Technical Background

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1.0 General Introduction

For over 30 years, TMC has specialized in providing precision working surfaces and vibration isolation systems for precision measurement laboratories and industry. To provide optimal performance, both precision "tops" and their supporting isolators must be designed to address the central issue: control of environmental noise.

1.1 Sources of Vibration

There are three primary sources of vibration (noise) which can disturb a payload: Ground vibration, acoustic noise, and "direct force" disturbances. Ground or seismic vibration exists in all environments throughout the world. This noise has various sources, from waves crashing on continental shorelines, the constant grind of tectonic plates, wind blowing trees and buildings, to manmade sources like machinery, HVAC systems, street traffic, and even people walking. TMC vibration isolation systems are designed to minimize the influence of these vibration sources.

Acoustic noise comes from many of the same sources, but is transmitted to the payload through air pressure waves. These generate forces directly on the payload. Even subsonic acoustic waves can disturb a payload by acting as a differential pressure on the diaphragms of pneumatic isolators. Air currents generated by nearby HVAC vents can also be a source of "acoustic" noise. TMC manufactures acoustic enclosures for OEM applications which protect payloads from this type of disturbance by providing a nearly airtight, heavy, energy-absorbing enclosure over the entire payload.

Acoustic noise can be measured, but its influence on a payload depends on many factors which are difficult to estimate (such as a payload's acoustic *cross-section*). The analysis of this type of noise source goes beyond the scope of this discussion.* In general, acoustic noise is the dominant noise source of vibration above 50Hz.

The third source of vibration are forces applied directly to the payload. These can be in the form of a direct mechanical coupling, such as vibration from a vacuum roughing pump being transmitted to the payload through a hose, or a laser water cooling line. They can also come from the payload itself. This is the case in semiconductor inspection equipment, where moving stages are used to position silicon wafers. The force used to accelerate the stage is also applied to the "static" portion of the payload in the form of a reaction force. Moving stages also shift the payload's overall center-of-mass (COM). Reducing these sources of vibration can be done passively, with TMC's MaxDamp line of isolators, or *actively* using feedback or feedforward techniques (active systems are discussed beginning on page 24). Payload-generated noise sources are usually of a well-known nature and do not require any measurements to characterize.

The influence of vibration transmitted to the payload can be minimized through good payload design. TMC offers a wide range of honeycomb optical tables, breadboards, and platform laminations. These are available in standard and custom shapes and sizes. All reduce the influence of environmental noise by having high resonant frequencies and exceptional damping characteristics (see Section 2).

1.2 Measuring Noise

Seismic (floor) noise is not usually known in advance, and must be measured. There are two types of seismic noise sources: periodic or coherent noise, and random or incoherent noise. The first requires the use of an *amplitude spectrum* while the second is analyzed using an *amplitude spectral density*. To determine the expected levels of vibration on a payload, these must be combined with the *vibration transfer function* for the isolation system supporting it.

1.2.1 Periodic Noise

Periodic noise usually comes from rotating machinery. By far the most common example are the large fans used in HVAC systems. These fans spin at a constant rate, and can generate a continuous, single-frequency vibration (and sometimes several harmonic frequencies as well). Another common source is air compressors. Unlike building fans, these cycle on and off according to demand. Compressors should be considered periodic, coherent noise sources, though they are *nonstationary*, meaning a measurement will change depending on whether the source is active or not. All periodic noise sources should be measured using an amplitude spectrum measurement, whether they are stationary or not.

An *amplitude spectrum* measurement is produced by taking the *Fourier transform* of data collected from a sensor measuring the noise. The most common sensor is an accelerometer, which will produce a spectrum with units of *acceleration* as a function of frequency. Accelerometers are popular because they have a "flat" frequency response, and random ground noise is usually fairly "flat"

* See Cyril M. Harris, Ed., Shock and Vibration Handbook, Third Ed. (The McGraw-Hill Companies, 1987)

in acceleration (see section 1.2.2 below). Amplitude spectrums can also be expressed as velocity or position amplitudes as a function of frequency. Most spectrum analyzers use the Fast Fourier Transform, or FFT. An FFT analyzer finds the amplitude of each frequency in the input data, and plots it. This includes the amplitudes and frequencies of any periodic noise sources. The amplitudes of periodic noise sources measured using an amplitude spectrum are independent of the length of the data record.

1.2.2 Random Noise

Random, or incoherent noise is measured using an *amplitude spectral density*. The difference is that the amplitude spectrum (above) is multiplied by the square-root of the data record's length before being displayed by the analyzer. The result is a curve which measures the random noise with units of [units] / $\sqrt{Hz}$, where [units] may be acceleration, velocity, or position. This *normalization* for the *measurement bandwidth* ensures that the measured noise level is independent of the length of the data record.† Without making this correction, for example, the level of random noise would appear to decrease by a factor of ten if the length of the data record were increased by a factor of 100. Note that periodic noise sources will appear to grow in amplitude as the data record gets longer when using the spectral density. Random ground noise levels vary greatly, but an "average" site may have 0.5 $\mu g / \sqrt{Hz}$ of noise between 1 and several hundred Hertz. Random noise can also be nonstationary. For example, stormy weather can significantly increase levels of random seismic noise. Figure 1 illustrates different noise levels in buildings.*

Velocity (-), Position (/), and Acceleration (\) for different VC:

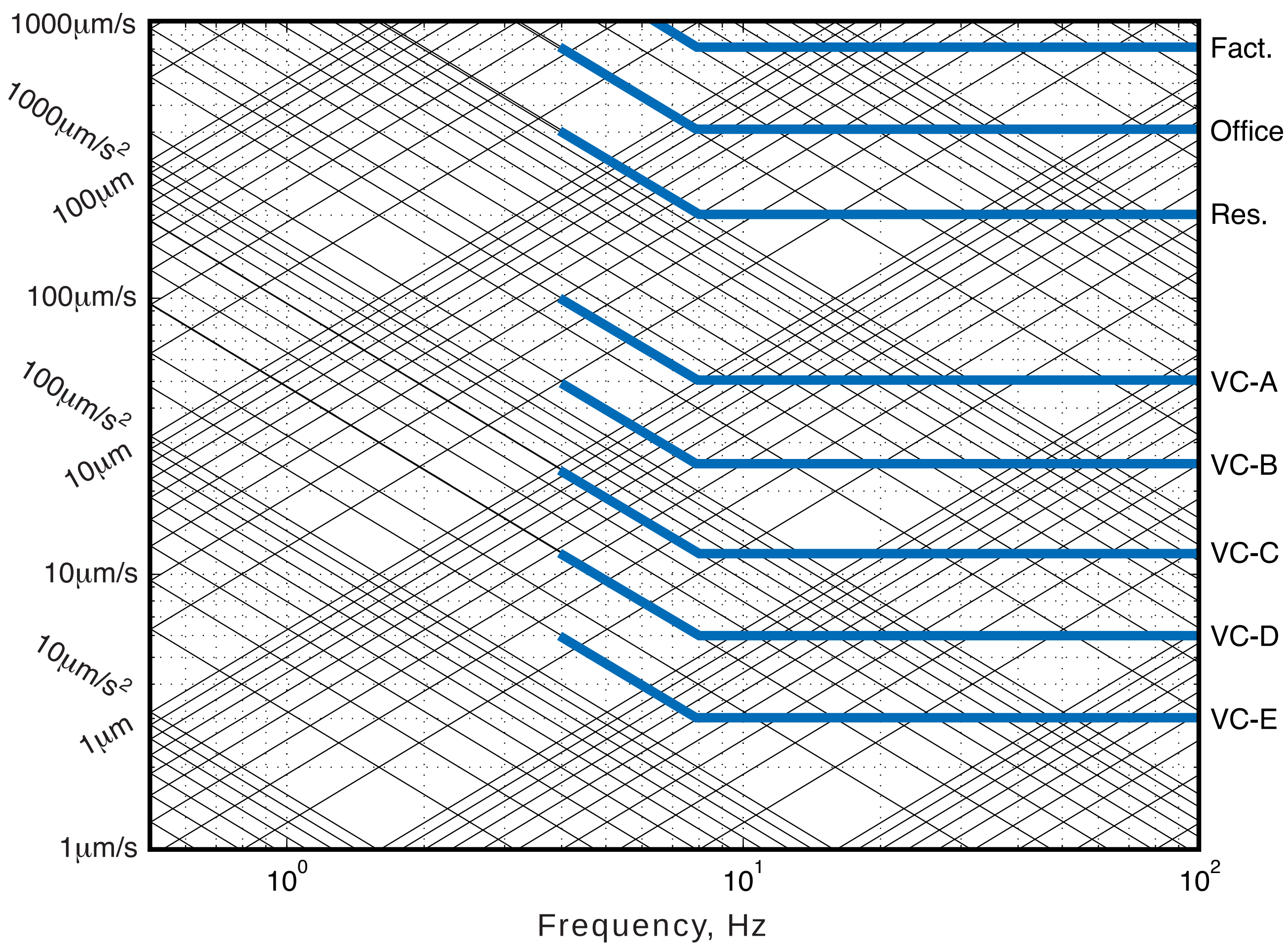

Figure 1

*Reprinted with permission from Collin Gordon Associates. VCA–VCE refer to accepted standards for vibration sensitive tools and instruments. The levels displayed are rms values measured in 1/3 octave band center frequencies. 1/3 octave plots are discussed in section 1.2.3.

†Other normalizations often apply such as corrections for "data-windowing" which is beyond the scope of this text. See "The Fundamentals of Signal Analysis" –Application Note Number 243. Hewlett Packard Corporation.

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1.2.3 Measuring RMS Values

Since most locations have a combination of both random and periodic noise sources, it is often desirable to come up with a single number which characterizes noise levels. This is usually done by quoting an RMS (Root-Mean-Squared) noise level within a specified range of frequencies.

Fortunately, this is easily done by integrating the power spectral density or PSD over the frequency range of interest. Since the PSD is just the square of the amplitude spectral density, we have the following expression for the RMS motion between the frequencies f_1 and f_2:

$$\sqrt{\int_{f_1}^{f_2}\left[\frac{Amp(f)}{\sqrt{Hz}}\right]^2 df} \qquad [1]$$

This formula correctly calculates the RMS value of the measurement taking into account both periodic and random noise sources. Most spectrum analyzers are capable of performing this integration as a built-in function. The contribution to this RMS value from any single periodic source can be measured using the amplitude spectrum (*not* the amplitude *density*), and dividing the peak value by $\sqrt{2}$. The contribution from several peaks can be combined by adding them in quadrature. RMS values are also sometimes expressed in "1/3 octave plots" in which a histogram of the RMS values calculated in 1/3 octave frequency bins is displayed as a function of frequency. An octave is a factor of two in frequency.

1.2.4 Characterizing Isolators

The noise level on a payload can be predicted by measuring the ground noise as described above, then multiplying those spectra by the *transfer function* for the isolation system. The transfer function is a dimensionless multiplier specified as a function of frequency, and is often referred to as the isolators' *transmissibility*. It is typically plotted as the ratio of table motion to ground motion as a function of frequency. It is common to express transmissibility in terms of decibels, or dB:

$$T_{dB} \equiv 20 \times \log_{10}\left(\frac{Payload\ Motion}{Floor\ Motion}\right) \qquad [2]$$

In practice, measuring the transfer function for an isolation system can be corrupted by other noise sources acting on the payload (such as acoustic noise). This is the primary reason why many measured transfer functions are noisy. To improve the quality of a transmissibility measurement, a "shake table" can be used. This is dangerous, however, as it can misrepresent the system's performance at low levels of vibration. The transfer function for pneumatic isolators is discussed below.

2.0 An Idealized Isolator

Figure 2 shows an idealized, one degree-of-freedom isolator based on a simple harmonic oscillator. It consists of three components: The isolated mass (*M*) represents the payload being isolated, and is shown here as a single block mass with no internal resonances.

