Designing for Quiet, Vibration-Free Operation

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DESIGNING FOR QUIET, VIBRATION-FREE OPERATION

A common problem associated with many motion and power transmission systems is the resulting noise and vibration that prevails when all of the components are linked together. On their own, the various components that make up the system pose no noise or vibration problems. Once these components are integrated into the final product, however, their individual dynamic characteristics can interact, resulting in a wide range of vibration-related problems. These can range from excessive noise levels or objectionable sound signatures to inaccuracies in a high precision positioning system. And, in the worst cases, a product may experience premature component failure from excessive cyclical stresses induced by unchecked vibration.

By using a few basic engineering analysis and design tools, many of these problems can be avoided before the product goes into production. These solutions can be easily implemented if they are considered early enough in the product's design cycle, ideally at the beginning. This article examines a few real-world vibration problems and reviews the theory and practical considerations of the associated fixes.

A few common problems/solutions

- Problem: Vibration from a stepper motor transmits into the product enclosure (a thin, rigid panel structure) and becomes broadcast as sound radiation. Solution: Vibration isolation.

- Problem: A hard disc drive (HDD) mounted in a data storage array produces rotational vibration that is interrupting an adjoining HDD in the same array. Solution: Vibration damping.

- Problem: A stepper motor's impulsive rotational steps excite a gear train/motor resonance in a high precision rotational positioning system. The amplification and duration of the resonant oscillation reduced system performance and accuracy. Solution: A tuned damper attached to the motor's output shaft.

Vibration Isolation

When a vibration source such as a stepper motor is mounted to a solid structure, it is common for vibration to be transmitted from the motor to the connected structure. This vibration often radiates from the structure's surfaces as sound, which forms the basis for many noise problems in mechanical systems.

Incorporating a vibration isolation system into the motor mounting scheme often provides the most effective way to reduce transmitted vibration levels in a structure. By definition, an isolation system must allow relative motion between the vibration source and the supporting structure. This is typically accomplished with some type of resilient connection between the two. In a properly designed isolation system, this resilient connection supports the static loads generated by and imposed on the vibration source while filtering the dynamic forces generated by the source. In contrast, a poorly designed system can amplify transmitted vibration. This phenomenon can be explained by a graph of the transmissibility characteristics of a basic, single degree of freedom isolatio system. (See Figure 1).

An isolation system's effectiveness is typically measured as *transmissibility* (Tr)—the ratio of acceleration transmitted to the mounting structure to the acceleration present in the source. When $Tr > 1$, the system is amplifying vibration. When $Tr < 1$, the system provides isolation. All passive isolation systems exhibit these two distinct frequency regions of amplification and isolation. The objective in isolation system design is to make sure the source's problem frequency ends up far enough into the isolation region to

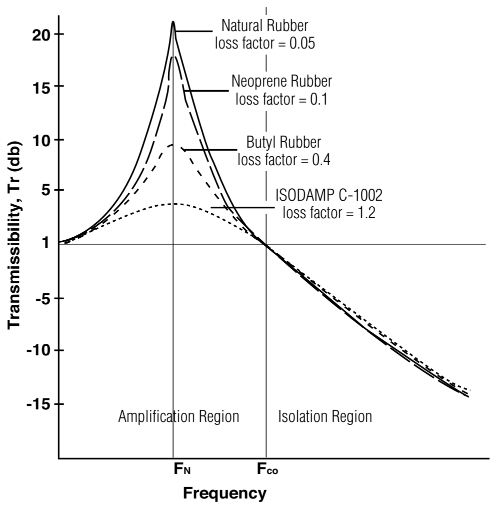


Figure 1

achieve the vibration reduction desired.

The point of peak transmissibility occurs at the isolation system's natural or *resonance frequency* (F_n). The location of this frequency within the spectrum is determined by the weight of the vibration source (W) and the isolation system stiffness (K_s) according to the following equation:

$$F_n=3.13\sqrt{\frac{\quad}{K_s}}$$

The amount of damping in the isolation system will determine the magnitude of peak transmissibility for the system. As damping increases, this peak value will decrease. For metallic spring-based mounting systems, peak transmissibility can reach 20 to 30. For lightly damped rubber systems, it can approach 10. With use of E-A-R's highly damped elastomers, this value can be minimized to 1.5. Elastomeric mount damping is often described in terms of a material loss factor (η^m), which is equivalent to twice the critical damping ratio. Using mounts with a high level of hysteretic damping can minimize amplification at resonance. This can prove especially important when a system cycles from high to low speeds during operation or generates impulsive type forces. (One should keep in mind that a stepper motor can exhibit a stepped impulsive forcing input, which can excite an isolation system's natural frequency.)

