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Study of active vibration isolation systems considering isolator – structure interaction

Pérez-Aracil, Jorge[1]; Pereira, Emiliano[2]; Díaz, Iván M.[3]; Reynolds, Paul[1]

ABSTRACT

Many technological applications require precise alignment between different devices to their correct development. The unwanted vibrations transmitted through the support to the payload can produce misalignment and this may entail high costs and loss of performance. Vibration isolation techniques have been proposed as a cost effective and more efficient alternative. This implies the insertion of an isolator between the vibration source and the payload. Depending on the configuration of the isolator, two different techniques can be used: 1) passive vibration isolation, 2) active vibration isolation. As it has been proved, the use of active vibration techniques within this application is best at improving overall performance. In this work, the initial research considerations on the improvement of the current state of active vibration isolation are presented. In addition, the interaction phenomenon between single – axis isolator and the flexible support system on which it is supported is studied. The dynamic requirements of the vibration isolation problem are identified and a study of the tuned parameters of a controller is included.

Keywords: *vibration isolation, active vibration control, isolator-structure interaction, dynamic modelling.*

1. INTRODUCTION

Precise alignment possesses a key role in many applications, being: research facilities as Diamond Light Source (DLS) [1], space applications [2]–[5] and private manufacturers. The main problem with these applications is the undesired vibrations, which may produce variations in the required position, and consequently, a loss in the performance of the system. Vibration isolation techniques have been presented as a more efficient solution [6]–[8], which implies the use of an isolator situated between the vibration source and the payload. Two different types of techniques can be used: 1) passive vibration isolation, 2) active vibration isolation. There are still a few issues that must deal with in the active vibration isolation (AVI) systems, such as structure – isolator interaction and more efficient control laws, that should be clearly geared to practical implementation.

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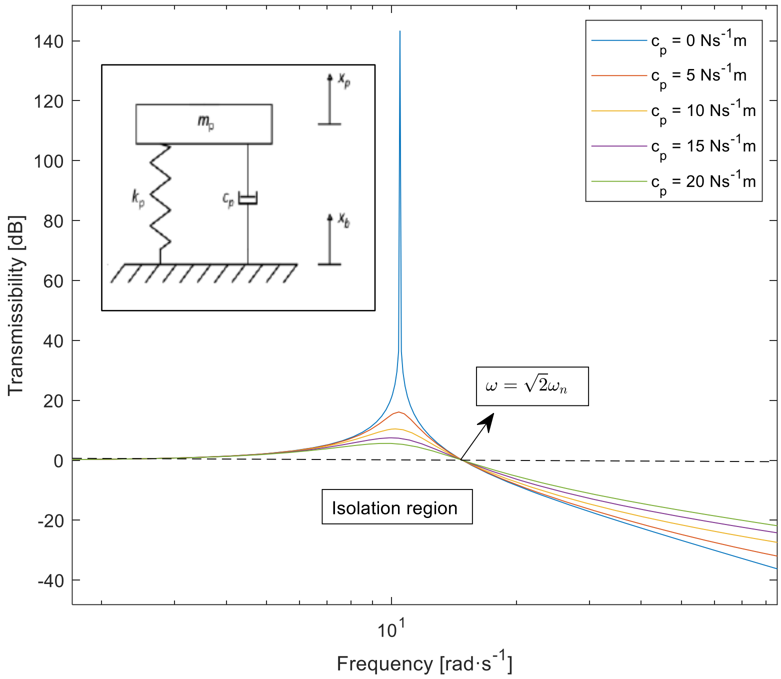

Figure 1. *Transmissibility magnitude curve of a passive isolator for different values of damping c_p.*

In passive vibration isolation (PVI) systems, the isolator only reacts to the relative displacement between the platform and the payload. The ratio of both displacements $(x_p(t)/x_b(t))$, see the schematic representation of an isolator in Fig. 1, is called *transmissibility* [9], and it is used to analyse the performance of the isolator. In Fig. 1, a representation of the transmissibility of a generic single – axis isolator has been represented in frequency domain for different values of damping. As it can be observed, the increment of damping reduces the transmissibility at the natural frequency of the isolator. However, isolation occurs for excitation frequencies $\omega > \sqrt{2}\omega_{n_p}$, ω_{n_p} being the natural frequency of the isolator, although it is important to note that the *roll-off* rate decreases as the damping increases. The response of the isolator could be improved making it softer, which implies a reduction of the natural frequency, increasing that the frequency band of the excitation that can be rejected. For that purpose, *negative stiffness* was introduced by Platus in 1992 [10]. This technique reduces the stiffness of the global system using nonlinearities as spring orientation or buckling [11]. Nonetheless, they are not suitable for all applications in which vibration isolation is required, since their implementation is not easy, and they do not present some important characteristic that active systems do, as detailed below.