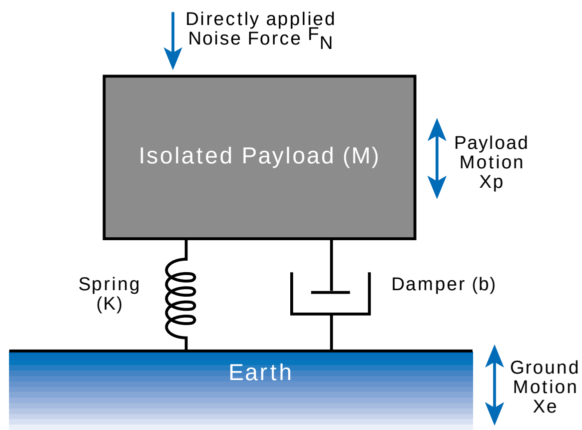


Figure 2

A spring (*k*) supports the payload, and produces a force on the payload given by:

$$Force = k \times (X_e - X_p) \qquad [3]$$

where X_e and X_p represent the (dynamic) position of the earth and payload respectively. The third component is the damper (*b*), which is represented schematicly as a dashpot. It absorbs any kinetic energy the payload (*m*) may have by turning it into heat, eventually bringing the system to rest. It does this by producing a force on the payload proportional and opposite to its velocity relative to the earth:

$$Force = b \times \left(\frac{dX_e}{dt} - \frac{dX_p}{dt}\right) \qquad [4]$$

The presence of X_e in both of these equations shows that vibration of the earth is transmitted as a force to the payload by both the spring (k) and the damper (b). Rather than use the parameters (M), (k), and (b) to describe a system, it is common to define a new set of parameters which relate more easily to the observables of the mass-spring system. The first is the *natural resonant frequency* ω_0:

$$\omega_0 = \sqrt{\frac{k}{M}} \qquad [5]$$

It describes the frequency of free oscillation for the system in the absence of any damping ($b = 0$) in *radians/second*. The frequency in cycles per second, or Hertz (Hz) is this angular frequency divided by 2π. One of two common parameters are used to describe the damping in a system: The *Quality factor Q* and the *damping ratio* ζ.

$$Q = \frac{\omega_0 M}{b} \quad \text{and} \quad \zeta = \frac{b}{2M\omega_0} \qquad [6]$$

It can be shown that the transmissibility for this idealized system is:

$$T \equiv \frac{X_p}{X_e} = \sqrt{\frac{1 + \left(\frac{\omega}{Q\omega_0}\right)^2}{\left(1 - \frac{\omega^2}{\omega_0^2}\right)^2 + \left(\frac{\omega}{Q\omega_0}\right)^2}} \qquad [7]$$

Figure 3 plots the transmissibility of the system vs. the frequency ratio ω/ω_0 for several values of the quality factor Q. The values of Q plotted range from 0.5 to 100. $Q = 0.5$ is a special case called *critical damping*, and is the level of damping at which the system will not overshoot the equilibrium position when displaced and released. The damping ratio ζ is just the fraction of the system's damping to critical damping. We use Q rather than ζ because $T \simeq Q$ at $\omega = \omega_0$, for Qs above about 2. There are several features which characterize the transmissibility shown in Figure 3:

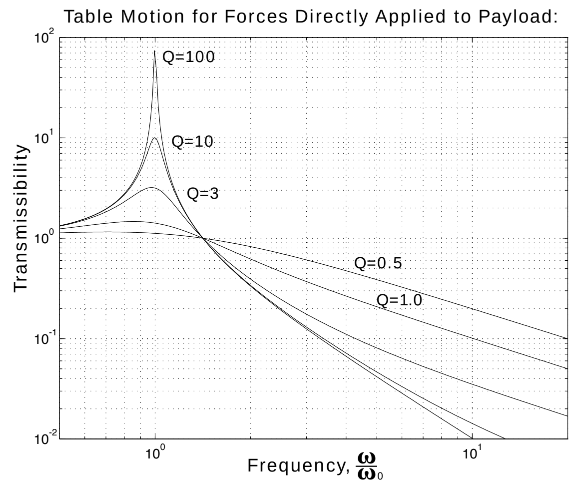


Figure 3

- In the region $\omega \ll \omega_0$, the transmissibility for the system is $\simeq 1$. This simply means that the payload tracks the motion of the earth, and no isolation is provided.
- In the region where $\omega \simeq \omega_0$, the transmissibility is greater than one, and the spring/damper isolator amplify the ground motion by a factor roughly equal to Q.
- As ω becomes greater than ω_0, the transmissibility becomes proportional to $(\omega_0/\omega)^2$. This is the region where the isolator is providing a benefit.
- In the region $\omega \gg \omega_0$, the best isolation is provided by the system with the smallest level of damping. Conversely, the level of isolation is compromised as the damping increases. Thus there is always a compromise between isolation in the region $\omega \gg \omega_0$ and $\omega \simeq \omega_0$.

The amplitude of motion transmitted to the payload by forces directly applied to it has a slightly different form than that expressed in Equation 7. This transfer function has units of displacement per unit force, so it should not be confused with a transmissibility:

$$\frac{X_p}{F_p} = \frac{Q}{M\left[Q^2\left(\omega_0^2 - \omega^2\right)^2 + \left(\omega\omega_0\right)^2\right]^{1/2}} \qquad [8]$$

Figure 4a plots this function vs. frequency. Unlike Figure 3, decreasing the Q reduces the response of the payload at all frequencies, including the region $\omega \gg \omega_0$.

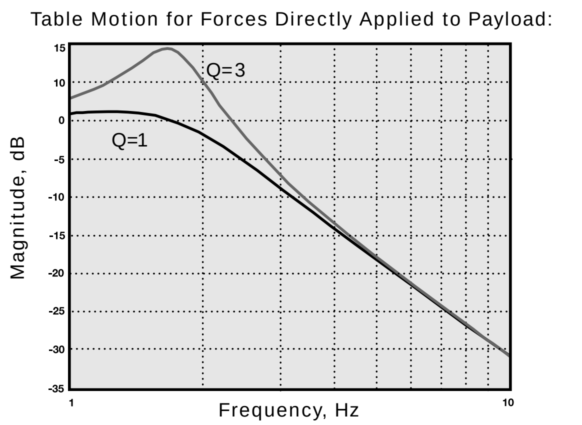


Figure 4a

TMC's MaxDamp isolators take advantage of this for applications where the main disturbances are generated on the isolated payload. Figure 4b shows the time-domain response of the payload corresponding to the curves shown in Fig. 4a. This figure also illustrates the decay of the system once it is disturbed. The envelope for the decay is $\exp(-\omega_0 t / 2Q)$.

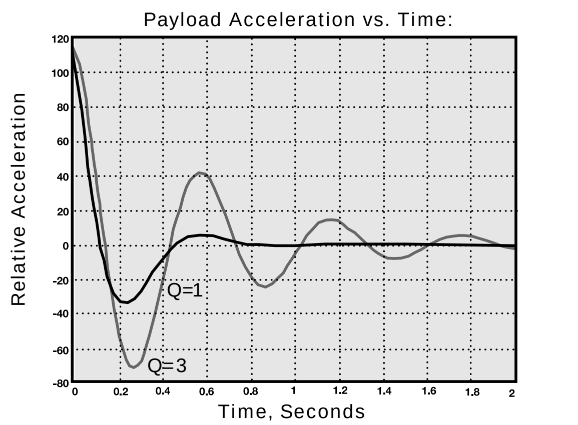


Figure 4b

There are some significant differences between real systems and the simple model shown in Fig. 2, the most significant being that real systems have six degrees-of-freedom (DOF) of motion. These DOF are not independent, but strongly couple in most systems. For example, "horizontal transfer functions" usually show two resonant peaks because horizontal motions of a payload drive tilt motions, and vice-versa. A detailed description of this type of coupling is beyond the scope of this catalog.

2.1 Pneumatic Isolators

Figure 5 shows a simplified pneumatic isolator. The isolator works by the pressure in the volume (V) acting on the area of a piston (A) to support the load against the

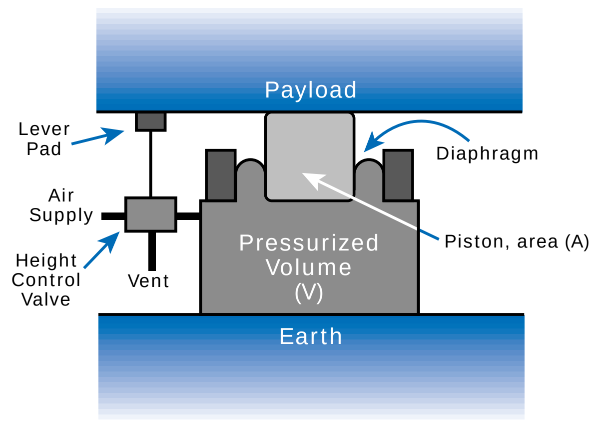


Figure 5

force of gravity. A reinforced rolling rubber diaphragm forms a seal between the air tank and the piston. The pressure in the isolator is controlled by a height control valve which senses the height of the payload, and inflates the isolator until the payload is "floating." There are many advantages to pneumatic isolators. It can be shown that the resonant frequency of the payload on such a mount is approximately:

$$\omega_0 \approx \sqrt{\frac{nAg}{V}} \qquad [9]$$

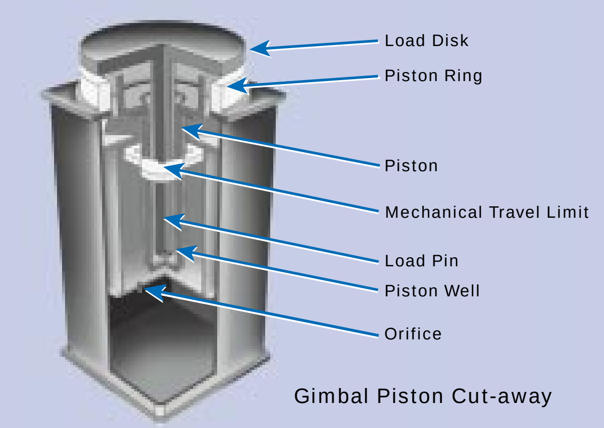
where g is acceleration of gravity (386 in/s^2 or 9.8 m/s^2) and n is the gas constant for air and equal to 1.4. Unlike steel coil springs, this resonant frequency is nearly independent of the mass of the payload, and the height control valve always brings the payload back to the same operating height.* Gas springs are also extremely light weight, eliminating any internal spring resonances which can degrade the isolator's performance.

The load capacity of an isolator is set by the area of the piston and the maximum pressure the diaphragm can tolerate, and is simply the product of these two numbers. It is common to rate the capacity at 80 psi of pressure. This allows a 4" piston to support a 1,000 lb load (for example). Though the simple isolator in Figure 5 will work, it has very little horizontal isolation, and has very little damping.

*Equation 9 assumes the isolator's pressure is high compared with atmospheric pressure. Lightly loaded isolators will exhibit a slightly higher resonant frequency.

3.0 Practical Pneumatic Isolators

Figure 6 shows a cutaway view of TMC's patented Gimbal Piston isolator. It uses two air chambers instead of one. These are connected by a small orifice. As the piston moves up and down, air is forced to move through this orifice, producing a damping force on the payload. This type of damping is very strong for large displacements of the piston, and less for small displacements. This allows for fast settling of the payload, without compromising small-amplitude vibration isolation performance. Damping of this type usually produces a $Q \approx 3$ for displacements on the order of a few mm.



Gimbal Piston Cut-away

Figure 6

The damping which an orifice can provide is limited by several factors. TMC's MaxDamp isolators use a different method: multi-axis viscous fluid damping (patent pending). These isolators can extend the damping to near critical levels for those applications which require it. For example, semiconductor inspection equipment often uses very fast moving stages to transport wafers. MaxDamp isolators allow the payload to settle very quickly after a stage motion, while still providing significant levels of vibration isolation. The isolator uses a very low outgassing, high-viscosity synthetic oil which is hermetically sealed within the isolator's single air chamber. A special geometry ensures the isolator damps both vertical and horizontal motions (in both X and Y directions) with equal efficiency.

Both the Gimbal Piston and MaxDamp isolators incorporate a simple and robust pendulum isolator to provide horizontal isolation. Like air springs, pendulums also produce an ω_0 which is payload-independent, and equal to $\sqrt{g/l}$, where l is the length of the pendulum. In the Gimbal Piston, the pendulum is actually the piston itself: The payload is supported by a *load disk* which transfers its burden to the bottom of the *piston well* through the *load pin* which contacts the bottom of the well with a pivoting thrust bearing. As the payload moves sideways, the piston well pivots like a gimbal in the plane of the diaphragm. Thus a pendulum is formed, whose length is equal to the vertical distance from the roll in the diaphragm to the bottom of the load pin.

TMC's Compact Sub-Hertz Pendulum (CSP™) system (patent pending) uses a different type of pendulum concept to extend horizontal resonant frequencies as low as 0.3 Hz. This isolator uses a geometrical lever effect to "fold" a 0.3 Hz pendulum into a package less than 16" (400 mm) high. An equivalent simple pendulum would have to be 110" (almost 3 meters) tall. The CSP is discussed further in the Advanced Products section of this catalog.

