and 3-story structures equipped with the floating system was reduced by an average of 87.33%, 62.21%, and 47.08%, respectively; meanwhile, under vertical loading, deflection was reduced by 11.1%. Li et al. [49] designed a negative stiffness vibration isolator (NSMSVI) utilizing a magnetic spring combined with a rubber membrane to achieve an even lower natural frequency. The experimental results showed that the NSMSVI achieved a minimum natural frequency of 1.5 Hz, with a maximum attenuation of −40 dB between 0 Hz and 100 Hz, and a maximum load-bearing capacity at its lowest stiffness. Recently, the thick layer damping rubber bearing (TLDRB) designed by Lu et al. [50] has undergone laboratory testing. The experimental data indicate that at a vertical preload pressure of 12 MPa, the horizontal equivalent stiffness of TLDRB slightly increases, and the dissipated energy increases with the applied displacement. Furthermore, TLDRB exhibits a higher equivalent viscous damping ratio compared to damping rubber bearings. The basic configuration diagram of NSMSVI is shown in Figure 7.

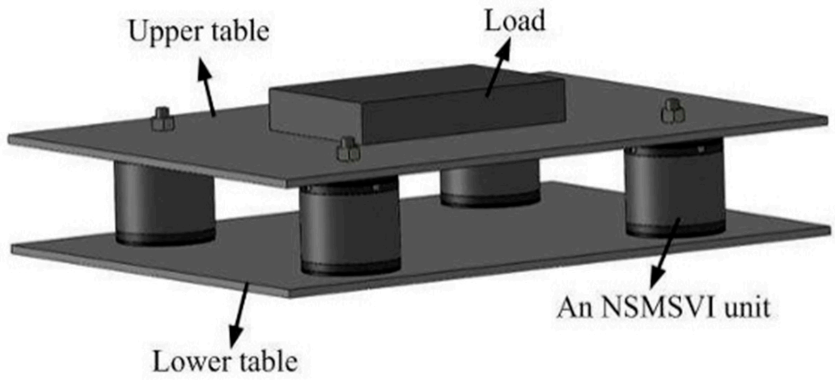


Figure 7. The basic configuration diagram of NSMSVI.

3.2.3. Magnetorheological Fluid Vibration Isolators

Magnetorheological fluid (MRF), with its rheological properties rapidly and reversibly controlled by an external magnetic field, is classified as a smart material. Due to their unique properties, MR fluids have garnered increasing attention in recent years and demonstrated broad application prospects in the automotive industry and in vibration control [51].

The magnetorheological fluid isolator has continuous damping adjustability, which means that it can adjust the damping force in real time according to the actual demand to meet the vibration challenges under different operating conditions [52,53]. In addition, its control mode is simple and intuitive, making it easy to operate and maintain. It is also compact and does not take up too much space, and it thus ideal for space-constrained applications. These advantages provide magnetorheological liquid isolators with a wide range of applications in aerospace, automotive engineering, building shock absorption and other fields. Dogruer [54] improved the basic MRF shock absorber design by designing a novel MRF shock absorber for use in the suspension system of a highly maneuverable multi-purpose wheeled vehicle, which incorporates non-magnetic stainless shims and an accumulator to compensate for the volume changes caused by piston movement, enhancing the basic MRF damper design. The experimental results showed that the use of non-magnetic stainless shims increases the force during the rebound phase, with a maximum power consumption of only 31.5 watts. Sun et al. [55] proposed a novel compact variable stiffness and damping damper for vehicle suspension systems that consists of two coaxial dampers in a compact structure and can independently vary its stiffness and damping. A series of experiments revealed that the stiffness of this shock absorber can vary within a range of 8.7 to 24.5 kN/m, thereby enhancing its vibration isolation capabilities. Although the MR fluid isolator exhibits stability at different temperatures during experimentation, the entire system remains susceptible to temperature sensitivity, MR fluid sedimentation, and poor stability, which may affect its performance.