The AVI systems are those that, in addition to the passive isolator, sensor, signal processor and an actuator are also part of the isolation system, showing important differences to PVI systems [12]. On the one hand, when these elements work together, some issues such as stability of the control system, real-time processing, inherent actuator dynamics and the influence of the support structure on its performance make AVI design and implementation a challenging task. On the other hand, AVI systems present many advantages when they are compared with PVI systems. Some of the advantages are: 1) quasi-zero static deflection, 2) the possibility to change the position of the payload, 3) the possibility to create a softer system than PVI systems which will in turn improve the response for low frequencies, keeping an optimal performance for any input and, 4) the capability to adapt the control to the possible changes in the system and the excitation [12].

The study of new ways of improving the current state in vibration isolation of multiple synchronized devices is the main objective of this research, and for that purpose different control strategies must be studied. Their implementation in real scenarios is an important point in the research development. For that, the use of electrodynamic shakers becomes very important. They are key devices in vibration control [13]–[16]. The possibility to change their configuration allows to implement AVI controllers, enabling performance to be evaluated.

This work studies a non-developed isolation problem, which consists of an electrodynamic shaker placed on a simply supported beam. The novelty is that the moving mass of the electrodynamic shaker is not negligible compared with the modal mass of the beam. In other words, there is an interaction between force exerted by the actuator and acceleration of the beam. The dynamic model of this system together with an example of controller is included in this work. The dynamic model of the system is developed and explained in Section 2, an example of application is done in Section 3 and conclusions and on-going works are presented in Section 4.

2. VIBRATION ISOLATION MODELLING

In this section, a dynamic model of an active isolator situated on a support structure has been made. The interaction isolator – support structure has been considered, the dynamic requirements are identified, and a controller is proposed.

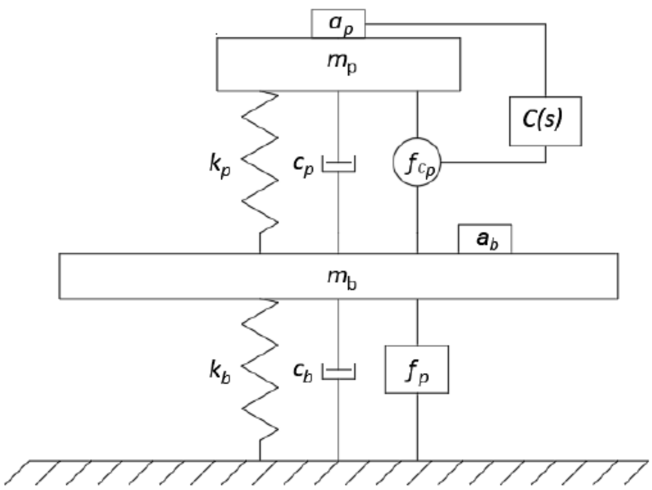

Figure 2. *Schematic of a single axis active isolator system mounted on a flexible support.*

The general schematic of the problem presented here is shown in Fig. 2. The dynamic parameters of the isolator and support system are represented by the subscripts p and b, respectively. Whereas, m_i, c_i, k_i represent the mass, damping and stiffness of system i, $f_{c_p}(t)$represents the control force of the isolator and $f_p(t)$ is the disturbance force, which is applied to the support system. The general control system is represented by $C(s)$, where s represents the Laplace variable, a_i represents the accelerometers situated on system i, which are used to measure the accelerations outputs of the system, $(\ddot{x}_p(t), \ddot{x}_b(t))$. The dynamic of the accelerometers is not considered in this work, since their dynamics are totally out to the frequency band of interest. The problem has two inputs: 1) the perturbation force $f_p(t)$ and the control force $f_{c_p}(t)$. The first one is supposed to be unknown, while the second one tries to mitigate the effects of the perturbation force in the system, using the measured outputs. Note that the objective of the control force $f_{c_p}(t)$ is to reduce the acceleration between the platform and the support system. Thus, the control force moves the mass m_p to minimize the transmissibility function $(\ddot{x}_p(t)/\ddot{x}_b(t))$. The equations of motion of the system can be derived from the second Newton's law. The differential equation of the platform, that relates the movement of the mass m_p, the control force $f_{c_p}(t)$ and the movement of the support structure $x_b(t)$ is given by:

$$f_{c_p}(t) - c_p\left(\dot{x}_p(t) - \dot{x}_b(t)\right) - k_p\left(x_p(t) - x_b(t)\right) = m_p\,\ddot{x}_p(t). \tag{1}$$

The differential equation of the support system is:

$$f_p(t) - f_{c_p}(t) + c_p\left(\dot{x}_p(t) - \dot{x}_b(t)\right) + k_p\left(x_p(t) - x_b(t)\right) - c_p\dot{x}_b(t) - k_b x_b(t) = m_b\,\ddot{x}_b(t). \tag{2}$$

The analysis of the isolation performance will be made in frequency domain. The Laplace transform of Eq. (1) gives:

$$\begin{aligned} s^2X_p(s) &= s^2\left(\frac{F_{c_p}(s)}{m_ps^2 + c_ps + k_p} + \frac{c_ps + m_p}{m_ps^2 + c_ps + k_p}X_b(s)\right) \\ &= G_p(s)F_{c_p}(s) + T_{pb}(s)s^2X_b(s). \end{aligned} \tag{3}$$

in which:

$$G_p(s) = \frac{s^2X_p(s)}{F_{c_p}(s)} = \frac{s^2}{m_ps^2 + c_ps + k_p} \quad \text{and} \quad T_{pb}(s) = \frac{X_p(s)}{X_b(s)} = \frac{c_ps + m_p}{m_ps^2 + c_ps + k_p}.$$

The transfer function $G_p(s)$ represents the relation between the acceleration $\ddot{x}_p(t)$ of the platform and the applied control force $f_{c_p}(t)$. No external disturbance force applied on the platform has been considered here. The term $T_{pb}(s)$ is the transmissibility between the platform and the support structure. The force that the isolator imparts in the platform mass m_p can be derived from Eq. (3):

$$G_{p_f}(s) = \frac{m_ps^2}{m_ps^2 + c_ps + k_p}. \tag{4}$$

The Laplace transform of Eq. (2) gives:

$$\begin{aligned} s^2X_b(s) &= s^2\left(\frac{F_p(s) - F_{c_p}(s)}{m_bs^2 + (c_p + c_b)s + k_p + k_b} + \frac{c_ps + m_p}{m_bs^2 + (c_p + c_b)s + k_p + k_b}X_p(s)\right) \\ &= G_b(s)\left(F_p(s) - F_{c_p}(s)\right) + T_{bp}(s)s^2X_p(s). \end{aligned} \tag{5}$$

in which

$$G_b(s) = \frac{s^2}{m_b s^2 + (c_p + c_b)s + k_p + k_b} \quad \text{and} \quad T_{bp}(s) = \frac{c_p s + m_p}{m_b s^2 + (c_p + c_b)s + k_p + k_b}.$$

The relation between the acceleration of the support structure $\ddot{x}_b(t)$ and the total force received by the support system (see Fig. 3) is given by $G_b(s)$. Whereas, $T_{bp}(s)$ is the transmissibility between the support structure and the platform. These two terms are usually neglected ($G_b(s)$ and $T_{pb}(s)$) in vibration isolation problems, since the isolator does not usually affect significantly the movement of the support system. Thus, these two terms have been considered with this framework. The transmitted force to the platform due the movement of the support structure is expressed as:

$$f_t(t) = c_p\left(\dot{x}_p(t) - \dot{x}_b(t)\right) + k_p\left(x_p(t) - x_b(t)\right) = m_p\ddot{x}_p(t). \tag{6}$$