Horizontal damping in most isolators comes from horizontal-to-tilt coupling: As a payload moves sideways, it also exercises the isolators in the vertical direction (through tilt), thereby providing damping. Some systems, like TMC's MaxDamp isolators, damp horizontal motions directly with fluidic damping.

At small amplitudes, small amounts of friction in the rolling diaphragm and the small resistance to flow presented by the damping orifice have an impact on the isolator's performance. For this reason it is important to use as small an excitation level as possible when measuring their transmissibility.

3.1 Number and Placement of Isolators

Three or more isolators are required to support a payload, the most common number being four. Since there can only be three valves in a system (see 3.3), two legs in a four post system must be connected as a master/slave combination. Although a master/slave combination forms an effective support point, the damping it produces is much different than a single (larger) isolator at that point would provide. TMC always recommends using at least four isolators (except for "round" payloads like NMR spectrometers). Placement of these isolators under a payload has a dramatic effect on the performance of systems.

For small rigid payloads, like the granite structures in semiconductor manufacturing equipment, it is best to place the isolators as close to the corners of the payload as possible. This dramatically improves the tilt stability of the system, reduces the motions of the payload caused by onboard disturbances, and improves both the *leveling* and *settling times* for the system. *Leveling time* is the time for the valving system to bring the payload to the correct height and tilt. *Settling time* is the time for a payload to come to rest after an impulse disturbance.

For extended surfaces, such as large optical tables, the isolators should be placed under the surface's nodal lines. This minimizes the influence of forces transmitted to the table through the isolators. This is discussed in 4.3. For either type of payload, it is always better to position the payload's center-of-mass in the same plane as the isolator's effective support points. This improves the stability of the system (see 3.4) and decouples the horizontal and tilt motions of the payload.

Uneven floors can be accommodated in several ways. Most TMC isolators have a ±0.5 inch travel range, and this provides enough flexibility for almost all applications. Some systems also provide leveling feet. If a floor is extremely uneven, then providing piers for the isolators may be required. Some free-standing isolators or other types of supports (like rigid tripods) must be grouted to the floor if the floor's surface has a poor surface quality. Quick-setting "ready-mix" concretes are well suited for this purpose.

3.2 Safety Features

The ease with which pneumatic isolators can lift payloads weighing several thousand pounds belies the severity of their burden. By tying isolators together with "tie bars," the risk of toppling such massive loads through accident or events like earthquakes is dramatically reduced. TMC's tiebars are heavy-gauge formed channels which use constrained-layer damping to prevent them from resonating. Such damping is hardly required, however, since the isolation efficiency of the isolators at those frequencies is extremely high. Systems can also be provided with earthquake restraint brackets which prevent the payload from shaking off the isolators in an extreme event.

Of great importance to safety are the travel limits built into TMC's isolators. Figure 6 shows an internal "key" (yellow) which prevents the system from overextending even when pressurized to 120 psi (830 k *Pa*) under "no load" conditions. Since there can be several thousands of pounds of force behind the isolator's piston, an isolator without such a travel limit can quickly become a cannon if suddenly unloaded. Protection such as chain-linked pressure reliefs does not provide the intrinsically high level of safety a mechanical travel limit does.

3.3 Leveling Valves

All rigid payloads, even those with ten isolators, use only three height control valves. Because three points define a plane, using a greater number of valves would *mechanically overconstrain* the system, and result in poor position stability (like a four-legged restaurant table), and a continuous consumption of air. Proper placement and plumbing of these three valves is crucial to optimizing the performance of a system.

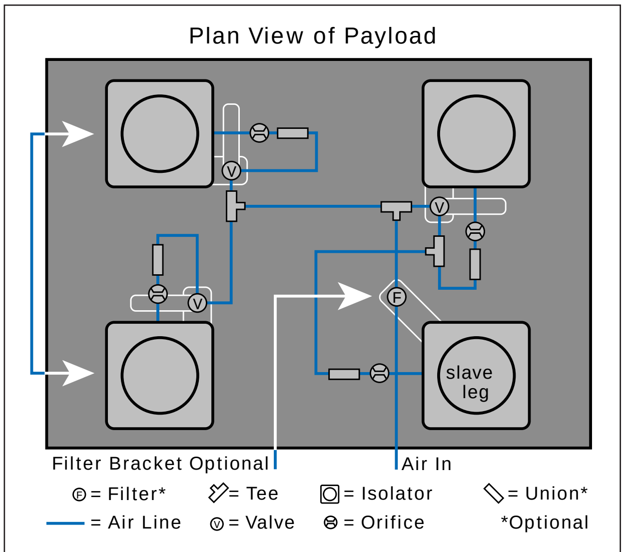


Figure 7a

Figure 7a and Figure 7b show the typical plumbing for a four-post and six-post system. A system contains three valves, a pressure regulator/filter (optional), some quick-connect tees and an orifice "pigtail" on each isolator. The pigtail is a short section of tubing with an orifice inserted inside. This section is marked with a red ring, and has a union on one end to connect to the height control valves' air lines. A mechanical valving system is a type of servo, and these orifices limit the "gain" of the servo to prevent oscillation. Some very high center-of-gravity systems may require smaller orifices to prevent instabilities. TMC uses fixed orifices rather than adjustable needle valves because of their long-term stability, and ease of use.

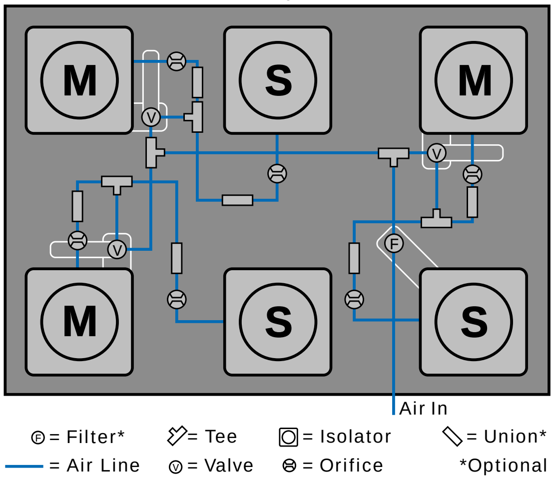


Figure 7b

In a system with four or more isolators, two or more of those isolators need to be tied together. Usually the valve is mounted near an isolator (for convenience), and that isolator is called the "master". The remote isolators(s) using that valve are called slaves. Choosing which legs are "master" and "slave" affects the stability of the system (See 3.4), and has a large impact on a system's dynamic behavior. Dynamic performance is particularly important in semiconductor inspection machines which have fast moving stages. There are several "rules of thumb" which can be applied to make the correct choice. These can conflict with each other on some systems. Some experimentation may be required to determine the optimal choice.

These rules, in approximate order of importance are:

1. The *effective support* point for a master and its slaves is at their geometric center. For a master with a single slave, this point is midway between the mounts. There are always only three "effective" support points for any system. Connecting these points forms a "load triangle." The closer the payload's center-of-mass (COM) is to the center of this triangle, the more stable the system will be. For example, on a four post system, the master/slave combination should support the lighter end of the payload.

2. A corollary to rule #1 is that the system should be plumbed so that the pressure difference between all isolators is minimized.

3. The *gravitational tilt stability* of a system is proportional to the square of the distance between the isolators. Therefore, for greatest stability, the master/slave combinations should be on the long side of a payload.

4. The tilt axis with the highest stiffness, damping and stability is the one parallel to the line between the master and slave legs (in a four post system). For moving stage applications, the main stage motion should be perpendicular to the line between the master and slave leg.

5. A moving stage can cause a cross-axis tilt because the valve for the master/slave legs is not co-located with the effective support point. For this reason, many systems should have the valve moved from the master leg to the effective support point.

6. A *control triangle* is formed by the three points where the valves contact the payload. Like the load triangle, the system will have the greatest stability and best positioning accuracy if the COM is inside this triangle. The valves should be mounted, and their "arms" rotated such that this triangle has the largest possible area.

7. Sometimes following the above rules results in a system with poor height and tilt positioning accuracy. In this case, an alternate choice for the master/slave combination(s) might be required.

In addition to valve location, there are several different types of valves which are available. TMC offers a *standard* and *precision* mechanical valve. The standard valve is less expensive, and has a positioning accuracy (dead band) of around 0.1" (2.5 mm). It has the property that the valve is tightly sealed for motions smaller than this. This makes it ideal for systems which must use pressurized gas bottles for an air supply. Precision valves offer a 0.01" (0.3 mm) or better positioning accuracy, but leak a very small amount of air (they use all-metal valve seats internally). This makes them less suitable for gas bottle operation. Finally, TMC offers the Precision Electronic Positioning System (PEPS®)** which has a $\simeq$ 0.0001" ($\simeq 2\ \mu m$) position stability. Refer to the discussion of PEPS in the Advanced Products section of this catalog.

For cleanroom applications, TMC offers versions of the mechanical valves made from stainless steel and/or supplied with a vented exhaust line.

3.4 Gravitational Instability

Like a pen balanced on its tip, payloads supported below their center of mass are inherently unstable: as the payload tilts, its center-of-mass moves horizontally in a way that wants to further increase the tilt. Fighting this is the stiffness of the pneumatic isolators, which try to

** patent pending

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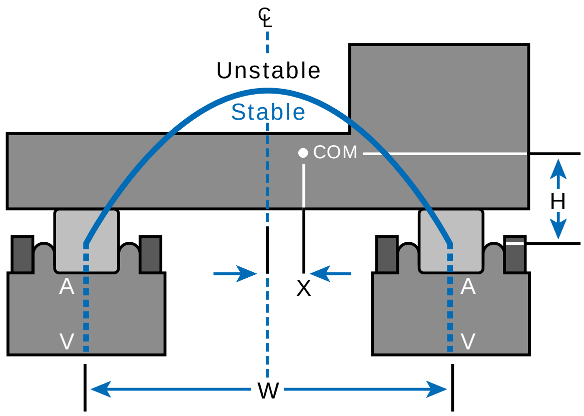

Figure 8

restore the payload to level. The balance of these two forces determines whether the system is *gravitationally stable* or not. Figure 8 shows a payload supported by two *idealized* pneumatic isolators. The width between the isolators' centers is W, the height of the payload's COM is H above the effective support point for the isolators, and the horizontal position of the COM from the centerline between the isolators is X. It can be shown that there is a region of stability given by the condition:

$$H < \frac{An}{V}\left(\frac{W}{2} - X\right)\left(\frac{W}{2} + X\right) \qquad [10]$$

or, for $X = 0$,

$$H < \frac{AnW^2}{4V} \qquad [11]$$

where n is the gas constant and is equal to 1.4.

This relationship is shown in Figure 8 as an inverted parabola which defines the stable and unstable regions for the COM location. The second equation clearly shows that the stability improves with the *square* of the isolator separation. This is important as it demonstrates that it is *not* the aspect ratio H/W that determines the stability of a system (as some references claim), and that the stable region is not a "triangle" or "pyramid." Unfortunately, real systems are not as simple as the one in Figure 8.

The ratio A/V in Equations 10 and 11 represents the stiffness of the isolators (see Equation 9 on page 14). In a two-chamber isolator, however, what is the proper V? Unlike the isolators in Fig. 8, which have a fixed spring constant, real isolators *have a spring constant which is frequency dependent*. At high frequencies, the orifice between the two chambers effectively blocks air flow, and V may be considered the top air volume alone. At the system's resonance, the "effective" air volume is somewhere between the top and total (top plus bottom) volumes. At low frequencies, the action of the height control valves gives the isolators an extremely high stiffness (corresponding to a very small V). Moreover, the height control valves also try to force the payload back towards level. These are only a few reasons why Equation 10 can't be applied to two chamber isolators. Instead, we assign three regions: stable, unstable, and borderline; the first two being based on the "total" and "top only" air volumes respectively. The stability region is also different for the axes parallel and perpendicular to the master/slave isolator axis.

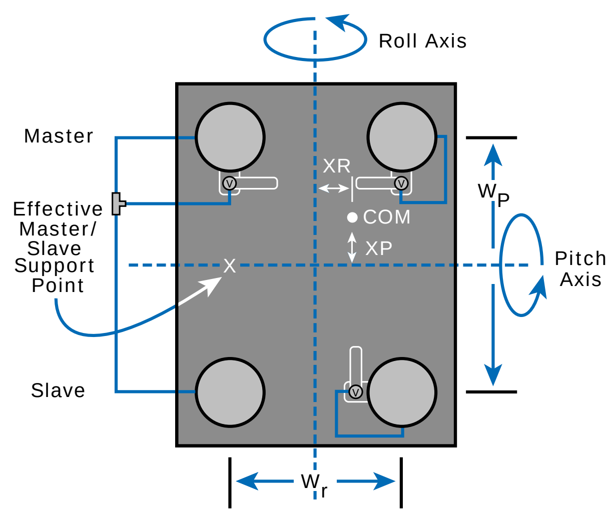

Figure 9

Figure 9 defines the two different axes for a four leg system. The pitch axis is less stable because the master/slave legs on the left of the figure offer no resistance to pitch at low frequencies (though they do resist pitch at frequencies above ≅1 Hz). To compensate for this, the master/slave combination is chosen such that W_p is greater than W_r . The region of stability is the volume defined by the inverted parabolas along the two axes.