Magnetorheological elastomers (MREs) are solid analogues of MRF that can effectively overcome the drawbacks of MR fluids [56]. In addition, magnetorheological elastomers also have an adjustable resonant frequency and low requirements for tightness. Chen [57] investigated the damping characteristics of MREs by using a dynamic mechanical analyzer (DMA) to test their properties. The study revealed that the damping ratio of MREs initially increases with an increase in the applied magnetic field, peaking at a critical value of 300 mT before declining, and that a low damping ratio can enhance the vibration reduction effect of MREs. Deng [58] designed an adaptive tuning vibration absorber (ATVA) based on MREs. They conducted experiments to investigate the natural frequency tuning capabilities of the ATVA by varying the applied magnetic field intensity. The experimental results showed that the natural frequency of the ATVA can be adjusted from 27.5 Hz to 40 Hz.

MRF isolators have advantages in energy recovery. Sapiński [59] designed an energy-harvesting linear magnetorheological (EH-LMR) shock absorber. This shock absorber utilizes an electromagnetic energy extractor to recover energy from external excitations and adapts to the excitations by altering its damping characteristics. Experiments have demonstrated that the proposed EH-LMR shock absorber can recover the kinetic energy that traditional shock absorbers typically dissipate. Ge and Liu [60] designed a novel single-tube hybrid energy-supplying magnetorheological shock absorber that combines a linear energy supply with a ball screw energy supply. When a vehicle travels at 40 km/h on a C-level road, the energy-feeding efficiency of the linear energy-feeding shock absorber is 9.3%, while the efficiency of the hybrid energy-feeding shock absorber increases to 14.6%, representing a 57% improvement. The internal structure of a magnetorheological damper is shown in Figure 8.

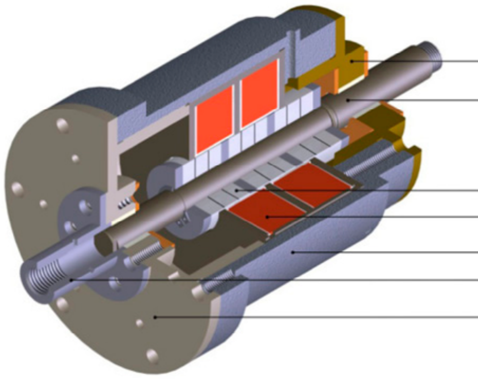


Figure 8. The internal structure of a magnetorheological damper.

3.2.4. Composite Damping Vibration Isolators

In addition to the above three types of isolators, there are other types of isolators. S.H. [61] tested the vibration characteristics of linear compressors and designed a flat plate dynamic isolator. Fan et al. [62] developed three viscoelastic damping materials, namely a bitumen-based damping material, a water-based damping coating and a butyl rubber damping material, to reduce the vibration and noise inside a train or vehicle. Cindy S. Barrera [63] designed a series of experiments to investigate the potential use of eggshell as a natural rubber reinforcement material. Chen [64] developed a single degree of freedom mass–spring–damper model to represent the rubber isolation system and used polynomial functions to describe the spring and damper coefficients related to the relative displacement. The excellent performance of rubber-spring compliant damping isolators is reflected in their high damping characteristics, simple yet efficient design, minimal footprint, and excellent durability that ensures long-term stable vibration isolation. Ren and his team [65] designed a symmetrical metal rubber and wire spring composite isolator and conducted random vibration tests to study the performance of the isolator in overall vibration isolation. The experimental results showed that the peak attenuation of the metal rubber composite isolator at the resonance point was more than five times that of the wire spring isolator, and that the random vibration response of the system had a large degree of attenuation in the frequency range of 0.12 to 2.00 kHz.

The comparison of vibration isolator types is shown in Table 1.

Table 1. Comparison of vibration isolator types.

