# Review of Active Vibration Isolation Strategies

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**Abstract**: This paper reviews recent patented developments in active vibration isolation. First of all, the fundamental limitations of passive vibration isolations are established, to understand the motivations to introduce active control in vibration isolation. Then, the main different active strategies are presented using simple systems and compared. Finally, several specific issues are listed and briefly discussed.

**Keywords:** Active vibration isolation, inertial reference, active stabilization, sky-hook spring.

## 1. INTRODUCTION

There is an increasing need for *vibration isolation* in many different fields: magnetic resonance imaging, semi-conductor industry, car suspension, microscopy, machine tools, aircraft, gravitational wave detectors. Even though the level of the vibrations to isolate and of the environmental disturbances can differ by several orders of magnitude, the strategies to achieve the required level of stability can be very similar. Depending on the application, vibration isolation may also be referred to as *vibration suppression*, *vibration cancellation*, *stabilization* or *immobilization*.

The objective of this paper is to review the recent patented developments in active seismic vibration isolation. Essentially precision engineering applications are considered, like lithography, where a very high precision is required, i.e. both a cancellation of the seismic vibrations and a good immunity to external disturbances.

Consequently, the dual problem of force transmission [1] lies out of the scope of this review. Specific systems for narrow band isolation systems for helicopters, like dynamic anti-resonant vibration isolator [2-4] are also not discussed in this paper.

First of all, the performances and fundamental limitations of passive vibration isolation strategies are established, to understand the motivations to introduce active control in vibration isolation. Then, the main different active strategies are presented using single degree of freedom (d.o.f.) systems and compared. Finally, several specific issues in practical realizations are listed and briefly discussed.

## 2. BACKGROUND AND MOTIVATIONS

Figure (**1a**) shows the simplest passive suspension, constituted of a spring $k$ in parallel with a dashpot $c$. The dynamic equation of the system is

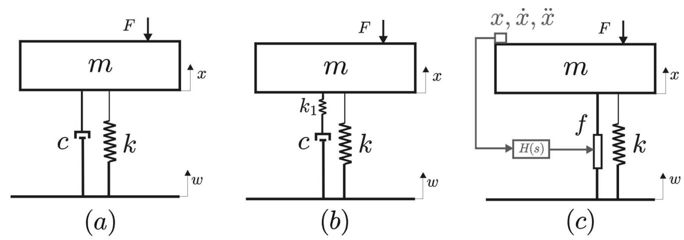


**Fig. (1)**. (**a**) Passive isolator; (**b**) Relaxation isolator; (**c**) Principle of an active isolator.

$$m\ddot{x} + c(\dot{x} - \dot{w}) + k(x - w) = F \quad (1)$$

where $m$ is the mass of the sensitive equipment to isolate, $x$ the displacement of the mass, $w$ the seismic excitation and $F$ represents the perturbations directly applied to the mass.

Using the Laplace transform, Eq.(1) becomes

$$X(s) = \frac{cs + k}{ms^2 + cs + k} W(s) + \frac{1}{ms^2 + cs + k} F(s) \quad (2)$$

or

$$X(s) = T_{wx}(s) W(s) + T_{Fx}(s) F(s) \quad (3)$$

$T_{wx}(s)$ is called the *transmissibility* of the isolator. It characterizes the way seismic vibrations $W$ are transmitted to the equipment $X$. $T_{Fx}(s)$ is called the *compliance*. It characterizes the capacity of disturbing forces $F$ to create displacement of the equipment $X$. Assuming that $F$ and $W$ are not correlated, the power spectral density of the mass displacement, $\Phi_x(\omega)$ is given by

$$\Phi_x(\omega) = |T_{wx}(\omega)|^2 \, \Phi_w(\omega) + |T_{Fx}(\omega)|^2 \, \Phi_F(\omega) \quad (4)$$

where $\Phi_w(\omega)$ and $\Phi_F(\omega)$ are respectively the power spectral density of the seismic vibrations and the power spectral density of the disturbing forces. In order to minimize the vibrations of a sensitive equipment, a general objective to

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design a good isolator is to minimize both $|T_{wx}(\omega)|$ and $|T_{Fx}(\omega)|$ in the frequency range of interest.