Considering Eq. (1) and Eq. (6), the Laplace transform of the transmitted force $f_t(t)$ to the platform from the platform movement is given by:

$$F_t(s) = m_p T_{pb}(s) s^2 X_b(s) = H(s) s^2 X_b(s) = m_p s^2 X_p(s). \tag{7}$$

where $H(s) = m_p T_{pb}(s)$. Thus, the control scheme can be:

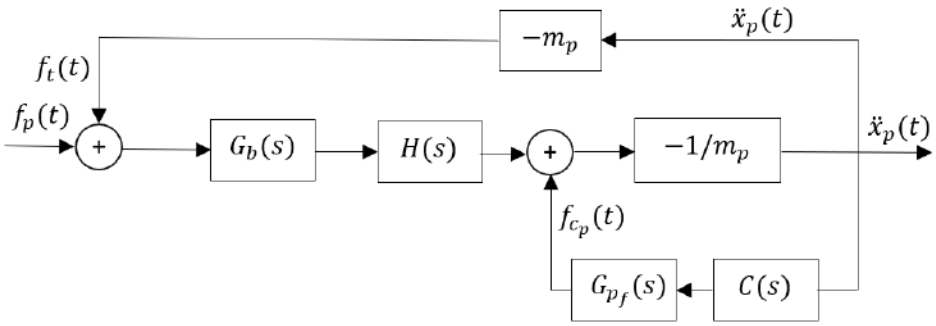

Figure 3. *Control scheme.*

Then, the closed – loop transfer function between the acceleration of the platform and the perturbation force can be derived as:

$$G_{CL}(s) = \frac{s^2 X_p(s)}{F_p(s)} = \frac{-G_b(s)H(s)}{m_p\left(1 + \dfrac{C(s)G_{p_f}(s)}{m_p} - -G_b(s)H(s)\right)}. \tag{8}$$

In this work, only vibration isolation respect to the support system is considered. It means that the platform does not need to keep a position respect a fixed global frame. For that purpose, instead of Eq. (8), the transmissibility between the acceleration in the support system and the platform is considered to analyze performance of the isolator. Based on the schematic of Fig. 3, the transmissibility can be expressed as:

$$T_{XX}(s) = \frac{X_p(s)}{X_b(s)} = \frac{-H(s)}{m_p\left(1 + G_{p_f}(s)C(s)/m_p\right)}. \tag{9}$$

Therefore, the controller $C(s)$ must achieve the two following objectives: 1) a reduction of the response at the resonance frequency of the isolator, 2) a reduction in the cut-off frequency to improve the response for frequencies lower than the natural frequency of the isolator. The first objective can be achieved by a damping increment. The second objective can be achieved if the corner frequency of the transmissibility curve is moved to the left, which is equivalent to actively reduce the system frequency. If the measured output is the platform acceleration, the general expression to achieve both objectives is given by:

$$C(s) = C_s + C_d(s) = -k_s - \frac{k_d}{s}. \tag{10}$$

in which k_d represents the gain of the integral part, which imparts damping to the system, and where k_s is the gain of the controller part proportional to the acceleration output.

3. EXAMPLE OF APPLICATION

In this section, the above developed theory is complemented with a numerical example of application. It allows to study the performance of the different proposed controllers that will be implemented in the active isolator, also it grants the possibility to understand the interaction phenomenon and to analyze stability problems before the experimental implementation. This motivates the study of the numerical model based on a real experiment.

