Fundamentals of Vibration Isolation

9

The Need for Optical Tables

In the modern research or application laboratory it is often necessary to conduct experiments or make measurements in a vibration-free environment. Optical systems with multiple components which must be individually mounted and aligned in a precise and rigid fashion are particularly vulnerable to vibration-induced performance degradation. Vibrational sources are common in virtually every environment. Vacuum pumps, elevator mechanisms, seismic activity, air conditioners, heat pumps, road and rail transport, and many other sources all contribute to the vibrational background noise that is coupled to the foundations and floors of the surrounding buildings.

Because visible light has a wavelength of approximately 0.5 μm, interferometry-based experiments (including holography) may be impossible to perform in the presence of vibrations of even submicron amplitude. Many laser applications involve focusing the beam to a waist of only a few microns in radius. If the position of this spot is critical to system performance, then vibrations with amplitudes in the micron range can ruin an experiment. Optical and mechanical machining or probing of semiconductor wafers requires similar stability. Furthermore, many experiments involve mechanical elements that move or vibrate and therefore must be vibrationally isolated from all other critically aligned elements.

To be useful, the surface on which an optical system is mounted must satisfy several basic requirements. It must provide a rigid base on which optics can be mounted and aligned reliably, with both long-term stability and no inherent vibrational resonances. It must successfully damp any vibrations caused by motorized or moving parts in the experiment, thus preventing these vibrations from influencing critical optical elements. Finally, it must isolate the experiment as a whole from ambient background laboratory vibrations. If these criteria are not met, undesirable effects often result—an individual component, or the system as a whole, may not function properly; valuable data may be buried in random noise; or data may be totally misunderstood and wrongly evaluated as a result of vibrationally induced noise.

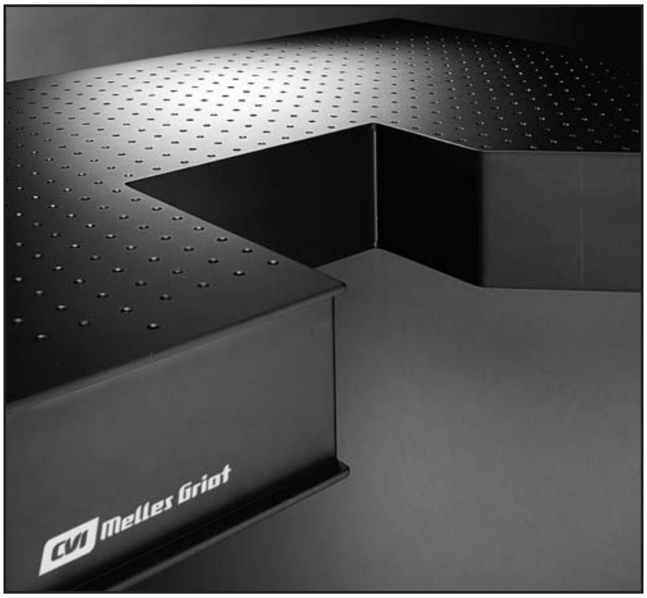
To help solve these problems, optical tables and breadboards of various designs have been developed. An ideal optical table is designed to maintain a rigid and flat upper surface without being overly massive. The table is then mounted seismically, usually on air springs, to prevent the coupling of ambient background vibration. In the past, tabletops have been made from granite, concrete, wood, steel, and a variety of unusual composite structures in attempts to improve performance while keeping weight at an acceptable level. Each has advantages and disadvantages, but it is now generally accepted that the best overall performance is achieved by using a composite structure consisting of a sandwich of metal honeycomb bonded between flat metal plates.

The ambient vibrational noise in the structure of a building is generally in the 4- to 100-Hz range; consequently, vibration-isolation mounts (supports) are designed to minimize transmitted vibrations at frequencies above a few cycles per second. Energy cannot be transferred from one type of vibration to another if separated by a large difference in frequency. The resonant frequency of CVI Melles Griot isolation supports is typically 5 Hz or less. The resonant frequency of a CVI Melles Griot tabletop is well above 100 Hz. This separation between the resonant frequencies of the tabletop and its support system ensures that no energy coupling can take place.

As with any other high-performance product, optical table design is an ongoing process, with new technology and ideas being incorporated continually to enhance performance. CVI Melles Griot product development has been focused particularly on three areas:

- Improving the performance of the tabletops by using advanced damping technology
- Developing a cost-effective table with performance well in excess of that needed for most applications
- Addressing the often overlooked issue of horizontal isolation in the vibration isolation supports

Most commercial isolation mounting systems for optical tables directly damp vertical motion by use of a damping orifice in the air springs. There is no direct horizontal isolation and damping. Instead, they rely on the inefficient coupling between the vertical and horizontal modes. CVI Melles Griot horizontally enhanced isolators incorporate a long pendulum mechanism designed specifically to provide direct horizontal damping.



Optical tables provide a stable flat working surface.

Fundamentals of Vibration Isolation

Sources of Vibration

Consider a laboratory in a location where a number of energy sources exist simultaneously. The experiment is being performed on a supported steel plate. A mechanical beam chopper has been placed on this plate, accompanied by an oscilloscope with a line-synchronous cooling fan, and a vacuum pump rests on the floor. The steel plate will be undamped and, if excited mechanically or acoustically, will "ring" for long periods of time. The chopper, scope, and pump will provide mechanical excitation because there is a direct mechanical link with no isolation. It is also highly likely that the pump will radiate acoustic energy and that the cooling fan will generate variations in local air density, thereby changing the path length of any interferometer. However, this is by no means the whole story. Some other potential sources frequently identified in laboratory installations are shown in figure 9.1. Therefore, it is common in laboratories to find an ambient noise spectrum where the inputs are most likely to be structural and acoustic. The accompanying table lists some of the most common sources of noise (acoustic and structural energy).

Typically the frequency content will range from 4 to 100 Hz and peak around 40 Hz. Further properties of the inputs are that they can be multiaxial, random, and periodic. It can be seen from this list that a laboratory needs to be carefully selected and managed efficiently as a "quiet room." Structural inputs are most likely to dominate, simply because the energy coupling efficiency between mechanically connected objects is high compared to acoustic excitation and because structural sources are direct displacement inputs.

Common Vibrational Sources

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

Thermal disturbances from air conditioning systems and cooling fans also can cause relative motion between components owing to material expansion and contraction as a function of temperature fluctuations. Most air conditioning systems will typically only maintain temperatures to within one degree per hour. Additionally, some experimental techniques will be

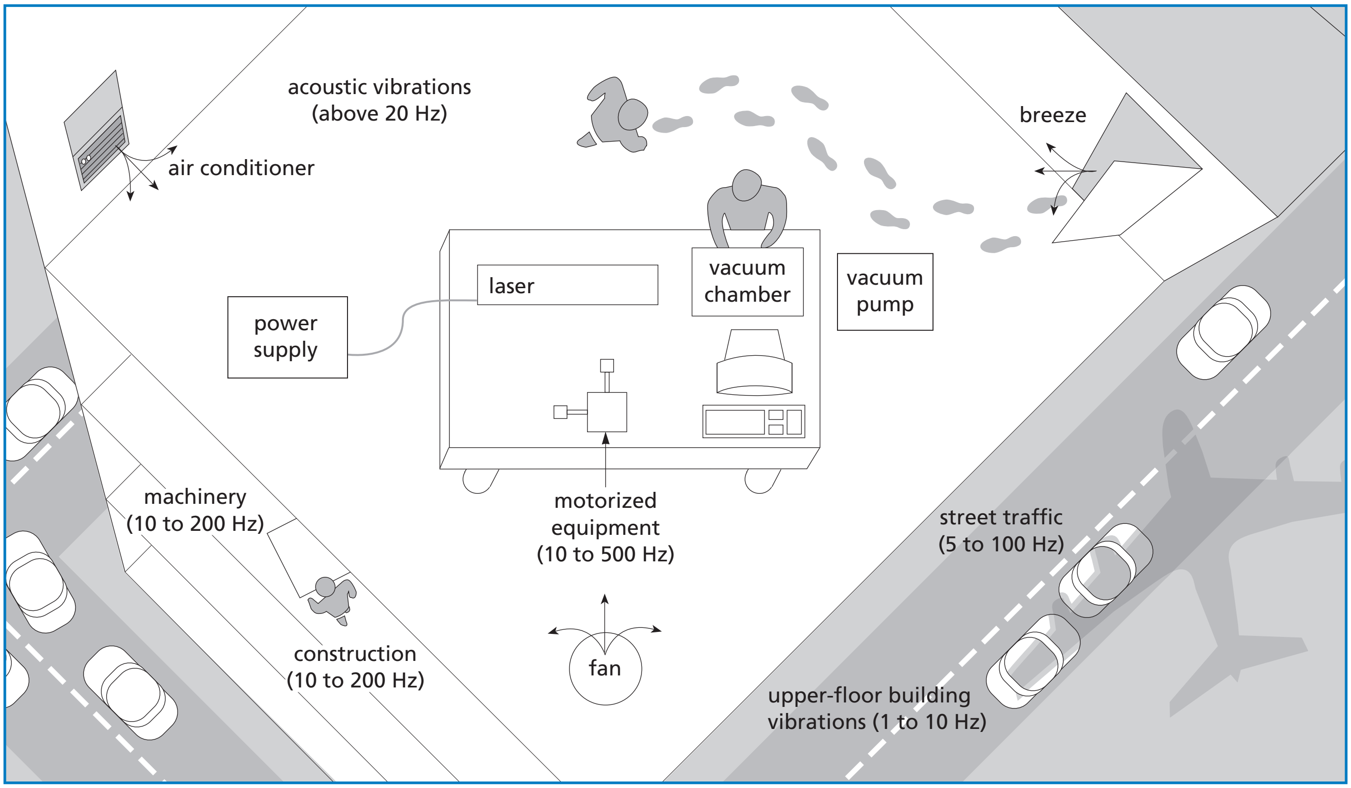


Figure 9.1 **Common environmental sources of noise**

sensitive to virtual movements as a consequence of changes to the refractive index of air and its density, both of which have a temperature dependency.

If the above energy sources are not considered fully, it is possible that sensitive electro-mechanical equipment can experience structural damage and reduced signal-to-noise ratios. Optical equipment can experience low-frequency jitter, resulting in blurred images, reduced resolution, and the appearance of line thickening.

VIBRATION CRITERIA

When determining the isolation from ambient vibration required for a specific facility, a common misconception is that the most expensive equipment will account for all eventualities. However as we have seen, without due care and attention to the planning of the whole laboratory, the potential performance of the platform can be seriously compromised.

To this end it is useful to first identify some vibration criteria, in terms of maximum permissible levels for the measurements to be made without significantly degrading the signal-to-noise ratio of the measurements. This is not difficult because most manufacturers of analytical equipment will specify limits in velocity and acceleration amplitudes with respect to frequency.

If specific data are not available, generic criteria can be used. Colin Gordon & Associates have widely published criteria backed by many years of research and experience, which can be used as a means of setting targets for the maximum permissible vibration levels on the working surface. These generic criteria are summarized in figure 9.2 and described in the table.

With a clear understanding of the potential noise sources and vibration criteria needed for high-quality results, it is now possible to construct a vibration-isolation system and control regime. If it is not possible to remove noise sources from the laboratory and its local environment, the system should attenuate all dynamic inputs in the range of from 4 to 100 Hz and simultaneously minimize the duration of any impulsive disturbance by providing a level of damping to the working surface. Thermally induced sources must also be minimized and, where necessary, the area above the working surface should be enclosed to negate air currents.

ASSESSING THE ENVIRONMENT AND APPLICATION

Laboratory environments are continuously subjected to ambient vertical vibrations in the region of from 4 to 40 Hz. In the wind, tall buildings may sway up to a meter with a horizontal frequency of from 1 to 10 Hz. Vibrations caused by equipment and machinery often have a somewhat higher frequency, but usually they are below 200 Hz. Vibrations, both natural and manmade—always with us— produce relative motions of components that may be imperceptible to the casual observer and yet are often disastrous to a wide range of precision experiments.