The magnitude of $T_{wx}(j\omega)$ is the solid curve shown in Fig. (**2a**). At low frequency, $T_{wx}(j\omega)=1$. An overshoot appears at the resonance frequency of the system. Then, for higher frequencies, the curve is decreasing with a slope of -2 in logarithmic scale. The *isolation* property, i.e. $|T_{wx}(j\omega)|$ smaller than 1, starts at $\sqrt{2}f_n$ (where $2\pi f_n=\sqrt{k/m}$).

In order to increase the passive isolation, the first idea is to reduce the value of $f_n$ as much as possible. However, this still leaves the overshoot in the sensitive low frequency range. The amplitude of the overshoot can be reduced by increasing the damping coefficient $c$. For example, this corresponds to an elastomeric mount. In this case, the price to pay is a degradation of the isolation at high frequency (dashed curve in Fig. (**2a**)). Now, the slope of the roll off is only -1. This is a *first tradeoff* in the design of a passive mount, between damping and isolation. To some extend, this problem can be dealt with, by introducing a second stiffness $k_1$ in series with $c$ (Fig. (**1b**)). It results in a high level of damping, and a roll off with a slope of -2 (dashed-dotted curve in Fig. (**2a**)).

However, as the resonance frequency of the system decreases, a second problem arises. For frequencies below the resonance, the compliance $T_{Fx}(s)$ is moving up with the inverse of $k$. This can be easily seen by taking $s\to 0$ in Eq. (2). In other words, the system becomes rapidly unacceptably sensitive to any external force $F$ directly applied on the equipment. This is the *second tradeoff* in the design of a passive mount, between isolation and robustness to external force. As we will see, these two tradeoffs do not exist anymore in an active suspension.

Figure (**1c**) shows the simplest version of an active isolator, comprising a motion sensor, an actuator $f$ and a control unit $H(s)$. Neglecting for simplicity the dynamics of the actuators and sensors, the general expression of the force delivered by the actuator is

$$f = g_a\ddot{x}+g_v\dot{x}+g_p x \tag{5}$$

where $g_a$, $g_v$ and $g_p$ are constant gains. In this case, Eq.(2) becomes

$$X(s)=\frac{cs+k}{(m+g_a)s^2+(c+g_v)s+(k+g_p)}W(s)+\frac{1}{(m+g_a)s^2+(c+g_v)s+(k+g_p)}F(s) \tag{6}$$

The effect of these gains on the closed loop transmissibility is illustrated in Fig. (**2b**) for each gain individually. An acceleration feedback has an effect of adding virtual mass (dotted curve in Fig. (**2b**)). Only few applications exist, as the same effect can be obtained passively by adding real mass. A velocity feedback creates a so called *sky-hook damper* [5-7], because the actuator delivers the same force as a dashpot linking the mass to an imaginary point fixed in the sky. It relaxes the first tradeoff by removing the overshoot at the resonance (dashed-dotted curve in Fig. (**2b**)), without any degradation of the isolation at high frequency. This strategy, or its semi-active version, is used for example in automotive engineering for car suspensions, or in civil engineering to isolate buildings from seismic vibrations. It can be realized using a classical hydraulic actuator [8] or with an electro-rheological fluid [9,10]. A possible improvement of the strategy is obtained by adding a PID controller on the relative displacement between the payload and the ground to avoid low frequency drifts [11,12]. In this case, the general expression of the control force is

$$f = sg_v x+(g_p^r+\frac{g_i^r}{s}+sg_d^r)(x-w) \tag{7}$$

where $g_p^r$, $g_i^r$ and $g_d^r$ are constant gains of the controller. The proportional gain creates a restoring force of constant $g_p^r$. It can be used to adjust the suspension resonance frequency: a negative value will decrease the resonance, i.e. increase the isolation. The derivative gain creates an

