Active Vibration Isolation of Launcher Vibration Environment

Conference Paper · September 2016

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ACTIVE VIBRATION ISOLATION OF LAUNCHER VIBRATION ENVIRONMENT

27-30 September 2016, Toulouse, France

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ABSTRACT

Sensitive payloads mounted on top of launchers are subjected to many sources of disturbances during the flight. The most severe dynamic loads arise from the ignition of the motors, gusts, the pressure fluctuations in the booster and from the separation of the boosters. In order to reduce the transmission of these dynamic forces, payloads can be mounted on passive isolators, which comprise passive elements to filter out high frequency loads, at the expense of harmful amplifications of the motion at low frequency due to suspension modes (where the payload bounce on the suspension stiffness). To some extent, the amplification can be by reduced increasing the damping with passive components. However, increasing the damping will in turn compromise the isolation, which is inherent to passive isolators.
Several prototypes of active suspensions have been developed, including piezoelectric actuators or magneto-rheological fluids. However, so far, only passive isolators have been used in flight, because none of the active prototypes have been capable to fulfill the stringent requirements of space flight.
This paper presents a novel concept of active mount for aerospace payloads, which offers several advantages, including ease of mounting, high damping authority on both suspension resonances and flexible resonances without compromising the isolation and large stability margins of the closed loop system. The concept is presented in the first part of the paper. It is studied numerically on a single degree of freedom model and compared to other state-of-the-art strategies.
The second part of the paper is dedicated to an experimental validation on a scaled test bench. It consists of a flexible structure, mounted on three active isolators, mounted at the other end to another flexible structure, this one being used for the base excitation. Experimental results are discussed in terms of transmissibility, compliance and deformation. It is shown that the feedback operation allows to damp both suspension and flexible modes, and significantly reduce the force transmitted, confirming numerical predictions from a finite element model of the set-up.

1. INTRODUCTION

Since the world's first satellite has been launched nearly sixty years ago, the technology of launchers and satellites has considerably improved. While in the early days, satellites were mounted on rigid supports, modern and sophisticated satellite equipments require a better isolation from launcher disturbances [1]. To this purpose, passive suspensions have been progressively introduced. Mounting the satellites on passive suspensions produces a filtering of high frequency vibrations [2-3]. The softer the suspension, the better the isolation.

However, this comes at the expense of large and harmful amplifications of the lower frequency suspension modes [4], and raises concerns of strength and clearance issues because of the larger compliance. This is a first fundamental limitation of passive isolation [5].
The amplification at resonance can be reduced by increasing the damping in the isolator. However, it comes in turn at the expense of a reduction of the isolation at high frequency. This is a second

fundamental limitation of passive suspensions. To some extent, the high frequency isolation can be recovered by using a relaxation isolator. However, it does not reduce the compliance, and has a very limited efficiency.

The only way to bypass the two aforementioned tradeoff is to use active vibration isolation systems. They do not show such amplifications at low frequencies and are capable of an overall damping performance. Considerable efforts on active [6-9] and semi-active vibration isolators [10,11] have been carried out. However, measurement noise and instability of the controllers in face of un-modelled dynamics of the payloads have often hindered performance, or the final design of the isolator exhibited excessive mass.

The most flight proven vibration isolation system is the CSA Softride [2], whose properties are detailed in Table 1. While very attractive on the grounds of its modularity and performance, the isolation is obtained at the cost of a large magnification of the launcher vibration around the main resonance of the system, where the payload is bouncing on the stiffness of the suspension. This feature is inherent to passive systems. In this project, our objective is to use an architecture similar to the flight-proven CSA Softride, combined with active control capability to reduce magnification due to the first structural resonances.

Table 1. Comparison of isolators from launcher vibration environment.

	CSA SOFT RIDE isolator [2]
First attenuation frequencies	[25Hz ; 500Hz]
Isolator modes	1^{st} longitudinal mode : ~22Hz
Vibration attenuation in dB	10-20 dB
Pre-stressed	No
Geometry	Multiple modules around a ring

This paper presents a novel concept of active mount for aerospace payloads, which offers several advantages, including ease of mounting, high damping authority on both suspension resonances and flexible resonances without compromising the isolation and large stability margins of the closed loop system. The concept is presented in the first part of the paper. It is studied numerically on a single degree of freedom model and compared to other state-of-the-art strategies.