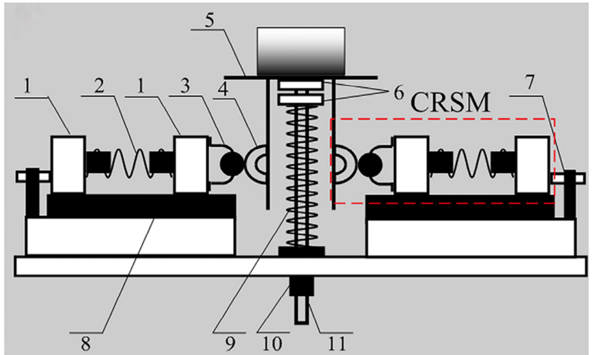
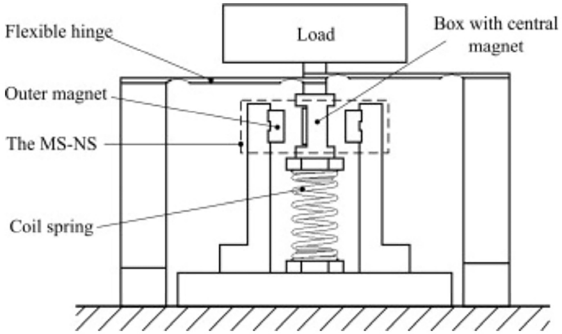
Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Spring Type Vibration Isolators	Cam–Roller–Spring Type		1. Low production cost; 2. Excellent adjustment performance; 3. Excellent lateral vibration isolation;	1. Complex structure; 2. Large installation space; 3. Discontinuity of transfer rate;	[30,31]
	Negative-Stiffness Spring Type		1. Excellent dynamic response; 2. high static load carrying capacity; 3. Tunability;	1. Difficulty in adjusting stiffness; 2. Restricted vibration range; 3. Poor long-term stability;	[35,37,42]

Table 1. *Cont.*

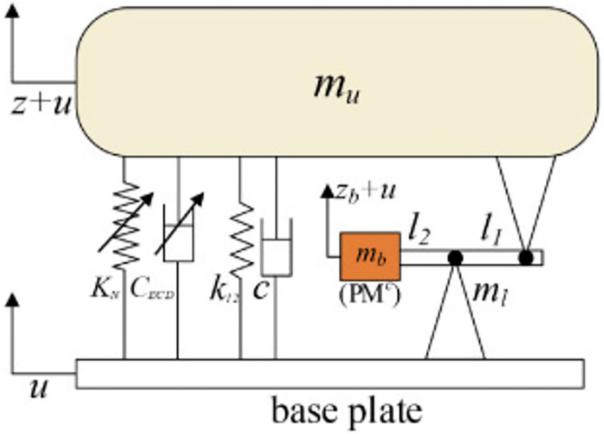
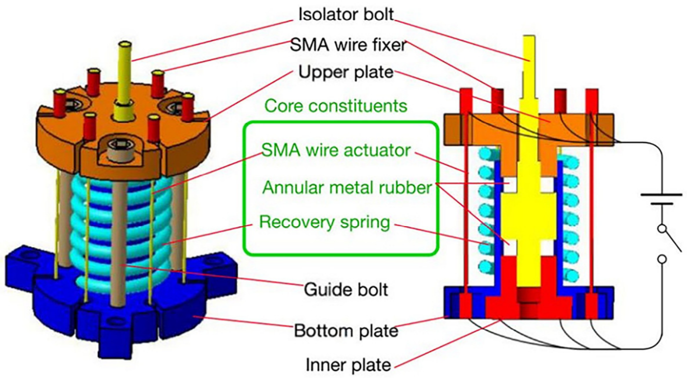
Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Spring Type Vibration Isolators	Spring Lever Type		1. Excellent vibration isolation; 2. Reducing static deformation; 3. Achieving higher energy recovery;	1. Large size and weight; 2. Affected by materials; 3. Environmentally sensitive;	[41]
Rubber Vibration Isolators	Annular Metal Rubbers		1. Simple structure; 2. Free frequency adaptation; 3. Satisfy a variety of use scenarios;	1. Higher production costs; 2. Poor thermal conductivity; 3. Poor durability;	[46]

Table 1. *Cont.*

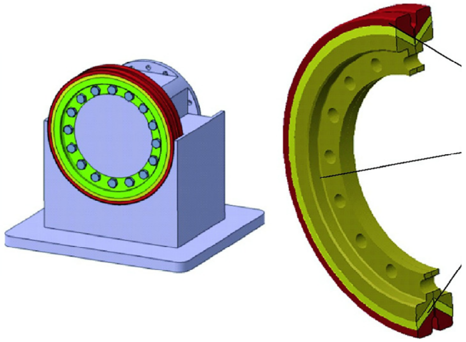
Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Rubber Vibration Isolators	NBR-FKM		1. Adjustable radial stiffness; 2. Strong environmental adaptability; 3. Excellent mechanical properties;	1. Material dependence; 2. Structural complexity; 3. Complex production processes;	[47]