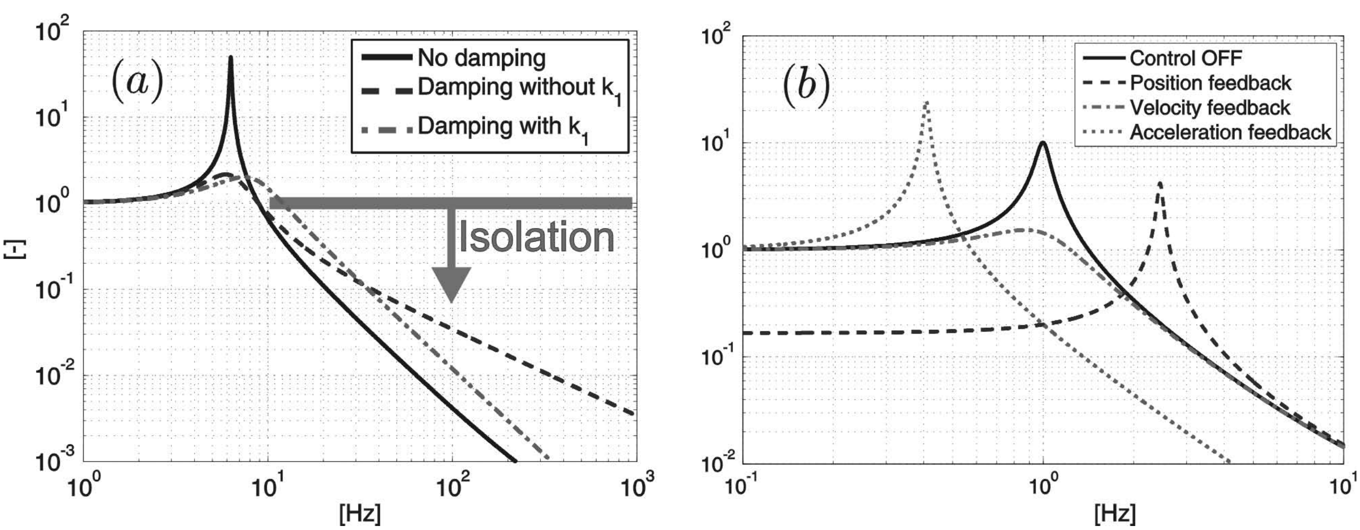


**Fig. (2).** Magnitude of $T_{wx}(j\omega)$ for :(**a**) the isolators shown in Figs. (**1a**) and (**1b**); (**b**) the isolator shown in Fig. (**1c**).

additional damping force of constant $g_d^r$. The integral gain $g_i^r$ is used to remove the drifts of the payload. It can be realized with a Magneto-Rheological (MR) fluid [13-15] or MR elastomer [16,17].

A position feedback creates a so called *sky-hook spring*, because the actuator delivers the same force as a spring linking the mass to an imaginary point fixed in the sky. It relaxes the second tradeoff by creating both an isolation at low frequency (dashed curve in Fig. (**2b**)), and a stiffening of the isolator which improves the robustness to external disturbances (see Eq.(6)). Most of the active isolation strategies are based on a combination of a sky-hook spring and a sky-hook damper. Three different configurations are studied in details in the next section.

## 3. PRACTICAL REALIZATIONS

A major difficulty to realize a sky-hook spring is to measure the position of the equipment. In practice, this is achieved by using an *inertial reference*. Such a sensor consists of an oscillator having an extremely low resonance frequency, equipped with a sensor measuring the relative displacement between the inertial mass and the equipment, as shown in Fig. (**3a**) [18,19]. Another possibility, shown in Fig. (**3b**), is to mount the inertial reference on the ground, and still sense the relative displacement between the inertial mass and the equipment [20-23]. A third configuration is shown in Fig. (**3c**). Here, the active support consists of an intermediate mass mounted on a piezoelectric actuator, and connected to the equipment by a layer of elastomer. It combines a position feedback on the intermediate mass, and a passive isolation provided by the upper stage.

These three configurations are studied in more details in the following section. They correspond to commercial products developed by the companies owning of the cited patents. These products are multi-purpose tables used when a quiet environment is required, e.g. for atomic force microscopy. This is the reason why they are often called *optical tables*.

