Vibration Isolation for Optical Science and Engineering

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ABSTRACT

The following tutorial discusses the concept of vibration isolation in optical tables. Vibration isolation is a function of the environment, application, and components of the isolation system. A general background on vibration and vibration isolation is provided. Additionally, the concept of additional figures of merit to quantify vibration isolation is introduced. Finally, a list of suppliers of vibration isolation systems is provided for reference.

Keywords: *Compliance, damping, dynamic deflection coefficient, isolation, optical tables, relative motion, vibration.*

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1. INTRODUCTION

All buildings and laboratories are subject to some amount of vibration, yet modern optical science and engineering includes numerous fields of work and study in which sensitive equipment requires a vibration-free environment to provide the highest-quality data results and at the same time avoid excessive wear. Thus, in disciplines such as lithography, optical and electron microscopy, and long-path laser applications, the use of vibration isolation systems is commonplace.

2. VIBRATION FUNDAMENTALS

The combined effect of numerous vibration sources closely approximates random vibration (amplitude and frequency). In such an environment, sensitive optical equipment may suffer rapidly from wear, and measurements may be degraded in the form of image noise and blur. To achieve vibration isolation, the logical first step is to determine the vibration environment and the degree of isolation required for the application. The vibration environment will relate to the selection of isolators and the sensitivity of the application will relate to the selection of the tabletop. Common sources of vibration and their associated frequencies and amplitudes are listed in Figures 1 (right) and 2 (below), describing various applications and the maximum allowable levels of vibration.

Source	Frequency (Hz)	Amplitude (in.)
Air Compressors	4–20	10^{-2}
Handling Equipment	5–40	10^{-3}
Pumps	5–25	10^{-3}
Building Services	7–40	10^{-4}
Foot Traffic	0.55–6	10^{-5}
Acoustics	100–10,000	10^{-2} to 10^{-4}
Air Currents	Labs can vary depending on class	Not applicable
Punch Presses	Up to 20	10^{-2} to 10^{-5}
Transformers	50–400	10^{-4} to 10^{-5}
Elevators	Up to 40	10^{-3} to 10^{-5}
Building Motion	46/height in meters, horizontal	10^{-1}
Building Pressure Waves	1–5	10^{-5}
Railroads	5–20	±0.15g
Highway Traffic	5–100	±0.001g

Fig. 1: Common Sources of Vibration[1]

Criterion Curve	rms Amplitude (μm/sec)*	Detail Size (μm)	Description of Use
Workshop (ISO)	800	N/A	Distinctly discernible vibration. Appropriate to workshops and nonsensitive areas
Office (ISO)	400	N/A	Discernible vibration. Appropriate to offices and nonsensitive areas
Residential Day (ISO)	200	75	Barely discernible vibration. Probably adequate for computer equipment, probe test equipment and low-power (to 20 ×) microscopes
Operating Theatre (ISO)	100	25	Vibration not discernible. Suitable in most instances for microscopes to 100 ×
VC-A	50	8	Adequate for most optical microscopes to 400 × , microbalances, optical balances, proximity and projection aligners
VC-B	25	3	Appropriate for optical microscopes to 1000 × inspection and lithography equipment (including steppers) to 3 micron line widths
VC-C	12.5	1	A good standard for lithography and inspection equipment to 1 μm detail size
VC-D	6	0.3	Suitable for the most demanding equipment, including electron microscopes (TEMs and SEMs) and E-beam systems.
VC-E	3	0.1	A difficult criterion to achieve in most instances. Assumed to be adequate for long-path laser-based interferometers and other systems requiring extraordinary dynamic stability

Fig. 2: Applications and Maximum Allowable Vibration Levels[2]

2.1 – Simple Harmonic Motion

Simple harmonic motion describes the sinusoidal pattern of a mass on a spring subjected to an external force. The resulting pattern of motion will resonate with a natural frequency, ω_0, as the spring stores and imparts energy to the moving mass. The natural frequency is proportional to the spring constant, k, and the mass, m.