Table 1. *Cont.*

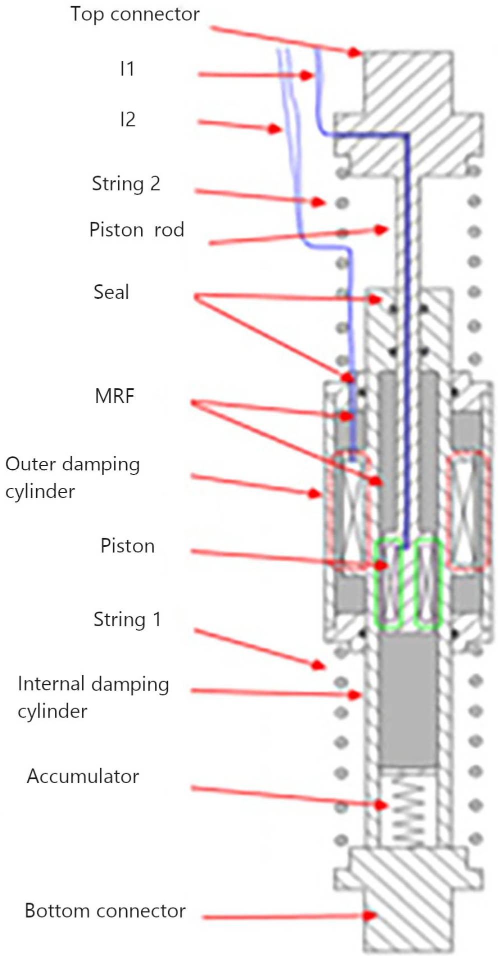
Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Magnetorheological fluid dampers	Magnetorheological Fluid Types		1. Damping is adjustable; 2. The control mode is simple; 3. Compact structure;	1. Magnetic particle deposits; 2. Poor system stability; 3. High production cost;	[51,55,56]

Table 1. *Cont.*

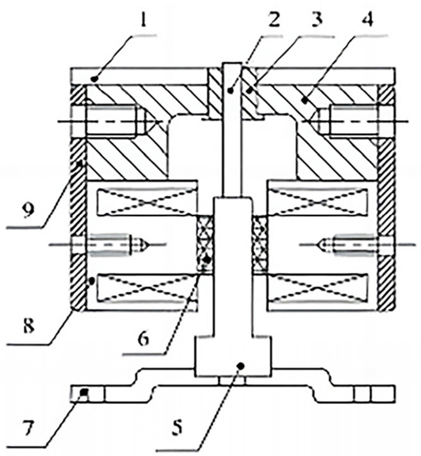
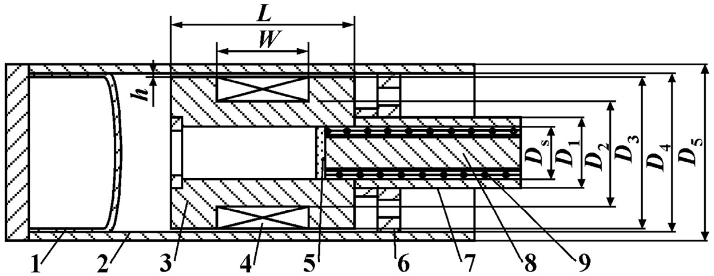
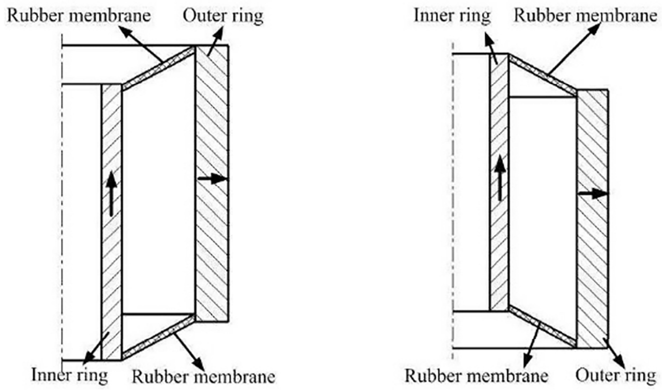
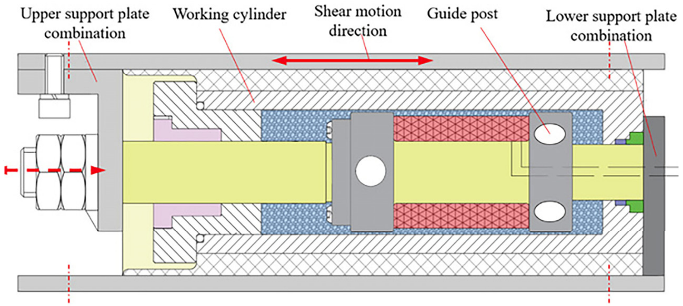
Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Magnetorheological fluid dampers	Magnetorheological Elastomer (MRE)		1. Adjustable resonant frequency; 2. Low requirement for sealing; 3. Compact structure;	1. Slow response at high frequencies; 2. Sensitive to temperature changes; 3. High production cost;	[57,58]
	Magnetorheological liquid type with energy recovery		1. Can achieve energy recovery; 2. No additional sensors required;	1. Heavier damper weight; 2. Poor stability; 3. High production cost;	[60]