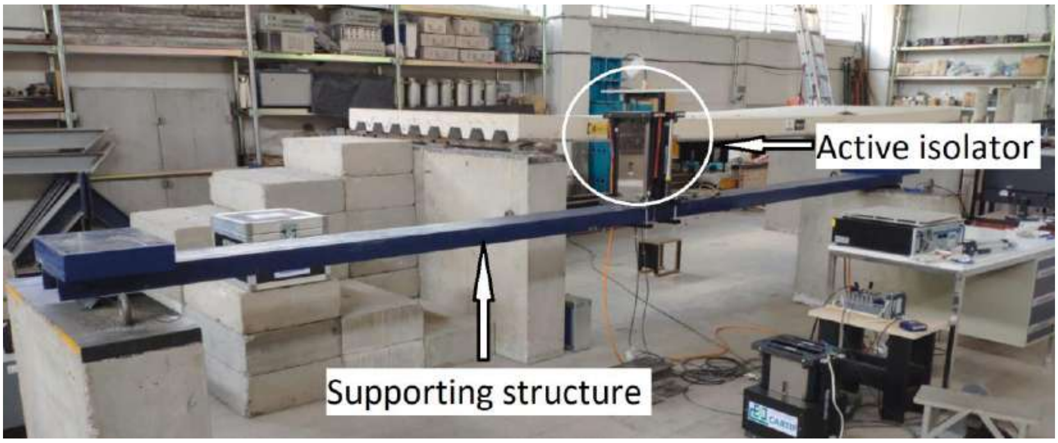

Figure 4. *General view of the AVI experimental test.*

A general view of the experimental test is shown in Fig. 4. This setup allows to introduce to the system a controlled excitation in order to derive experimentally the different transfer functions expressed in Section 2 and validate the proposed framework. An active isolator is situated on the center of a simply – supported beam. The expression of Eq. (9) is used to assess the isolator performance, since it represents the ratio between the acceleration of the platform (measured at the top of the isolator) and the beam. Both accelerometers are vertically aligned.

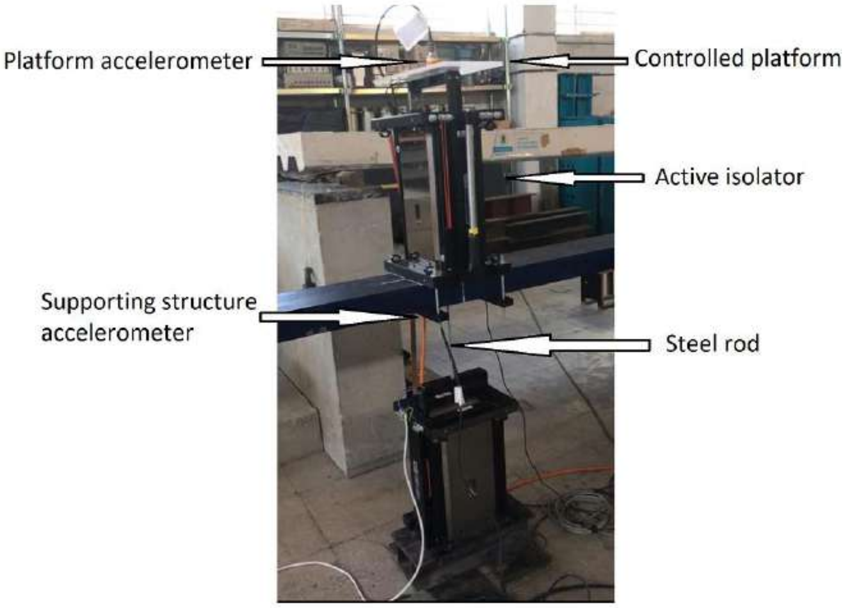

Figure 5. *Experimental implementation: active isolator, beam support, accelerometers and the electrodynamic actuator used to generate the perturbation force.*

If an electrodynamic shaker is situated below the support system and it is connected to the beam with a steel rod, see Fig. 5, then the dynamic model of the system is like that shown in Fig. 2: an isolator is mounted on a beam, that acts as support system, and it is perturbed by a force which is imparted by a shaker.

Table 1. Dynamic parameters of the system.

Property	Value	
	Active isolator	Support structure
Mass	3 kg	127 kg
Damping	2 Nsm^{-1}	12 Nsm^{-1}
Stiffness	330 N/m	113664 N/m
Frequency	10.48 rad/s	29.91 rad/s

The used values for this numerical example, based on the above shown configurations, are shown in Table 1. The frequency of both systems is calculated using the expression $\omega_{n_i} = \sqrt{k_i/m_i}$, [17]. One of the main aims of the control system is to reduce the response at the isolator natural frequency. For that purpose, the integral part of Eq. (10) should be used, which corresponds with (k_d/s).