The condition for *absolute stability* is:

$$H < \left(\frac{An}{2V}\right)_{Tot.}\left(\frac{W_p}{2} - X_p\right)\left(\frac{W_p}{2} + X_p\right)$$

and [12]

$$H < \left(\frac{An}{V}\right)_{Tot.}\left(\frac{W_r}{2} - X_r\right)\left(\frac{W_r}{2} + X_r\right)$$

and the formula for *absolute instability* is:

$$H > \left(\frac{An}{2V}\right)_{Top}\left(\frac{W_p}{2} - X_p\right)\left(\frac{W_p}{2} + X_p\right)$$

and [13]

$$H > \left(\frac{An}{V}\right)_{Top}\left(\frac{W_r}{2} - X_r\right)\left(\frac{W_r}{2} + X_r\right)$$

with the volume between being "possibly" or "marginally" stable. The ratios *A/V* are not universal and should be confirmed for different capacities and models of isolators, but are approximately 0.1 in^{-1} for $(A/V)_{Top}$ and 0.05 in^{-1} for $(A/V)_{Tot}$. Figure 10 illustrates what the marginally stable region looks like for two chamber isolators. Unfortunately, the COM of many systems ends up in this indeterminate region. These rules do not account for the actions of the height control valves, which will always improve a system's stability. If the payload has a mass which can shift (a liquid bath or a pendulum) these rules can also change.

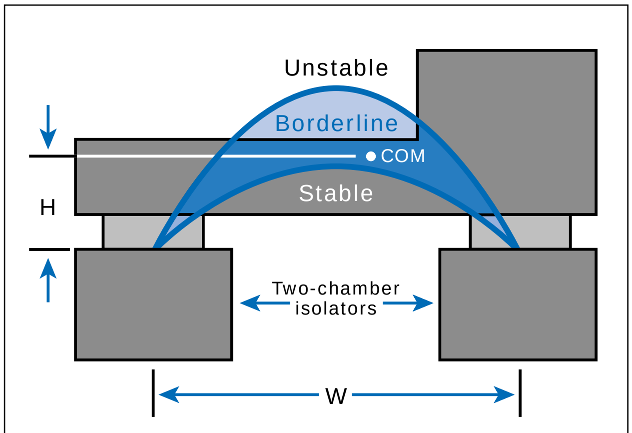

Figure 10

Equations 14 & 15 give "rules of thumb" for calculating the stability of a system. As with all such rules, it is only an approximation, based on an "average" isolation system. It is always best to use as low a COM as possible.

$$\frac{W^2}{H} > 60 \text{ inches} \quad \text{or} \quad \frac{W^2}{H} > 1.5 \text{ meters} \qquad [14]$$

Because MaxDamp isolators use a single air chamber, they are more stable, and the rule becomes:

$$\frac{W^2}{H} > 25 \text{ inches} \quad \text{or} \quad \frac{W^2}{H} > 0.64 \text{ meters} \qquad [15]$$

Note that the effective support point for TMC's Gimbal Piston isolators is approximately 7 inches below the top of the isolator. For lightly loaded isolators, these rules underestimate system stability. If your system violates these equations, or is borderline, the stability can be improved using counterweights, special volume isolators, different isolator valving, etc. Contact a TMC sales engineer for advice on the best approach.

4.0 High Performance Table Tops

Table tops are the platform for conducting many types of measurements and processes. They can serve as a mechanical reference between different components (such as lasers, lenses, film plates, etc.) as well as simply providing a quiet work surface. Tops typically use one of three constructions: a composite laminate, a solid material (granite) or a lightweight honeycomb. The choice of construction depends on the type and size of the application.

Figure 11 shows a typical laminated construction. These are usually 2 to 4 inches thick and consist of layers of steel and/or composite materials epoxy-bonded together into a seamless stainless steel pan with rounded edges and corners. A visco-elastic adhesive can be used between the plates to enhance the damping provided by the composite layers. All bonding materials are chosen to prevent delamination of the assembly due to heat, humidity, or aging. The ferromagnetic stainless steel pan provides a corrosion-resistant, durable surface which works well with magnetic fixtures. "Standard" sizes for these tops range from 24" square to 6' x 12', and can weigh anywhere from 100 - 5,000 lbs. This type of construction is not well suited to applications which require large numbers of mounting holes (tapped or otherwise). The ratio of steel to lightweight damping composite in the core depends primarily on the desired mass for the top.

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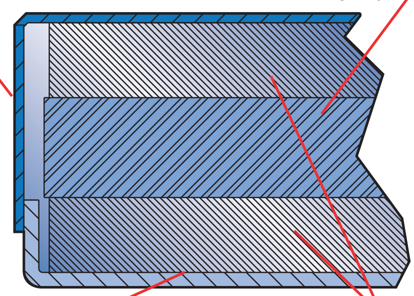


Figure 11

There are many applications in which a heavy top is of benefit: it can lower the center-of-gravity for systems in which gravitational stability is an issue. If the payload is dynamically "active" (like a microscope with a moving stage), then the increased mass will reduce the reaction motions of the top. Lastly, steel is very strong, and very high mass payloads may require this strength.

Granite and solid-composite tops offer a relatively high mass and stiffness, provide moderate levels of damping, and are cost effective in smaller sizes. Their nonmagnetic properties are desirable in many applications, and they can be lapped to a precise surface. Mounting to granite surfaces is difficult, however, and granite is more expensive and less well damped than laminate tops in larger sizes. The highest performing work surfaces are honeycomb-core tables.

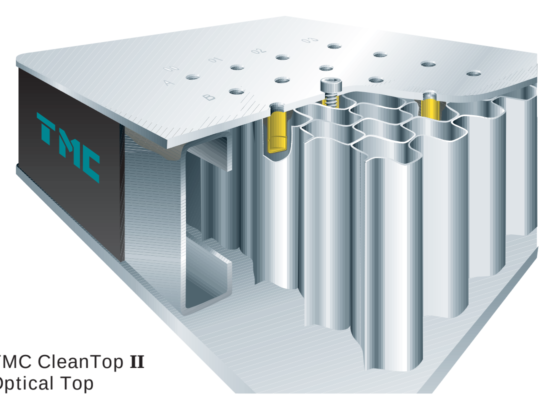
4.1 Honeycomb Optical Tables

Honeycomb core table tops are very lightweight for their rigidity, and are preferred for applications requiring bolt-down mounting or larger working surfaces. They can be made in any size from 1 foot on a side, and a few inches thick, to 5 x 16 feet and over 2 feet thick. Larger tops can also be "joined" to make a surface which is almost unlimited in size or shape. The smaller surfaces are often called "breadboards," and the larger sizes "optical tops" or "optical tables."

Honeycomb core tables were originally developed for high-precision optical experiments like holography. They evolved due to the limitations of granite surfaces, which were extremely heavy and expensive in larger sizes, and were difficult to securely mount objects to. The goal was to develop a work surface with the stability of granite without these drawbacks.

Honeycomb core tables are rigid for the same reasons as a structural "I-beam." An I-beam has a vertical "web" which supports a top and bottom flange. As weight is applied to the beam, the top flange is put in compression and the bottom in tension, because the web holds their separation constant. The primary stiffness of the beam comes from this compression and extension of the flanges. The web also contributes to the stiffness by resisting shear in its plane, but its most important function is resisting vertical compression (keeping the flanges separate). The same thing happens in an optical table (see figure 12). The skins of the table have a very high resistance to being stretched or compressed (like the flanges of the I-beam). The honeycomb core is extremely resistant to compression along its cells (serving the same role as the I-beam's web). As the core *density* increases (cell size decreases), the compressional stiffness of the core and its shear modulus increase, and the mechanical coupling to the skins improves – improving the performance of the table.

Optical tables are also much better than granite surfaces in terms of their thermal properties. Because of their metal construction and very low heat capacity (due to their relatively light mass), honeycomb core tables come to thermal equilibrium with their environment much faster than their granite counterparts. The result is a reduction in thermally-induced distortions of the working surface.



TMC CleanTop **II** Optical Top

Figure 12

4.2 Optical Table Construction

There are many other benefits to using a honeycomb core. The open centers of the cells allow an array of mounting holes to be placed on the table's surface. These holes may be capped to prevent liquid contaminants from entering the core, and "registered" with the core's cells. During the construction of TMC optical tops, the top skin is placed face down against a reference surface (a lapped granite block), and the epoxy, core, sidewalls, and bottom skin and damping system built up on top of it. The whole assembly is clamped together using up to 30 tons of force. This forces the top skin to take the same shape (flatness) of the precision granite block. Once the epoxy is cured, the table's top skin keeps this precise flatness (typically ±0.005") over its entire surface.

TMC's patented CleanTop II design allows the core to be directly bonded to the top and bottom skins of the table. This improves the compressional stiffness of the core, and reduces the thermal relaxation time for the table. The epoxy used in bonding the table is extremely rigid without being brittle yet allows for thermal expansion and contraction of the table without compromising the bond between the core and the skins.

Honeycomb core tables can also be made out of a variety of materials, including nonmagnetic stainless steel, aluminum for magnetically sensitive applications, and "super invar" for applications demanding the highest grade of thermal stability. Lastly, the individual cups sealing the holes in the top skin (unique to TMC's patented CleanTop II design) are made of stainless steel or nylon to resist a wide range of corrosive solvents.

The sidewalls of the optical table can be made out of many materials as well. Some of TMC's competitors' tops use a common "chipboard" sidewall which, though well damped, is not very strong, and can be easily damaged in handling or by moisture. TMC tables use an all-steel sidewall construction with constrained-layer damping to provide equally high levels of damping with much greater mechanical strength.

4.3 Honeycomb Optical Table Performance

The performance of an optical table is characterized by its static and dynamic rigidity. Both describe how the table flexes when subjected to an applied force. The first is its response to a static load, while the second describes the "free oscillations" of the table.

Figure 13 shows how the static rigidity of a table is measured. The table is placed on a set of line contact supports. A force is applied to the center of the table, and the table's deflection (δ) measured. This gives the static rigidity in terms of *μin/lbf* (or *μm/N*) This rigidity is a function of the table's dimensions and the physical properties of the top and bottom skins, sidewalls, core, and how they are assembled.

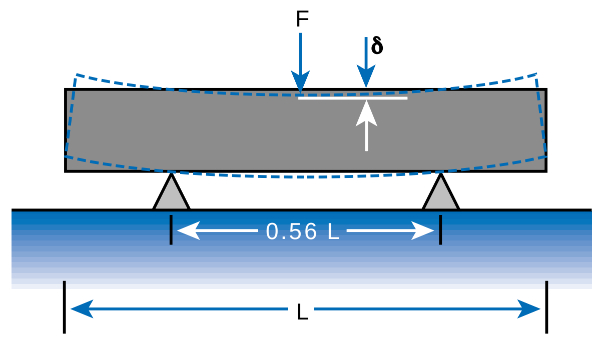


Figure 13

4.3.1 The Corner Compliance Curve

Dynamic rigidity is a measure of the peak-to-peak motion of a table's oscillations when it is excited by an applied impulse force. When hit with a hammer, several *normal modes* of oscillation of the table are excited, and each "rings" with its own frequency. Figure 14 shows the four lowest frequency modes of a table. Dynamic compliance is measured by striking the table with an impact testing hammer (which measures the level of the impact's force near the corner of the table). The table's response is measured with an accelerometer fastened to the top as close to the location of the impact as possible. The signals are fed to a spectrum analyzer which produces a *corner compliance curve*. This measures the deflection of the table in *μin/lbf* (or mm/N) for frequencies between 10 and 1,000 Hz.