Table 1. *Cont.*

Type	Structure	Modelling Diagram	Advantages	Disadvantages	Reference
Composite type damper	Rubber Spring Vibration Isolators		1. High damping performance; 2. Simple structure, small footprint; 3. Good durability;	1. Performance affected by temp; 2. Poor high frequency vibration isolation; 3. Low load-carrying capacity;	[49,65]
	Rubber magnetorheological Isolators		1. Excellent tangential load capacity; 2. Widely adjustable range of damping; 3. Response adjusted quickly;	1. Significant temperature effects; 2. Complex design and high production costs;	[66]

3.3. Active Vibration Isolation

Unlike passive vibration isolation, active vibration isolation control technology encompasses a wide range of interdisciplinary techniques. In a typical active vibration isolation control system, mechanical and electronic components work harmoniously together. The primary elements of any active vibration isolation control system are the mechanical structures that generate unwanted vibrations, sensors to detect these vibrations, a controller that intelligently uses sensor inputs to generate appropriate control signals, and actuators to counteract the effects of disturbances on the structure. Active vibration isolation control refers to the process of applying a certain control strategy based on the detected vibration signals, performing real-time calculations, and then driving the actuators to exert a certain influence on the control target, ultimately achieving the goal of suppressing or eliminating vibrations. Depending on the control algorithm, active vibration isolation control can be broadly classified into two categories: feedforward active vibration isolation control [67,68] and feedback active vibration isolation control [69]. A block diagram of the combined feedforward/feedback control for active vibration isolation is shown in Figure 9.

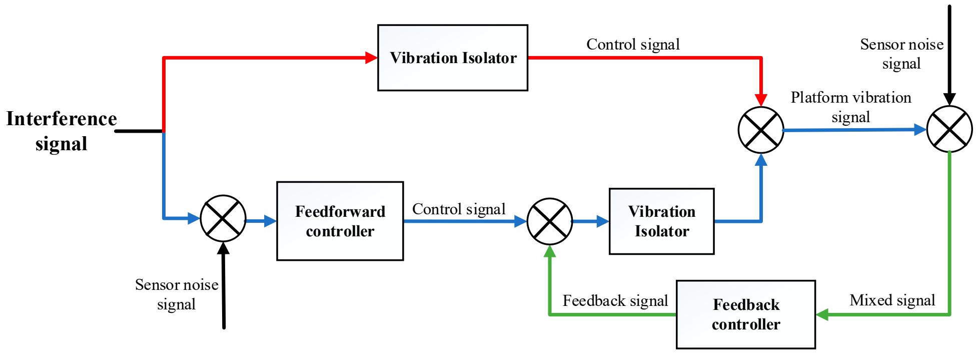


Figure 9. Control signal diagram of combined feedforward/feedback control for active vibration isolation.