Before choosing a vibration-isolation system it is important to determine both the severity of the environment and the sensitivity of the application. The sensitivity of the experiment primarily relates to the selection of the tabletop; the severity of the environment primarily relates to the

Application and Interpretation of Criterion Curves

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

**Data courtesy of Colin Gordon Associates*

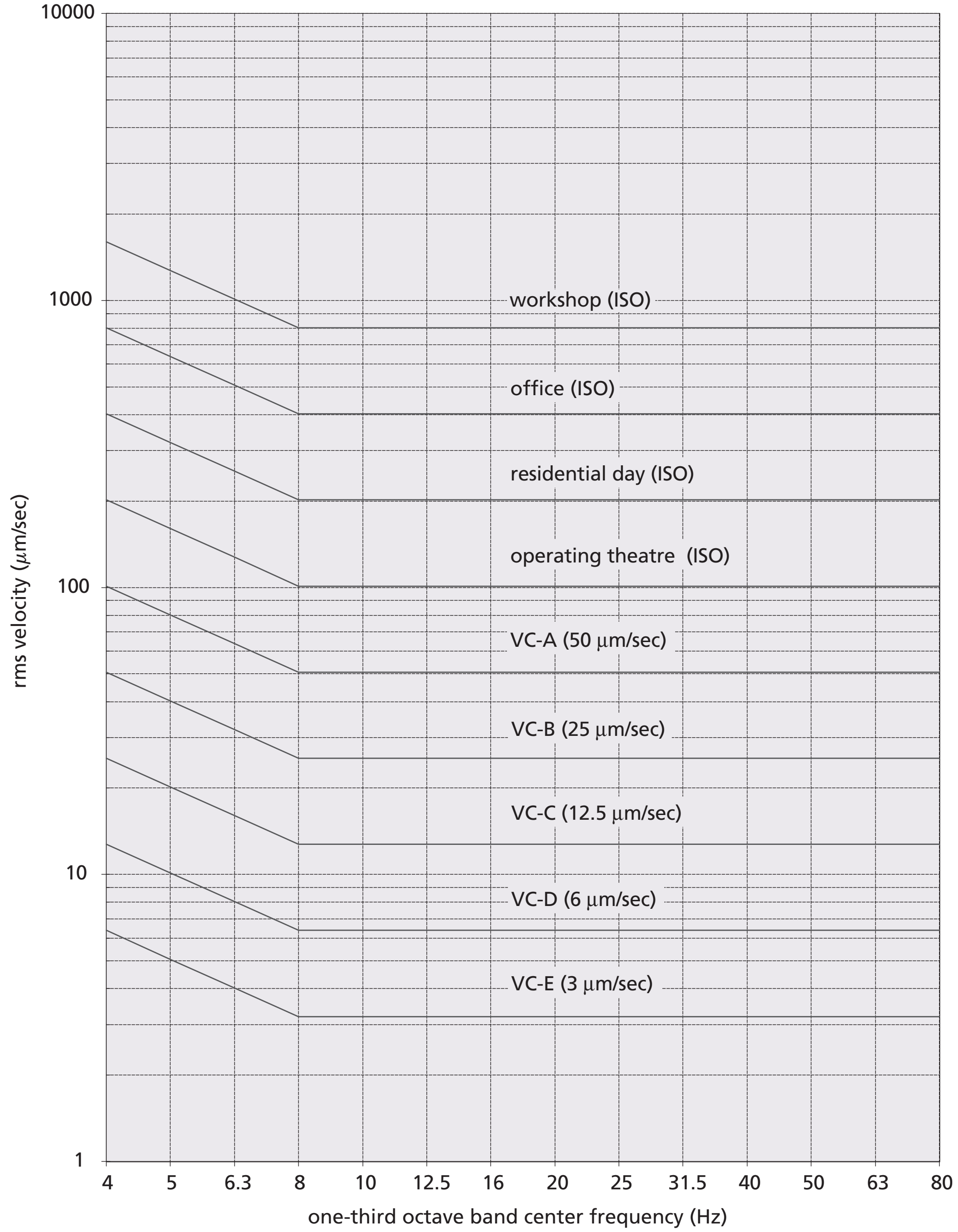


Figure 9.2 **Generic vibration criterion**

selection of the isolation system. After both aspects have been considered separately, the overall requirements can be reviewed together to ensure that the tabletop is compatible with the mounting system.

A SIMPLE GRAPHICAL MODEL

The following examples show typical experiments and their environmental situations. They provide simple guidelines for selecting the most appropriate vibration-isolation equipment. In each case, two arrow gauges are used to characterize the overall requirement: one represents the sensitivity of the experiment, and the other represents the severity of the environment, as shown in figure 9.3. Both arrows range from a low of 0 to a high of 10. For the experiment gauge, 10 represents an exceedingly vibration-sensitive application, such as long-exposure holography, whereas 0 represents an application where only a large working surface is required. For the environment gauge, 10 represents a site in a high, steel-frame building in a heavily trafficked area, whereas 0 represents a remote basement location with no passing traffic. Using this model, considerations of the experiment in its intended environment can be interpreted graphically as the relative position of the two gauges. The following examples illustrate the use of these gauges.

Case A, shown in figure 9.4, is typical of a less-demanding experiment conducted in an environmentally sound location. Case B, shown in the same figure, typifies a highly sensitive experiment conducted in the most demanding location. However, in reality, it is more common to experience the situations shown in cases C and D.

Case C (figure 9.5) is typical of a delicate interferometer located in a basement and mounted on a seismic block, hence incurring negligible environmental disturbances.

Case D (figure 9.6) represents a relatively insensitive experiment located on the fourth floor of a building. This experiment needs considerably better environmental isolation than the previous case.

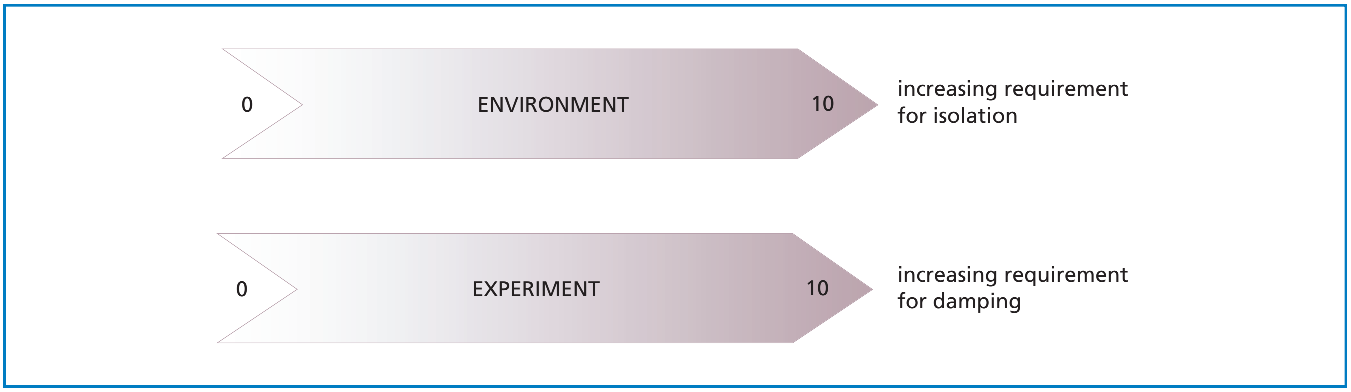


Figure 9.3 **Simple arrow gauges representing environmental and experimental conditions**

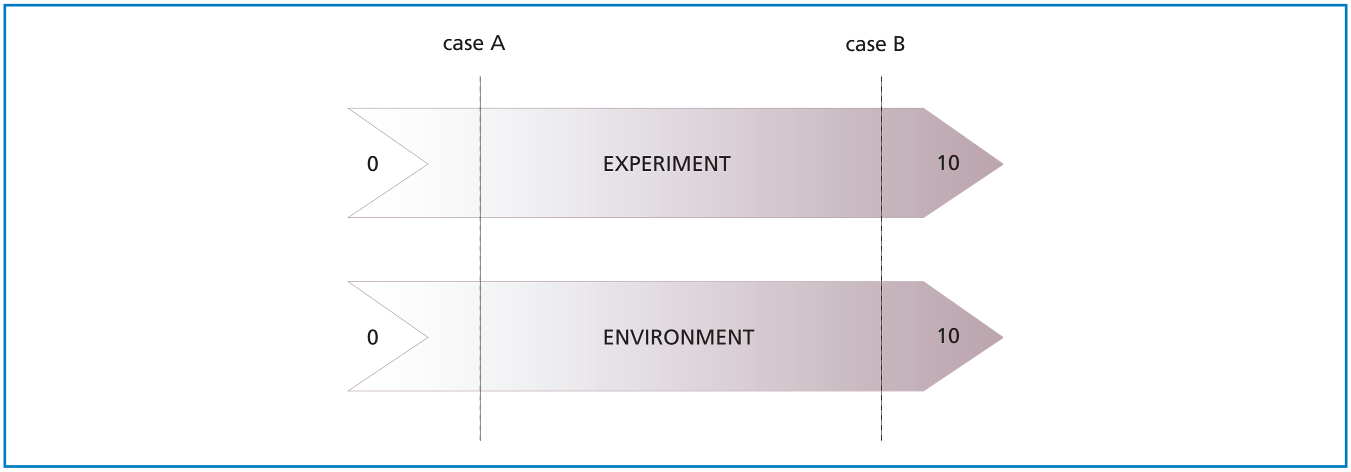


Figure 9.4 **Cases A (low sensitivity/low severity) and B (high sensitivity/high severity)**

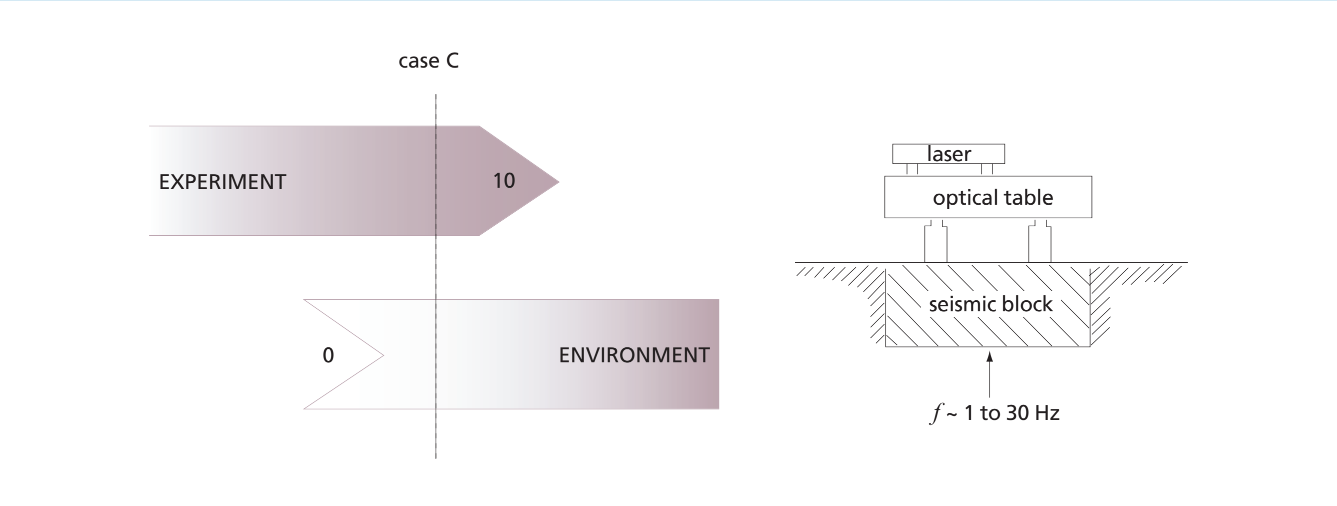


Figure 9.5 **Case C (high sensitivity/low severity)**

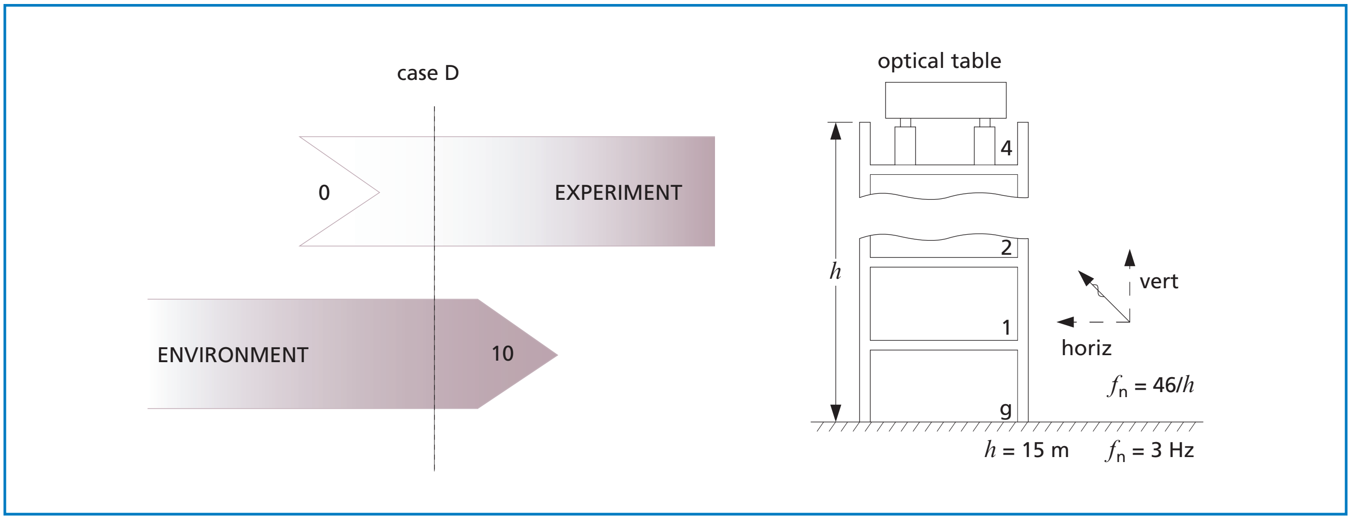


Figure 9.6 **Case D (medium sensitivity/medium severity)**

Theory of Tabletop Vibration

The primary goal of optical table design is to eliminate relative motion between any components on the surface. Before the design of optical tables can be discussed, however, it is necessary to examine the underlying theory and to understand the appropriate nomenclature: compliance, vibration, resonance, and damping.