The second part of the paper is dedicated to an experimental validation on a scaled test bench. It consists of a flexible structure, mounted on three active isolators, which are in turn mounted at the other end to another flexible structure, used for the base excitation. Experimental results are discussed in terms of transmissibility, compliance and deformation. It is shown that the feedback operation allows to damp both suspension and flexible modes, and significantly reduce the force transmitted, confirming numerical predictions from a finite element model of the set-up.

2. PROPOSED CONCEPT OF ACTIVE ISOLATOR

The proposed concept of active isolator is composed of a compliant metallic structure, in which a sensor and a piezoelectric actuator is integrated. An example of this configuration is shown in Fig. 1. As for flight-proven isolator, it consists of an elliptic metallic structure that plays the role of suspension. Additionally, a piezoelectric stack actuator has been placed inside the structure, along the horizontal axis of the ellipse. The stack is further divided in two parts, respectively used as actuator and as force sensor, and both constitute a perfectly collocated pair.

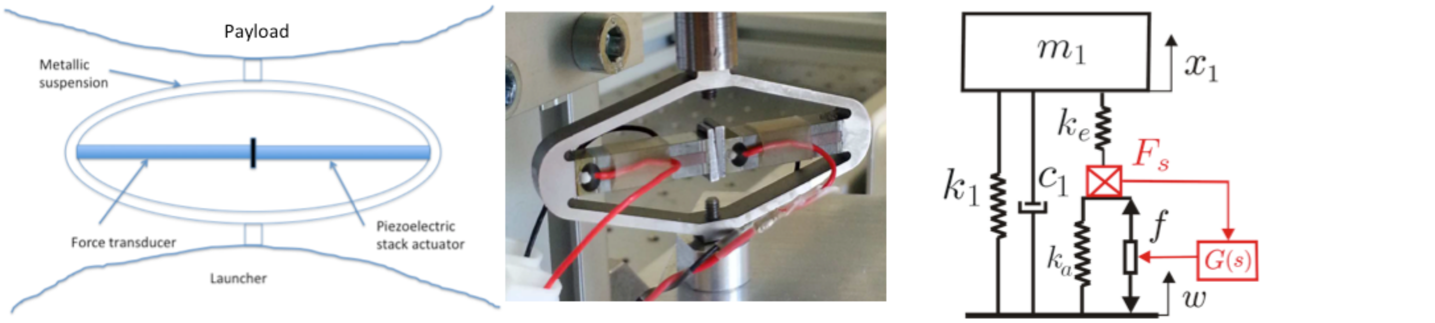


Fig. 1. From left to right: Concept of active isolator; Picture of an APA 100M from Cedrat-Technologies used for the experiments (one piezoelectric stack is used as force sensor and the other one is used as actuator); Simplified model of a one d.o.f. payload mounted on such isolator.

In each mount, the actuator will be driven only by using the signal from the force sensor of the same mount. Both suspension resonances and payload resonances are actively damped by applying viscous damping forces obtained by integrating the signal from the force sensors [12]. Such control strategy, known as Integral Force Feedback (IFF) has been extensively discussed in [13,14]. In the proposed configuration, it can also be noticed that the softening effect inherent to force control is limited by the metallic suspension, which will also continue to work as a passive isolator in case of failure of the piezoelectric stack.

The advantages of this mount are:

1. Modularity: The number and position of mounts will depend on the mass and inertia of the payload and the specific mechanical environment.
2. The elliptical shape is working both as a suspension spring and as a mechanical amplifier for increasing the actuator range.
3. Each unit is stand-alone (does not require any additional sensor).
4. The control architecture use decentralized feedback loops in each active mount, which offers excellent stability margins.

Although the full design of the suspension will require a complete study, the objective of this preliminary work is to test experimentally the concept using a commercial isolator. To this purpose several set-up of increasing complexity have been developed: one d.o.f payload mounted on one isolator, flexible payload mounted on three isolators and the same structure mounted on a flexible launcher are presented respectively in section two, three and four.