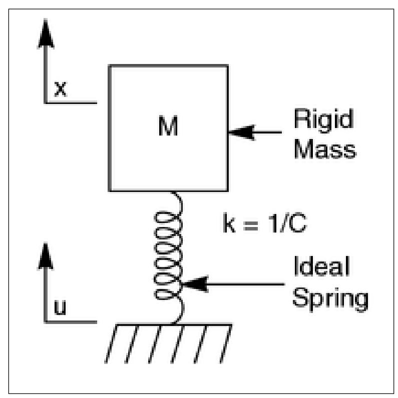

Fig. 3: SHO[3]

A mass connected to an ideal linear spring will move as a simple harmonic oscillator (SHO), where the motion is described as:

$$\mathbf{mx'' + k(x - u) = 0}$$

In the equation above, $|x|$ is the mass motion amplitude and $|u|$ is the spring-end motion amplitude. The transmissibility, T, is the ratio of $|x|$ to $|u|$ and can be written as:

$$\mathbf{T = |x| \,/\, |u| = 1 / (1 - (\omega^2/\omega_0^2))}$$

Where ω describes the range of frequencies that the system is subjected to.

Plotting the transmissibility as a function of the frequency ratio defines the characteristics of a simple harmonic oscillator, which can be divided into three distinct regions: 1) $\omega \ll \omega_0$ (far below resonant frequency), $\omega^2/\omega_0^2 \cong 0$ and $T \cong 1$, so mass motion is very similar to that of the spring; 2) $\omega \cong \omega_0$ (near resonant frequency), motion of the mass asymptotically goes to infinity, in theory only; 3) $\omega \gg \omega_0$ (far above resonant frequency), $u \cong 1 - \omega^2$ and the motion of the spring is not transmitted to the mass, i.e. the spring acts as an isolator. Figure 4 below shows a plot of transmissibility as a function of frequency ratio.

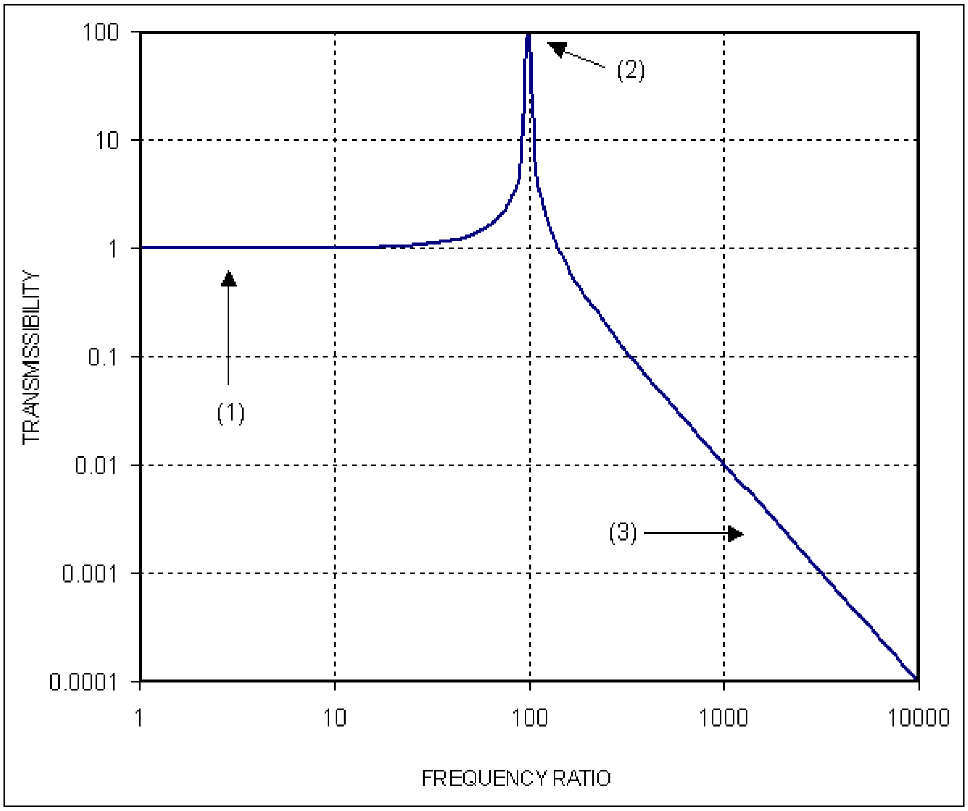

Fig. 4: Transmissibility vs. Frequency Ratio, SHO

2.2 – Damped Simple Harmonic Motion

A simple harmonic oscillator that includes an element that dissipates energy from the mass-spring system is called a damped simple harmonic oscillator (DSHO). In such a system, motion can be described by the following equation:

$$\mathbf{mx'' + bx' + k(x - u) = 0}$$

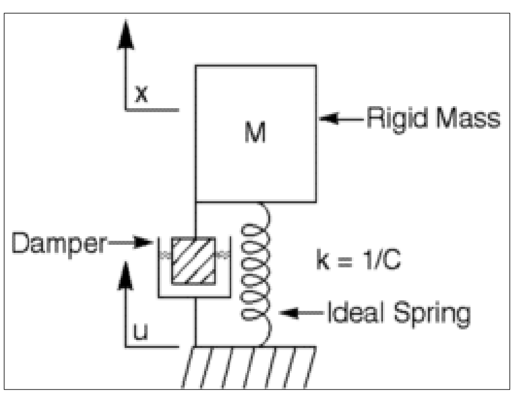

Fig. 5: DSHO[4]

The damping element adds a term proportional to the velocity of the mass to the motion equation. The transmissibility of a damped system subjected to an external force can be described by an equation slightly more complicated that that of the SHO:

$$\mathbf{T = |x| \,/\, |u| = [(1 + (2\zeta\omega/\omega_0)^2) \,/\, (1 - (\omega^2/\omega_0^2)^2 + (2\zeta\omega/\omega_0)^2)]^2}$$

With the inclusion of a damping term (ζ, defined as the ratio of actual damping, C, to critical damping, C_c), the graph of transmissibility as a function of the frequency ratio can be plotted for various levels of damping, as seen in Figure 6 below.

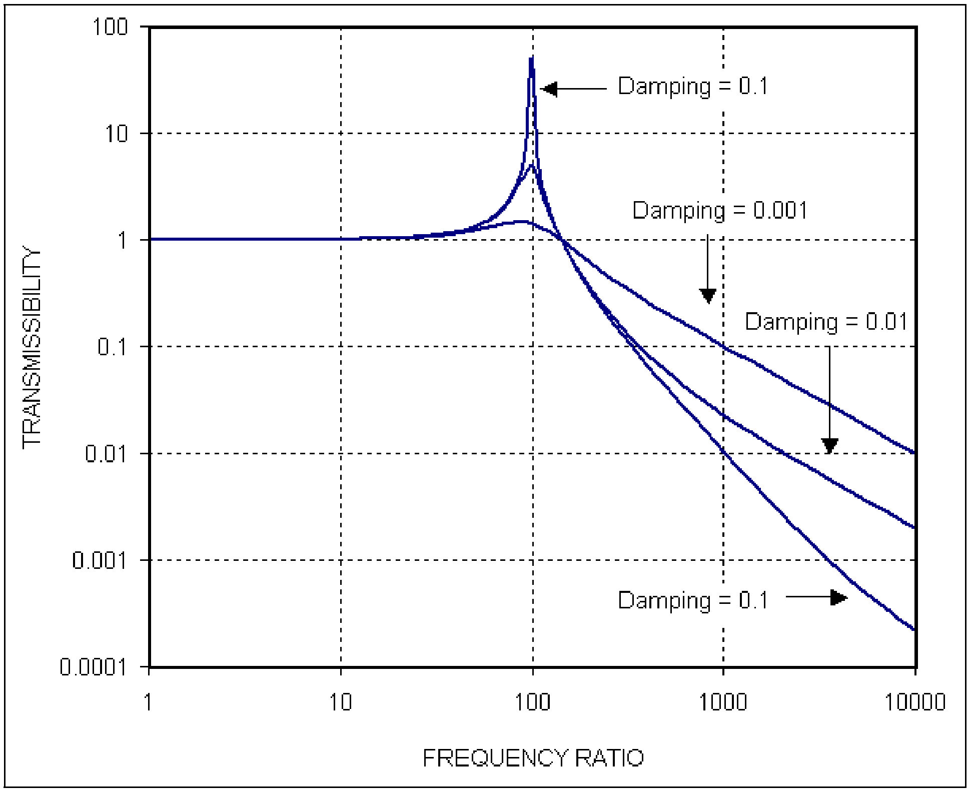

Fig. 6: Transmissibility vs. Frequency Ratio, DSHO

As the value of the damping term increases, the amplitude of T at resonance decreases; however, the decline of T at higher frequency ratios is less. For $\zeta = 0$, i.e. no damping, the curve is identical to that of the SHO, displayed in Figure 4 above.