COMPLIANCE

A common problem in engineering and physics is the deformation of a body or structure in response to external forces. These forces may be static or dynamic. The design of an optical tabletop is a good example of this problem. A static force, such as that caused by a large, localized mass placed on the table, can cause the table to sag. A dynamic force, such as the acoustic vibrations in the air, the vibrations of a small motor sitting on top of the table, or vibration induced from the building into the tabletop through its mounting points, can cause the tabletop to vibrate and deform.

Compliance is the most widely used transfer function for the vibrational response of an optical table. In the case of a constant (static) force, this is defined as the ratio of the linear or angular displacement to the magnitude of the applied force. In the case of a dynamically varying force (vibration), the compliance is the ratio of the excited vibrational amplitude (angular or linear displacement) to the magnitude of the forcing vibration. Any deflection of the tabletop is evidenced by the change in relative position of the components mounted on the table surface. Therefore, by definition, lower compliance values mean a better table because the deflection of the surface on which components are mounted is minimized.

Compliance, used to measure deflection at different frequencies, is measured in units of displacement/unit force, i.e., meters/Newton.

Compliance Curves

To understand compliance, consider the hypothetical structure, shown in figure 9.7, with only one vibrational degree of freedom (i.e., a structure with only one direction of deformation). This could be a steel bar, firmly anchored at one end, which can bend only in one plane.

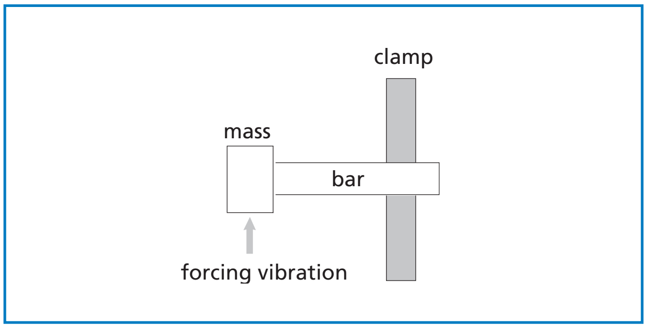


Figure 9.7 **A simple one degree of freedom system: a constrained bar vibrating in only one plane**

All periodic vibrations can be expressed as combinations of sine and cosine functions, together with appropriate amplitude, frequency, and phase. Therefore, consider a single frequency sinusoidal vibration applied to the bar. From Newton's laws, the general equation of motion is

$$ma + cv + kx = F_0 \sin ft \qquad (9.1)$$

where the left-hand side pertains to the forced system and the right-hand side pertains to the forcing function, and

a is the acceleration

v is the velocity

x is the displacement

m is the mass being moved or deflected

c is the damping

k is the stiffness

$F_0 \sin ft$ is a sinusoidally varying force with frequency f, maximum amplitude F_0, and time t.

The general expression for compliance of a system such as this is given by

$$\text{compliance} = \frac{x}{F} = \frac{1}{\left[\left(k - mf^2\right) + \left(cf\right)^2\right]^{1/2}} . \qquad (9.2)$$

If we restate the above equation in words,

$$\text{compliance} = \frac{1}{\sqrt{(\text{stiffness} - \text{mass effects}) + \text{damping}}} . \qquad (9.3)$$

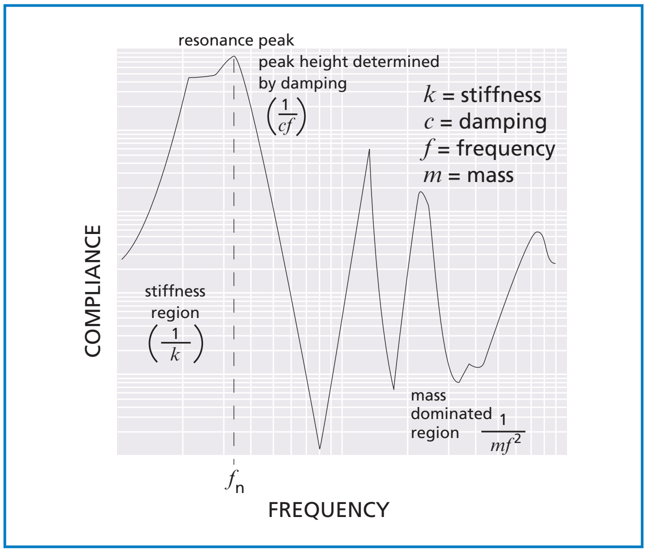


Figure 9.8 **Compliance versus frequency for a one degree of freedom system**

CVI Melles Griot

A plot of compliance versus frequency shows that the compliance of a rigid body can be separated into three parts: stiffness, resonance effects, and mass effects, as shown in figure 9.8.

Compliance at Low Frequencies—Stiffness

At low frequencies, the stiffness term dominates the compliance equation. When a low-frequency forcing vibration is applied to the unattached end of our bar, it bends in response. The amount of deflection is determined by the stiffness of the bar which ultimately depends on its shape, the tensile modulus of elasticity (Young's modulus) of the bar material, and the method of mounting and/or constraining the bar.

Any solid body has a fixed equilibrium (rest) geometry which is what distinguishes a true solid from a liquid. The equilibrium geometry is that which minimizes the potential energy. When forces are applied to a solid body it can be deformed from this equilibrium shape; however, the potential energy of the body rises and is manifested as resistive forces, which act to restore the equilibrium geometry.

When the bar is deflected, the restoring forces attempt to return it to its equilibrium position; however, the momentum of the bar causes it to overshoot the equilibrium position. Restoring forces then act in the opposite direction to return the bar to its equilibrium position. Again, momentum causes the bar to overshoot. This oscillation is an example of a simple harmonic oscillator. The oscillation occurs with a characteristic frequency f_n—the resonant frequency—given by

$$f_n = \sqrt{\frac{k}{m}}\ . \qquad (9.4)$$

In the absence of damping, this oscillation would persist forever. When a body is vibrating at its resonant frequency, the energy in the vibration alternates between potential and kinetic. At the maximum deflection or displacement, the velocity is zero, the acceleration is maximum, the potential energy is maximum, and the kinetic energy is zero. At the equilibrium, or zero displacement, the opposite is true.

In a real system, such as an optical table, resonant vibrations rarely approximate harmonic vibrations with such simple mathematics; however, the discussion above is still valid.

Compliance at Resonance

When the forcing vibration is at the resonant frequency, each maximum in the velocity of the forcing vibration coincides with a maximum in the acceleration of the excited vibration. This adds to the acceleration of the bar, which thereby accumulates vibrational energy and actually amplifies the forcing vibration. This is in accordance with equation 9.2, which can be solved to show

$$\text{compliance} = \frac{1/k}{\left[\left(1 - f^2/f_n^2\right)^2 + \left(2\,\zeta f/f_n\right)^2\right]^{1/2}} \qquad (9.5)$$

where f is the frequency of the forcing function and ζ is the damping ratio. From this equation, it can be seen that when the frequency of the forcing function is close to the resonant frequency, the compliance is determined solely by the damping term and can be quite large.

Another way to picture resonance is to consider what happens as the forcing function is raised slowly from zero, as shown in figure 9.9. At forcing frequencies near zero, the bar bends synchronously with the forcing vibration. As the frequency is increased, the bar starts to lag behind the forcing vibration because it has momentum and cannot reverse direction instantaneously in response to the periodic direction changes in the applied force (i.e., vibration). The bar eventually halts and reverses direction due to its own restoring forces. The same thing happens at the opposite extreme of the motion.

The phase lag continues to increase with frequency until it is exactly 90 degrees at the resonant frequency. At this point, the balance of the bar momentum and its restoring forces causes the maximum in the bar acceleration to coincide with (and be in the same direction as) the maximum in the velocity of the forcing vibration, resulting in amplification of the vibrational input.

At forcing vibration frequencies greater than the resonant frequency, the phase difference between forcing and excited vibrations in this theoretical single-resonance system is 180 degrees.

In a real structure such as an optical table, there are many possible resonant modes of vibration. The phase of an excited vibration is also dependent upon where on the table it is measured, and upon what type of vibration is being excited.

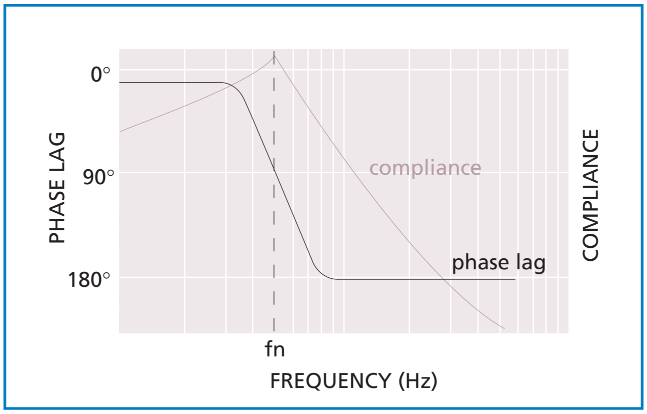


Figure 9.9 **The variation of phase lag between excited and forcing vibrations for a system with a single resonance**

MODIFYING A RESONANT FREQUENCY

Ignoring damping for the moment, the compliance curve is dominated by stiffness at low frequencies and by mass effects above the resonant frequency. The frequency of resonance can be determined from the difference between these two terms in the equation for compliance, and as we saw in equation 9.4, is given by

$$f_n = \sqrt{\frac{k}{m}} \ .$$

The quantity (k/m) is known as the stiffness-to-mass ratio. The resonant frequency of a simple system with one degree of freedom can be altered by changing this stiffness-to-mass ratio. Decreasing the mass and/or increasing the stiffness shifts the resonance to a higher frequency and vice versa, as shown in figure 9.10.

The balance of the mass and stiffness is a simple, yet important, concept in optical table design because the goal is to push the table resonances to higher frequency and lower amplitude. Clearly, increasing the stiffness of the table by proportionally increasing the mass will not shift the resonance. The designer's goal is to maximize the stiffness while minimizing the mass.

DAMPING

Damping refers to any process that causes an oscillation in a solid body to decay to zero amplitude. It is a very important phenomenon in vibration suppression or isolation in real systems because it causes energy to be diverted from vibration to other energy sinks.

Damping is a resonant effect, inasmuch as it significantly affects the compliance function at or near resonance, i.e, when $f \cong f_n$. The height of the compliance peak at resonance is determined primarily by the amount of damping. In the absence of any damping, the peak would be infinitely high, and the system would vibrate ad infinitum; however, the natural damping of the materials themselves prevents this.

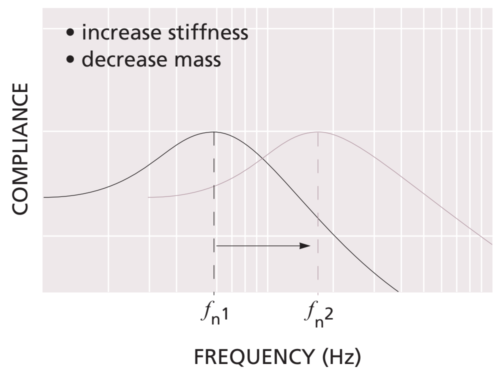


Figure 9.10 **Shifting the resonant frequency by changing the stiffness to mass ratio**

The microstructure of a well-damped material is such that deformations cause strains in the material, which rapidly degrade the energy as heat (i.e., internal energy). Materials such as wood and rubber have a large amount of natural damping. Metals manifest a small amount of internal damping caused by a small amount of friction at grain boundaries.