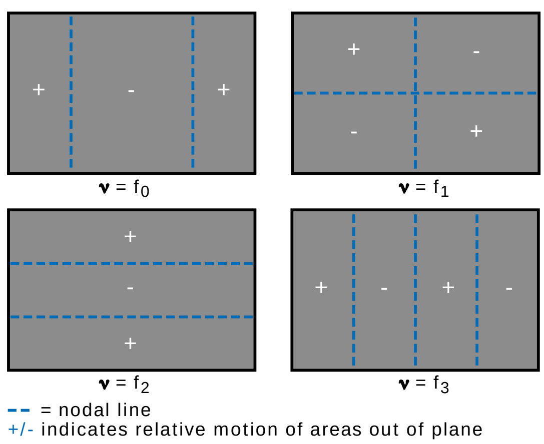


Figure 14

Each normal mode resonance of the top appears as a peak in this curve at its resonant frequency. The standard way to quote the dynamic compliance of a top is to state the peak amplitude and frequency of the lowest frequency peak (which normally dominates the response). Figure 15 shows the compliance curve for a table with low levels of damping (to emphasize the resonant peaks). The peaks correspond to the modes shown in Fig. 14. The curve with a slope of $1/f^2$ is sometimes referred to (erroneously) as the "mass line," and it represents the rigid-body motion of the table. "Mass line" is misleading because the rigid-body response of the top involves rotational as well as translational degrees of freedom, and therefore also involves the two moments of inertia of the table in addition to its mass. For this reason, this line may be ten times or more above the line one would calculate using the table's mass alone.

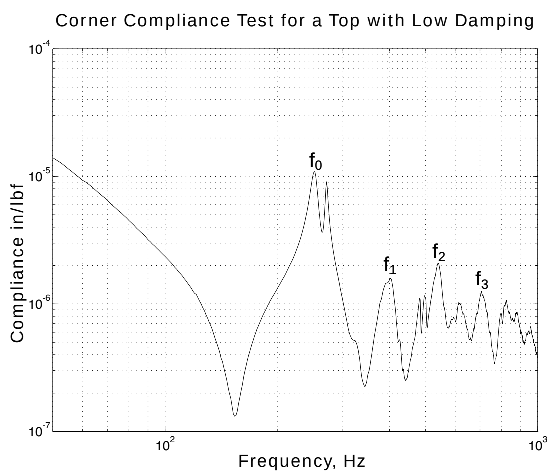


Figure 15

Figure 15: f_0-f_3 show the four lowest resonances of the table.

The compliance curve is primarily used to show how well a table is damped. The higher the level of damping, the lower the peak in the compliance test, and the quicker the table will ring down after an impact disturbance. There are two ways to damp the modes of a table: *narrow-band* and *broad-band* damping. The first uses tuned mechanical oscillators matched to the frequencies of the normal mode oscillations to be damped. Each matched oscillator can remove energy at a single frequency. TMC uses *broad-band damping*, where the mode is damped by coupling the table to second mass by a lossy compound. This damps all modes and all frequencies.

Tuned damping has several problems. If the frequency of the table changes (from placing some mass on it), then the damper can lose some of its effectiveness. Also, several dampers must be used, one for each mode (frequency) of concern. This compounds the matching problem. Each of these dampers are mounted in different corners of the table. This results in *different compliance measurements for each corner of a table.* Consequently, the quoted compliance curve may only apply for one of the four corners of a top. In addition, tuned dampers are strongly limited in how far they can reduce the Q. It is difficult, for example, to get within a factor of 10 of critical damping using reasonably-sized dampers.

In broad-band damping, the secondary masses are distributed uniformly through the table, producing a compliance curve which is corner-independent. It is also insensitive to changes in table resonant frequencies, and will damp *all* modes – not just those which have matched dampers. In fact, TMC's highest grade tables can have near *critical damping* of the lowest modes (depending on aspect ratios, thicknesses, etc.).

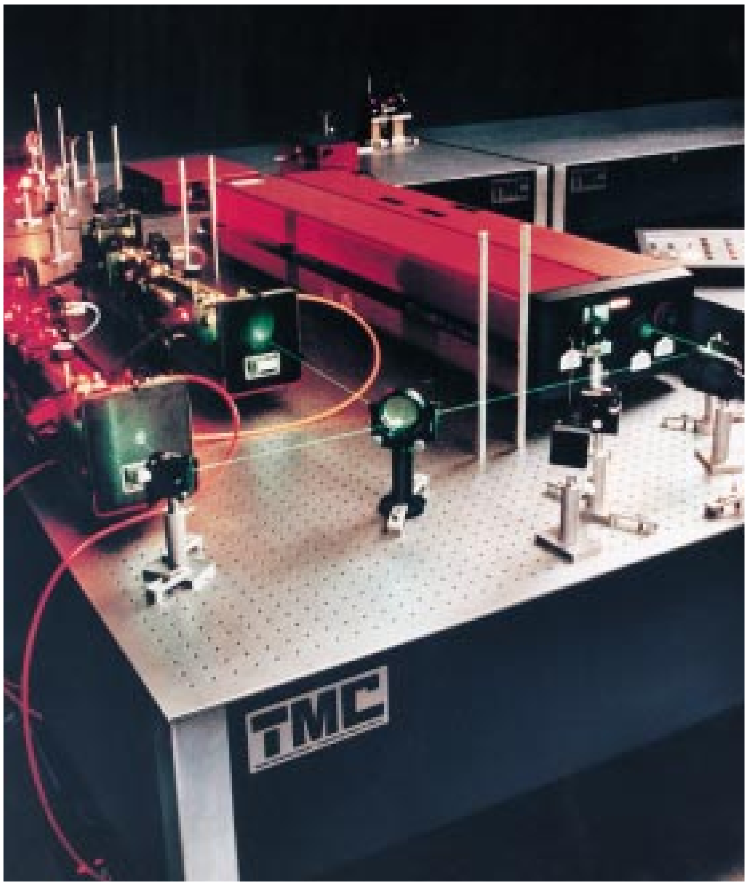
4.3.2 Compliance Curves as a Standard

Although used as a standard for measuring table performance, the corner compliance curve is far from a uniform and unambiguous figure of merit. The problem is not only with tables using tuned damping. All measurements are extremely sensitive to the exact location of the test impact, and the monitoring sensor. TMC measures compliance curves by placing the sensor in a corner 6" from the sides of the table, and impacting the table on the inboard side of the sensor. Since the core of the table is recessed from the edge of the table by 1-2", impacting the table closer to the corner produces "edge effects." The result is a test which is inconsistent from corner to corner or even impact to impact. On the other hand, measuring further from the corner can bring the sensor and the impact point dangerously close to a nodal line for the first few modes of the table (Figure 14). This is so sensitive that a few *inches* can have a dramatic effect on the measured compliance for a top.

It is also important to properly support the table being tested. TMC supports tables at four points, along the two nodal lines 22% from the ends of the table. Either pneumatic isolators, or more rigid rubber mounts can be used for this test (though rubber mounts may change the damping of higher-order modes). Though this is fairly standard with manufacturers, the customer must be aware that the compliance test will only represent their setup if they support their top in this way.

Nodal shapes present a major problem in the uniformity of the corner compliance curve as a standard figure of merit, since there is no industry or government standard for testing (like TMC's 6" standard for sensor locations). Part of the problem is the measurement point – near nodal line(s) for the modes – is a position where the resonance amplitude varies the most; from *zero* at the node to a maximum at the table's edge. The ideal place to make a compliance measurement would be where the mode shape is "flat." For example, this would be the center of the table for the first mode in Figure 14. Here, the measurement is almost independent of the sensor or impact locations *for the first mode only*. For many higher modes, however, this is dead center on nodal line(s), producing essentially meaningless results. Rather than bombard customers with a separate test for each mode shape, for better or worse, the corner compliance test has become the standard.

In recent years, some attempts have been made to produce other figures of merit. TMC does not use these because they compound the uncertainty of the compliance test with several other assumptions. So-called "Dynamic Deflection Coefficients" and "Maximum Relative Motion"[†] take information from the compliance curve, and combine it with an assumed input force spectrum. Unfortunately, the "real" relative motion you observe will also depend on the way your table is supported. If, for example, your top is properly supported by the isolators at the nodal lines of the lowest mode (0.53 L apart), then there is *no* excitation of the lowest mode from the isolators (on which these figures of merit are based). Likewise, if you support a top improperly, the mode can be driven to large amplitudes. Moreover, the "assumed" input depends on two very poorly defined factors: floor noise and isolator efficiency. Even if these are well defined, it is much more likely that acoustic sources of noise will dominate at these frequencies (typically 100-1000 Hz). For all these reasons, we consider these alternate figures of merit essentially meaningless, and do not use them.



[†]These particular figures of merit were developed by Newport Corporation of Irvine, CA.

5.0 Active Vibration Isolation Systems

This section will assist engineers and scientists in gaining a general understanding of active vibration isolation systems, how they work, when they should be applied, and what limitations they have. Particular attention has been given to the semiconductor manufacturing industry, since many applications have arisen in this field.

5.1 History

Feedback control systems have existed for hundreds of years, but have had their greatest growth in the 20th century. During World War II, very rapid advances were made as the technology was applied to defense systems. These developments continued, and even today most texts on control systems feature examples like fighter aircraft control and missile guidance systems.

Active vibration isolation systems were an extension of the electromechanical control systems developed for defense. As early as the 1950s, active vibration cancellation systems were being developed for applications like helicopter seats. Thus, active control systems specifically for vibration control have been around for over 40 years! In the precision vibration control industry, active vibration isolation systems have been available for nearly 20 years. There are many reasons why they haven't come into wide use.

Active vibration isolation systems are relatively complex, costly, and often provide only marginal improvements in performance compared with conventional passive vibration isolation techniques. They are also more difficult to set up, and their support electronics often require adjustment. Nonetheless, active systems can provide function which is simply not possible with purely passive systems.

Two things have lead to the renewed interest in active vibration control systems in recent years. The first is the rapid growth of the semiconductor industry, and the desire to produce more semiconductors, faster, and at a lower cost. Lithography and inspection processes usually involve positioning the silicon wafer relative to critical optical (or other) components by placing the wafer on a heavy and/or fast moving stage. As these stages scan from site to site on the wafer, they cause the whole instrument to "bounce" on the vibration isolation system. Even though the motion of the instrument may be small after such a move (a few mm), the resolution of the instrument is approaching, and in some cases going below 1 nm. Instruments with this type of resolution are inevitably sensitive to even the smallest vibration levels. Active systems help in these cases by reducing the residual motions of an isolated payload after such stage motions occur.

The second change which has made active systems more popular has been the advancement in digital signal processing techniques. In general, an active system based on analog electronics will outperform a digitally-based system. This is due to the inherent low noise and wide bandwidths available with high-performance analog electronics (a relatively inexpensive operational amplifier can have a 30 bit equivalent resolution and a "sampling rate" of many MHz). Analog electronics are also inexpensive. The problem with analog-based systems is that they must be manually adjusted and cannot (easily) deal with non-linear feedback or feedforward applications (see Section 5.4.3). Digital controllers have the potential to automatically adjust themselves, and to deal with non-linear feedback and feedforward algorithms. This allows active systems to be more readily used in OEM applications (such as the semiconductor industry). They can also be programmed to perform a variety of tasks, automatically switch between tasks on command, and can have software upgrades which "rewire" the feedback system without lifting a soldering iron. Though they can be more expensive to manufacture and develop, that cost differential is becoming small.

To further the reader's understanding of the costs and benefits of these systems, we have provided a brief introduction to the terminology and techniques of servo control systems.

5.2 Servos & Terminology

Although the terminology for active systems is fairly universal, there are some variations. The following discussion introduces the terminology used by TMC, and should help you with the concepts involved in active systems. The basis for all active control systems is illustrated in Figure 16.

It contains three basic elements:

- The block labeled "G" is called the *plant*, and it represents the behavior of your mechanical (or electronic, hydraulic, thermal, etc.) system *before any feedback is applied*. It represents a *transfer function*, which is the ratio of the block's output to its input, expressed as a function of frequency. This ratio has both a magnitude and a phase, and may or may not be unitless. For example, it may represent a *vibration transfer function* where the input (line on the left) represents ground motion and the output (line to the right) represents the motion of a table top.

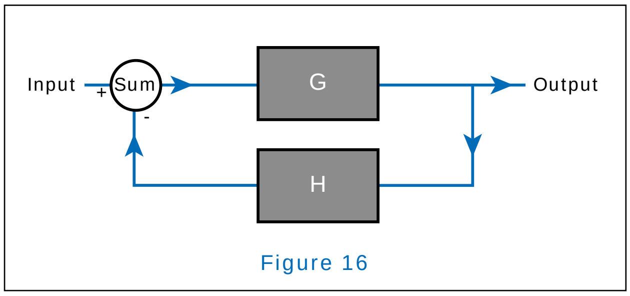


Figure 16: A basic feedback loop consists of three elements: The Plant,Compensation, and Summing Junction.