3. SINGLE DEGREE-OF-FREEDOM ISOLATOR

Fig. 1 shows a picture of the amplified piezoelectric stack used for the experiments. It is an APA 100M from Cedrat-Technologies, where the piezoelectric actuator has been divided in two parts: one is used as actuator, and the other one is used as force sensor. A small mass has been mounted on the actuator to represent a single d.o.f. payload. A simplified sketch of the system is shown in Fig. 1 (right). In this model, k_1 represents the stiffness of the metallic suspension when the stack is removed; k_a is the stiffness of the actuator; k_e is a stiffness used to adjust the pole of the isolator; w is the motion imposed by launcher; F_s is the force sensor; f is the actuator; G(s) is the controller; x_1 is the motion of the payload; c_1 is a viscous damper used to match experimental results. Fig. 2 shows the matrix of transfer function from input (w,f,F) to output (F_s, x_1) in open loop (blue curves) and closed loop (green curves). The following numerical values have been used for the simulation:

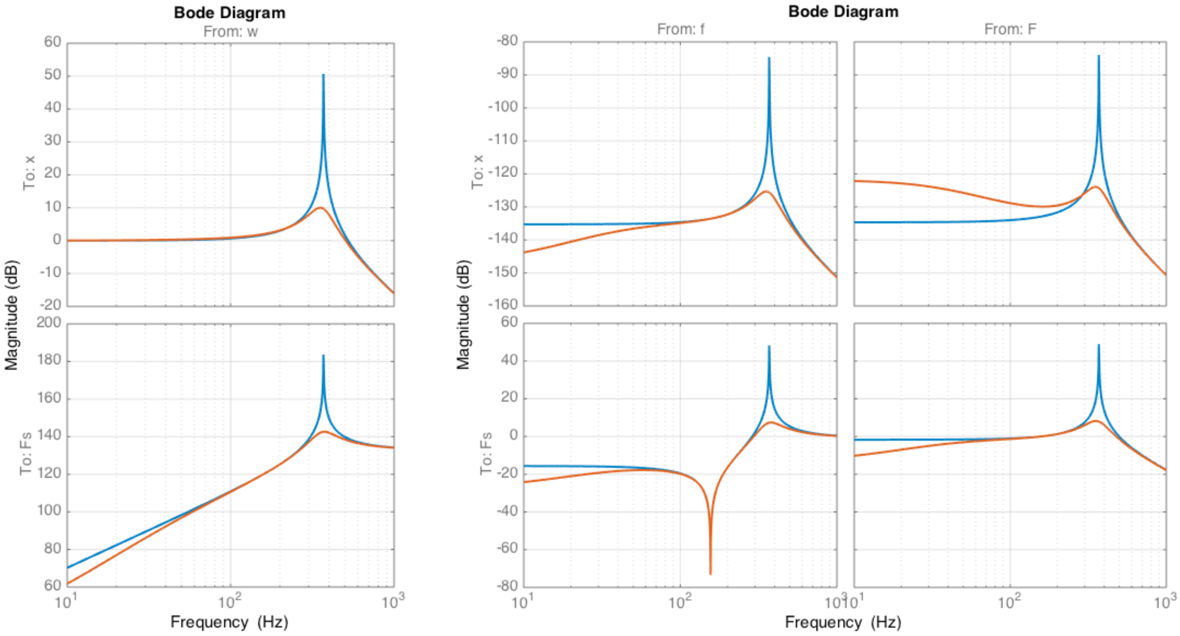


Fig. 2. Matrix of transfer functions from input (w,f,F) to output (F_s, x_1) in open loop (blue curves) and closed loop (green curves).

Firstly, it can be noticed on the transmissibility that the resonance peak is almost critically damped, without compromising the high-frequency passive isolation for frequencies higher than the resonance of the suspension. Secondly, the degradation of the compliance induced by the feedback operation is limited at $1/k_1$ as anticipated. Thirdly, the fraction of the force transmitted to the payload that is measured by the force sensor is reduced not only at the resonance, but also in a broad frequency range at low frequencies.