### 3.1. Inertial Reference on the Equipment

This strategy is basically disclosed in [24-26]. From Fig. (**3a**), the dynamic equations of this first configuration are

$$m_r\ddot{x}_r + k_r(x_r - x) + c_r(\dot{x}_r - \dot{x}) = 0 \tag{8}$$

and

$$m\ddot{x} + c(\dot{x} - \dot{w}) + k(x - w) = F + f \tag{9}$$

where $f$ is the force delivered by the actuator

$$f = -H(s)(x - x_r) \tag{10}$$

and $H(s)$ is the compensator. In the Laplace domain, Eq.(8) becomes

$$\frac{X_r}{X} = \frac{k_r + sc_r}{m_r s^2 + sc_r + k_r} = G_r \tag{11}$$

and Eq.(9) can be rewritten as

$$X = \frac{k + cs}{ms^2 + cs + k}W + \frac{1}{ms^2 + cs + k}F + \frac{1}{ms^2 + cs + k}f \tag{12}$$

Defining $G$ as

$$G = \frac{1}{ms^2 + cs + k} \tag{13}$$

and using Eq.(10) and Eq.(11), Eq.(12) becomes

$$X = (k + cs)GW - H(s)G(X - G_r X) + FG \tag{14}$$

or, after basic manipulations

$$X(s) = \frac{G(k + cs)}{1 + GH(1 - G_r)}W + \frac{G}{1 + GH(1 - G_r)}F \tag{15}$$

From Eq.(15), we see that, above the resonance of the sensor, $G_r$ is very small, and both $T_{wx}$ and $T_{Fx}$ vary as $\frac{1}{G^{-1} + H}$. They are shown respectively in Figs. (**4a**) and (**4b**) using the following numerical values: $m = 100\,kg$,