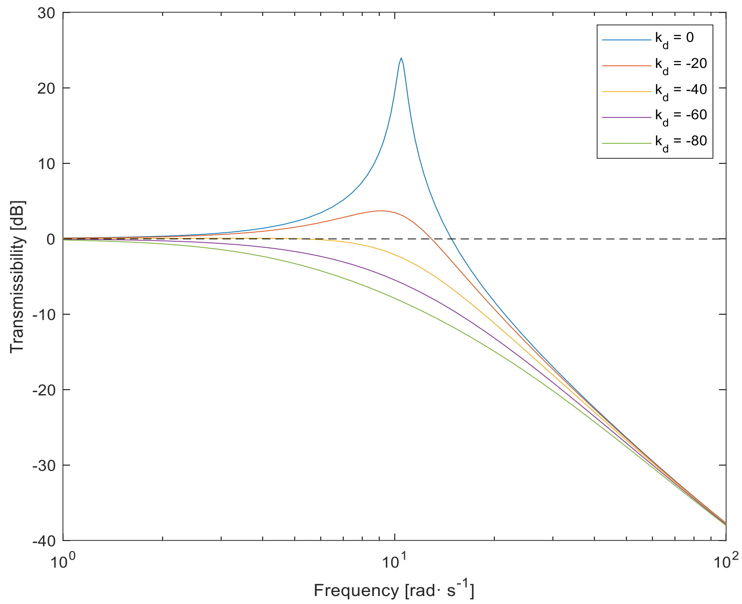

Figure 6. *Transmissibility for different values of* k_d.

As the absolute value of the gain k_d increases, the ratio $(x_p(t)/x_b(t))$ is reduced. However in a traditional damper, the exerted control force is proportional to the relative velocity between the platform $\left(\dot{x}_p(t) - \dot{x}_b(t)\right)$, in this case the applied control force $f_{c_p}(t)$ is proportional to the velocity of the platform $\dot{x}_p(t)$. This control law is called *sky – hook* damper [9], [17].

The Fig. 6 shows that in passive mode ($k_d = 0$), isolation occurs for frequency values greater than $\sqrt{2}\omega_n$, as mentioned above. However, if the active damping increases, isolation can be reached for $\omega < \omega_{n_p}$.

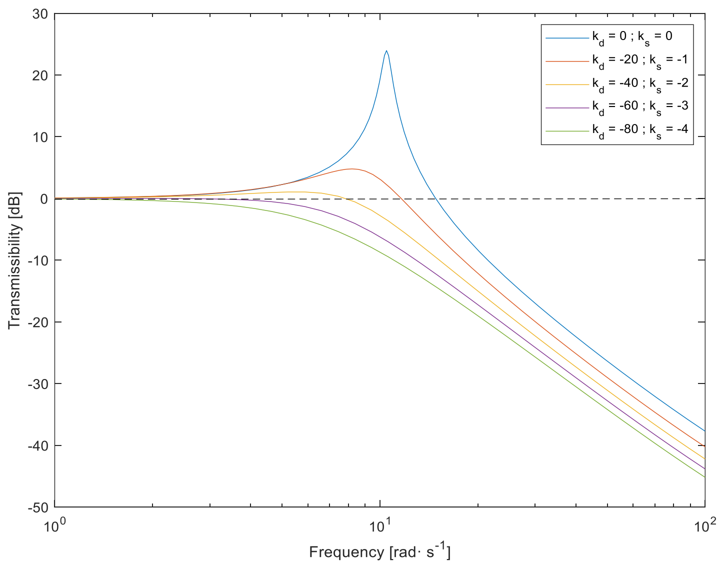

Figure 7. *Transmissibility for different values of* k_d *and* k_s.

It is important to consider that, in a real experiment, there is an upper limit for the gain k_d, due to the limitations of the involved devices. Whereby, to improve the performance of the isolator, an actively reduction of its natural frequency and an addition of damping can be reached using both terms of Eq. (10). The result is shown in Fig. 7. The corner frequency is reduced as the absolute value of the gain k_s increases, and the peak value becomes more flatter as k_d is increased.

4. CONCLUSIONS

Active vibration isolation systems present many advantages compared with PVI systems. The use of feedback systems allows to raise different control schemes according to the requirements of each problem. The necessity of keeping a precise alignment between different devices, when they are affected by vibration from several sources, leads to study all the possible scenarios. In this work, a single – axis isolator placed on a simply – supported beam, considering the interaction between them, has been analysed. The main difference respect to a simultaneous positioning and vibration isolation problem has been identified. Also, a generic controller has been proposed, which improves the response of the isolator respect to the support system by increasing the damping and by reducing the natural frequency of the system (i.e., reducing the cut-off frequency of the transmissibility function).