In many instances a structure may consist of supporting elements which are quite rigid, with little natural damping, and tend to ring. If the structure is being designed to avoid vibrations, additional damping is required. There are many methods of introducing damping into a system, and typically they rely on using friction to degrade the vibrational energy into heat.

Real structures have multiple resonant modes. Usually these modes are coupled, making the form of the resonant vibration quite complex, particularly in the case of nonsymmetrical structures. In addition, each vibrational mode often produces a whole series of harmonic vibrations. Nonetheless, the theory described in the preceding notes is still valid. To summarize:

- At low frequencies, below the first resonant frequency, the compliance of a real structure is determined by the stiffness. Stiffer structures are less compliant (i.e., they vibrate less for a given forcing vibration).
- At frequencies above the region of the first resonance, the compliance of a rigid structure is given roughly by $1/mf^2$ although resonant effects are superimposed upon this.
- The compliance at the various resonant peaks is dependent mainly on the amount of damping at these resonant frequencies and by the effective mass associated with the vibrational mode. However, the relative height of these peaks is still approximately proportional to $1/f^2$.

At higher frequencies, the compliance is totally dominated by the mass (inertia) of the structure. From equation 9.2, the compliance at high frequencies becomes

$$\text{compliance} = \frac{1}{mf^2} \ . \qquad (9.6)$$

Although in certain very simple vibrational systems (e.g., a single mass suspended from a spring) it may be fairly easy to evaluate the mass involved in the vibration, in most real-world cases, determining the mass can be very complex and is best determined by computer programs. For example, an optical table has several types of bending and flexural resonant modes of vibration, and different points on the table have different vibrational amplitudes. Furthermore, at nodal points, there is no vibrational amplitude at all.

ANTIRESONANCES

In systems, such as an optical table, with several vibrational degrees of freedom, the compliance curve will often show negative peaks between the resonant peaks. These are usually overlooked in most discussions of optical tables, yet they give an insight into the concept of compliance phase.

The phase of the vibrational response of the table changes from 0 to 180 degrees as the compliance curve passes through a resonance. The *stiffness* part of the curve is in phase with the forcing function and the *mass* part of the curve is exactly out of phase with the forcing function. Now consider the case of a system with two resonances (of similar compliance) at different frequencies as shown in figure 9.11. The mass part of one resonance occurs at the same frequency as the stiffness part of the other resonance; therefore, it is possible for the net compliance at this point to be very small. The negative peak, termed an *antiresonance*, and may almost reach zero if the two terms are very similar in magnitude.

COMPLIANCE OF A REAL TABLE

The concept of an ideal rigid body is useful when considering optical table performance. This theoretical structure does not resonate and therefore has no compliance peaks. When plotted on a log:log scale, an ideal rigid body has a compliance proportional to $1/f^2$ and is represented by a straight line with a slope of -2. It represents the ultimate design goal when manufacturing optical tables—the nearer the actual curve fits the straight line the better the dynamic stiffness.

A compliance curve of a real table is shown in figure 9.12. The measurements were taken at the corner of the tabletop. Conventionally, this type of data is presented on a double logarithmic scale—vertically because of the large range of compliance values and horizontally in order to display a wide range of frequencies while highlighting the important region around the low-frequency resonances.

Several aspects of this curve merit special comment. The initial portion of the plot, before the first table resonance, is determined primarily by the table supports, not by the table itself. Notice how the peaks at compliance resonances decrease in size toward higher frequency. As the frequency increases, the denominator in the compliance expression (see equation 9.2) becomes large and therefore the compliance is reduced. This means that, as the frequency increases, a given excitation force produces a smaller amplitude excitation in the table. The higher-frequency vibrations of a table are the various combinations and overtones of the fundamental vibrations of the table, discussed below.

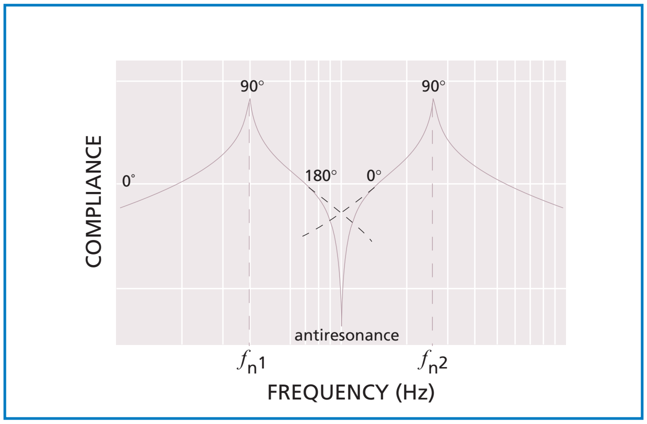


Figure 9.11 **An antiresonance, caused by two vibrational components of equal magnitude that are 180 degrees out of phase**

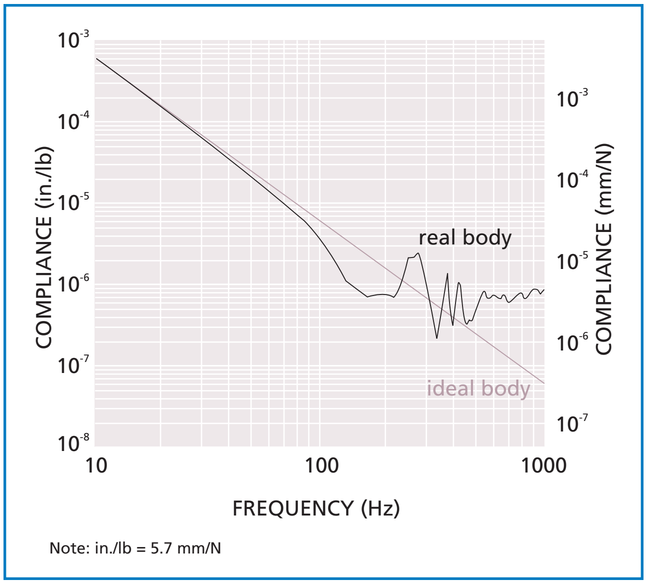


Figure 9.12 **A typical compliance curve of a high-performance CVI Melles Griot optical table**

The low-frequency peaks are the most important because they are the largest peaks, corresponding to the weakest points in the compliance spectrum; and typical vibrations from laboratory equipment are usually below 150 Hz. These peaks should be at the highest frequency possible in order to keep the compliance in the 0 to 150 Hz region as low as possible.

In this particular curve, notice the inverted peak at around 120 Hz. This is a typical antiresonance caused by the phase effects described earlier.

It should be noted also that a single compliance curve does not describe the performance of an optical table. A few modes may have very low, or zero, amplitudes at the corner of measurement. In general, however, a compliance curve obtained at the corner of an optical table is a reasonable measure of the worst-case performance of any part of the table surface.

It is always difficult to specify a piece of equipment by a convoluted curve or spectrum as opposed to a number or series of numbers. Consequently, two values are used to specify the performance of an optical table: the frequency of the first resonance, which should be as high a value as possible, and the peak compliance, the height of the highest (usually the first) peak.

TABLE STIFFNESS

As previously discussed, the measured low-frequency compliance of an optical table is entirely a function of the table support structure used during the test. The low-frequency behavior of an optical table is usually expressed as the stiffness or deflection under a measured static load. This can be measured as either a displacement of the center of the table or as an angle of deformation of the tabletop. The actual values depend on the relative position of the supports.

TABLE THICKNESS

Compliance also varies with table thickness. The graph in figure 9.13 shows that, for a 1.5 × 3-m table, if the thickness is increased from 210 to 420 mm, the compliance improves from 6×10^{-5} mm/N to 1.4×10^{-5} mm/N.

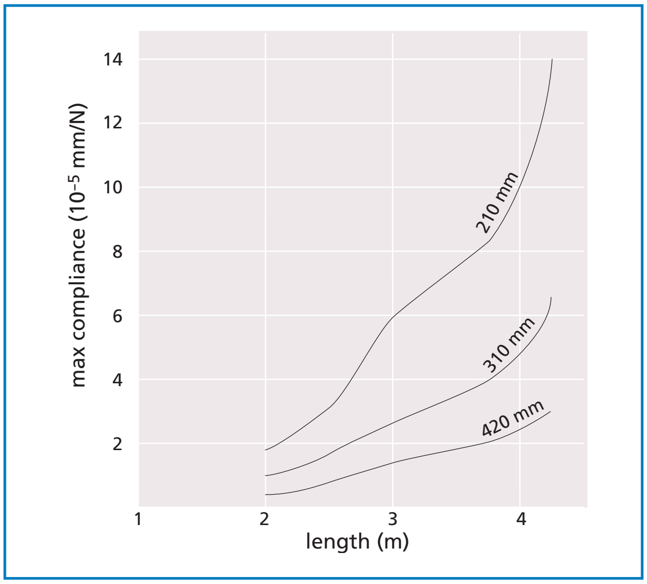


Figure 9.13 **Compliance vs length for different thickness tabletops**

Table compliance improves with thickness

Many applications require a high degree of stability

Tabletop Design

There are two basic goals in designing an optical tabletop. The natural resonances of the table should be as high as possible, since the excited vibration amplitude decreases sharply with increased frequency ($1/mf^2$), and the table should be well damped to further decrease the amplitude of the low-frequency resonances. Both of these goals must be accomplished without making the table overly massive and heavy. Increasing the resonant frequency is achieved by maximizing the stiffness-to-mass ratio; decreasing the resonant amplitude is achieved by enhancing the natural damping of the tabletop.

TABLETOP VIBRATIONS

A typical rectangular tabletop is capable of several types (modes) of low-frequency vibration, as shown in figure 9.14: bending along the long axis of the tabletop, bending along the short axis of the tabletop, and torsional bending.

Each of these independent modes has a characteristic resonant frequency, which gives rise to a corresponding peak in the compliance curve. To avoid any resonant effects from induced floor or tabletop vibrations, the first bending and twisting peaks (shown in figure 9.15) should occur at as high a frequency as possible. The peak associated with the short bending mode usually occurs at a much higher frequency and therefore can be ignored.

Each mode has a series of higher-frequency harmonics (overtones) associated with it. Each successive harmonic has more nodes (null amplitude points) and antinodes (points of maximum amplitude) and a higher resonant frequency than the previous. Because different vibrational modes have their nodes and antinodes at different places on the table, and because it is possible to have several modes oscillating simultaneously, the resultant compliance is a complex function which varies greatly over a table's surface. Furthermore, deflections under time-varying loads (dynamic operation) can be much higher than when under static loading. The nodes and antinodes for the first bending and torsional modes on a typical optical table are shown in figure 9.16.

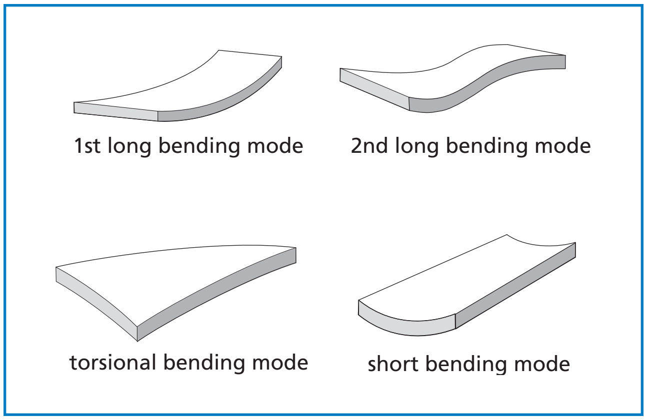


Figure 9.14 **Vibrational modes of an optical tabletop**

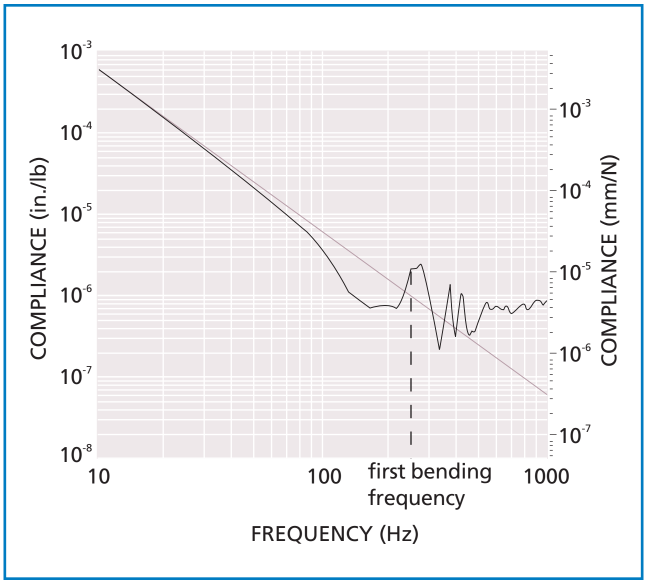


Figure 9.15 **First bending-mode peak**

TABLETOP MATERIALS

There are three basic requirements for an optical tabletop:

- It must have good static and dynamic stiffness (see figure 9.17).
- Its first resonance must be higher than 100 Hz.
- It must be well damped.