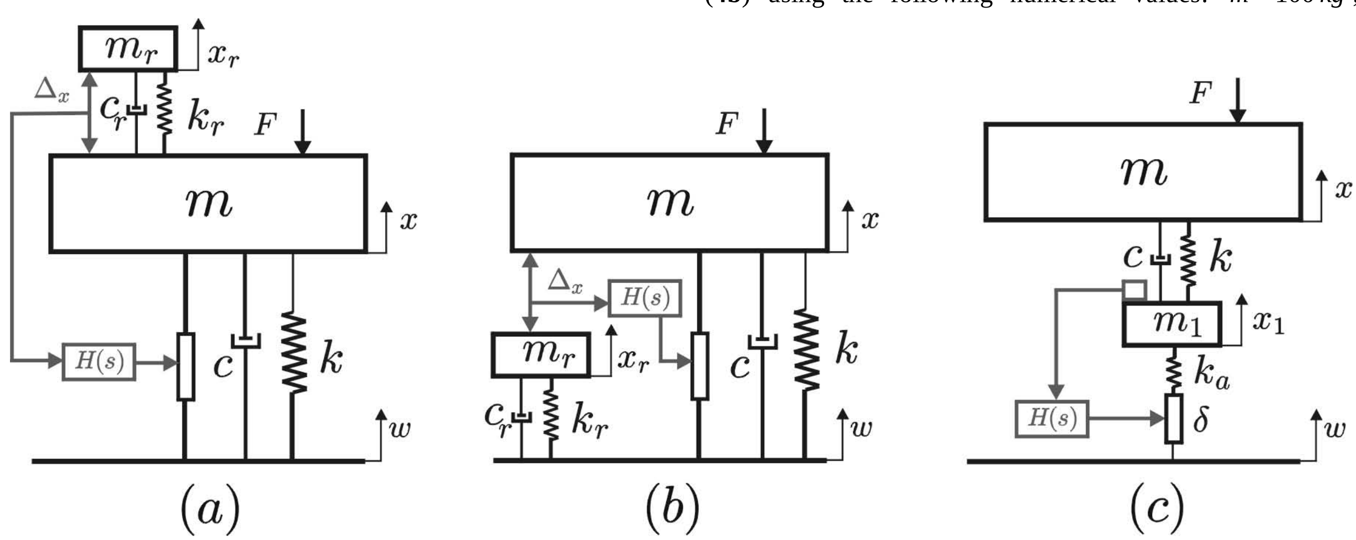


**Fig. (3)**. Three practical realizations of the sky-hook spring.

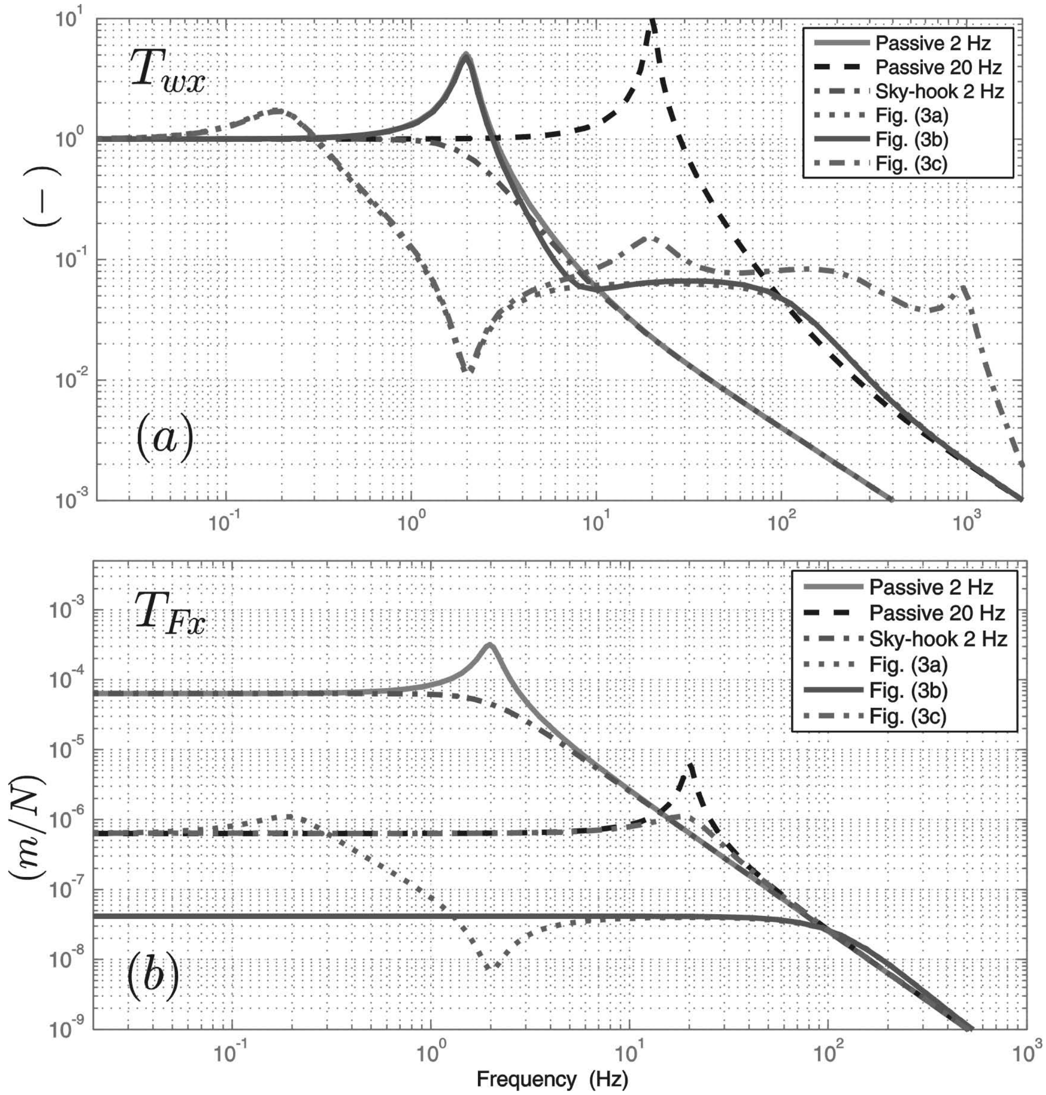


**Fig. (4)**. Various passive and active isolation strategies. (**a**) Magnitude of the transmissibility $T_{wx}(j\omega)$; (**b**) Magnitude of the compliance $T_{Fx}(i\omega)$.

$m_r = 0.1\,kg$; The resonance frequency of the reference mass is $2\,Hz$ and has $10\,\%$ of critical damping; The resonance frequency of the isolator is $20\,Hz$ and has $5\,\%$ of critical damping. The controller has a lead at high frequency to improve the stability and a lag at low frequency to reduce the overshoot. One sees that the resonance of the sensor creates a zero in both curves, and that both curves are actively decreased above this zero.