The point at which the amplification region meets the isolation region defines the system's *crossover frequency* (F_{co}), which typically occurs at 1.414(F_n). Frequencies above F_{co} will be isolated at a rate of about -12 dB per octave of increase in frequency for a given isolation system.

In designing an effective isolation system, proper mount selection will require a basic understanding of the components in the motion system (operational conditions and dimensional information) and access to mount manufacturer's performance data. Here is a list of the primary items required to specify an isolation mount:

1. Weight of motor or vibration source
2. Number of mounting points
3. Problem frequency to be isolated from mounting structure
4. Desired level of vibration reduction or transmissibility
5. Miscellaneous information on mounting scheme and orientation of source, e.g., type of mounting hardware, attachment point dimensions, available space, mounting plane orientation, service temperature range.

Since most commercial suppliers of isolation mounts supply stiffness data as a selection criteria, one must use the items listed above to determine the maximum allowable isolator stiffness. Once known, this value can be compared to the catalog values listed. Maximum stiffness thus can be calculated.

$$K=W\left[\qquad\qquad\right]$$ Where:

K = Maximum Mount Stiffness (lbs./in.)

$$\left(\frac{F}{3.13\sqrt{\frac{1}{TR}+1}}\right)^2$$

W = Weight (lbs.) per Isolator

F = Problem Frequency (Hz)

TR = Desired Transmissibility

As a side note,

$$TR = 1 - \left(\frac{\%\,\text{Isolation Efficiency}}{100}\right)$$

As an example, if a five-pound moto with four mounting points has a forcing frequency of 475 Hz and an isolation efficiency of 80 percent is desired, the above equation yields a maximum mount stiffness of 4,798 pounds-per-inch. The next step is to scan the available catalogs and find a mount of the right size and configuration that has a stiffness of this value or less. It should be noted that the softer the mount, the greater the dynamic and static deflections will be.

Static deflection can now be estimated by using Hooke's Law, $x = \frac{F}{K}$, where F is any static force (including the source weight), K is the system stiffness and x is the calculated static deflection. For the above example, the static deflection associated with the motor's weight is 0.001-inch.

Other considerations may apply to an isolation system, depending upon the nature of the problem. For example, in the case of mounting a computer hard disk drive in a desktop work station, it is important not only to isolate the higher frequency spindle and actuator vibrations but also to limit the lower frequency resonant deflections generated by actuator arm motion. E-A-R's ISODAMP highly damped elastomers minimize this resonant deflection yet allow the flexible isolator to provide the necessary level of isolation at higher frequencies. The same considerations would apply to a high-speed, position-critical stepper motor mounting system.

As a last note on isolation systems, it is very important to provide a high rigidity mounting point for isolators. Since the mounting structure's attachment points also exhibit flexibility, the isolationsystem is actually behaving like two springs connected in parallel. Since the system has been designed around the isolator's characteristics, it is best if at least the isolation mounts carry 90 percent of the total deflection. As a rule of thumb, mounting points should exhibit 10 times the isolator's stiffness.

Vibration Damping

Occasionally, incorporating an isolation system into a motion system's design proves not possible or practical. This usually occurs when the vibration source cannot be allowed to move relative to the mounting structure or foundation. When these situations arise, a structural damping treatment can often solve the problem.

Structural resonances are inherent in any system that possesses mass and stiffness. They will often amplify and efficiently conduct system vibration from one component to another if they are excited. If a vibration source produces vibration in the frequency range of a system resonance, it will often be the dominant component of troublesome vibration. By incorporating an effective damping system into the surrounding structure, the resonant buildup of energy can be dramatically reduced, thereby reducing the amount of vibration transmitted to other components or structures in the system.

All materials exhibit certain levels of damping described by a material loss factor (η_m). Most structural materials have so little internal damping, however, that their resonant behavior makes them effective in radiating noise and transmitting vibration. This applies to all solid, metallic materials, which exhibit material loss factors from 0.0001 to 0.002, and most rigid structural plastics, which exhibit material loss factors from 0.01 to 0.05. In contrast, damping materials exhibit material loss factors well in excess of 1.0.

When designing a structural damping treatment it is common to achieve a 10–fold increase in system loss factor. This increase in damping levels can provide a 20 dB reduction in resonant vibration and the estimated “large panel” radiated noise (which assumes that the wavelength of the bending wave is small relative to panel dimensions). Actual noise reductions will vary depending upon the dominance of this resonant noise within the overall system.

