Tutorial
Selection of vibration isolators

OPTI 521
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Abstract

Controlling or isolating vibration caused by a ground motion or an acoustic noise is critically important in precise optical fabrications and measurements. There are two methods for controlling or isolating vibration, passive and active vibration isolation. The goal here is to compare passive and active vibration isolation and examine applications for both vibration isolation, such as an optical table and a stage for an optical lithography.

1. Introduction

For precise optical fabrications and measurements, a vibrational circumstance can be tremendously important and critical issue. Vibrational disturbances consist of periodic or nonperiodic forces and include acoustic noise.

A particularly important condition occurs when the frequency of the periodic force nearly or exactly corresponds to the natural frequency f_n of an equipment. In the region around f_n, the transmissibility is over 1, which means the amplified vibration of a ground motion transmits to the equipment. If the transmissibility is lower than 1, the equipment is isolated from the ground motion. The natural frequency f_n of the equipment is given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

By this equation, the natural frequency can be small, if one designs the equipment with very low stiffness and very heavy. But it is not a reasonable way. Instead of changing the design of the equipment, the natural frequency also can be decreased if a vibration isolation system is employed strategically to the equipment.

There are two methods for controlling or isolating vibration, passive and active vibration isolation. In the following chapters, general issues, types, and examples about these tow isolations are discussed.

2. Passive vibration isolation

2-1. General issue

The passive vibration isolation system basically consists of a mass M, a spring K and damper (dash-pot) C shown in Figure 2[1]. In this case, one assumes that a mass M is a rigid body for simplifying the system and can be replaced by an equipment. The vibration which a unit transmits to a supporting structure or the vibration which a unit feels when it is being excited by a vibrating structure can be reduced or attenuated by this isolator.

Isolation is attained primarily by maintaining the proper relationship between the disturbing frequency and the system's natural frequency. The damped natural frequency of the system is described by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}\left(1-\left(\frac{C}{C_C}\right)^2\right)}$$

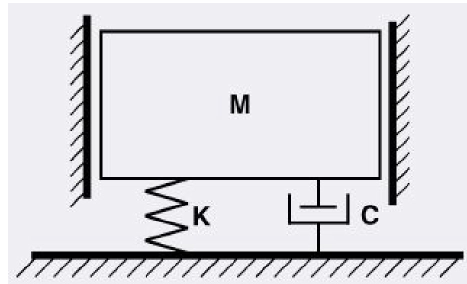

Figure1 Schematic of a passive isolator

C is a damping coefficient with a unit of [Kg-sec/m]. C_C is a critical damping with a unit of [Kg-sec/m]. C/C_C is a damping factor with unitless.

In this case, the equipment on the isolator is assumed to be a rigid body. But, in the real system, the equipment should have its resonant frequency. Then, it is desirable to select the proper isolator so that its natural frequency will not coincide with any critical frequencies of the equipment.

Figure 2 shows that a typical transmissibility curve for an isolated system where f_d = disturbance frequency and f_n = isolation system natural frequency.

Damping is advantageous when the mounted equipment is operating at or near its natural frequency because it helps to reduce transmissibility. Increasing a damping factor C/C_C reduces an isolation system natural frequency. Then a frequency ratio f_d/f_n increase and transmissibility decreases.

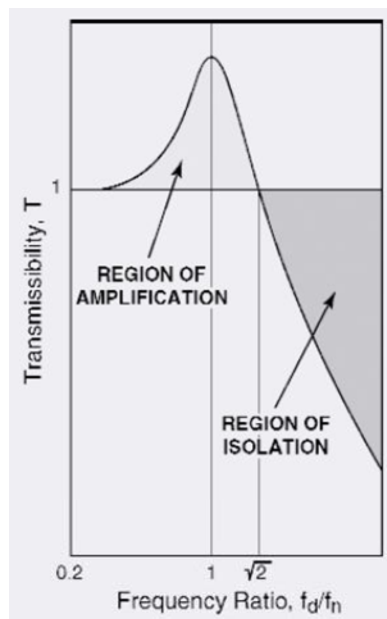

Figure2 Transmissibility curve for an isolated system

2-2. Selection of passive vibration isolators

First of all, a disturbing frequency f_d should be defined. If there is more than one disturbing frequency, one should focus on the lowest frequency. When the lowest frequency is isolated, then all of the other higher frequencies will also be isolated. The disturbing frequency sometimes can be measured directly, but in almost cases, one should estimate the disturbing frequency from several known data or criterion. For typical military and aerospace environments, one can choose the disturbing frequency from

Table 1 indicated by Vukobratovich[2]. Also, one can define the disturbing frequency from generic vibration criterion (VC) in appendix A[3].