For the sake of comparison, the transmissibility and compliance of three other isolators are also represented in the figure: two passive isolators with a resonance frequency of $2\,Hz$ and $20\,Hz$, and a sky-hook isolator with a frequency of $2\,Hz$.

### 3.2. Inertial Reference on the Ground

Another possibility consists of mounting the inertial reference directly on the ground, but still measure the difference between the two masses, as shown in Fig. (**3b**). This strategy is described in [27,28]. A similar system is also disclosed in [29,30]. In this case, the dynamic equations are

$$m_r\ddot{x}_r + c_r(\dot{x}_r - \dot{w}) + k_r(x_r - w) = 0 \tag{16}$$

and

$$m\ddot{x} + c(\dot{x} - \dot{w}) + k(x - w) = F + f \tag{17}$$

where $f$ is the force delivered by the actuator

$$f = -H(x - x_r) \tag{18}$$

and $H(s)$ is the compensator. In the Laplace domain, Eq.(16) becomes

$$\frac{X_r}{W} = \frac{sc_r + k_r}{ms^2 + sc_r + k_r} = G_r \tag{19}$$

and Eq.(17) can be rewritten as

$$X = \frac{k + cs}{ms^2 + cs + k}W + \frac{1}{ms^2 + cs + k}F + \frac{1}{ms^2 + cs + k}f \tag{20}$$

If we define $G$ as

$$G = \frac{1}{ms^2 + cs + k} \tag{21}$$

then, using Eq.(18) and Eq.(19), Eq.(20) can be rewritten as

$$X = [(k + cs) + G_r H(s)]GW - H(s)GX + FG \tag{22}$$

or, after basic manipulations

$$X(s) = \frac{G(k + cs + HG_r)}{1 + GH} W + \frac{G}{1 + GH} F \tag{23}$$

The transmissibility and the compliance are shown respectively in Figs. (**4a**) and (**4b**) for comparison. The numerical values are the same as before, and $H(s)$ has a lead filter at high frequency to increase the stability. Above the resonance frequency of the sensor, $G_r \to 0$ and Eq.(23) is identical to Eq.(15), i.e. both $T_{wx}$ and $T_{Fx}$ vary also as $\frac{1}{G^{-1} + H}$. However, in this case, the resonance of the sensor creates a pole in both transmissibilities, i.e. a lower bandwidth at low frequency than the strategy of Fig. (**3a**).

On the other hand, for frequencies lower than the resonance of the sensor, $G_r \to 1$, $T_{wx} \to 1$, but $T_{Fx}$ is still $\frac{1}{G^{-1} + H}$. In other words, at very low frequency, and for the same plant, the strategy of Fig. (**3b**) has the advantage over the strategy of Fig. (**3a**) to be more robust to external forces (see Fig. (**4b**)). The reason is that a force, applied on the equipment at a frequency below the resonance of the sensor, cannot be sensed in the configuration (a), while it can still be sensed (and thus counteracted) in the configuration (b).

A variation of this strategy is presented in [31], where one soft part of the equipment is used as inertial reference. It still works the same, assuming that this part has a much lower resonance frequency than the other part of the equipment.

### 3.3. Hard Mount

The third strategy, shown in Fig. (**3c**), is an example of two-stages active mount. Several versions and embodiments are disclosed in [32-36]. It is constituted of an intermediate mass mounted on a stiff piezoelectric stack and linked to the equipment through a layer of elastomeric material.

In this case, the control law is based on a measurement of the absolute velocity of the intermediate mass with a geophone, and integrate the signal to have the position. The role of the upper stage is to decouple the equipment from the base, and improve the passive isolation at high frequency. From Fig. (**3c**), the dynamic equations of the system, expressed in the Laplace domain, are given by

$$ms^2 X + k(X - X_1) + cs(X - X_1) = F \tag{24}$$

$$m_1 s^2 X_1 + k_a(X_1 - W - \delta) + k(X_1 - X) + cs(X_1 - X) = 0 \tag{25}$$

where $\delta = -H(s)X_1$ is the elongation of the piezoelectric stack actuator, $k_a$ its stiffness, and $H(s)$ is the compensator.