Fig. 3 compares the theoretical (solid curves) and experimental (dashed curves) closed loop transfer functions between the sensor and the actuator obtained for various values of the control gain. It clearly shows a high efficiency of the isolator and a good match between the model and the experiment. The experimental data are further compared with the model in Fig. 4, which shows the trajectory of the closed loop poles in the complex plane.

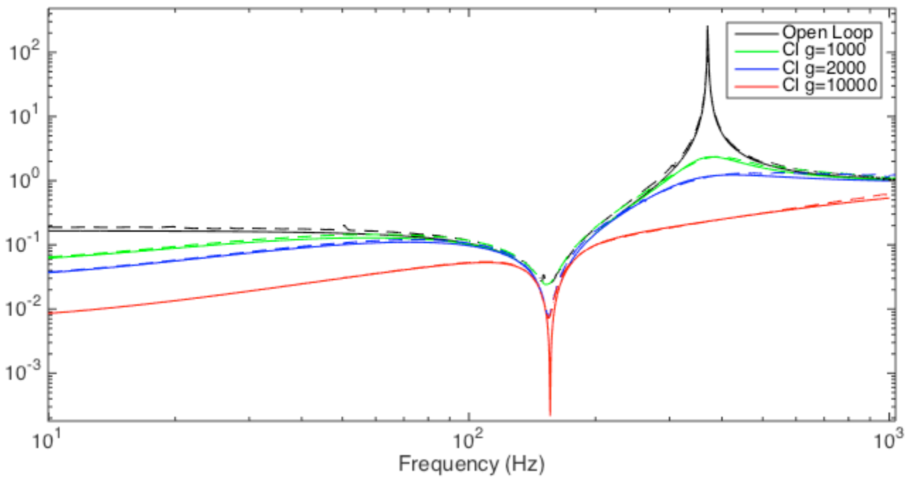


Fig. 3. Open loop and closed loop transfer function between the actuator and the force sensor. Comparison between the model (solid curve) and the experiment (dashed curve).

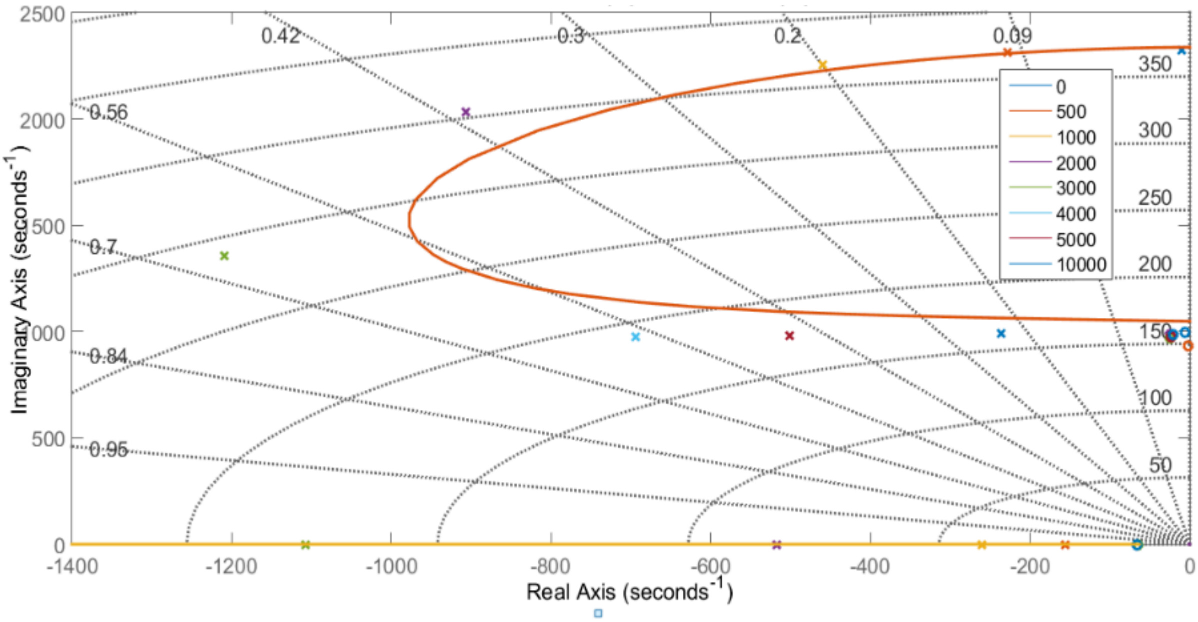


Fig. 4. Single d.o.f. system. Comparison between the theoretical (solid curve) and the experimental (crosses) root-locus.