Table1

Vibration Power Spectral Densities for Typical Military and Aerospace Environments

Environment	Frequency (f) (Hz)	Power spectral density
Navy warships	1–50	0.001 g^2/Hz
Minimum integrity test	20–1000	0.04 g^2/Hz
per MIL-STD-810E	1000–2000	−6 dB/octave
Typical aircraft	15–100	0.03 g^2/Hz
	100–300	+4 dB/octave
	300–1000	0.17 g^2/Hz
	≥ 1000	−3 dB/octave
Thor-Delta launch vehicle	20–200	0.07 g^2/Hz
Titan launch vehicle	10–30	+6 dB/octave
	30–1500	0.13 g^2/Hz
	1500–2000	−6 dB/octave
Ariane launch vehicle	5–150	+6 dB/octave
	150–700	0.04 g^2/Hz
	700–2000	−3 dB/octave
Space shuttle	15–100	+6 dB/octave
(orbiter keel location)	100–400	0.10 g^2/Hz
	400–2000	−6 dB/octave

Source: From Vukobratovich, D., in *Handbook of Optomechanical Design*, CRC Press, Boca Raton, FL, 1997, p. 65, chap 2.

Next, one should determine the minimum isolator natural frequency f_n defined by

$$f_n = \frac{f_d}{\sqrt{2}}$$

Determining f_n makes it possible to choose the passive vibration isolator. When the passive vibration isolator is chosen, the damping factor C/C_C can be known. Thus, one can calculate transmissibility T given by

$$T = \sqrt{\frac{1+\left(2\frac{f_d}{f_n}\frac{C}{C_C}\right)^2}{\left(1-\frac{f_d^{\,2}}{f_n^{\,2}}\right)^2+\left(2\frac{f_d}{f_n}\frac{C}{C_C}\right)^2}} \qquad T_{\max} = \frac{1}{2\frac{C}{C_C}}$$

By the transmissibility, we can estimate the performance of the passive vibration isolator. Also, the selection can be narrowed down by operating temperature range, size of the isolator, the mount capacity, etc. In general, there are several types of the passive vibration isolators, such as elastomeric isolators, metal springs, Spring-Friction dampers, and spring combined by air damping etc.

2-3. Example of passive vibration isolator

Thorlabs shows a selection matrix for optical tables and optical table supports based on the working environment and application[4] as shown as in Table 2 and 3.

Optical Table and Optical Table Supports Selection Guide

	Quite Environment[1] (PSD < 10^{-10} g^2/Hz)	Typical Laboratory Environment[2] (PSD ~10^{-9} to 10^{-8} g^2/Hz)	Noisy Environment[3] (PSD <10^{-7} g^2/Hz)
Less Demanding Applications • Pulsed Laser • General Spectroscopy • Velocimetry • Multimode Fiber Coupling	Standard, StandardPlus, Performance or PerformancePlus Series of Optical Tables with Rigid Optical Table Supports	Performance or PerformancePlus Series of Optical Tables with Passive Optical Table Supports	Performance, PerformancePlus, Ultra, or UltraPlus Series of Optical Tables with Active Optical Table Supports
General Applications in Photonics • Bioimaging • Raman Spectroscopy • Micropositioning and Machining	Performance or PerformancePlus Series of Optical Tables with Passive Optical Table Supports	Performance or PerformancePlus Series of Optical Tables with Active Optical Table Supports	Ultra or UltraPlus Series of Optical Tables with Active Optical Table Supports
Demanding Applications • Nanopositioning • Submicron Precision • Phase Related • Holography • Single Mode Fiber Alignment	Performance, PerformancePlus, Ultra, or UltraPlus Series of Optical Tables with Active Optical Table Supports	Performance, PerformancePlus, Ultra, or UltraPlus Series of Optical Tables with Active Optical Table Supports	Ultra or UltraPlus Series of Optical Tables with Active Optical Table Supports

[1]The lab floor consists of a subterranean slab in a remote environment.
[2]The lab is in the basement or ground floor of building.
[3]The lab is on the upper floors of a building or near significant sources of vibrations.