In this case, the ratio is unitless. If the input is a force, and the output a position, then the transfer function has units of (m/N). The transfer function of G has a special name: the *plant transfer function*.

- The block labeled "H" is called the *compensation*, and generally represents your servo. For a vibration isolation system, it may represent the total transfer function for a sensor which monitors the plant's output (an accelerometer), some electronic filters, amplifiers, and lastly an actuator which produces a force acting on the payload. In this example, the response has a magnitude, phase, and units of (N/m). Note that the *loop transfer function* for the system, which is the product (GH), must be unitless. The loop transfer function is the most important quantity in the performance and stability analysis of a control system, and will be discussed later.
- The circle is a *summing junction*. It can have many inputs which are all summed to form one output. All inputs and the output have the same units (such as force). A plus or minus sign is printed next to each input to indicate whether it is added or subtracted. Note that the output of H is always subtracted at this junction, representing the concept of negative feedback. The output of the summing junction is sometimes referred to as the *error signal* or *error point* in the circuit.

It can be shown that the *closed-loop transfer function* for the system is given by Equation 16. This is perhaps the single most important relationship in control theory. The denominator 1+GH is called the *characteristic equation*, since the location of its roots in the complex plane determine a system's stability. There are several other properties which are immediately obvious from the form of this equation.

$$\frac{Output}{Input} = \frac{G}{1+GH} \qquad [16]$$

First, when the *loop gain* (the magnitude $|GH|$) is much less than one, the closed-loop transfer function is just the numerator (G). For large loop gains ($|GH| \gg 1$), the transfer function is reduced or suppressed by the loop gain. Thus the servo has its greatest impact on the system when the loop gain is above *unity gain*. The frequency span between the *unity gain frequencies or unity gain points* is the *active bandwidth* for the servo. In practice, you are not allowed to make the loop gain arbitrarily high between unity gain points and still have a stable servo. In fact, there is a limit to how fast the gain can be increased near unity gain frequencies. Because of this, the loop gain for a system is usually limited by the available bandwidth.

Another obvious result from Equation 16 is that the only frequencies where the closed-loop transfer function can become large is where the magnitude of $|GH| \approx 1$, and its *phase* becomes close to 180°. As the quantity GH nears this point, its value approaches (-1), the denominator of

Equation 16 becomes small, and the closed-loop response becomes large. The difference between the phase of GH and 180° at a unity gain frequency for GH is called the *phase margin*. The larger the phase margin, the lower the amplification at the unity gain points. It turns out, however, that larger phase margins also decrease the gain of the servo within its active bandwidth. Thus, picking the phase margin is a compromise between gain and stability at the unity gain points. Amplification at unity gain will always happen for phase margins less than 90°. Most servos are designed to have a phase margin between 20° and 40°. Amplification at a servo's unity gain frequencies appear like new resonances in the system.

5.3 Active Vibration Cancellation

The previous section provided a qualitative picture of how servos function, and introduced the broad concepts and terminology. In reality, most active vibration cancellation systems are much more complex than the simple figure shown in Figure 16. There are typically 3 to 6 degrees-of-freedom (DOF) controlled: three translational (X, Y, and Z motions), and three rotational (roll, pitch, and yaw). In addition, there may be many types of sensors in a system, such as height sensors for leveling the system, and accelerometers for sensing the payload's motions. These are combined in a system using *parallel* or *nested* servo loops. While these can be represented by block diagrams like that in Figure 16, and are analyzed using the same techniques, the details can become quite involved. There are, however, some general rules which apply to active vibration cancellation servos in particular.

Multiple Sensors – Although you can have both an accelerometer measuring a payload's inertial motion, *and* a position sensor measuring its position relative to earth, *you can't use both of them at any given frequency*. In other words, the active bandwidth for a position servo cannot overlap with the active bandwidth for an accelerometer servo. Intuitively, this is just saying that you can't force the payload to track two independent sensors at the same time. This has some serious consequences.

Locking a payload to an inertial sensor (an accelerometer) makes the payload quieter, however the accelerometer's output contains no information about the earth's location. Likewise, locking a payload to a position sensor will force a payload to track earth more closely – including earth's vibrations. You can't have a payload both track earth closely and have good vibration isolation performance! For example, if you need more vibration isolation at 1 Hz, you must increase the gain of the accelerometer portion of the servo. This servo must have a low-frequency cutoff, which will be pushed down in frequency as this gain is increased. This means that the servo which positions the payload with respect to earth must have its gain lowered. The result is a quieter platform, but one that takes longer to move back to its nominal position when disturbed. This is discussed further in Section 5.6.

Gain Limits on Position Servos – As mentioned above, position sensors also couple ground vibration to a payload. This sets a practical limit on the unity gain frequency for a height control servo (like TMC's Precision Electronic Positioning System – PEPS®). To keep from degrading the vibration isolation performance of a system, the unity gain frequency for PEPS is limited to less than 3 Hz. This in turn limits its low-frequency gain (which determines how fast the system re-levels after a disturbance). Its main advantages are more accurate positioning (up to 100 times more accurate than a mechanical valve), better damping, better high-frequency vibration isolation, and the ability to electronically "steer" the payload using feedforward inputs (discussed later). It will not improve how fast a payload will re-level.[1] PEPS can also be combined with TMC's PEPS-VX™ system which uses inertial payload sensors to improve vibration levels on the payload.

Structural Resonances – Another important concern in active vibration isolation systems is the presence of structural resonances in the payload. These resonances form the practical bandwidth limit for *any vibration isolation servo which uses inertial sensors directly mounted to the payload*. Even a fairly rigid payload will have its first resonances in the 100-500 Hz frequency range. This would be acceptable if these were well-damped. In most structures, however, they are not. This limits the bandwidth of such servos to around 10-40 Hz. Though a custom-engineered servo can do better, a generic off-the-shelf active vibration cancellation system rarely does.

5.4 Types of Active Systems

Although we have alluded to "position" and "acceleration" servos, in reality these systems can take many different forms. The basic performance of the servo in Figure 16

[1] This is an approximate statement, since PEPS is a linear system, and mechanical valves are very non-linear. PEPS generally levels faster for small displacements, and slower for large ones.

can be augmented using *feedforward*. The following Sections introduce the most common configurations and briefly discuss their relative merits.

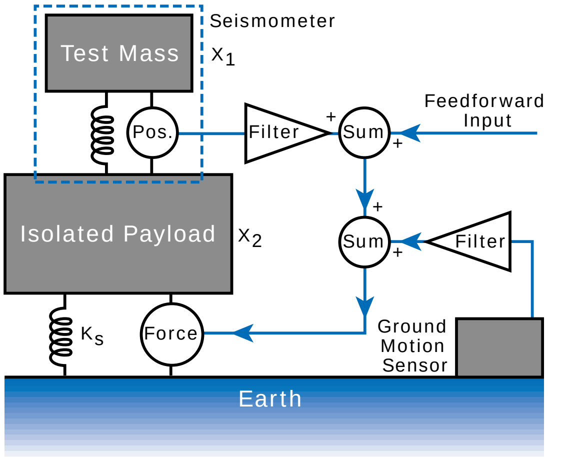


Figure 17

Figure 17: The basic inertial feedback loop uses a payload sensor and a force actuator, such as a loudspeaker "voice coil" to affect the feedback. Feedforward can be added to the loop at several points.

5.4.1 Inertial Feedback

By far the most popular type of active cancellation system has been the inertial feedback system, illustrated in Figure 17. Note that the pneumatic isolators have been modeled here as a simple spring. Neglecting the *feedforward* input and the *ground motion sensor* (discussed in Section 5.4.3), the feedback path consists of a seismometer, filter, and force actuator (such as a loudspeaker "voice coil"). The seismometer measures the displacement between its *test mass* and the *isolated payload*, filters that signal, then applies a force to the payload such that this displacement ($X_1 - X_2$) is constant – thereby nulling the output of the seismometer. Since the only force acting on the test mass comes from the compression of its spring, and that compression is servoed to be constant ($X_1 - X_2 \approx 0$), it follows that the test mass is actively isolated. Likewise, since the isolated payload is being forced to track the test mass, it must also be isolated from vibration. The details of this type of servo can be found in many references.[2]

The performance of this type of system is always limited by the bandwidth of the servo. As mentioned previously, structural resonances in the isolated payload limit the bandwidth in practical systems to 10-40 Hz (normally towards the low end of this range). This type of system is also "AC coupled" since the seismometer has no "DC" response. As a result, these servos have *two unity gain frequencies* – typically at 0.1 and 20 Hz. This is illustrated in greater detail in Section 5.6. As a result, the servo reaches a maximum gain of around 20-40 dB at ~2 Hz – the natural frequency of the passive spring mount for the system. The closed-loop response of the system has two new resonances at the ~0.1 and ~20 Hz unity gain frequencies. Due to the small bandwidth of these systems (only around two decades in frequency), the gain is not very high except at the natural (open-loop) resonant frequency of the payload. The high gain there completely suppresses that resonance. For this reason, it is helpful to think of these systems as *inertial damping systems*, which have the property of damping the system's main resonance without degrading the vibration isolation performance (passive damping can also damp this resonance, but significantly increases vibration feedthrough from the ground).

5.4.2 More Bandwidth Limitations

As mentioned earlier, these systems are limited to a low upper unity gain frequency by structural resonances in the payload. They are also limited, however, in how low their lower unity gain frequency can be pushed by noise in the inertial sensor. This is described in detail in the reference noted in Footnote 2. Virtually all commercial active vibration cancellation systems use *geophones* for their inertial sensors. These are simple, compact, and inexpensive seismometers used in geophysical exploration. They greatly outperform even high-quality piezoelectric accelerometers at frequencies of 10 Hz and below. Their noise performance, however, is not adequate to push an inertial feedback system's bandwidth to below ~0.1 Hz. To break this barrier, one would need to use much more expensive sensors, and the total cost for a system would no longer be commercially feasible.

Another low-frequency "wall" which limits a system's bandwidth arises when the inertial feedback technique is applied in the horizontal direction (note that a six degree-of-freedom [DOF] system has three "vertical" and three "horizontal" servos. Horizontal DOFs are those controlled using horizontally driving actuators – X, Y, and twist [yaw]). This is the problem of *tilt to horizontal coupling*. If you push a payload sideways with horizontal actuators and it tilts, then the inertial sensors read the tilt as an *acceleration*, and try to correct for it by accelerating the payload – which, of course, is the wrong thing to do. This effect is a fundamental limitation which has its roots in Einstein's *Principle of Equivalence* – which states that it

[2] See for example, P.G. Nelson, Rev. Sci. Instrum., 62, p.2069 (1991).

is impossible to distinguish between an acceleration and a uniform gravitational field (which a tilt introduces). The only solution to this problem is to not tilt a payload when you push it. This is *very* difficult to do, especially in geometries (like semiconductor manufacturing equipment) which are not designed to meet this requirement. Ultimately, one is forced to use a combination of horizontal *and vertical* actuators to affect a "pure" horizontal actuation. This becomes a "fine tuning" problem, however, which even at best yields marginal results. TMC prefers another solution.

Passive Horizontal Systems – Rather than use an active system to get an "effective" low resonant frequency, we have developed a *passive* isolation system capable of being tuned to as low as 0.3 Hz in the horizontal DOFs. Our Compact Sub-Hertz Pendulum System (CSP™) is not only a more reliable and cost effective way to eliminate the isolator's 1-2 Hz resonance, but it also provides better horizontal vibration isolation up to 100 Hz or more – far beyond what is practical for an active system. Unfortunately, such passive techniques are very difficult to implement for the vertical direction, so TMC recommends the use of systems like our PEPS-VX™ active cancellation system to damp the three "vertical" DOFs. PZT-based active systems, such as TMC's STACIS™, use another approach which allows for active control of horizontal DOFs (see Section 5.4.4).

5.4.3 Feedforward

The performance of the inertial feedback system in Figure 17 can be improved with the addition of *feedforward*. In general, feedforward is much more difficult than feedback, but it does offer a way to improve the performance of a system when the feedback servo is limited in its bandwidth. There are two types of "feedforward" systems which are quite different, though they share the same name.