Future work will be related with the analysis of different scenarios, in which several isolators and different control algorithms will be involved, with the main purpose of keeping the relative position of them. With these objectives, several experimental works will be developed.

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REFERENCES

[1] R. P. Walker, “Overview of the Status of the Diamond Project,” presented at the European Particle Accelerator Conference (EPAC), Edinburgh, Scotland, 2006, p. 5.

[2] A. Preumont *et al.*, “A six-axis single-stage active vibration isolator based on Stewart platform,” *J. Sound Vib.*, vol. 300, no. 3–5, pp. 644–661, Mar. 2007.

[3] X. Yang, H. Wu, B. Chen, S. Kang, and S. Cheng, “Dynamic modeling and decoupled control of a flexible Stewart platform for vibration isolation,” *J. Sound Vib.*, vol. 439, pp. 398–412, Jan. 2019.

[4] C. Wang, X. Xie, Y. Chen, and Z. Zhang, “Investigation on active vibration isolation of a Stewart platform with piezoelectric actuators,” *J. Sound Vib.*, vol. 383, pp. 1–19, Nov. 2016.

[5] M. Li, Y. Zhang, Y. Wang, Q. Hu, and R. Qi, “The pointing and vibration isolation integrated control method for optical payload,” *J. Sound Vib.*, vol. 438, pp. 441–456, Jan. 2019.

[6] C. Collette, S. Janssens, K. Artoos, and C. Hauviller, “Active vibration isolation of high precision machines,” *Diam. Light Source Proc.*, vol. 1, no. MEDSI-6, Apr. 2011.

[7] F. Matichard *et al.*, “Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 2: Experimental investigation and tests results,” *Precis. Eng.*, vol. 40, pp. 287–297, Apr. 2015.

[8] F. Matichard *et al.*, “Advanced LIGO two-stage twelve-axis vibration isolation and positioning platform. Part 1: Design and production overview,” *Precis. Eng.*, vol. 40, pp. 273–286, Apr. 2015.

[9] A. Preumont, *Vibration control of active structures*. New York, NY: Springer Berlin Heidelberg, 2018.

[10] D. L. Platus, “Negative-stiffness-mechanism vibration isolation systems,” presented at the San Jose - DL tentative, San Jose, CA, 1992, pp. 44–54.

[11] E. Palomares, A. J. Nieto, A. L. Morales, J. M. Chicharro, and P. Pintado, “Numerical and experimental analysis of a vibration isolator equipped with a negative stiffness system,” *J. Sound Vib.*, vol. 414, pp. 31–42, Feb. 2018.

[12] J. E. Ruzicka, “Active Vibration and Shock Isolation,” presented at the National Aeronautic and Space Engineering and Manufacturing Meeting, 1968.

[13] I. M. Díaz and P. Reynolds, “Robust saturated control of human-induced floor vibrations via a proof-mass actuator,” *Smart Mater. Struct.*, vol. 18, no. 12, p. 125024, Dec. 2009.

[14] I. M. Díaz and P. Reynolds, “Acceleration feedback control of human-induced floor vibrations,” *Eng. Struct.*, vol. 32, no. 1, pp. 163–173, Jan. 2010.

[15] I. M. Díaz, E. Pereira, M. J. Hudson, and P. Reynolds, “Enhancing active vibration control of pedestrian structures using inertial actuators with local feedback control,” *Eng. Struct.*, vol. 41, pp. 157–166, Aug. 2012.

[16] E. Pereira, I. M. Díaz, E. J. Hudson, and P. Reynolds, “Optimal control-based methodology for active vibration control of pedestrian structures,” *Eng. Struct.*, vol. 80, pp. 153–162, Dec. 2014.

[17] D. Karnopp, M. J. Crosby, and R. A. Harwood, “Vibration Control Using Semi-Active Force Generators,” *J. Eng. Ind.*, vol. 96, no. 2, p. 619, 1974.