3.3.1. Feedforward Active Vibration Isolation Control

Feedforward control is a control strategy that predicts potential changes in the system output based on the system input or disturbance information and proactively adjusts the control inputs before these changes occur [70]. The technology of feedforward active vibration isolation control utilizes the adaptive feedforward principle to inject currents containing higher order harmonics with adjustable phases and amplitudes into the linear compressor motor drive, thereby achieving complete force cancellation. The output of the load sensor (force sensor) is used to measure the higher order harmonic forces, and the desired phases and amplitudes of the injected harmonic signals are estimated in real time, as shown in Figure 10.

Feedforward active vibration isolation control mainly focuses on improving the vibration isolation performance and computational efficiency of the system, and experts at home and abroad [71–75] have continuously improved the vibration isolation efficiency, robustness, and system vibration isolation performance through a variety of means. Elliott et al. [71] investigated the application of multi-channel feedforward control under random disturbances and proposed an instant steepest descent algorithm for adaptive controller tuning, which demonstrated faster convergence compared to traditional algorithms. Zhou et al. [72] designed a feedforward compensation controller utilizing a finite impulse response (FIR) filter and improved the filter-x least mean squares (Fx-LMS) algorithm for gradient estimation. Experiments validated that the proposed method significantly reduced base disturbances. L. van de Ridder et al. [73] designed a Coriolis mass flowmeter integrated with multi-degree of freedom (multi-DOF) active vibration isolation, and achieved over 40 dB of vibration isolation by utilizing an adaptive feedforward control strategy based on the filtered-x least mean squares (FxLMS) algorithm to determine the optimal feedforward controller parameters. However, such feedforward controllers relying on the spring–damper model perform poorly in isolators with low suspension frequencies. In 2015, M.A. et al. [74] proposed a feedforward control strategy that restricts low-frequency control actions, achieving nearly zero phase shift at high frequencies. The feedforward controller implemented through a self-tuning infinite impulse response (IIR) filter can effectively reduce the performance limitations caused by parameter estimation errors while maintaining computational efficiency. In 2018, they introduced another self-tuning generalized finite impulse response (FIR) filter as a feedforward controller [75], which requires many adaptive parameters to accurately estimate low-frequency poles and slightly damped resonances. In the same year, they proposed a multi-input–multi-output (MIMO) disturbance feedforward controller and developed a disturbance feedforward control strategy specifically for active vibration isolation systems with internal isolator dynamics. The research results show that this controller and control strategy significantly reduce vibrations in industrial environments, enhancing vibration isolation performance [76,77]. Eduardo et al. [78] proposed a novel feedforward active vibration control scheme and overcame the issue of undesirable vibrations being introduced by the adaptive controller itself through the proposed CVA-FxNLMS algorithm. Through these studies, research in the field of feedforward vibration isolation control has primarily focused on improving the vibration isolation performance and computational efficiency of systems. By leveraging adaptive algorithms, FIR filters, multi-DOF isolators, self-tuning IIR filters, and MIMO controllers, continuous improvements have been made in vibration isolation efficiency, robustness, and overall system performance.

3.3.2. Feedback Active Vibration Isolation Control

Feedback control is a control strategy that operates by continuously monitoring the actual output of a system and comparing it with the desired output. Based on the difference between the two, known as the error signal, feedback control dynamically adjusts the control input to drive the system output towards the target value. In vibration isolation systems, feedback control is commonly used to reduce system responses caused by external vibrations, such as floor vibrations and mechanical shocks [79]. Feedback active vibration isolation control technology employs classical feedback theory to design linear feedback controllers, with force sensors serving as critical feedback elements. In the feedback active vibration isolation control scheme (as shown in Figure 9), the transfer function of the servo compensator is derived in detail to ensure the implementation of notch filtering at each harmonic frequency point. The control law expression for feedback active vibration isolation control is as follows:

$$I_{b2} = -G_f(\Delta F) \tag{10}$$

where ΔF is the total reaction force acting on the linear compressor housing; G_f is the transfer function of the servo compensator used for vibration control; and I_{b2} is the current generated by the servo compensator.