The geophone is modelled by a high pass filter at $2\,Hz$. The equations of motion are rewritten in function of the position mass $x$ and the ground motion $w$. From Eq.(24) we have

$$X = G_1 X_1 + \frac{G_1}{k + sc} F \tag{26}$$

where

$$G_1 = \frac{sc + k}{ms^2 + cs + k} \tag{27}$$

and from Eq.(25)

$$X_1 = G_2 X + \frac{k_a G_2}{cs + k} W \tag{28}$$

where

$$G_2 = \frac{sc + k}{m_1 s^2 + sc + k + k_a(1 + H)} \tag{29}$$

By replacing Eq.(28) in Eq.(26), we get

$$X = \frac{G_1}{(1 - G_1 G_2)(k + sc)} \{k_a G_2 W + F\} \tag{30}$$

At low frequency ($s \to 0$), we have

$$G_1 \to 1 \quad \text{and} \quad G_2 \to \frac{k}{k + k_a(1 + H)} \tag{31}$$

By replacing in Eq.(30), and taking $s \to 0$, we have

$$T_{wx} = \frac{1}{1 + H} \quad \text{and} \quad T_{Fx} = \frac{1}{k} + \frac{1}{k_a(1 + H)} \tag{32}$$

Thus, by feeding back the absolute displacement of the intermediate mass (and not the relative displacement), it also realizes a *sky-hook spring*, but with a much higher stiffness than in the two previous cases. Again, $T_{wx}$ and $T_{Fx}$ are shown in Figs. (**4a**) and (**4b**) for comparison. The numerical values are still the same as for the previous cases. Additionally, $m_1 = 5\,Kg$, and $k_a = 197\,MN/m$. $H(s)$ has a lag at low frequency and a lead at high frequency to increase stability. The transmissibility shows an anti-resonance at the resonance of the sensor (as for the strategy shown in Fig. (**3a**)), and two peaks: one at $20\,Hz$, corresponding to the resonance of the equipment on the stiffness of the elastomer, and one at $1000\,Hz$ corresponding to the resonance of the intermediate mass on the stiffness of the piezoelectric actuator. The overall stiffness of this isolator can be easily increased by replacing the elastomer with a less compliant material. In [37], the elastomer is replaced by Shape Memory Alloy (SMA) for space applications, to avoid shocks during satellite launchings.

A first improvement of the strategy is obtained by mounting an offload spring in parallel with the piezoelectric actuator to reduce the static force applied on it [38,39].

A second improved version of the two stages system is disclosed in [40], where a force actuator is inserted in the upper stage, between the intermediate mass and the equipment. The general expression of the force is

$$f = (g_p^x + \frac{g_i^x}{s} + sg_v^x)sx + (g_p^{x_1} + \frac{g_i^{x_1}}{s} + sg_v^{x_1})sw \qquad (33)$$

where $g_p^x$, $g_i^x$, $g_v^x$, $g_p^w$, $g_i^w$, $g_v^w$, are the constant gains of the controller. This second force can further decrease both the transmissibility and the compliance of the isolator. In this case however, two additional sensors are required, one to measure the payload velocity $sx$ and the other one to measure the ground velocity $sw$. As the performances of this feed-forward part rely on the knowledge of the system, it is better used in an adaptive configuration, like the well known *Least Mean Square* algorithm [41].

In [42], another system with an intermediate mass is disclosed, also having a force actuator between the intermediate mass and the payload, and a pneumatic spring to compensate for the load.

## 4. FIELDS OF APPLICATIONS

In principle, the isolators presented in the previous section can be realized with many different types of actuators and sensors, and can be rendered arbitrarily soft or stiff. The choice of the most suitable strategy, and of the physical elements depends on the application. It has to be guided by: the frequency range of interest, the level of isolation required, resolution required, the type of environment (radiation, magnetic field), the robustness of the feedback loop to uncertainties, the modal density of the payload, the space available, and the price.

However, depending on their stiffness, active isolators can be roughly classified into three categories as a function of the stiffness of the support: Soft, medium and stiff strategies. For each of them, Table **1** provides examples of practical realizations, their main advantages and disadvantages, and examples of fields of applications [43-65].