3. VIBRATION ISOLATION

For effective vibration isolation, an optical table must perform effectively in three basic areas. First, the table needs to offer a rigid surface on which optical and mechanical parts can be mounted and aligned. Second, the table must adequately damp vibrations inherent to the experiment or application. Finally, successful elimination of external vibrations must be met. The first two criteria are inherent to the optical table. The third criteria are inherent to the isolators.

3.1 – Tabletop

The tabletop should provide a stiff, flat surface for mounting of optical and mechanical components without being overly massive. Currently, it is generally accepted that the best overall performing tabletops are constructed of metal honeycomb sandwiched between flat metal plates[5]. The honeycomb structure offers numerous benefits over other materials:

Fig. 7: TMC CleanTop™ II Optical Tabletop Cutaway[6]

- A high rigidity-to-mass ratio
- Open cells allow for an array of mounting holes
- Rapid response to reach thermal equilibrium

The natural frequency of the table should be significantly higher than that of the isolators it rests on to ensure that energy coupling does not take place. Considering that the ambient vibrations in the majority of laboratory application is in the range of 1 – 100Hz, the isolators should have a low natural frequency – on the order of 5Hz – and the optical table should have a natural frequency significantly above 100Hz.

3.2 – Isolators

The isolators in a vibration isolation system serve to eliminate vibration from the environment in which the optical system is utilized. This isolation is achieved by using a mounting system with an extremely low resonant frequency.

Although there are numerous materials and designs of isolators, the majority of isolators used in modern optical tables with vibration insulation employ pneumatic isolators. The pneumatic isolator is essentially a flexible diaphragm on a column of compressed air. The air is compressed or allowed to expand in response to vibrations. A pneumatic isolator's resonant frequency is given by:

$$\omega_n = [(r \times A \times g) / V]^{0.5}$$

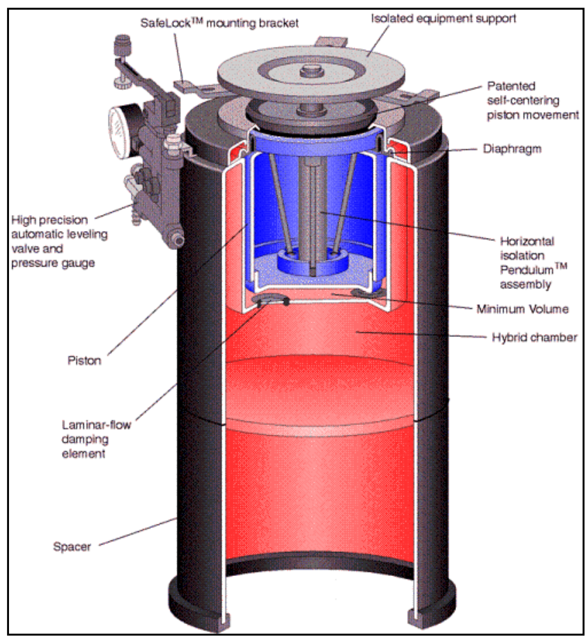

Fig. 8: Newport Corp. Pneumatic Isolator Cutaway[7]

Where r is the specific heat ratio for the air in the spring, A is the area of the piston, g is gravity, and V is the volume of the cylinder.

In practical application, isolators are often somewhat more complex than the simple concept described above. In addition to vertical vibration isolation, damping, horizontal vibration isolation, and load leveling are also integrated into the isolator system.

3.3 – Performance Specification

The major manufacturers of vibration isolation systems for optical applications agree that the most import measures of vibration control in optical tables are the tabletop's dynamic and static rigidities. Both characteristics define how an optical table will flex in response to an applied load.

3.3.1 – Dynamic Rigidity

The tendency of a structure to move as a result of an applied force is known as compliance, which is defined mathematically as:

$$\mathbf{C = |x| \,/\, |F|}$$

Where $|F|$ is an applied force, and $|x|$ is the resulting displacement.

For any structure, a compliance curve can be generated, describing the displacement of a point on a structure due to an applied force as a function of frequency. The dynamic rigidity is a measurement of a structure's resistance to bending when subjected to vibration, and is derived directly from the compliance curve.

Related to the compliance curve is the maximum amplification at resonance, Q, which is defined as the maximum compliance value of the highest peak above the ideal rigid body line. Effective damping is achieved with a low Q value, indicating that the stability of the structure will be better.