4. FLEXIBLE PAYLOAD MOUNTED ON THREE ISOLATORS

The second set-up considered consists of a heavier payload, mounted on a set of three isolators. The payload consists of two masses, connected through three flexible blades with a tunable length. The set-up is shown in Fig. 5, along with a simplified sketch showing only the vertical d.o.f. of the payload, mounted on one isolator.

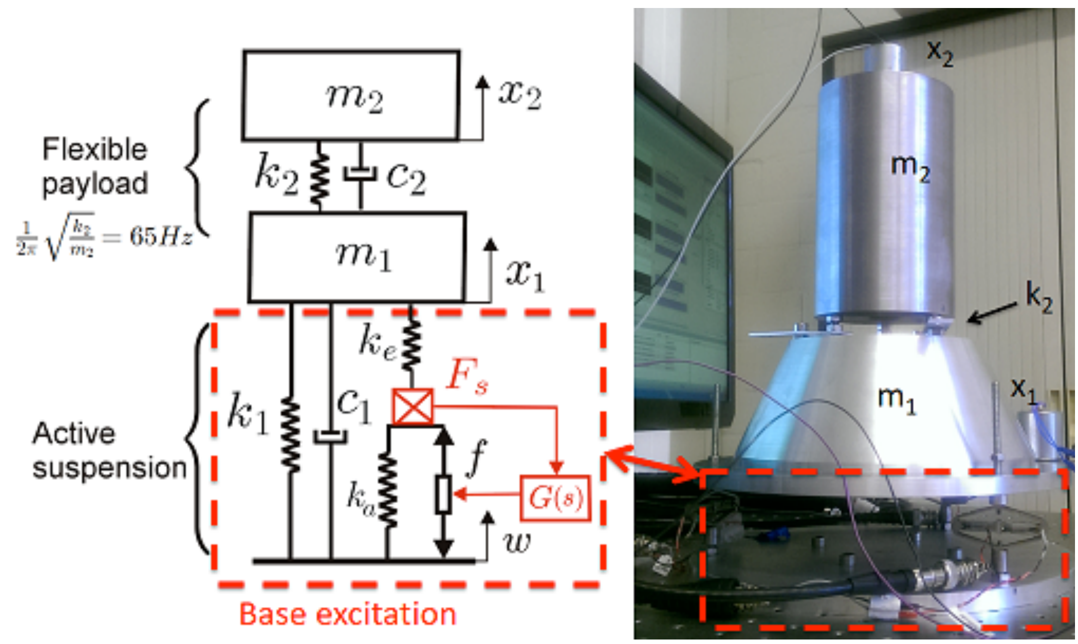


Fig. 5. Right: Picture of the experimental set-up. It consists of a flexible payload mounted on a set of three isolators. Left: simplified sketch of the set-up, showing only the vertical direction.

The flexible resonance of the payload in the vertical direction has been tuned around 65 Hz. It corresponds to the resonance of m_2 while m_1 is blocked. The first six mode shapes and corresponding resonance frequencies are shown in Fig. 6. The set-up has been mounted on a passive optical table. A shaker has been mounted on the table top besides the set-up in order to excite it from the base as for the launcher.