Table2 A selection matrix for optical tables and optical table supports based on the working environment and application.

Optical Table Supports Comparison Table

	Active Vibration Isolation Supports	Standard Passive Vibration Isolation Supports	Heavy Duty Passive Vibration Isolation Supports	Rigid, Non-Isolating Supports
Vertical Resonant Frequency	1.25 Hz	4.5 Hz	4.5 Hz	N/A
Vertical Transmissibility at Resonance	10 db	22 dB	22 dB	N/A
Isolation Efficiency at 10Hz	97.5 %	74 %	74 %	N/A
Isolation Type	Active	Passive	Passive	N/A
Damping Efficiency	Best	Good	Good	N/A
Load capacity (set of 4)	5500 lbs (2500 kg)	600 - 2425 lb (275 - 1100 kg)	1200 - 4850 lb (550 - 2200 kg)	5500 lbs (2500 kg)

Table3 . A comparison table for the optical table supports made by Thorlabs.

These tables indicates that

- Less demanding applications do not need a vibration isolator in quiet environment.
- The passive vibration isolator cannot be applied to noisy environment or demanding applications.
- The active vibration isolator can be applied to every application. (If the budget is approved.)

3. Active vibration isolation

3-1. General issue and comparison with passive vibration isolator

The active vibration isolation system essentially uses a feedback or a feedforward circuit which consists of an accelerometer, a controller, and an electromagnetic transducer in addition to a mass, spring and damper. Since the passive vibration isolation system has a resonant frequency f_n, it can isolate a vibration over square root of 2 times f_n. On the other hand, the active vibration isolation system basically does not have a region of amplification, thus it can isolate a vibration in low frequency.

An accelerometer is attached on the table including the passive vibration isolators. The accelerometer can detect a vibration transmitted from the passive vibration isolators. For a feedback control, a controller analyses the frequency and amplitude of the vibration and outputs a feedback signals. An electromagnetic transducer receives the feedback signal and creates a force canceling the vibration. This system can be applied to 6 degree of freedom. In the case of a feedforward control, the accelerometer is located on the floor.

As mentioned above in the chapter 2-1, the passive vibration isolator has a natural frequency. Then, if a floor has a frequency equal to the natural frequency, the vibration is amplified and transmitted to the equipment. Then, the passive vibration is only available in the region over square root of 2 times f_n. If one tries to reduce the natural frequency of the passive vibration isolator, it becomes compliant. This means that it takes a long time to stabilize a vibration of the equipment caused by someone's touching to this equipment. For example, after an adjustment of the equipment or an exchange of a sample, it takes a long time so that one can use this equipment. On the other hand, there is no the region of amplification on the active vibration isolators ideally, and it can isolate a vibration from very low frequency. In addition, since the damping time is very short, the equipment can be stabilized in a short time. In a real active vibration isolator, since it also uses a passive vibration isolator and feedback or feedforward control is not perfect, there might be the region of amplification.

3-2. Selection of active vibration isolators

The active vibration isolators are used for isolating tremendously low frequency, like about 1 Hz or under 1 Hz, and used with the passive vibration isolators. There are two radically different approaches about the active vibration isolators, feedback and feedforward control[5]. A feedback control is used in general but a feedforward control is used in acoustic field frequently.

The principle of feedback is presented in Figure 3.

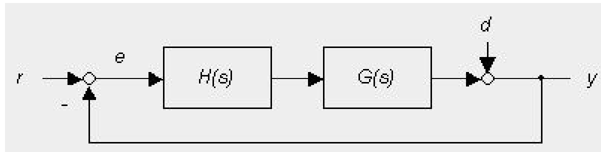

Figure 3. Principle of feedback control

The output y of the system is compared to the reference input r and the error signal, e = r-y, is passed into a compensator H(s) and applied to the system G(s). The design problem consists of finding the appropriate compensator H(s) such that the closed-loop system is stable and behaves in the appropriate manner. The bandwidth ω_C of the control system is limited by the accuracy of the model. There is always some destabilization of the flexible modes outside ω_C.