Feedback active control has significant advantages in suppressing vibrations in different frequency ranges, improving vibration isolation and system robustness. Through theoretical analysis, system modeling, algorithm design, and experimental verification, researchers [80–84] have continuously optimized the control algorithm, and successfully achieved broadband vibration attenuation from a low frequency to high frequency, providing an important technical reference and solution for industrial and scientific research fields requiring high precision and high stability. Wang et al. [80] proposed a Stewart platform equipped with piezoelectric actuators for micro-vibration isolation. In the active control loop, they combined the direct feedback of integrated force with adaptive feedback based on FxLMS, achieving a 30 dB attenuation of periodic disturbances and a 10–20 dB attenuation of random disturbances within the frequency range of 5–200 Hz. Cheng et al. [81] introduced an active control method into quasi-zero stiffness vibration isolators (QZS-VIs) and proposed a time-delay dual-feedback control method that integrates cubic displacement feedback and cubic velocity feedback to enhance the vibration isolation performance of QZS-VIs. The results indicated that the proposed dual-feedback control method can effectively suppress vibrations in the resonance region without compromising the performance in the non-resonance region. Ren et al. [82] presented an adaptive feedback control algorithm that utilizes filters to extract components with slower convergence rates from the error signal and controls them independently for rapid attenuation. Experiments demonstrated that when disturbances contain multiple harmonics, the proposed algorithm achieves better vibration attenuation than traditional LMS-based adaptive feedback. Huang et al. [83] proposed the combination of acceleration feedback with PD position feedback to suppress the transmission of periodic vibrations, thereby improving the performance of active vibration isolation systems (AVISs) equipped with electromagnetic actuators. For a ground-tested vibration isolation mounting system, the vibration isolation performance improved from 24.47 dB to 2.4 dB at the resonance frequency, achieving a −34 dB attenuation at 100 Hz. Xavier Mininger et al. [84] conducted experiments on a single-phase structure to control vibrations in switched reluctance motors (SRMs) using active vibration control methods with piezoelectric materials. The study revealed that active control techniques utilizing positive position feedback (PPF) controllers can achieve more significant vibration suppression effects.

Feedback vibration isolation control can be implemented in various ways, including direct feedback, time-delay dual-feedback, enhanced adaptive algorithms, a combination of acceleration and PD position feedback, positive position feedback, control considering inertia changes, and AMD with NAF control algorithms [85,86]. By combining different types of feedback and continuously optimizing feedback control algorithms, it is possible to suppress vibrations across different frequency ranges, thereby enhancing the vibration isolation performance and system robustness.

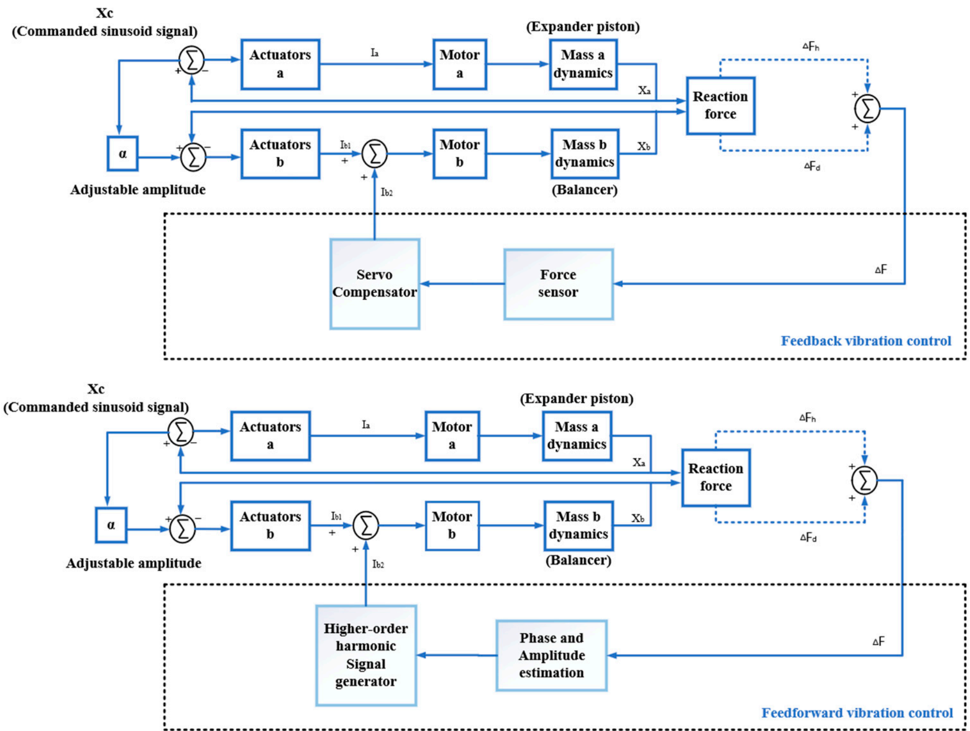


Figure 10. Feedforward active vibration isolation control block diagram and feedback active vibration isolation control block diagram.