Vibrational Feedforward – This scheme involves the use of a ground motion sensor, and is illustrated in Figure 17. Conceptually it is fairly simple: if the earth moves up by an amount Δz, the payload feels a force through the compression of the spring equal to $K_S \Delta z$. The ground motion sensor detects this motion, however, and applies an equal and opposite force to the payload. The forces acting on the payload "cancel," and the payload remains unaffected. "Cancel" is in quotes because it is a greatly abused term. It implies *perfect cancellation* – which never happens. In real systems, you must consider *how well* these two forces cancel. For a variety of reasons, it is difficult to have these forces match any better than around 10% – which would result in a factor of 10 improvement in the system's response. Matching these forces to the 1% level is practically impossible. The reasons are numerous: the sensor is usually a geophone, which doesn't have a "flat" frequency response. Its response must be "flattened" by a carefully matched conjugate filter. Then the *gain* of this signal must be carefully matched so the force produced by the actuator is exactly equal in magnitude to the forces caused by ground motion. These gains, and the properties of the "conjugate filter," must remain constant to within a percent with time and temperature – which is difficult. Gain matching is also extremely difficult if the system's mass distribution changes – which is not unusual in a semiconductor equipment application. Lastly, the cancellation level is limited by the sensor's inherent noise (noise floor).

Another limiting factor to vibrational feedforward is that it becomes a *feedback* system if the floor is not infinitely rigid (which it isn't). This is because the actuator, in pushing on the payload, also pushes against the floor. The floor will deflect with that force, and that deflection will be detected by the sensor. If the level of the signal produced by that deflection is large enough, then an unstable feedback loop is formed.

Because of the numerous problems associated with vibrational feedforward, TMC has not pursued it. Indeed, though available from other vendors, we know of no successful commercial application of the technique. It is possible, however, with ever more sophisticated DSP controllers and algorithms, that it will be more appealing in the future. The technique which does show significant promise is *command feedforward*.

Command Feedforward – Also shown in Figure 17, command feedforward is *only useful in applications where there is a known force being applied to the payload, and a signal proportional to that force is available*. Fortunately, this is the case in semiconductor manufacturing equipment where the main disturbance to the payload is a moving stage handling a wafer.

The concept here is very simple: A force is applied to the payload of a known magnitude (usually from a stage acceleration). An electronic signal proportional to that force is applied to an actuator which produces an equal and opposite force. As mentioned earlier, there is a tendency in the literature to overstate the effectiveness of this technique. Ridiculous statements claiming "total elimination" of residual payload motions are common. As in vibrational feedforward, there is a gain adjustment problem, but *all* issues concerning sensor noise or possible feedback paths are eliminated. This is true so long as the signal is a true *command* signal from (for example) the stage's motion controller. If the signal is produced from an encoder reading the stage position, then it is possible to form an unstable feedback loop. These systems can perform very well, suppressing stage-induced payload motions by an order of magnitude or more, and will be further discussed in Section 5.7.

5.4.4 PZT-Based Systems

Figure 18 shows the concept of a "quiet pier" isolator such as TMC's STACIS™ line of active isolators (patent #5,660,255). It consists of an *intermediate mass* which is hard mounted to the floor through a piezoelectric transducer (PZT). A geophone is mounted to it, and its signal fed back to the PZT in a wide-bandwidth servo loop. This makes a "quiet pier" for supporting the payload to be isolated. Isolation at frequencies above the servo's active bandwidth is provided by a ≃20Hz elastomer mount. This elastomer also prevents piers from "talking" to each other through the payload (a payload must rest on several independent quiet piers). This system has a unique set of advantages and limitations.

The vibration isolation performance of the STACIS system is among the best in the 0.2-20 Hz frequency range, subject to some limitations (discussed below). It also requires much less tuning than inertial feedback systems, and the elastomer mount makes the system all but completely immune to structural resonances in the payload. Alignment of the payload with external equipment (docking) is a non-issue because the system is essentially "hard mounted" to the floor through the 20 Hz elastomers. The settling time is very good because the response of the system to an external force (a moving stage) is that of the 20 Hz elastomer mount. This is comparable to the best inertial feedback systems. The stiffness of the elastomer mount also makes STACIS almost completely immune to room air currents or other forces applied directly to the payload, and makes it capable of supporting very high center of gravity payloads.

Unfortunately the PZT has a range of motion which is limited (around 20-25 μm). Thus the servo saturates and "unlocks" if the floor motion exceeds this peak-to-peak amplitude. Fortunately, in most environments, the floor motion will never exceed this amplitude. To obtain a good vibration isolation characteristic, the active bandwidth for the PZT servo is from ~0.1 to ~200 Hz. This high bandwidth is only possible if the isolator is supported by a very rigid floor. The isolator needs this because it depends on the intermediate mass moving an amount proportional to the PZT voltage *up to a few hundred Hertz*. If the floor has a resonance within the active bandwidth, this may not be true. Most floors have resonances well below 200 Hz, but this is acceptable as long as the floor is *massive* enough for its resonance not to be significantly driven by the servo. The proper form of the floor specification becomes *floor compliance*, in μin/lbf (or μm/N). In general, this requires STACIS be mounted directly on a concrete floor. It generally will work mounted on raised floors, or in welded steel frames only if the support frame is carefully designed to be very rigid. Another problem is "building sway" – the motion at the top of a building caused by wind. This is often more than 25 μm on upper floors, so the system can saturate if not used in but the first few floors of a building (depending on the building's aspect ratio and construction).

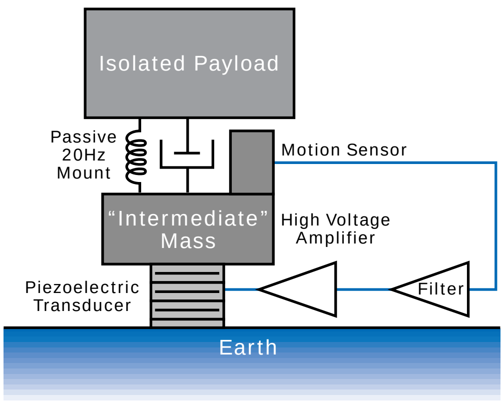


Figure 18

Figure 18: This method involves quieting a small "intermediate mass" with a high-bandwidth servo, then mounting the main payload on that "quiet pier" with a passive 20 Hz rubber mount.

In multi-DOF systems, each pier controls three translational degrees-of-freedom. With several isolators in a system, tilt and twist are also controlled. Each isolator requires five PZTs and three high-voltage amplifiers.

5.4.5 Exotics

There are many other types of active vibration isolation systems.

The first broad class of "alternate" active systems are the hybrids. One of these is a hybrid between a quiet pier and a simple pendulum isolator. Here, a three post system contains only three PZTs which control the vertical motion at each post actively (thus height, pitch, and roll motions of the payload are actively controlled). The "horizontal" DOFs are isolated using simple pendulums hanging from each 1-DOF quiet pier. This system has only about one-fifth the cost of a full-DOF quiet pier system because of the many fewer PZTs. On the other hand, the pendulum response of these systems in the horizontal direction is sometimes less than desirable.

There are also hybrids of the STACIS/quiet pier type of system with inertial feedback systems to improve the dynamic performance of the elastomer mount. These systems have additional cost and must be tuned for each application.

There are many non-linear schemes, of which the "grab-and-release" method is the most frequently thought of. Here the concept is to "grab" the payload just before a disturbance, and "release" it right after. This sounds reasonable, but a detailed modeling of the payload and the compliance of the supporting frame show that this scheme is very hard to implement without driving high-frequency resonances in the system. Also, when you "grab" a system, you are not holding it infinitely rigid, but you are merely increasing the stiffness of its mounting. For example, a "released" system might be 1.5 Hz horizontally, but "grabbed" means holding the payload rigidly to a welded frame. Even very stiff frames have resonances which are usually well below 50 Hz when loaded with heavy payloads. A typical value would be 20 Hz. If you release the payload at its point of *maximum velocity* for that 20 Hz resonance, the payload moves almost as much as if you didn't "grab" it at all. Releasing it at its point of *maximum deflection* imparts a sharp force impulse to the payload. Neither is good. The other problem with this type of non-linear control is that it becomes very payload- and frame-specific, which makes it undesirable for commercial production.

5.5 Types of Applications

Broadly, there two different types of applications: vibration critical or settling time critical. These are *not* the same, and each has different solutions. Some applications may be both, but since their solutions are not mutually exclusive, it is fair to think of both types independently. It is important to note, however, that since the solutions are independent, *so are their costs*. Therefore you should avoid buying an active system to reduce vibration if all you need is faster settling times, and vice-versa.

5.5.1 Vibration Critical Applications

Vibration critical applications are actually in the minority. This means the number of applications which need better vibration isolation than a *passive* system can provide is quite small. Passive vibration isolation systems by TMC are *extremely* effective at suppressing ground noise at frequencies above a few Hertz. There are only two types of applications where the vibration isolation performance of a passive isolator is a problem.

First, it is possible that the level of ground noise is so high that an instrument which is functional in most environments becomes ground noise sensitive. This usually only happens in buildings with very weak floors, or in tall buildings where building sway becomes an issue. This is an unusual situation, since most equipment (such as semiconductor inspection machines) usually come with a "floor spec" which vendors are very hesitant to overlook.

The second type of applications are those with the very highest degree of intrinsic sensitivity. Prime examples are atomic force and scanning tunneling microscopes (AFMs and STMs). These have atomic scale resolutions, and are sensitive to the *smallest* payload vibrations.

In both these situations the isolation performance of passive mounts is usually adequate, except for the frequency range from about 0.7 Hz to 3 Hz where a passive mount *amplifies* ground motion. This is a convenient coincidence, since active systems (such as the inertial feedback scheme) are good at eliminating this resonant amplification (but not much else). Again, it is important to avoid an active vibration cancellation system *unless you have an application which you are sure has a* ***vibration isolation*** *problem that cannot be solved with passive isolators*. Most semiconductor equipment today has a different issue: settling time.

5.5.2 Settling Time Critical Applications

Settling time critical applications are those where the vibration isolation performance of a passive pneumatic isolator is completely adequate, but the *settling time* of the isolator is insufficient. It is easy to determine if yours is such a system. If it works fine after you let the payload settle from a disturbance (stage motion), then you only have a settling time issue. (See Section 5.8). Before continuing, however, it is important to understand what is meant by "settling time."

Settling Time? – The term *settling time* is one of the most abused terms in the industry, primarily because it lacks a widely accepted definition. A physicist might define the settling time as the time for the energy in the system to drop by $1/e$. This is a nice, model-independent definition. Unfortunately, it is not what *anybody* means when they use the term. The most common definition is the "time for the system to stop moving." This is the worst of all definitions since it is non-physical, model and payload dependent, subjective, and otherwise completely inadequate. Nonetheless, it can be used with some qualifications.

In theory, a disturbed harmonic oscillator's motion decays exponentially – which is infinitely long lived. When used as a vibration isolator, one could think of the time when a system "stops moving" as the time required for the RMS motion of the system to reach a constant value, where the system's motion is dominated by the feedthrough of ground vibration. This is neither what people mean by settling time, nor is it model independent, since the "time to stop moving" depends on the magnitude of the initial disturbance, and the level of ground noise. In fact, ***there is no definition of "settling time" as a single specification which can be used to define system performance in this context – passive or otherwise***.

This is the definition used by TMC: *Settling time is the **time** required for a payload subjected to **a known input** to decay below a **critical acceleration** level*. This is an exact definition that requires ***three*** numbers: the *known input* is the initial acceleration of the payload immediately after the disturbance (stage motion) stops. The *critical acceleration level* is the maximum acceleration level the payload can tolerate and still successfully perform its function. The *settling time* is the period for the exponential decay of the initial acceleration to go below the critical acceleration level. Notice that we use a critical *acceleration level* and <u>not</u> a maximum displacement. It is not displacement of a payload which corrupts a process, but acceleration, since acceleration is what introduces the internal stresses in a payload which distort the structure, stage positioning, optics, etc. Of the three numbers, this is the most critical to understand, since it fundamentally characterizes the rigidity of your instrument.

For the product specifications in this catalog, the critical acceleration and input levels are unknowns. For this reason, we quote our settling time specifications as the time required for a 90% reduction in the initial oscillation amplitude.

5.6 The Problems with Inertial Feedback

Though inertial feedback systems can be used to reduce the settling time and improve vibration isolation performance, they have several significant drawbacks. As already mentioned, implementing a horizontal inertial feedback system is strongly limited by the tilt to horizontal coupling problem (Section 5.4.2). Another problem is that these systems have relatively poor *position* settling times.

Figure 19 shows the response of a payload to an external disturbance. It is based on a model of an idealized 1-DOF system, and is only meant to qualitatively demonstrate the performance of a multi-DOF system. Both curves represent the same active system, except the first plots the ratio of displacement to applied force, and the second plots the ratio of acceleration to applied force, both as a function of frequency. The only difference is that the first graph has been multiplied by two powers of frequency to produce the second. The curves show what a position sensor and an accelerometer would measure as this system was disturbed. Please note that the magnitude scales on these graphs have an arbitrary origin and are only meant for reference.