For many years, scientists performed delicate optical experiments on home-built tables. Tabletops have been constructed from granite, concrete, wood, and steel. Also, many elaborate composite structures have been used in attempts to improve performance while keeping weight at a realistic level. Each technique has disadvantages. Granite and concrete slabs tend to absorb water vapor and deform accordingly. In fact, due to residual chemical changes, concrete changes shape slightly for many years. Steel has a high density and tends to ring at several vibrational frequencies, with very little natural damping. In many ways, wood is a surprisingly good choice for an optical table; however, it has a tendency to warp with time, particularly in the presence of moisture.

Clearly, the best material for optical table construction should be a composite, a matrix of materials that combine the stiffness of pure metal, the damping of rubber, and the low density of a material like wood. It is now accepted generally, that the best combination of materials for optical table construction is a metal honeycomb sandwich. The top and bottom layers are steel or aluminum, and the large core is a low-density metal honeycomb, usually steel. The plates provide the stiffness or rigidity, and the honeycomb is a material which combines excellent dynamic stiffness with very low density. The honeycomb also exhibits fairly good natural damping; however, in cases where ultimate performance is required, additional damping is needed.

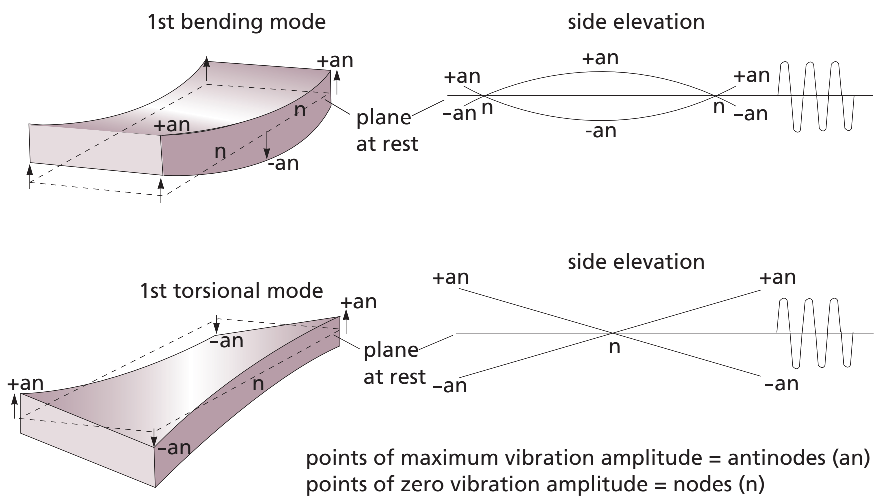


Figure 9.16 **Vibrational nodes**

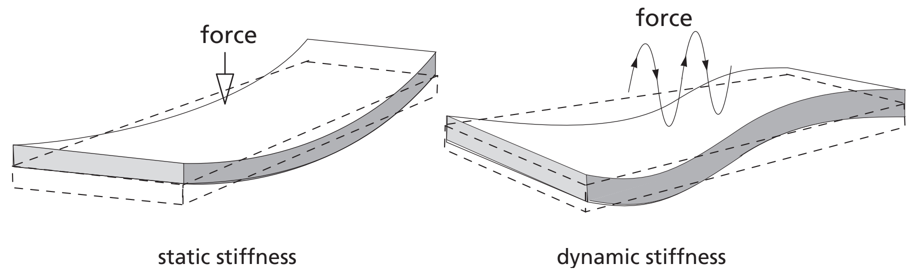


Figure 9.17 **Static and dynamic stiffness**

Stiffness of Hollow versus Solid Structures

Consider a solid bar of a homogeneous material such as steel (figure 9.18). If this bar is bent, then one edge is stretched and the other is compressed. The core of the bar is affected much less by this deformation. Consequently, the major contributions to the restoring force arise from the edges of the bar, and a hollow bar would be almost as resistant to this bending deformation as a solid bar. Therefore, the strength of a structure is dependent not only on the material of the structural elements, but also on their shape and position. This is an important concept in optical table design.

Now consider the case of a simple I beam. Under load, the top flange is under tension and the bottom flange is under compression, but at the "neutral axis" the stress is zero. The flanges carry the bending stresses and the web resists the shear forces. The farther apart the flanges are, the lower the deflection for a given load, i.e. the higher the stiffness.

STEEL AND HONEYCOMB SANDWICH

An optical table requires the rigidity of steel, yet a solid steel table would be ridiculously massive and would ring badly. As previously discussed, the important vibrations of an optical table are those occurring at lower frequencies (bending and torsional vibrations). These all involve vertical displacements of various portions of the table, which may be in phase and out of phase.

A hollow steel table would be very rigid for its mass because the sides and the top and bottom skins would act as an extended I beam. However,

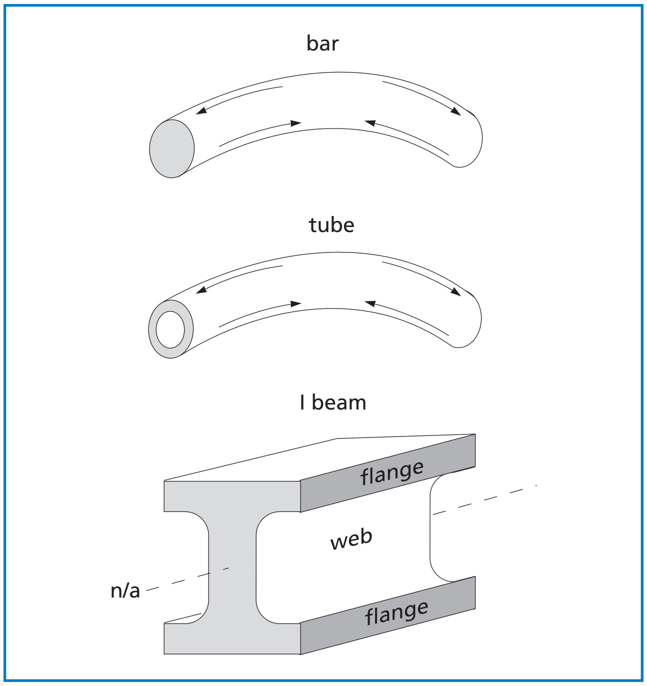


Figure 9.18 **Tubes and I-beams have a large stiffness-to-mass ratio**

it would tend to ring owing to the absence of damping and would also sag in the middle under heavy load conditions.

Consequently, optical tables should not be hollow, but should be filled with a material that combines rigidity and damping with reasonable stiffness in the vertical direction. The most successful materials for this purpose are honeycomb materials made from steel, aluminum, phenolic, graphite, Kevlar, or even wood pulp.

Honeycomb Theory

Consider bending a thin sheet of material such as paper. The sheet is very resistant to deformation within the plane of the sheet, as shown in figure 9.19. Now suppose the tabletop is filled with many linked sheets all lying vertically. If the sheets are all parallel, the table is much more resistant to deformation in one bending direction than the other. In one case the initial motion is parallel to the sheets, and in the other it is perpendicular to them.

Suppose now that the sheets are linked and arranged at 90 degrees to one another. In the case of a rectangular top, the sheets could be arranged so that the core is very resistant to the two bending (strong) vibrations. However, there is a plane at 45 degrees to both the strong planes. The effectiveness of the core in resisting a vibration in this plane will be reduced by cosθ where θ equals 45 degrees. Any vibration, such as torsion, which involves vibration in or close to this plane will not be resisted.

Clearly, it is advantageous to have the sheets in the structure at intermediate angles. For both theoretical and practical fabrication reasons, the best honeycomb is a hexagonal or sinusoidal cellular structure. When this is incorporated into a tabletop, the worst case would be a vibration at 30 degrees to the two strong planes.

All CVI Melles Griot standard tables consist of a sandwich structure: thick steel top and bottom plates with a metal honeycomb core. CVI Melles Griot honeycomb is fabricated from strips of precision crimped steel which are then bonded together with a high tensile strength epoxy adhesive. The structure is shown in figure 9.20.

Bonding Materials

In any sandwich structure, the ultimate performance depends on more than the quality of the materials used in the sandwich components, in this case thick steel outer plates and the metal honeycomb core. It also depends on the bonding materials used to bind the elements together to produce an integral composite structure. CVI Melles Griot uses a hot pressing method and a modified structural epoxy adhesive to bond the honeycomb to the steel plates. This aerospace-grade adhesive possesses high tensile strength as well as extremely high shear and peel strengths. The bonds are cured for 18 hours under vacuum to ensure that the adhesive reaches its ultimate properties.

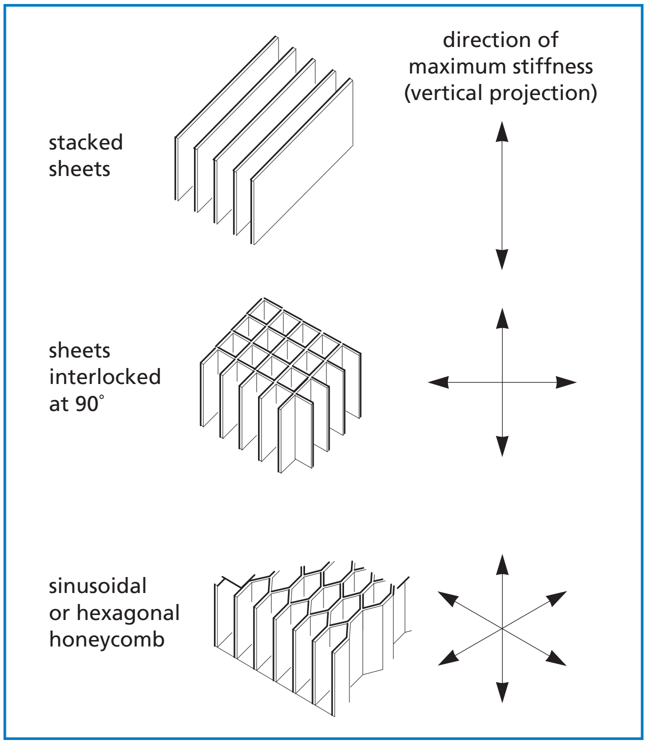


Figure 9.19 **Planes of stiffness**

Fundamentals of Vibration Isolation

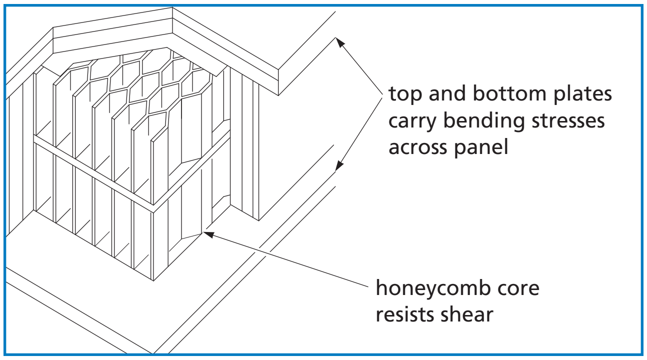


Figure 9.20 **Construction of a honeycomb panel**

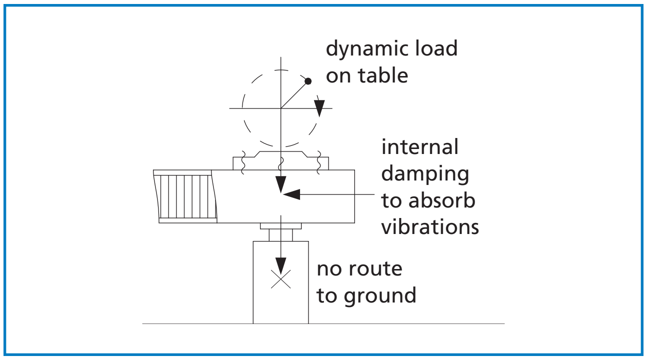


Figure 9.21 **Damping and vibration isolation**

DAMPING TECHNIQUES

Damping refers to any process that causes an oscillation in a solid body to decay to zero. Damping is a resonant effect, inasmuch as it significantly affects, the compliance function at or near resonance.