Soft strategies offer the advantage of a broadband isolation, but are very sensitive to external disturbances. Medium strategies still benefit from the passive isolation at high frequencies. They are widely used for optical applications and lithography. However, their relatively low dynamic stiffness can engender an insufficient compatibility with other tasks, like a micro-positioning stage, and may still be insufficiently robust to external disturbances. In stiff strategies, the isolation is only provided by the feedback operation [66-68]. The high stiffness provides a high robustness to external disturbances, but also a strong coupling. On the other hand, the strong connection between the ground and the equipment has to be properly dealt with, in order to avoid stability issues.

## 5. MULTI-DIMENSIONAL ACTIVE ISOLATION

In the previous sections, the main strategies used for active vibration isolation have been presented on single d.o.f. systems. The transposition of these strategies to a real system having six d.o.f., or more, is not straightforward, because several problems arise. One of them is the coupling between the d.o.f.s. To address this problem, active legs can be mounted either perpendicular to each other [69], or inclined like in the so called *Stewart platforms* [70-72]. Cross-couplings can also reduced numerically, using singular value decompositions [73-76], or modal decompositions [77,78]. The performances of the controller can also be improved using collocated actuators and sensors, to avoid stability problems from higher modes [79]. For example, in a piezoelectric stack mount, one part can be used as sensor, the other one as an actuator [80].

## 6. SENSOR LIMITATIONS

As mentioned in the previous section, the presence of high order modes can significantly reduce the performances

**Table 1. Comparison of Active Vibration Isolation Strategies**

| **Frequency** | **Low (e.g. 2 *Hz* )** | **Medium (e.g. 20 *Hz* )** | **High (e.g. 200 *Hz* )** |
|---|---|---|---|
| Examples | Pneumatic actuator [43-47]; Pneumatic + inertial actuator [48]; Fluid [11,49] | Electromagnetic in parallel with a soft element [50-52,42]; Piezoelectric in series with a soft element (rubber) | Piezoelectric in series with a stiff element (flexible joint) |
| Advantages | Broadband isolation | Passive isolation at high frequency | Extremely robust to external disturbances |
| | | High stability | Robust to magnetic fields |
| Disadvantages | Very compliant | Rather low dynamic stiffness | Noise transmission |
| | Noisy | Low compatibility with positioning stages | Strong coupling |
| Applications | Nuclear magnetic resonance [53]; Semiconductor industry [54-57]; Car suspension [11,58-61]; Scanning probe microscopy [62] | MEMS [63]; Optical tables [33-36] | Aicraft [64,65] |

of the active vibration isolation, and it can be important to account for these modes in the development of the strategy. However, typical sensors used for the measurement of low frequencies (e.g. geophones, seismometers) have a cutoff frequency around $200\,Hz$. Above this frequency, accelerometers have a much better sensitivity. In [81], a procedure to combine the signals from two sensors is disclosed, in order to increase the bandwidth of the controller.

At very low frequency, inertial sensors mounted on the equipment cannot distinguish between a tilt of the equipment and a lateral motion. This is known as the *tilt to horizontal coupling*. An original solution has been disclosed in [82], where the sensor is mounted on a rotational joint.

## 7. CONCLUSIONS

This paper has presented recent patented developments in the field of active vibration isolation. First of all, the motivations for such strategies have been explained. It has been shown that active suspensions do not suffer from the two fundamental tradeoffs inherent to passive suspensions. Then, the main active strategies have been presented and compared using single d.o.f. models. Finally, specific issues in the design of active suspensions have been highlighted, like tilt-to-horizontal coupling and cross-talks.

## 8. CURRENT & FUTURE DEVELOPMENTS

The technology is now developed enough for the active vibration isolation tables to reduce the transmission of the seismic vibrations by a factor up to 100 in the low frequency range. Placed in a classical laboratory, it results in a motion of the table in the nanometer range, mainly limited by the hardware noise floor. Some directions for future improvements of these performances are:

- Improvement of the hardware spectral noise floor (actuator, sensor, amplifier)
- Combining vibration isolation with posture control or pointing [83]
- Maintain the performances for a long service life
- Reduce the sensitivity to the environmental disturbances (forces, magnetic field, temperature, radiation, humidity)

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