The principle of feedforward is presented in Figure 4.

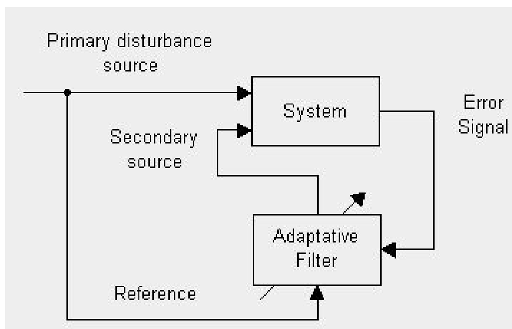

Figure 4. Principle of feedforward control

When a signal correlated to the disturbance is available, the feedforward control can be used. It was originally developed for noise control. The feedforward strongly relies on the availability of a reference signal. The reference signal is passed through an adaptive filter, the output of which is applied to the system by secondary sources. The filter coefficients are adapted in such a way that the error signal at one or several critical points is minimized. Unlike the feedback control, which can only attenuate the disturbances near the resonances, the feedforward control works for any frequency and attempts to cancel the disturbance completely by generating a secondary signal of opposite phase.

The brief summary and comparison about the feedback and feedforward control are listed in Table 4.

Table 4. Comparison of control strategies

Type of control	Advantages	Disadvantage
Feedback	No model needed	Effective only near resonances
	Guaranteed stability when collocated	Limited bandwidth (bandwidth << sampling frequency)
	Global method	Disturbances outside bandwidth are amplified
	Attenuates all disturbances within bandwidth	Spillover (dampling decreases when bandwidth increases)
Feedforward	No model needed	Reference needed
	Wider bandwidth (bandwidth = sampling frequency/10)	Local method (response may be amplifier in some part of the system)
	works better for narrow band disturbance	Large amount of real time computations

3-3. Example of active vibration isolator

Canon develops a lithography system with active vibration isolators as described in the following[6].

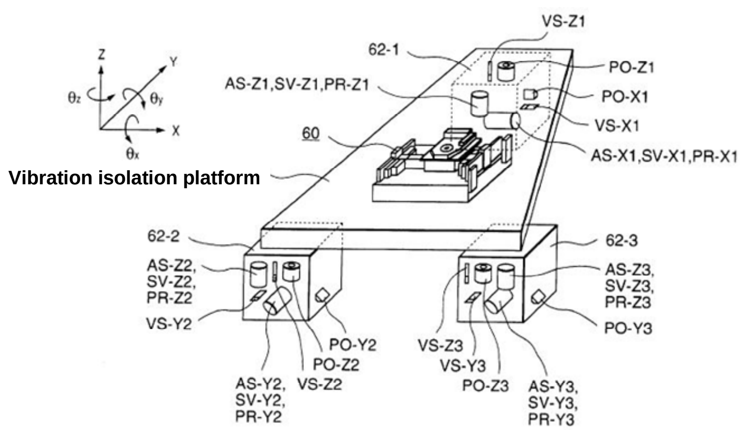

Figure 5. the mechanical structure of the active vibration isolator

Figure 5 illustrates the mechanical structure of an active vibration isolator for controlling the movement and attitude of a generally triangular vibration isolation platform with six degrees of freedom. An optical barrel can be mounted on the vibration isolation platform. It is supported by active supporting legs, which can control two vertical and horizontal axes. The active supporting leg has vibration measurement sensors outputting acceleration and velocity, position measurement sensors, pressure sensors, servo valves, and air-spring actuators.