In theory, a resonating structure without damping has an infinitely high compliance peak at its natural frequency. Damping systems suppress these peaks by converting the energy of vibration into heat, which causes the amplitude of the disturbance to decay rapidly to zero.

As previously stated, the equipment mounted on the tabletop can be a major source of vibration (e.g., an out-of-balance laser-beam chopper blade). Once a table is mounted on its pneumatic isolators it is floating in a free condition, and any vibrations induced on the top cannot find a route to the ground. Therefore, these vibrations must be damped by the table's own internal damping (see figure 9.21).

The damping of an optical table arises from several sources. The materials themselves have some natural damping, particularly in the case of the metal honeycomb. Producing a table with state-of-the-art performance, suitable for the most demanding applications, requires additional damping. Some manufacturers use dashpot dampers for this purpose. These oil-filled dashpots are placed at positions in the table where the amplitude of the excited vibration is the greatest, usually at the corners. The dashpot, used in this way, acts as a tuned damper. The natural frequency of the paddle moving within the damper is chosen to match the largest compliance peak for the particular table. Other resonance peaks are not greatly affected. Oil-filled dashpots, however, can leak if the tabletop is tilted excessively or stored on its side.

In CVI Melles Griot tables, the damping method involves positioning dry dampers at strategic points in the core of the table. These dampers are specially shaped pieces of inhomogeneous material which act as though they contain a spectrum of masses, separated by a continuous spectrum of distances in an elastomeric polymer. The effect is dramatic, greatly reducing the height of the low-frequency resonance compliance peaks, sometimes by more than an order of magnitude.

CVI MELLES GRIOT OPTICAL TABLETOP CONSTRUCTION

CVI Melles Griot optical tabletops provide an ultrastable environment for optical research and product development.

Minimum Relative Tabletop Motion

CVI Melles Griot StableTop™ optical tables, shown in figure 9.22, incorporate a triple-plate double-honeycomb design, which provides unsurpassed stiffness and excellent dynamic rigidity. The top layer consists of a stainless steel top plate and a secondary steel plate, separated by a unique plated-steel honeycomb-like structure. The second layer includes the main steel honeycomb structure and the stainless steel bottom plate. The main honeycomb core is fabricated from strips of precision-formed plated steel, which is bonded together with a high tensile strength epoxy adhesive. The thinner, honeycomb-like top structure is made from cylindrical-shaped steel cups, which enhance the overall stiffness of the tabletop and provide improved support of the top plate.

Low Dynamic Deflection

A tabletop with state-of-the-art performance suitable for the most demanding applications requires enhanced damping. CVI Melles Griot tables use a unique damping method to reduce the amplitude of tabletop resonances. Strategic positioning of tuned and broadband dampers ensures superior performance by reducing the compliance curve resonance peak, which further reduces relative tabletop motion.

Excellent Surface Flatness

Tabletop flatness is critically important during many experimental setups. Lack of local flatness requires readjustment of components for height variations across the tabletop and can cause component wobble. CVI Melles Griot tabletops provide unsurpassed flatness due to high-precision magnetic stainless-steel plates which are specially handled to maintain superior flatness throughout the manufacturing process. A

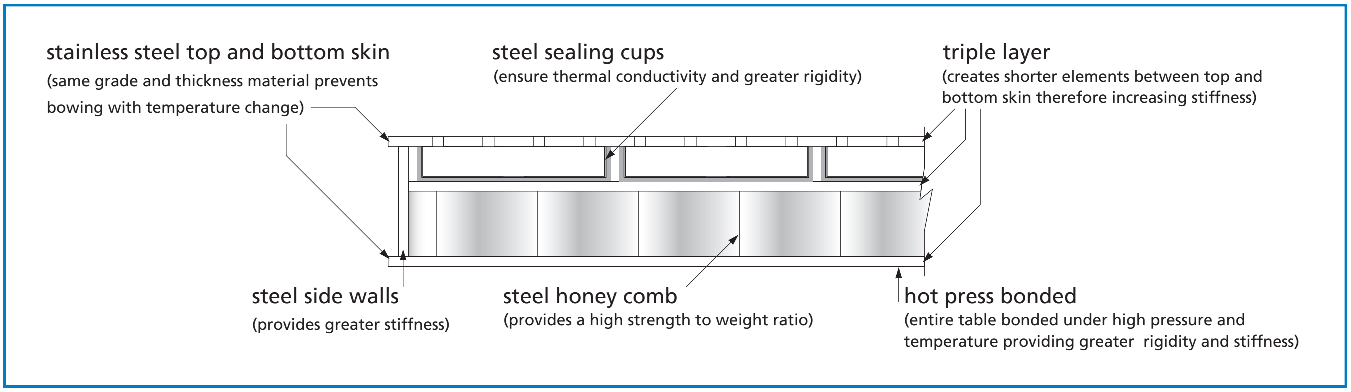


Figure 9.22 **CVI Melles Griot tabletop with triple-plate, double honeycomb core construction**

unique thermal bonding process ensures that stress is not induced during manufacture, thereby retaining the flatness of the top plate.

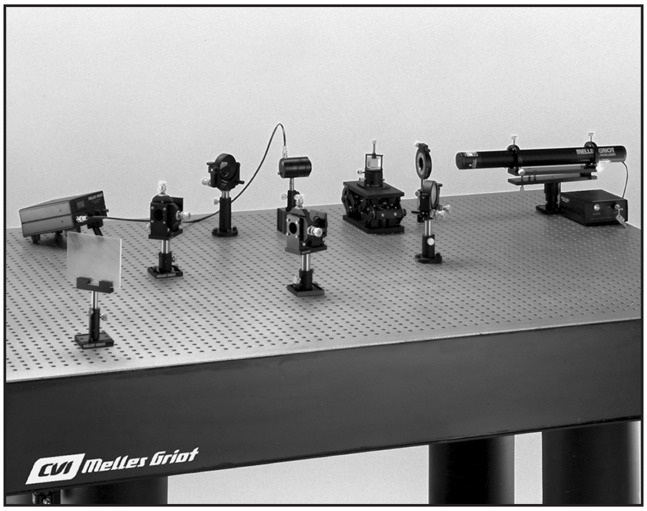
Athermalized Design

The top and bottom plates of CVI Melles Griot tabletops, made of magnetic stainless steel, have identical thermal and mechanical characteristic. This unique athermalized design eliminates any thermal bowing effects caused by temperature variations.

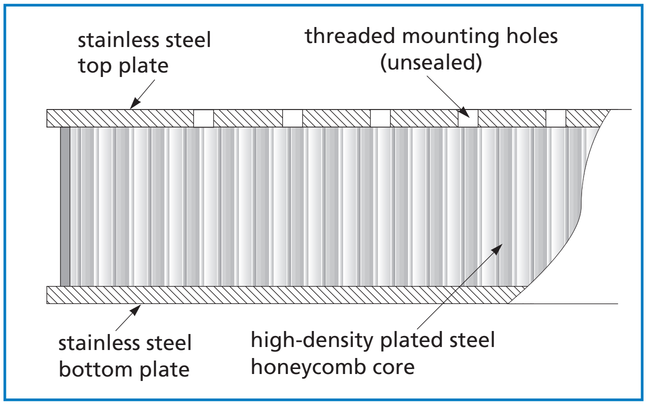
EVALUATING TABLETOP PERFORMANCE

The performance of an optical table is quantified by its compliance. At CVI Melles Griot, the compliance of our tables is measured with a dynamic signal analyzer. The test setup is shown in figure 9.23.

An impulse hammer is used to apply a measured force to the top surface of the table; transducers attached to the surface sense the resultant vibrations. The signals from the transducers are interpreted by the analyzer and used to produce a frequency response spectrum—a compliance curve. During the development phase of an optical table, compliance curves are recorded at many points on the tabletop; however, compliance is always greatest at the corners. The compliance curves and data published by CVI Melles Griot are recorded at the corner of the table and, therefore, represent the worst-case data.



A typical setup on an optical table



Schematic of a two layer breadboard

Fundamentals of Vibration Isolation

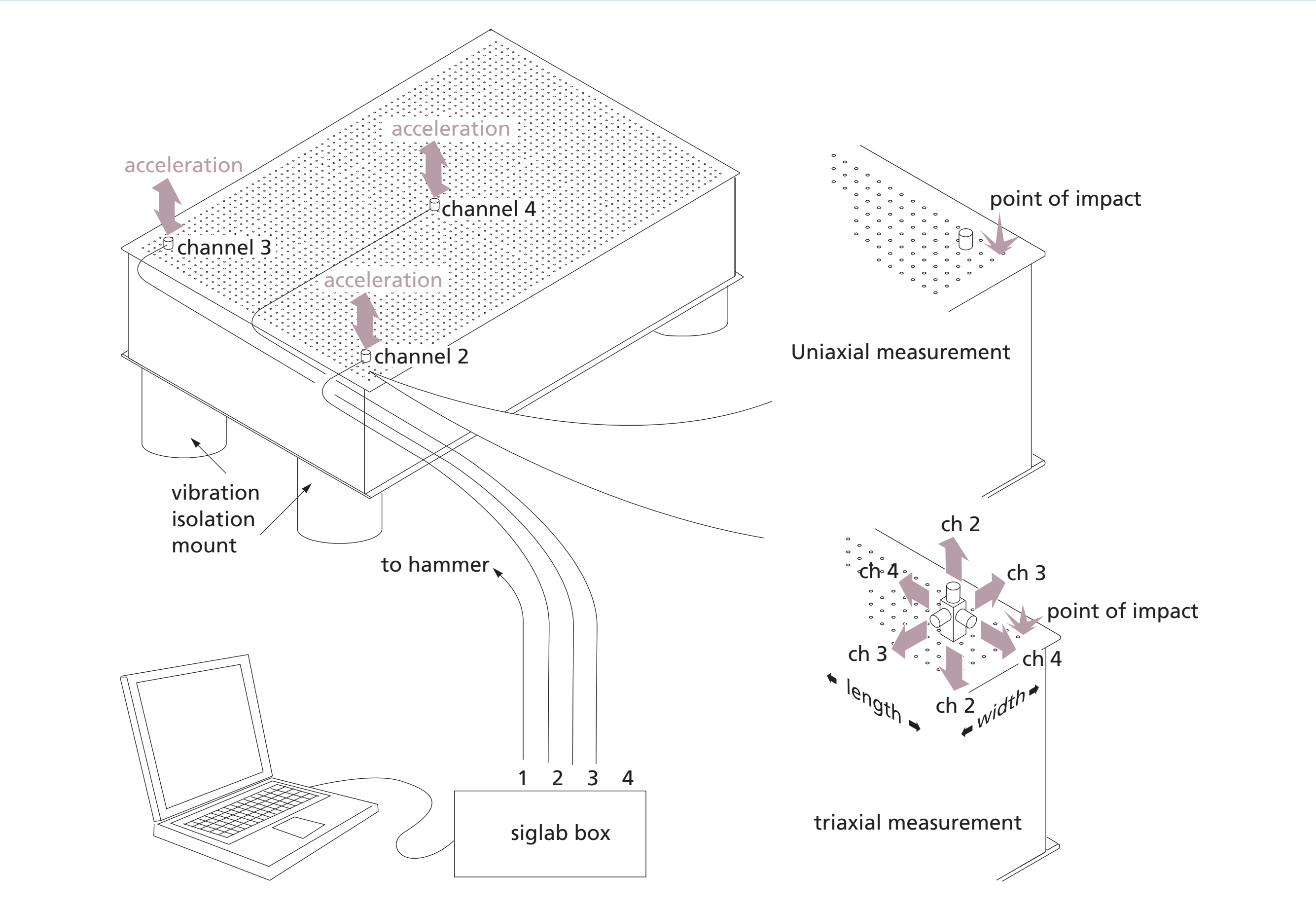


Figure 9.23 **Evaluating the performance of an optical tabletop**

Theory of Vibration Isolation

Vibration isolation or elimination on an optical surface is a two-part problem. As discussed previously, an optical tabletop is designed to have zero or minimal response to a deflective force or vibration. This in itself is not sufficient to ensure a vibration-free working surface. The rigid tabletop may still vibrate without deforming (i.e., vibrations of the table on the mounting system). These vibrations are constrained translations and/or rotations of the optical table.