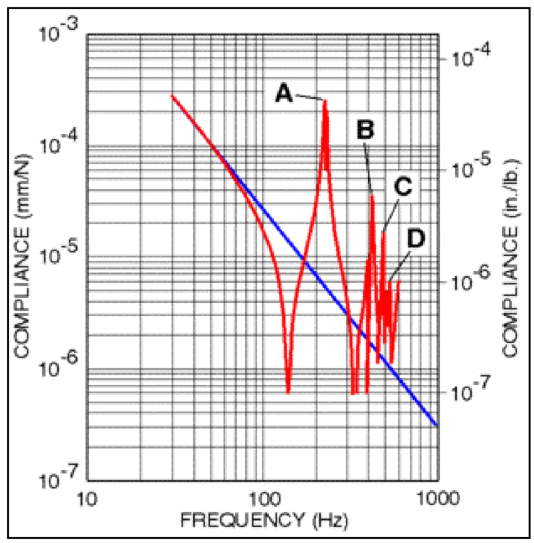

Fig. 9: Undamped Optical Table Compliance Curve[8]

3.3.2 – Static Rigidity

Every surface will exhibit some amount of downward deflection when subjected to an applied load. The static rigidity measures a structure's stiffness under an applied load. Static rigidity is proportional to the applied load, geometry of the optical table, Young's modulus of the table skin, and shear modulus of the table core. Obviously, it is preferable to have a minimum amount of sag in an optical table, so as not significantly affect the alignment of components.

3.4 – Relative Motion & Dynamic Deflection Coefficient

Newport Corporation, a well-known supplier of vibration isolation systems, makes a compelling case for additional figures of merit besides dynamic and static rigidity. The first figure of merit is known as relative motion, and is derived as follows:

A tabletop's acceleration response to random vibration is given by:

$$\mathbf{G_{rms} = [(\pi/2) \times \omega_n^{\,2} \times Q \times (PSD)]^{0.5}}$$

Where PSD (Power Spectral Density) describes how the power of a signal or time-series is distributed with frequency. PSD can be measured directly or estimated quite reliably, assuming random vibration.

The relative displacement of the tabletop is given by:

$$\boldsymbol{\delta = (G_{rms} \times g) / (2\pi \times \omega_n)^2}$$

Combining these two equations gives the displacement response of a tabletop to random vibration:

$$\boldsymbol{\delta = (1/(32\pi^3))^{0.5} \times g(Q/\omega_n{}^3)^{0.5} \times (PSD)^{0.5}}$$

Finally, the relative motion formula is completed as:

$$\textbf{Maximum Relative Motion @ } \boldsymbol{\omega_n = g(1/(32\pi^3))^{0.5} \times (Q/\omega_n{}^3)^{0.5} \times (PSD)^{0.5} \times T \times 2}$$

The second term in the maximum relative motion formula, $(Q/\omega_n{}^3)^{0.5}$, defines the second figure of merit, the dynamic deflection coefficient, which is derived from the structure's minimum resonant frequency and maximum amplification at resonance.

It is interesting to note that the maximum relative motion and dynamic deflection coefficient are not universally accepted figures of merit. Although utilized by both Newport Corporation and Melles Griot Inc., other suppliers of vibration isolation systems either stick to more traditional figures of merit[9], or simply dismiss maximum relative motion and dynamic deflection coefficient as meaningless[10].

4. SUPPLIERS

Below is a non-exhaustive list of suppliers that provide optical tables, isolators, and vibration isolation systems:

- Newport Corp.	(www.newport.com)
- Melles Griot Inc.	(www.mellesgriot.com)
- Technology Manufacturing Corp.	(www.techmfg.com)
- Kinetic Systems Inc.	(www.kineticsystems.com)
- Standa Ltd.	(www.standa.lt)
- New Focus Inc.	(www.newfocus.com)
- Elliot Scientific Ltd.	(www.elliotscientific.com)
- Spiers Robertson	(www.speirsrobertson.com)

In addition to an array of products, many of these suppliers offer technical data related to vibration isolation.

5. SUMMARY

Numerous disciplines in optical science and technology require vibration isolation for precision testing and measurement. The degree of vibration isolation is a function of the environment, application requirements, and optical table / isolators capabilities. A solid understanding of each of these parameters is essential to enable an engineer to select the proper components of a vibration isolation system, balancing cost and performance.

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