Another factor in assessing the effectiveness of a damping treatment involves comparing *driven* versus *resonant* vibration. Damping treatments rarely attenuate forced, non-resonant vibration. They are sometimes effective, though, since damping treatments add both mass and dynamic stiffness to a structure. For non-resonant forced vibration, adding mass or stiffness to a structure is the preferred method for reducing structural vibration levels.

Damping treatments typically consist of a viscoelastic polymer sheet with a high material loss factor, applied to a structural component. The two methods of passive structural damping are termed *extensional* and *constrained-layer* damping. Figure 2 illustrates the buildup of layers associated with each technique.

Extensional damping presents the simplest form of material application. The damping material is simply attached to the structure with a strong bonding agent. Alternatively, the material may be troweled onto the surface, or the structure may be dipped into a vat of liquefied material that hardens via a curing cycle. Once applied, the damping material dissipates vibration energy internally via extension and compression, which are induced by the vibratory flexing (bending

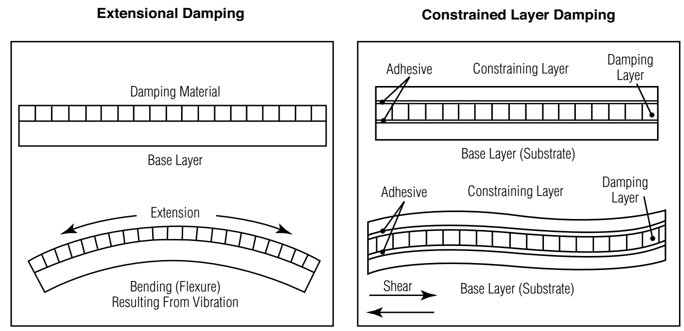


Figure 2

waves) of the structure.

Extensional damping performance increases with damping layer thickness. The damping material’s properties (Young’s Modulus and Material Loss Factor) also affects damping performance. System loss factor values are complicated to calculate, so the material’s manufacturer makes the best source of information on how a particular material and treatment thickness will affect a specific structure. A computer program typically is used to quickly calculate system loss factors and large panel noise reduction values for a variety of material and thickness options.

Figure 3 illustrates the effect of damping layer thickness relative to structural layer thickness on overall system loss factor. In addition, the figure indicates that damping performance is temperature-dependent and an important consideration for damping treatment design.

Stiffer structures usually require constrained-layer damping (CLD). With CLD, adding the damping material and a structural constraining layer to the base structural layer forms a “sandwich” buildup. When this system flexes during vibration, shear strain is induced in the damping material.

Due to the damping material’s properties, a portion of the strain energy carried by the damping layer dissipates internally as low-grade heat. For CLD treatments, the

Figure 3

method of attaching the layers does not matter as long as adequate surface contact and coupling occurs. The layers may be bolted, riveted or glued to obtain satisfactory performance. Adhesives should possess high shear stiffness, as softer adhesives will not adequately transfer shear strain to the middle damping layer.

Unlike extensional damping treatments, where a thicker treatment is always better, a thinner damping layer often provides better results in CLD systems. As with extensional systems, the damping material manufacturer can provide the best information on preferred installation and recommended buildup options for various damping material thicknesses and constraining layer materials and thicknesses. System loss factors for CLD systems are extremely cumbersome to calculate and require a specialized computer program along with extensive dynamic material property data. Thus it is important to work with materials that have been well quantified in terms of their dynamic properties. Since these properties are not typically found in common material databases, it often proves necessary to work with manufacturers that specialize in damping materials and have obtained this level of data on their materials exclusively for this purpose. E-A-R Speacialty Composites provides "reduced frequency nomograms" to characterize the material properties of its damping materials.

Figure 4 illustrates the effect of CLD on a variety of panel thickness combinations. For a given base structure thickness, the values obtained with CLD exceed those obtained with extensional damping (damping material properties and thickness are the same as in Figure 3). As a rule of thumb, if the constraining layer is to be the same material as the base structure, its thickness should be half the thickness of the base structure.

As a last note on structural damping treatments, the placement of the treatment is very important. Typically 100 percent coverage is not practical or possible. Therefore, it is important to place the damping treatment in an area that is exhibiting high flexural strain. Understanding where these areas are in complex structures will often require knowledge of mode shapes associated with troublesome frequencies. On a simple, less complex structures these areas can generally be determined via inspection.

Tuned Dampers

For position-critical equipment driven by stepper motors, such as scanners, sheet feeders, printers or robots, speed and accuracy gauge performance. But some inaccuracy- or instability-is inherent in such equipment. Because of inertia, theoretical seek-and-stop stepper motor functions are really seek-and-settle operations.