Typically, the entire table system is subjected continually to vibrational impulses from the laboratory floor. These vibrations may be caused by large machinery within the building or even by wind or traffic-excited building resonances (swaying).

SEISMIC MOUNTING

Vibrations of a floor in a building can be divided into two basic components: vertical and horizontal. Typically, vertical components range from 10 to 50 Hz, and horizontal components range from 1 to 20 Hz. To prevent such vibrations from disturbing an experiment, it is important to mount the table in such a way as to isolate it vibrationally from the laboratory floor (i.e., mount the table in such a way that its instantaneous position is independent of the periodic motions of the laboratory floor). This type of isolation is termed "seismic" mounting. When an object is truly seismically mounted with respect to the floor, the motions of the object and the floor are completely uncoupled and separate. The term seismic is linked with the study of earthquakes, in which the magnitude of an earthquake is estimated from the motion of the ground with respect to a seismically mounted indicator.

TRANSMISSIBILITY

The simplest model used in a theoretical treatment of mechanical vibration is the ball and spring as shown in figure 9.24. Consider a ball suspended from an enormous mass by a spring. For now, we shall ignore pendulum motions and consider a system having only one degree of freedom—capable only of vertical extension.

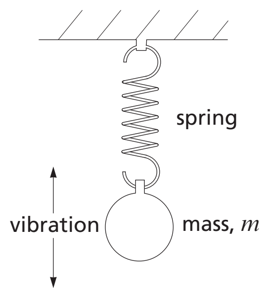


Figure 9.24 **A ball and spring example of a vibration isolation/transmission system with one degree of freedom**

In the absence of vibrational impulses, the ball is stationary at its rest position. Suppose the object from which the ball is suspended is not infinitely large or not infinitely stiff so that the point at which the spring is anchored starts to vibrate. Some of the vibration energy may be felt at the ball, causing it to vibrate at the same frequency. The frequency of this motion is given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (9.7)$$

where f_n is the resonant frequency of the oscillation, m is the mass moving during the oscillation, and k is the spring constant (related to the stiffness of the spring).

The flow of vibrational energy is expressed in terms of a transfer function. A transfer function is a method of quantifying how efficiently a forcing vibration can produce an excited vibration. The transfer function most applicable here, transmissibility, is defined as the ratio of the dynamic output to the dynamic input, i.e., the ratio of the amplitude of the transmitted vibration to that of the forcing vibration.

RESONANCE

In the example just outlined, the transmissibility of the spring is very dependent on the frequency of the forcing vibration. The idealized transmissiblity of this system is given by

$$T = \sqrt{\frac{1+\left(2\zeta f/f_n\right)^2}{\left(1-f^2/f_n^2\right)^2+\left(2\zeta f/f_n\right)^2}} \qquad (9.8)$$

where: f is the frequency of the forcing function, f_n is the natural resonant frequency of the system, and ζ is the damping ratio.

The graph shown in figure 9.25 is an idealized plot of transmissibility as a function of frequency. Note the similarity to the compliance transfer function previously discussed in the table vibration text. Again there are three distinct regions on the curve.

At low or zero frequencies, the ball and spring in the previous example move synchronously with the table and with the same amplitude; i.e., the transmissibility is unity as defined by the preceding equation. The system behaves as though the spring were rigid and there were no vibrational isolation. As the frequency increases a resonant condition is approached. The ball has mass and momentum when moving and cannot change direction instantaneously in response to a force whose direction is rapidly changing, i.e., the forcing vibration. The vibration of the ball starts to lag behind the forcing vibration, i.e., they are no longer in phase. Eventually, this phase lag becomes exactly 90 degrees and at this point, the system is vibrating at its natural or resonant frequency, as shown in figure 9.26.

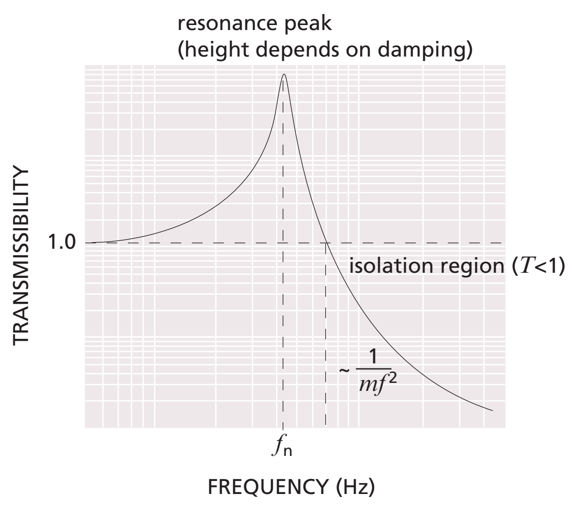


Figure 9.25 **A typical transmissibility vs frequency curve for a system with one degree of freedom**

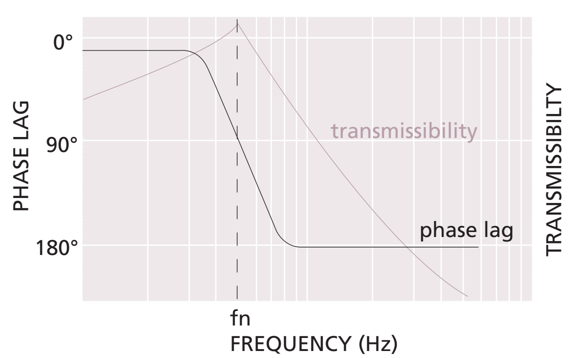


Figure 9.26 **The phase relationship between excited and forcing vibrations changes rapidly near resonance.**

An example of a structure at natural or resonant frequency is a simple tuning fork. It is the frequency at which the excited momentum and kinetic energy of the motion are in perfect dynamic equilibrium with the restoring force (potential energy). The system oscillates between high kinetic energy-zero potential energy and high potential energy-zero kinetic energy. At its resonant frequency, the system accumulates vibrational energy and increases in amplitude during the time the forcing vibration is applied. At resonance, the system acts as a vibrational amplifier because of the 90-degree phase lag: the maximum velocity of the forcing vibration coincides with maximum acceleration of the excited vibration. As defined by the equation for transmissibility, the amplitude of this resonant motion does not grow to infinity because of damping.

At frequencies much higher than the resonant frequency, the response of the ball is determined solely by the mass term, which is much larger than the stiffness term. In other words, the spring is relatively soft and the vibrational force travels slowly along it in the form of compression/extension waves. This slow transmission effectively spreads out the oscillatory nature of the forcing vibration. Essentially, the ball experiences a time-averaged force owing to the fast moving vibration and, unless the vibration involves a net displacement, this time-averaged force tends to zero with increasing vibrational frequency. As the transmissibility approaches zero, the position of the ball is not affected by the vibration in the large mass. This is a classic example of seismic mounting.

DAMPING

Damping refers to any process that causes an oscillation in a system to decay rapidly to zero amplitude. It is a very important phenomenon in vibration suppression or isolation. Damping causes the energy to be diverted from vibration to other energy sinks. The damping ratio of a system (ζ) is defined as the ratio of actual damping (C) to critical damping (C_c). Critical damping is the minimum amount of damping in a system necessary to prevent resonant oscillation following application of an impulsive force.

Damping is a resonant effect in that, primarily, it affects the transmissibility function at or near resonance.

At f/f_n=1, resonance, the transmissibility becomes

$$T = \sqrt{\frac{1}{(2\zeta)^2}+1} \qquad (9.9)$$

where ζ is the damping ratio, (C/C_c).

For a lightly damped system, i.e., $\zeta < 0.1$, the maximum transmissibility is

$$T = \frac{1}{2\zeta}\ . \qquad (9.10)$$

The height of the transmissibility peak at resonance is determined primarily by the amount of damping. In the absence of damping, the peak would be infinitely high. Also, a system with no damping would vibrate resonantly ad infinitum, even when the vibrational source (forcing function) is removed. Clearly, all real systems contain damping to some degree. Using the suspended ball as an example, the simplest form of damping would be to immerse the ball in a viscous oil. Viscous drag would convert the vibrational energy to heat by friction.

SEISMIC MOUNTING OF OPTICAL TABLES

The preceding discussion suggests in principle, a way in which an optical table could be vibrationally isolated (seismically mounted). If an optical tabletop could be suspended from the ceiling by springs, then at frequencies much higher than the natural resonance of the spring isolation system, ambient building vibrations would not be transmitted to the tabletop. Obviously, this solution has no practical value, but it clarifies the general principles involved.

Typically, ambient vibrations in a building tend to be in the range from 10 to 50 Hz for vertical components and from 1 to 20 Hz for horizontal components. To mount an optical table seismically under these conditions requires a mounting system with a very low resonant frequency (i.e., a spring with a small spring constant). Unfortunately, even with the use of composites, optical tables are relatively massive—typically 500 kg for a 1.2 × 2.4-m table. Such a system would involve highly extended springs and a very large travel range. However, it is interesting to note that if stiff springs were used in such a system, then the resonant frequency for pendulum motion could easily be less than 1 Hz; i.e., the tabletop would be well isolated from horizontal vibrations of the building.

The ideal mounting system, shown in figure 9.27, has a rigid tabletop mounted to the floor in such a way that it can vibrate in vertical and horizontal directions, both with low resonant frequencies. Even though these resonant frequencies can be made lower than most building vibrations, it is important that the height, and hence the width, of the resonant peak in the transmissibility curve be reduced. This is achieved by adding a damping mechanism which ensures minimum oscillation at resonance and short settling times.

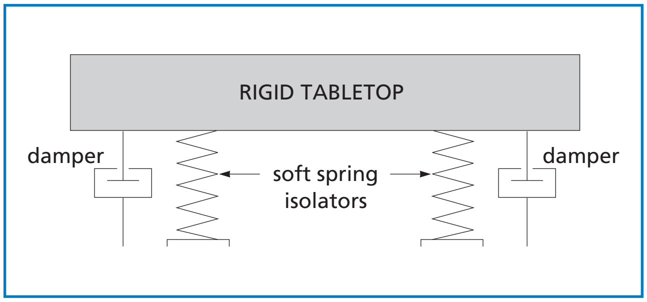


Figure 9.27 **Ideal seismic mounting of an optical table, consisting of weak spring supports with added damping**

Rigid, passive and self leveling table supports

Low noise, low vibration compressor can be used when compressed air is not available

Fundamentals of Vibration Isolation

Tabletop Isolator Design

True seismic mounting of an optical tabletop requires a mounting system which addresses two issues.

- Ambient building vibrations occur at low frequencies; therefore the mounting system must have very low vertical and horizontal resonant frequencies in order to function as a seismic isolator.
- These motions must be damped in order to suppress the resonant peaks.

The aim of vibration-isolation supports is to reduce motion and vibration from the building and surrounding environment, and to isolate these vibrations from the tabletop surface. Such vibrations may result from building seismic activity, nearby machinery, or pedestrian traffic. Isolation from these vibrations is achieved by using a mounting system with a very low resonant frequency, for example, soft-spring pneumatic systems.

AIR-SPRING VERTICAL ISOLATION

The simplest, most successful optical table supports are legs containing air springs, which have been successfully employed in solving vibration-isolation problems for over 50 years. Typically, an air spring is a flexible diaphragm on top of a rigid cylinder of compressed air as shown in figure 9.28. A heavy object can be placed on the air spring, and, as the object vibrates or as the floor vibrates with respect to the stationary object, the air is alternately compressed and allowed to expand.

Air springs differ from their conventional mechanical counterparts because they have a very low spring constant compared to their small range of travel. A mechanical spring with a low spring constant usually exhibits a large-amplitude vibration because of the small restoring force. In an air spring, the relationship between restoring force and deflection varies with the shape and material of the air spring envelope. The resonant frequency of an air spring isolator is given by

$$f_{\mathrm{n}} = \sqrt{\frac{rAg}{V}} \qquad (9.11)$$

where r is the specific heat ratio for the gas (air) in the spring, A is the piston area, g is the acceleration caused by gravity, and V is the volume of the air bag (or cylinder).

From the above equation, we can conclude that the stiffness of the spring (and hence the natural frequency of a mass supported on the spring) is dependent upon the height of the spring (volume of air), but it is independent of the load. Consequently, if the load is changed but the pressure is adjusted to maintain a constant height, then the resonant frequency remains constant. This is highly desirable for optical tables.