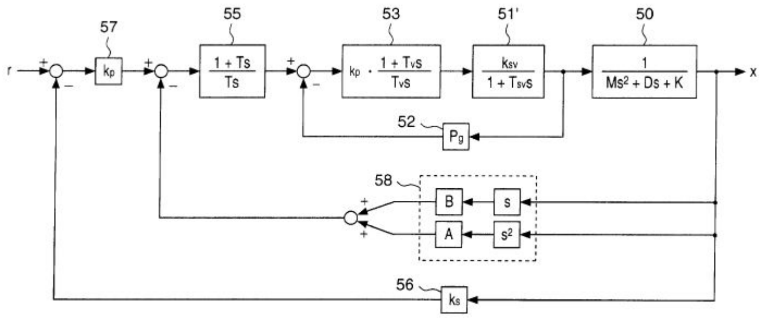

Figure 6. Block diagram of performing control

As Figure 6 shows, this system has three feedback controls for isolating vibration. The first feedback loop feeds back an output of pressure sensor, which measures a working force applied by air-spring actuators to the vibration isolation platform. The first feedback loop can control the working force applied by an air spring actuator.
The second feedback loop has Proportional Plus Integral (PI) (#55 in Figure 6). It produces damping and spring effects to the vibration isolation platform.
The third feedback loop feeds back an output from position measurement sensor. Then the position of the vibration isolation platform can be designated.

By this patent,

- The combination of feedback loop makes the equipment stable from vibration.
- The active vibration isolator is necessary for precision optical fabrication.

4. Conclusion

This paper provides two methods for controlling or isolating vibration, passive and active vibration isolation. For both isolation, the general issues, types, and examples are discussed. The passive vibration isolator can be selected by an environmental vibration criterion. The active vibration isolator can be selected by a requirement of low frequency isolation and types of control. Also, as example of both vibration isolation, optical tables by THROLAB and the stage for lithography by Canon is examined.

5. Reference

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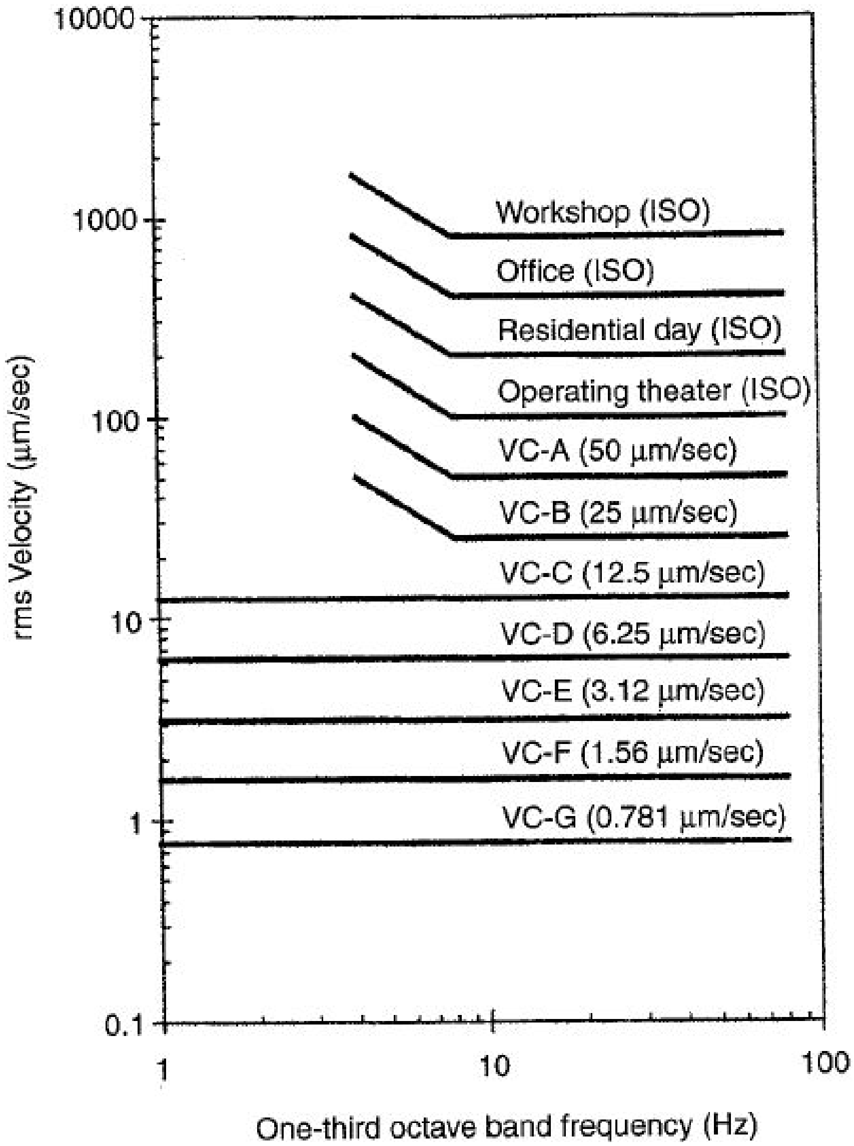
3. P. R. Yoder, in Opto-Mechanical Systems Design, CRC Press, Roca Raton, FL, 2006, p. 50-51
4. THORLAB, http://www.thorlabs.com/newgrouppage9.cfm?objectgroup_id=1105
5. Micromega-dynamics, http://www.micromega-dynamics.com/vibration-strategies.htm
6. S. Wakui, "Active vibration isolator, exposure apparatus, exposure method and device manufacturing method", United States Patent 6,286,644 B1 (2001)