Unlike most electric motors, stepper motors rotate in finite steps, typically with 400 or 800 steps possible in one revolution. When a stepper motor surges through the right number of steps and tries to stop, inertia causes the driven component to overshoot its target. Resonant oscillation continues until motion

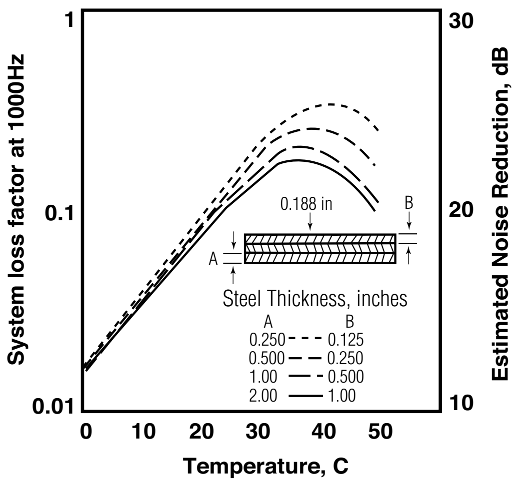


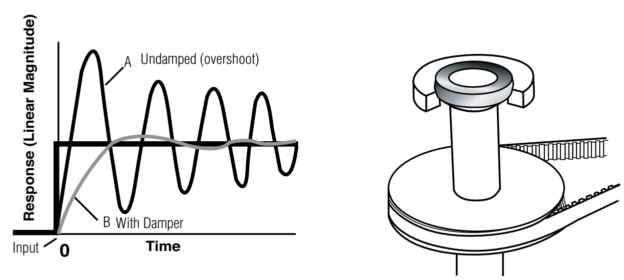
Figure 4

stepper motor surges through the right number of steps and tries to stop, inertia causes the driven component to overshoot its target. Resonant oscillation continues until motion stops or until the movements are small enough for the driven component to perform its function.

For these stepper motor applications, optimizing system performance means minimizing the magnitude and duration of thesettling oscillations without impeding normal operation. Tuned dampers that incorporate highly damped viscoelastic elastomers can provide a cost-effective solution to excessive settling time and system instability. As shown in Figure 5, an optimized tuned damper can reduce the time required for complete decay of oscillations.

In a rotational tuned damper design, an annular shaped mass ring is attached to an elastomeric coupling ring, which is attached to the drive shaft of the system. (See Figure 5.) It is important to use an elastomeric material with high internal damping. Both the mass and the elastomer ring move with the rotor. But inertia causes the mass to lag slightly behind the rotor motion. This small difference in rotational angle generates shear strain in the elastomer ring, which produces the necssary damping.

The effectiveness of a tuned damper lies in the characteristics of the damped elastomer ring and the amount of added mass. The added, or inertial, mass typically represents around 25 percent of the total effective rotating polar inertia before the addition of the damper. This value can be approximated for a system by adding up the polar moments of inertial mass for all of the driven rotational components, including the motor shaft assembly. Once the inertial mass is determined, the elastomer ring geometry must be calculated to provide the proper torsional stiffness. This will tune the coupled inertial mass resonance midway between the system resonance, with the inertial mass installed and locked in place, and the system resonance with no mass installed. This process can be iterative, requiring several rounds of prototyping.



Single degree of freedom system (transient response)

Turned damper, with ISODAMP C-1002, mounted on stepper motor output shaft

Figure 5

Design Tips

When possible, prototype a motion system with actual production components as early as possible in the product's design cycle. If production components differ greatly from those of the prototypes (in terms of specific mass, stiffness and damping), be aware that this will affect the final system.

- Think about potential vibration issues at the beginning. Most solutions are easy to implement if they are incorporated early in the design cycle.
- To obtain the most cost-effective solution, employ vibration isolation and damping treatments as close to the targeted vibration source as possible.
- Using E-A-R's highly damped mounting systems can protect a design from many of the common problems associated with lightly damped systems, e.g., excessive source motion due to impulse forcing, excessive sway under shock input, cycling through resonance as a motor changes speeds.
- When vibration isolation is not practical, implement a structural damping treatment to reduce vibration transmission from structure to structure.
- Apply damping treatments in areas of high strain energy and deflection.
- Work closely with material and component manufacturers to determine how environmental factors will affect the chosen treatment, e.g., temperature, corrosive solvents, off-gassing characteristics.
- A tuned damper can increase the performance of high-speed, position-critical motion systems.
- Active systems were not discussed in this article. They represent, however, an entirely different approach and range of solutions for vibration control. An active system can often solve a problem that cannot be solved using passive methods. The disadvantage for these systems is size and cost. Often passive and active systems are used together.

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