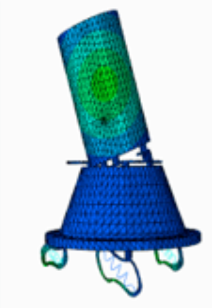
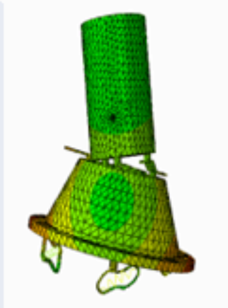
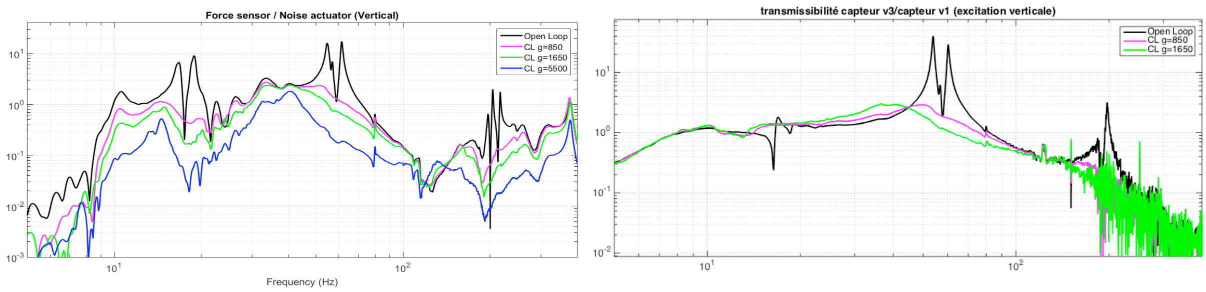
17 Hz		First tilt mode (dual-mode symmetric)	77 Hz		First twist mode
53 Hz		Second tilt mode (dual-mode symmetric)	164 Hz		Third tilt mode
57 Hz		Suspension mode	175 Hz		Flexible mode

Fig. 6. Mode shapes and corresponding resonance frequencies of the first 6 modes.

Two test campaigns have been conducted: one when the shaker excites the table in the vertical direction, and one when the shaker excites the table in the horizontal direction. For both campaigns, decentralized control loops have been used, as explained in sections 1 and 2. Typical experimental results are shown respectively in Figs. 7 and 8. For both figures, the same quantities are shown: Top left : Transfer function between the shaker noise and one force sensor ; Bottom left : integrated (downwards) RMS value of the force measured by one force sensor ; Top right : transmissibility between the table top w and m_2 ; bottom right : Transmissibility between the table top w and m1. For both campaigns, one sees that both the suspension modes and the flexible modes of the payload can be critically damped. Furthermore, the fraction of the force transmitted to the payload that is measured by the force sensor is also significantly reduced by up to two orders of magnitudes around the resonances, and the RMS value is reduced by a factor 5.



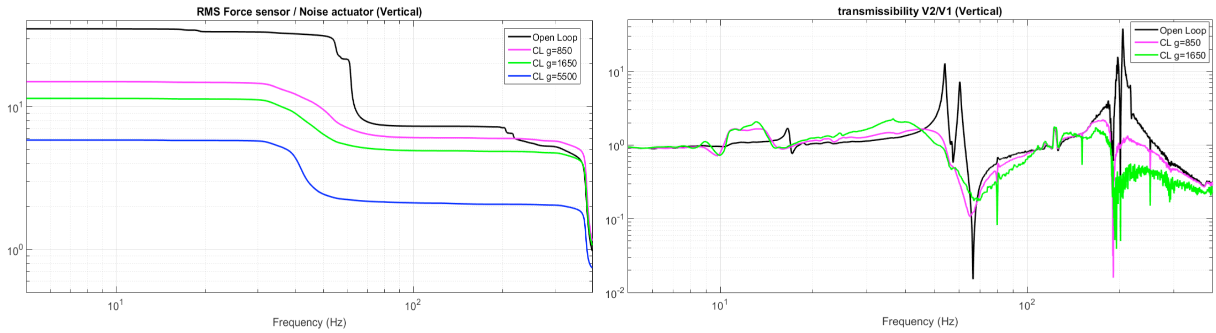


Fig. 7. Vertical excitation. Top left: Transfer function between the shaker noise and one force sensor; Bottom left: integrated (downwards) RMS value of the force measured by one force sensor; Top right: transmissibility between the table top w and m_2 ; bottom right : Transmissibility between the table top w and m_1.