Appendix A



Criterion	Definition
VC-A	256 µg between 4 and 8 Hz; 50 µm/sec (2000 µin./sec) between 8 and 80 Hz
VC-B	128 µg between 4 and 8 Hz; 25 µm/sec (1000 µin./sec) between 8 and 80 Hz
VC-C	12.5 µm/sec (500 µin./sec) between 1 and 80 Hz
VC-D	6.25 µm/sec (250 µin./sec) between 1 and 80 Hz
VC-E	3.12 µm/sec (125 µin./sec) between 1 and 80 Hz
VC-F	1.56 µm/sec (62.5 µin./sec) between 1 and 80 Hz
VC-G	0.78 µm/sec (31.3 µin./sec) between 1 and 80 Hz

Appendix A

TABLE 2.3
Application and Interpretation of the Generic Vibration Criterion (VC) Curves Shown in Figure 2.10

Criterion curve	Amplitude[1] (µm/s) (µm/a)	Detail size[2] (µm)	Description of use
Workshop (ISO)	800 (32,000)	N/A	Distinctly perceptible vibration. Appropriate to workshops and nonsensitive areas
Office (ISO)	400 (16,000)	N/A	Perceptible vibration. Appropriate to offices and nonsensitive areas
Residential day (ISO)	200 (8,000)	75	Barely perceptible vibration. Appropriate to sleep areas in most instances. Usually adequate for computer equipment, probe test equipment, and microscopes less than 40x
Operating theatre (ISO)	100 (4, 000)	25	Vibration not perceptible. Suitable in most instances for microscopes to 100x and for other equipment of low sensitivity
VC-A	50 (2,000)	8	Adequate in most instances for optical microscopes to 400x, microbalances, optical balances, proximity and projection aligners, etc
VC-B	25 (1,000)	3	Appropriate for inspection and lithography equipment (including steppers) to 3 µm line widths
VC-C	12.5 (500)	1–3	Appropriate standard for optical microscopes to 1000x, lithography and inspection equipment (including moderately sensitive electron microscopes) to 1 µm detail size
VC-D	6.25 (250)	0.1–0.3	Suitable in most instance for demanding equipment, including many electron microscopes (SEMs and TEMs) and e-beam systems
VC-E	3.12 (125)	<0.1	A challenging criterion to achieve. Assumed to be adequate for the most demanding of sensitive systems including long path, laser-based, small target systems, E-beam lithography system working at nanometer scales, and other systems requiring extraordinary dynamic stability
VC-F	1.56 (62.5)	N/A	Appropriate for extremely quiet research spaces; generally difficult to achieve in most instance, especially cleanrooms. Not recommended for use as a design criterion, only for characterization
VC-G	0.78 (31.3)	N/A	Appropriate for extremely quiet research spaces; generally difficult to achieve in most instances, especially cleanrooms. Not recommended for use as a design criterion, only for characterization

Notes: (1) As measured in one-third octave bands of frequency over the range 8 to 80 Hz (VC-A and VC-B) or 1 to 100 Hz (VC-C through VC-G).

(2) The detail size refers to linewidth in the case of microelectronics fabrication, the particle (cell) size in the case of medical and pharmaceutical research, etc. It is not relevant to imaging with probe technologies, atomic force microscopy, and nanotechnology.

Source: Courtesy of the Institute of Environmental Sciences and Technology, Rolling Meadows, IL, USA.