4. Vibration Isolation Optimization Methods for Linear Compressors

Different types of linear compressors require different vibration isolation optimization control strategies. In this regard, Yang et al. [87] proposed a simple yet effective vibration isolation optimization control system based on momentum cancellation to actively reduce the compressor vibrations in Stirling refrigerators. By adjusting the displacement signal in the refrigerator's expansion module, the actuator and displacer move in opposite directions, thereby reducing the vibration force. This system significantly reduces the fundamental vibration of the refrigerator while maintaining low power consumption. Ali Hassann [88] introduced an active tuned mass damper (ATMD) system using a coil motor for vibration isolation optimization control to reduce the compressor vibrations within specific frequency bands in free-piston Stirling engines. The natural resonance frequency of the ATMD is adjusted through relative or absolute position and velocity feedback, along with direct measurements of the disturbance frequencies, to track the excitation frequency. Experiments demonstrated that the proposed ATMD system can reduce the compressor vibrations in Stirling engines within a broadband range of 45 Hz to 55 Hz. Jiten H. Bhatt et al. [89] presented a novel model predictive control (MPC) scheme for optimizing the vibration isolation of compressors in single-piston linear cryocoolers. Experiments showed that the MPC control scheme has a better performance compared to traditional PI controllers, making it more suitable for optimizing the vibration isolation of compressors in cryocoolers. The comparison of vibration isolation optimization methods is in Table 2.

Table 2. Comparison of vibration isolation optimization methods.

Type	Methods	Effect	Reference
Passive isolation	Tuned Dynamic Absorber + High-Damping Isolator	Vibration reduced by 3×, self-induced force 20× lower	[90]
	"Spring–mass–spring" balancer	Reduces resonance	[91]
	Ultra-lightweight undamped tuned dynamic absorber	Ultra-low damping ratio, more than 100× vibration attenuation	[92]
Active isolation	Narrowband feedback control	Excellent isolation from high harmonic frequencies	[93]
	Combined with adaptive dynamic vibration absorbers	Faster response and excellent damping control accuracy	[94]

The above table lists the specific vibration isolation design optimization methods used in passive and active vibration isolation, respectively. The two types of vibration isolators have their own advantages: passive vibration isolators are simple in structure and have good stability, while active vibration isolators have adjustability. However, both types of vibration isolation can have a good vibration isolation performance after their design is optimized. In addition, vibration energy recovery has greater feasibility and development potential in the optimal design of active vibration isolators [95–112].

5. Summary and Outlook

This study provides a comprehensive overview of the latest advancements in vibration control and optimization for linear compressors. It commences by elucidating the sources of vibration in linear compressors and their corresponding measurement methods. Subsequently, it introduces and compares SDOF and MDOF vibration isolation models. An analysis is conducted on the working principles and current research status of both active and passive vibration isolation. Recognizing that different types of linear compressors necessitate distinct approaches to vibration isolation optimization control, the study explores effective optimization methods for vibration control in linear compressors from multiple dimensions, including structural optimization, material selection, and shock absorber design. When combined with active control, these isolators hold significant potential for application in vibration energy recovery strategies.

To further mitigate the negative impacts of vibration in linear compressors, special attention should be paid to the following aspects during the optimization of vibration control:

1. Given the diversity of linear compressor types, the optimal vibration isolation control strategies also vary. It is necessary to develop tailored vibration isolation optimization control methods.
2. In the field of active vibration isolation control, both feedforward and feedback control strategies exhibit unique characteristics, with feedforward excelling in predicting vibrations and feedback excelling in responding to them. However, regardless of the control strategy employed, the core problem lies in designing efficient and stable control algorithms. In particular, for the cutting-edge topic of vibration energy recovery, it is of the utmost importance to develop algorithms that are both robust and possess rapid convergence properties.

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