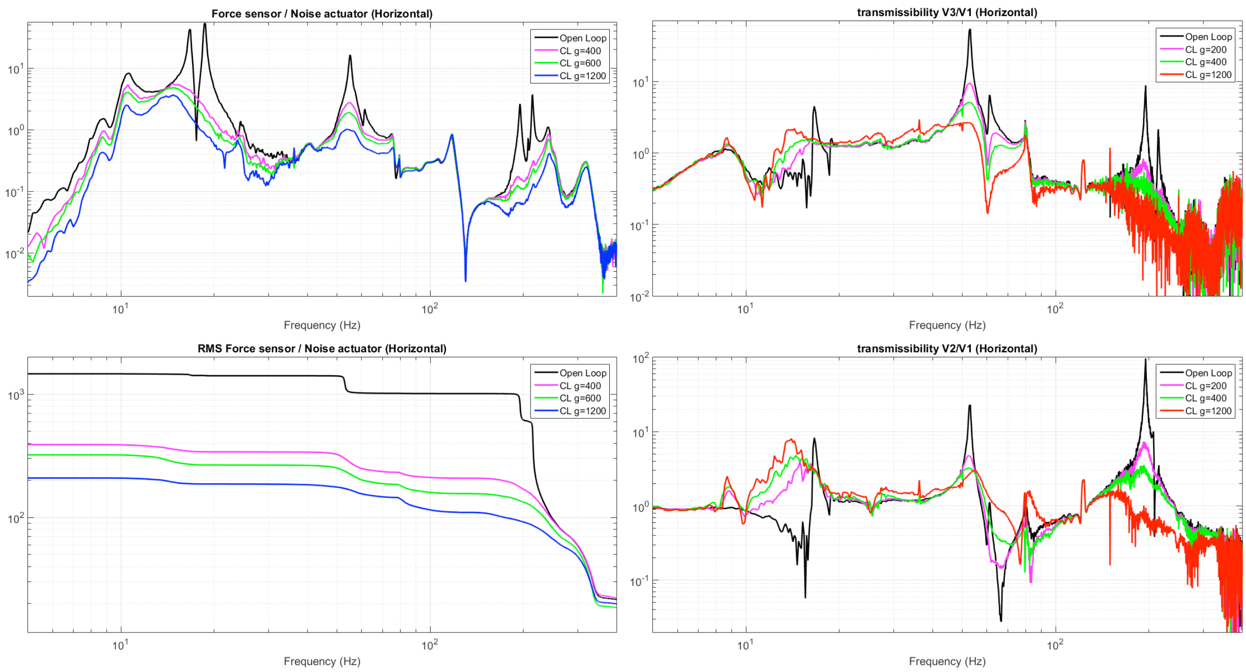


Fig. 8. Horizontal excitation. Top left: Transfer function between the shaker noise and one force sensor; Bottom left : integrated (downwards) RMS value of the force measured by one force sensor ; Top right : transmissibility between the table top w and m_2 ; bottom right : Transmissibility between the table top w and m_1.

5. MOUNTING ON FLEXIBLE STRUCTURE

The third set-up considered in this study is shown in Fig. 9. It is the same as in the previous section, except that a flexible support has been introduced between the isolator and the table top. The objective of this experiment is to test the robustness of the controller in presence of flexibilities on both sides of the isolator.