The simplest table support system has several legs (usually four, depending on table size). Each leg consists of a stiff rubber bag of compressed air, and the weight of the table rests on a metal disc on top of the bag. When the isolators are unpressurized, the table rests on the top of the outer casing of the leg. Pressurizing the air spring floats the table, thus providing seismic mounting. With four air springs involved, there are several vertical vibrational modes (i.e. resonant vibrations) of the table on the support system. These are the various in- and out-of-phase combinations of vibrations of the individual isolator legs. These motions include vertical motion, pitching and rolling. Their resonant frequencies are shifted only slightly relative to that of an individual isolator with a quarter of the table weight on it.

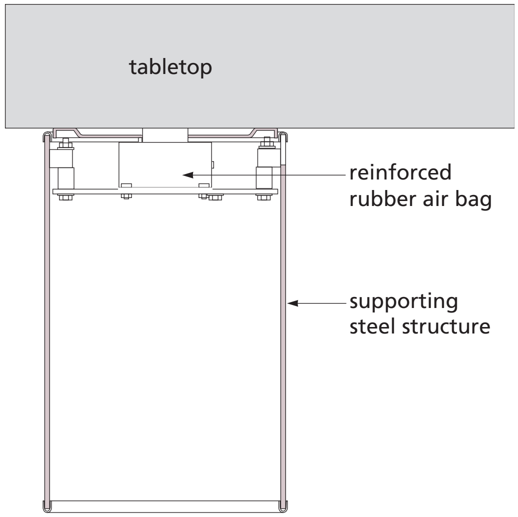


Figure 9.28 **A simple air spring**

CHOOSING AN ISOLATION SYSTEM

The simple air spring design described previously is more than adequate for use in lower vibration environments, or for less sensitive experiments, and it is a vast improvement on rigid support systems such as steel legs. However, three issues are not directly addressed: enhanced damping, horizontal vibrations, and load leveling.

Enhanced Damping: With the simple air spring, damping is limited to that provided by viscoelastic deformation of the walls of the air bag. As a result, the vibration peak at the resonant frequency can be quite large.

Horizontal Isolation: Horizontal vibration may make the table rock on its legs. The horizontal resonant frequency depends upon the nature and shape of the linkage between the table and the air bags, but is typically in the 3- to 5-Hz range. Because ambient horizontal vibrations tend to occur at lower frequencies than vertical floor vibrations (1 to 20 Hz versus 10 to 50 Hz), they are more difficult to isolate. However, with the exception of upper-floor locations in tall buildings, horizontal floor motions are generally much smaller then vertical vibrations.

Load Leveling: With the simple air spring, there is no provision for self-leveling and automatic height adjustment. When large loads are placed on the tabletop it will be lowered slightly, and if the load is not placed centrally, the tabletop will no longer be level.

Ground Floor Locations

In a ground floor situation, predominately vertical floor borne vibrations in the 4 to 50 Hz range are likely to be encountered. These are readily attenuated using a passive isolation system.

CVI Melles Griot Pump & Go™ passive isolators, shown in figure 9.29, are ideal for removing floor vibrations in the critical 10- to 50-Hz f.requency range. They provide simple, effective vibration isolation with excellent horizontal and vertical stability without any air consumption. Each isolator consists of a cylindrical reinforced rubber air mount in the top of a cylindrical steel leg. An integral part of the air mount is a steel plate on which the bottom of the optical table rests. The pressure of the isolator can be adjusted by admitting or releasing air through a standard Schraeder valve on the side of each isolator.

The passive air mount design is ideal for most general optical table uses, providing low-frequency isolation coupled with excellent stability in both vertical and horizontal directions. The thick wall construction ensures maximum safety and overload protection with an economical design. This passive air mount continues to support and isolate even with no air pressure. It provides low transmissibility at resonance, particularly when compared to conventional rubber or neoprene isolators. The transmissibility above 10 Hz is below 0.3 and results in low values of relative tabletop motion for the assembled system.

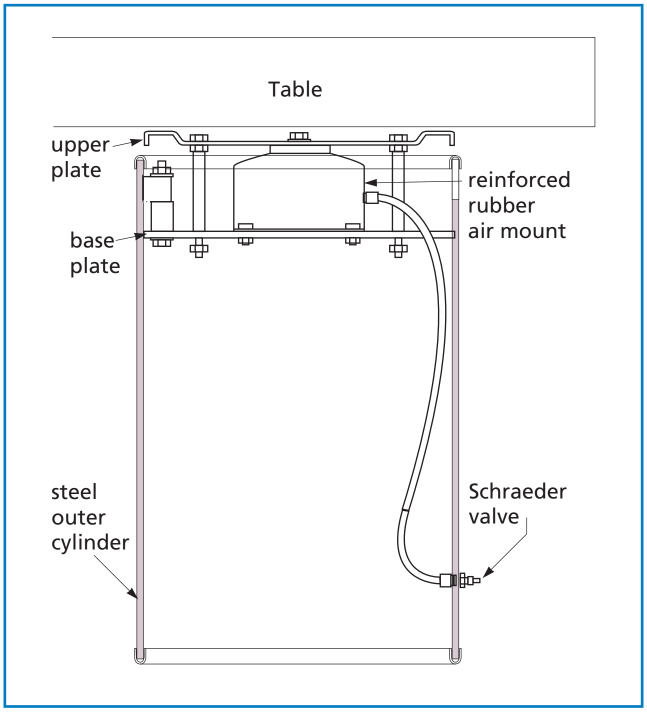


Figure 9.29 **Construction of Pump and Go™ isolator**

Upper-Floor Locations

If an experiment is conducted on an upper floor, the building may sway, causing both vertical and horizontal vibrations. CVI Melles Griot Super-Damp™ self-leveling isolators, shown in figure 9.30, are large-diameter free-standing systems that provide maximum stability and safety, without cumbersome tie bars. The low vertical and horizontal transmissibility of these isolators results in the least possible relative tabletop motion. The proprietary design uses no liquids that could leak or degrade over time.

Vertical damping is achieved by the use of a dual chamber, damped pneumatic spring. The table is supported by the air pressure in these chambers. A piston, clamped to the bottom of the table, is sealed to the upper chamber with a rolling rubber diaphragm, allowing virtually friction-free motion between piston and chamber. Floor or tabletop motion forces air to flow from one chamber to the other through a high-resistance damper.

This restriction of airflow damps oscillatory motion between the floor and table, dramatically reducing settling time. The volume ratio of the chambers has been optimized to maximize damping performance for our complete range of tabletops while preserving a low resonant frequency.

As stated previously in the discussion on seismic mounting in the vertical direction, isolation of an optical table from low-frequency ambient vibrations is best achieved by having a soft coupling between the floor and tabletop, so that the resonant frequency is lower than most of

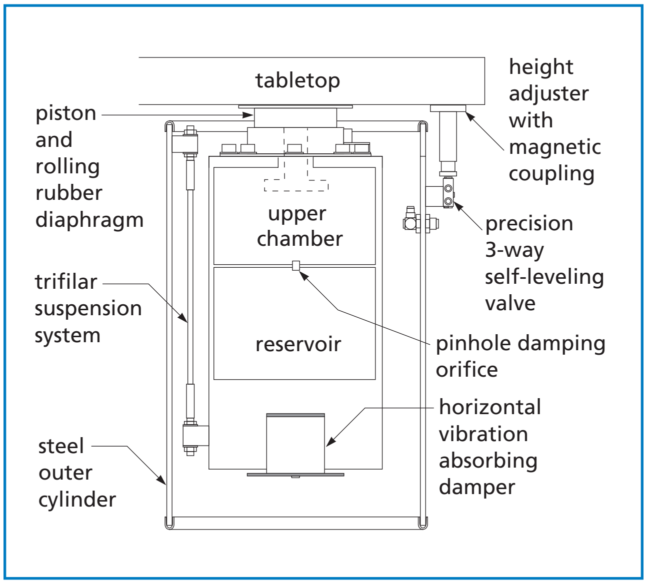


Figure 9.30 **Construction of SuperDamp™ isolator**

the ambient vibrations. It is also desirable for the transmissibility of the mounting system to be independent of loading.

A simple method of mounting a tabletop so that it can execute horizontal motion with a low resonant frequency is to suspend the table, rather like a pendulum. Furthermore, the resonant frequency of a pendulum is independent of the mass suspended from it and is given by

$$f_n = \sqrt{\frac{g}{l}} \tag{9.12}$$

where g is the acceleration caused by gravity and l is the length of the pendulum. CVI Melles Griot SuperDamp™ isolators damp horizontal vibrations by supporting the pneumatic vertical isolator on a trifilar suspension system. This innovative pendulum design uses gravity to provide the restoring force after horizontal disturbances. Horizontal oscillations at the system's resonant frequency are damped by linking the base of the vertical isolator to the outer cylinder with an oil-free vibration-absorbing damper.

To allow for changes in load distribution, SuperDamp isolators also feature a self-leveling system which incorporates precision threeway valves that do not compromise vertical isolation when the system is at rest. Because these valves are actuated by tabletop movement, the system returns to its original level position within ±0.25 mm (0.01 in.) after disturbances. The valves also compensate automatically for any changes in tabletop load distribution.

Additionally, this system allows the table height to be adjusted over a range of 26 mm (1.5 in.) and can be used to compensate for an uneven floor. These isolators require a constant supply of air. When the air supply is removed, the tabletop rests securely on top of the legs with the isolation system disabled. Three isolator heights are available to create a working table height of 910 mm (36 in.). Nonstandard heights are available upon request.

Less-Sensitive Applications

Many environments and applications do not warrant vibration isolation. In such cases, it is necessary only to have a nonisolating support structure with two basic features: an economical support structure strong enough to support an optical tabletop under heavy loading and a manual leveling adjustment on each support to compensate for uneven floors. CVI Melles Griot offers free-standing non-isolating supports each with three adjustment screws to level or adjust the table height.

INSTABILITY AND OSCILLATION

Systems with a high center of gravity (CG) can experience stability problems when supported on pneumatic isolators, particularly when the tabletop is narrow and/or thick. A high CG, combined with narrow isolator spacing, can cause static instability, and the table tends to rock slowly backwards and forwards about an axis midway between the two isolators. Furthermore, a dynamic instability can be experienced in the leveling system, causing the table to rock quickly from side to side.

Static Instability

Most table systems have a center of gravity which is above the height of the diaphragms, and narrow spaced isolators located at either end of the table. Any disturbance of the tabletop causes the CG to move away from the centerline between the isolators, and the weight of the table and equipment assists this sideways motion, causing the table to tilt further.

This tilting action is resisted by the stiffness of the isolators: stiffer isolators mean more stability. However, as we concluded previously, it is the softer isolators that are more effective in removing vibration. A trade-off must therefore be made between isolation and stability.

It is generally accepted that in order to avoid static instability and oscillation caused by excessive rocking, the CG, including that of the table, should be within the pyramid shown in figure 9.31.

The base of this pyramid is defined by connecting the center point of each isolator, and the height is equal to half the shortest distance between isolators. Static instability is not dependent on the air pressure in the system or (for active systems) the adjustment of the leveling valves. The only solution is either to lower the CG (moving equipment below the table surface by use of accessory shelves) or to increase the isolator spacing.

Dynamic Instability

In an active isolation system, table height is controlled by leveling valves, which let in or exhaust air as required. If the in-let or exhaust rate is too fast, then oscillation occurs. This oscillation can be removed by reducing the airflow to and from the isolators.

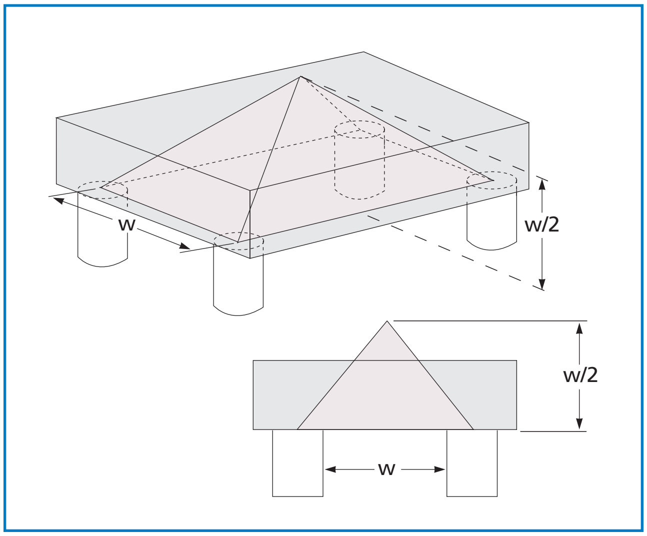


Figure 9.31 **Stability diagram for a tabletop**