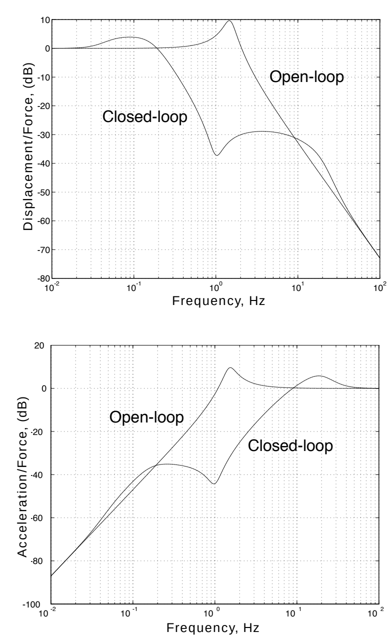


Figure 19

Figure 19: The curves show that the position response is dominated by a low-frequency resonance, while the acceleration response is dominated by a high-frequency peak. Note that the peak in the open-loop response is the same.

The curves show that the position response is dominated by a low-frequency resonance, while the acceleration response is dominated by a high-frequency peak. This is a counter-intuitive result, since the peak in the open-loop (purely passive) response is at the same frequency in both cases.

The good news is twofold: As promised, this system does a good job of suppressing the open-loop resonance in the system. In fact, it is even providing a substantial amount of additional isolation in the 0.5-5 Hz frequency range. Note that this model is idealized, and may overestimate the performance which you might actually achieve – especially for horizontal DOFs. The second piece of good news is that the acceleration curve is dominated by a well-damped resonance at around 20 Hz . If we assume the amplitude of the acceleration decays as:

$$Amplitude = A_0 \times e^{-t/\tau} \qquad [17]$$

Where A_0 is the initial amplitude and $\tau = Q / (\pi v)$. If the quality factor Q is approximately 2, then $\tau \approx 32$ ms. Quite good. For any payload which is sensitive to acceleration (which most are), the settling time for this system will be improved by an order of magnitude by this servo.

The problem with this system is illustrated in the first set of curves. They show the position response is dominated by a peak at ~0.1 Hz . Assuming the same Q as above, this means the decay constant τ is approximately 6.5 seconds! Even though the servo has been designed with a large phase margin to get the Q down to 2, the low frequency of the peak means it takes a long time to settle in position. Although payloads are most sensitive to accelerations, there are two notable cases where a long position settling time is a problem.

First, a long position settling time in the roll or pitch DOF of a payload can *look* like a horizontal acceleration. This is due to Einstein's Principle of Equivalence: As a payload tips, then the direction that gravity acts on the payload changes from purely vertical to some small angle off vertical. The Principle of Equivalence tells us that this is identical to having a level payload which is being *accelerated* by an amount equal to the tip angle (in radians) x g . In other words, each *mrad* of tilt turns into a *mg* of horizontal acceleration. Many instruments, such as electron microscopes, are sensitive to this.

Another significant problem is *docking* the payload. This is a common process where the payload must be periodically positioned relative to an off-board object with extreme accuracy – typically 20 to 200 µm. It can take an inertial feedback system a very long time to position to this level. There are two possible solutions to this. The first is to run the servo at a lower gain setting, sacrificing some isolation performance (which may not be needed), for a better position settling time. The second approach could be to turn off the servo for docking. Servos, unfortunately, *don't like* to be turned on and off rapidly – especially when their nominal gain is as high as the one illustrated here. This solution can be lumped into the "exotic" category of Section 5.4.5 under non-linear designs. It becomes very payload dependent, and is generally too unpredictable to use commercially. For this reason, TMC's PEPS-VX system uses the lower-gain approach.

It is also important to consider noise forces other than those caused by ground motion. Acoustic, "air current" and other sources can all introduce more noise on a payload than ground vibrations. Though inertial feedback systems reduce the influence of these noise sources, the noise on your payload may be higher than the ground motion multiplied by the transmissibility for the active system.

5.7 The Feedforward Option

For settling time sensitive applications, there is another option which is less expensive and avoids the problems associated with the inertial feedback method. As discussed in Section 5.4.3, *command feedforward* can be used to reduce the response of a payload to an external disturbance. You can use this technique with or *without* using the inertial feedback scheme in Figure 17. This section deals with the latter option.

5.7.1 Feedforward Pros

There are many advantages to using a feedforward *only* system. Some of these are:

- You don't spend extra money on improved vibration isolation performance which you don't need. The system is less expensive because you avoid the cost of six inertial sensors, and a feedback controller.
- The position stability of the payload is improved because it is now represented by the *open-loop* curves of Figure 19. There are also no issues about docking, since "turn-on transients" of the inertial feedback system are avoided. The feedforward system can remain on, and the payload docked (using products like TMC's AccuDock), with no problems.
- Since feedforward does not use any feedback, it is completely immune to resonances on the isolated payload.
- Using adaptive controllers, the amount of feedforward can be tuned to ensure at least a factor of ten reduction in the response of the payload to a disturbance (stage motion). This is comparable to what a well-tuned inertial feedback system can do.

5.7.2 Feedforward Cons

Despite being more robust, less expensive and easier to setup, there are still some disadvantages to the feedforward only option. Some of these are:

- As mentioned in Section 5.4.3, you are required to match the force capability of your disturbance (moving stage). The electromagnetic drivers which can do this are expensive, difficult to align, have a high power consumption, and have some stray magnetic fields which can cause problems in some applications.
- For moving $X-Y$ stages, the feedforward problem is *non-linear* due to twist couplings. For example, a payload will twist clockwise if there is an X-acceleration when the stage is in the full $-Y$ position, but counterclockwise when the stage is in the full $+Y$ position. Therefore there are feedforward terms proportional to $X\ddot{Y}$ and $Y\ddot{X}$. This suggests the use of a DSP-based controller.
- To keep the system running well, there should be a self-adaptive algorithm which keeps the gains properly adjusted. This is done by monitoring the motion of the payload and correlating it with the feedforward command inputs. This type of algorithm is *non-linear* and can be *unstable* under certain circumstances. In particular, with pure sinusoidal stage motion, stage accelerations become indistinguishable from payload tilting due to the shifting weight burden caused by the stage (the Principle of Equivalence again). Unable to make the right choice, the controller will always make the wrong choice.
- This method requires some work on the customer's or stage manufacturer's part to provide an appropriate set of command feedforward signals. These can be either analog or digital in form, but they must come from the stage motion controller.
- The isolation from floor vibration is no better than it is for a passive system (though, as mentioned, you may not need any improvement).

5.8 When Will You Need an Active System?

Determining your need for an active isolation system varies depending on whether you have a vibration or settling time critical application. Both can be difficult, and in either case you need to know something about your system's susceptibility to vibrational noise.

In vibration critical applications, it is insufficient to simply ask "does my system work?" If your system doesn't work with passive systems, or if the performance is inadequate, then you need to identify the source of the problem. For AFM/STM type applications, it may be obvious – the raw output of the stylus is dominated by a 1.5 Hz noise, and that is correlated with the payload motion, and you *know* your isolators have their resonance at that frequency. Other times it may be much less clear. For example, you may see a 20 Hz peak in your instrument, and that correlates with noise on the payload – but is it coming from the ground? Many HVAC systems in buildings use large fans which operate in this frequency range. If they do, they produce both acoustic noise and ground noise which are correlated with noise on the payload. So what is the source of the problem? Ground noise or acoustics? It can be impossible to tell. Keep in mind, however, that if your problem is at 20 Hz, most active systems won't help you (like the inertial feedback system) since they don't have any loop gain at that frequency. For an active system to make sense, you need to convince yourself that your problem comes from noise in the 0.5-10 Hz frequency range.

Settling time critical applications are more straightforward. To determine if you need an active system (which we assume to be feedforward only), there are three steps:

- **Step one**: Determine the *critical acceleration level* for your process (as discussed earlier in Section 5.5.2). A simple way to do this might be to move your stage and wait different amounts of time before making a measurement. If you know how long you need to wait and know the acceleration level of the payload after the stage stops, then you can derive this number. For a new instrument, the critical acceleration level can be very difficult to determine, and you might have to rely on calculations, modeling, and estimates.
- **Step two**: Estimate the initial acceleration level of the payload by multiplying your stage acceleration by the ratio of your stage mass to total isolated payload mass.
- **Step three**: Compare the numbers from steps one and two. If the critical acceleration level is above the initial payload reaction, then any TMC passive system should work for you. If it is below, then you need to compare the ratio of the initial to critical acceleration levels, and use Equation 17 to determine if the system can settle fast enough.
- If your allowed settling time is insufficient to get the attenuation you need, then you might want to try a system with higher passive damping. TMC's MaxDamp™ isolators have a decay rate up to five times faster than conventional pneumatic isolators (a Q-factor five times lower). This does sacrifice some vibration isolation, but is often a good tradeoff.
- If MaxDamp isolators won't work, then you will need an active system (passive isolation systems have run out of free parameters to solve the problem).

There are certain extreme examples which can determine your need very quickly. For example, if your critical level is below the initial payload acceleration and you want "zero" settling time, then you need an active system. However, if the ratio of the initial to critical level is more than 10 (with "zero" time), then you will either be forced to re-design your instrument or allow for a non-zero settling time. Active systems are *not* panaceas – they can't solve all problems.

5.9 General Considerations

If you are designing a new system, there are several general considerations which will make your system function optimally, whether it is active or not.

You should always use four isolators to support a system (rather than three), and they should be as widely separated as possible. This dramatically improves both the tilt stability and tilt damping in a system with only a marginal cost increase. It simplifies the design of the frame connecting the isolators, reduces the frame fabrication costs, gives better access to the components under the payload, and improves the overall stiffness of the system (assuming that your instrument has a square footprint).

You should use a center-of-mass aligned system whenever possible. This means putting the plane of the payload's center of gravity (CG) in the same plane as the moving stage's CG, and both of these should be aligned with the effective support point for the pneumatic isolators. This greatly reduces the pitch and roll of the payload with stage motions, and can reduce the cost of an active system by making it possible to use lower force capacity drivers in the vertical direction. Note that the "effective support point" for most isolators is slightly below the top of the isolator. Consult a TMC sales engineer for the exact location of this point for different isolation systems. A system's performance will also be improved by designing the payload such that the isolators support roughly equal loads.

The cost of the isolation system can be reduced by several means. The moving mass should be reduced as much as possible – this reduces the forces required to decelerate it, and thus reduces the cost of the magnetic actuators in the active system. You should also make the payload as rigid as possible to reduce the system's overall susceptibility to payload accelerations. Lastly, you can increase the static mass of the system, which will improve the ratio of static to active mass, and thus reduce the payload's reactions to stage motions.

It is quite possible that all of these steps, taken together, will allow you to avoid the use of an active system entirely.

5.10 Conclusions

The challenges created by Moore's Law* will require improved collaboration between systems engineers, integrators, stage manufacturers, and semiconductor tool manufacturers. There also needs to be a significant improvement in the awareness of the problem. This is simply a legacy of the by-gone days where "blind integration" of systems was sufficient. System engineers need to significantly shift their design goals for systems, since the conflict with high system throughputs and vibration isolation systems are fundamental, and active systems only improve the performance of systems by a certain factor. If the methods of design are not changed, then there may be a day in the not too distant future when even active systems won't work. Then you're really out of luck, since there *is no* next generation technology to turn to. Indeed, TMC already sees specifications which cannot be met even with the most optimistic assumptions about active system performance.

Active systems are *expensive*. The costs are driven by components like the magnetic or PZT actuators. Their prices are high because of the cost of their materials (rare earth NdFeB magnets or piezoelectric ceramics). The cost of power amplifiers, even when using the most mass-produced audio electronics components, can be expensive. When considering costs, it is important to realize that there is no such thing as an *incremental* active solution. The active system, if you need one, must match the forces generated by your stage motions. A system capable of less simply won't work.

TMC is striving to improve active isolation systems. Our goal is to make them more reliable, easier to install, maintain, and configure, and to make them self-configuring whenever possible. This will reduce system costs, engineering times, and speed the production of your systems.

TMC has a staff of sales engineers who can help you with any questions raised in this presentation, or to assist you in the design of an isolation system.

The Technical Background section of this catalog was prepared with assistance from Dr. Peter G. Nelson, Manager of Research and Development for TMC.

TMC isolators may be designed into a tool or applied as a point-of-use isolator.

*Gordon Moore, co-founder of Intel Corp., has pointed out that the density of semiconductors (in terms of transistors/area) has roughly doubled every 18 months, on average, since the very earliest days of commercial semiconductor manufacturing (even 1960 or earlier!).

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