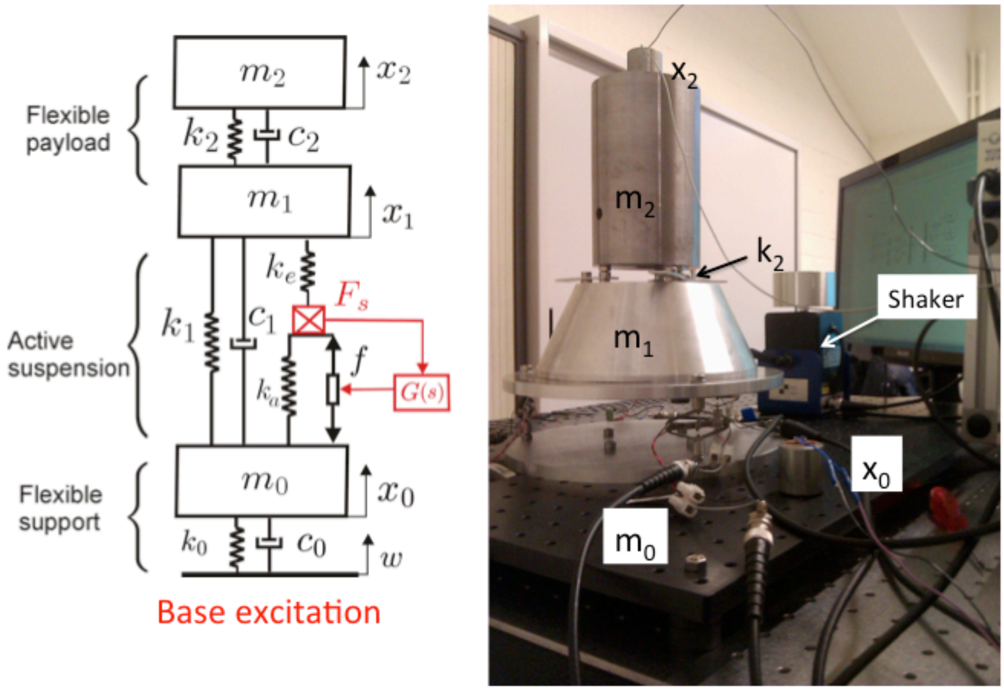


Fig. 9. Flexible payload mounted on a set of three isolators, including a flexible structure below the isolator: picture and simplified sketch showing only the vertical direction.

The same experimental campaign has been conducted: base excitation in both the vertical and horizontal direction with a shaker mounted on the table top, as shown in Fig. 9. As an illustration, Fig. 10 shows the transmissibility between the flexible support and the payload for various values of the control gain, when the structure is excited in the vertical direction. This figure has to be compared with the right part of Fig. 7. Although the modal density is larger, one can still clearly identify the first and the second resonances, respectively around 60 Hz and 190 Hz. The authority on these modes remains unchanged, and no stability issue has been encountered.

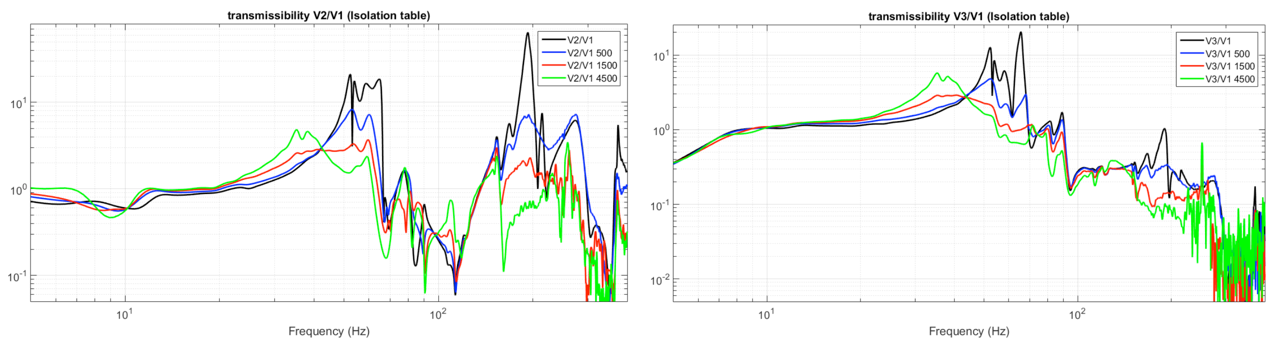


Fig. 10. Transmissibilities x_1/x_0 and x_2/x_0 between the flexible support and the payload for various values of the control gain, when the structure is excited in the vertical direction.

6. CONCLUSION

A concept of active isolator of launcher vibration has been proposed in this paper. The isolator comprises a metallic suspension and a pair of actuator and sensor collocated. The advantages of this concept are that the number and position can be adjusted easily and that it combines a high authority and a good robustness.

The concept has been tested experimentally using a commercial actuator on three set-ups of increasing complexity, ranging from a single d.o.f. payload to a flexible payload, isolated by three active mounts installed on a flexible structures.
In a future work, a dedicated design will be developed and the controller will be further studied, e.g. including dedicated algorithms [15] or combining active and passive damping to reduce the power consumption [16].

7. REFERENCES

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