The Basics of Vibration Isolation Using Elastomeric Materials

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Introduction

When motorized equipment, such as electric motors, fans or pumps, is mounted to a solid structure, energy can be transferred from the equipment to the structure in the form of vibration. This vibration often radiates from the structure as audible noise and potentially reduces performance or damages equipment. Most portable electronics, CD drives and vehicle-mounted electronics are especially sensitive to vibration and shock and must be isolated from that energy to ensure proper performance.

Isolation mounts reduce the transmission of energy from one body to another by providing a resilient connection between them. Selecting an improper mount for an application, however, can actually make the problem worse. The incorrect mount may reduce the high frequency vibration, but resonant conditions at lower frequencies can actually amplify the induced vibration. During an impact, the mount deflects and returns some of the energy by rebounding. Preventing this energy return can extend product life and prevent performance problems such as skipping in a CD drive and read/write errors on a hard disk drive.

Adding *damping* to a resilient mount greatly improves its response. Damping reduces the amplitude of resonant vibration by converting a portion of the energy into low-grade heat. Damping also dissipates shock energy during an impact. This reduces the amount of deflection required to absorb the shock, providing protection in smaller spaces.

This property is especially significant when designing shock protection for portable electronics, which become increasingly "miniaturized" with each new model. A highly damped material can provide the required impact protection in a smaller envelope than would be required for an undamped material.

Natural Frequency

All mounting systems have a *natural frequency* (f_n)—the frequency at which the system will oscillate if it is displaced from its static position and released. For example, consider a weight suspended from a rubber band, similar to the single degree of system model in Figure 1. If the mass is pulled down from its resting state, stretching the rubber band, and then released, the mass will move up and down at a certain frequency. This is the natural frequency. The natural frequency, f_n, is dependent upon the stiffness of the spring, K, and the mass of the load that it is supporting (M), and can be determined by the following equations:

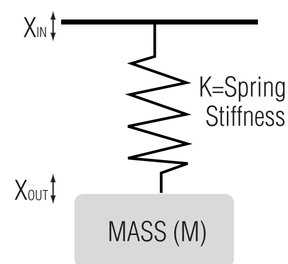


Figure 1

$$f_n={}^{1}/_{2}\pi\sqrt{{}^{K}/_{M}}$$

where K is the stiffness in newtons per meter (N/m) and M is the mass in kilograms (Kg), *or*

$$f_n=3.13\sqrt{{}^{K}/_{W}}$$

where K is the stiffness in pounds per inch (lb/in) and W is the weight of the mass in pounds.

The natural frequency may also be determined using the static deflection that the mass induces on the spring in the equation

$$f_n = \frac{1}{2}\pi\sqrt{G/d}$$

where G is the acceleration due to gravity (9.8 m/s^2 or 32.2 ft/s^2) and d is the static displacement in meters or inches. This equation is true only for an undamped system.

Damping

Controlling the natural frequency provides one means to control vibration. *Damping* provides another. Damping is the dissipation of energy, usually by releasing it in the form of low-grade heat. For example, dry friction, the most common damping mechanism, is the reason an object sliding on a surface will slow down and stop. Some mechanical devices use *viscous damping* as a means of energy dissipation. In these systems, fluid losses caused by a liquid being forced through a small opening provide the necessary energy loss. The shock absorbers on an automobile are an example of viscous dampers. Mathematical models for viscous damping are well established and provide a means for analysis. Viscous damping capability is characterized by the damping ratio, C/C_c or ζ.

Most elastomeric engineering materials for vibration isolation use a mechanism known as *hysteretic damping* to dissipate energy. When these materials are deformed, internal friction causes high energy losses to occur. The *loss factor* is used to quantify the level of hysteretic damping of a material. The loss factor (η) is the ratio of energy dissipated from the system to the energy stored in the system for every oscillation. It is often useful to relate the loss factor to the damping ratio so viscous damping models can be used for analysis. The damping ratio can be approximated from the loss factor by the following formula, which is more accurate at lower damping levels than at higher ones.

$$\eta \approx 2C/C_c \approx 2\zeta$$

A loss factor of 0.1 is generally considered a minimum value for significant damping. Compared to this value, most commonly used materials, such as steel, aluminum and most rubbers, do not have a high level of damping. Other specialized materials can have very high damping. Here are some materials and their approximate loss factor.

Material	Approximate Loss Factor
Aluminum	.007-.005
Steel	.05-.10
Neoprene	.1
Butyl Rubber	.4
ISODAMP® C-1002 thermoplastic	1.0

Vibration Isolation

The performance of an isolation system is determined by the *transmissibility* of the system—the ratio of the energy going into the system to the energy coming from the system. This can be expressed in terms of acceleration, force or vibration amplitude. Transmissibility (T) is equal to

$$T = \left| A_{out}/A_{in} \right| = \sqrt{\frac{1+\left(2\zeta f_d/f_n\right)^2}{\left[1-\left(f_d/f_n\right)^2\right]^2+\left[2\zeta f_d/f_n\right]^2}}$$

Where:
- T = Transmissibility
- A_{out} = Energy out of system (transmitted force)
- A_{in} = Energy into system (Disturbing force)
- ζ = Damping ratio
- f_d = Driving frequency
- f_n = Natural frequency

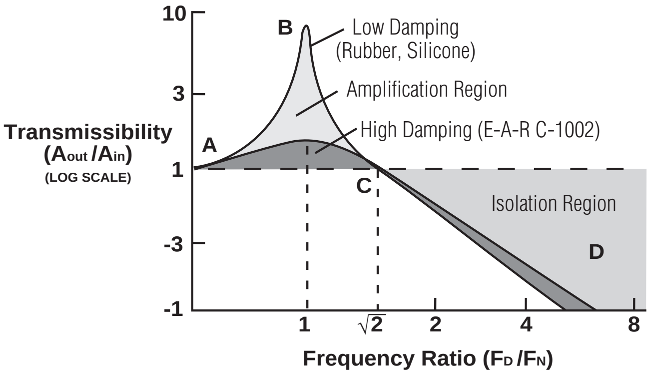


Figure 2

Figure 2 shows two typical transmissibility curves, one for a highly damped material (ζ≈0.5), another for a material with much lower damping (ζ≈ 0.05)

At very low frequencies ($f_d/f_n << 1$), the input vibration virtually equals output (the transmissibility is equal to 1), and input displacement essentially equals that of output.

If the driving frequency equals the natural frequency ($f_d/f_n = 1$), the system operates at resonance. If damping is ignored in the equation for transmissibility that was given earlier, a system that is operating at resonance will have a transmissibility approaching infinity. As damping increases, the transmissibility at resonance decreases. Figure 3 shows the relationship between peak natural frequency at resonance and loss factor. Of course, all real world systems have some level of inherent damping, but this demonstrates the important role that damping can play in vibration isolation. When a vibration isolation mount with very little damping is used at or near resonance, the energy amplification can create many problems, ranging from a simple increase in noise levels to catastrophic damage to mechanical equipment.

When the frequency ratio equals the square root of two ($f_d/f_n = \sqrt{2}$), transmissibility will once again drop to 1. This is known as the *crossover frequency*, and the area below this frequency is known as the *amplification region*. Above this frequency lies the isolation region, where transmissibility is less than 1. As a goal, the isolator designer tries to design a mounting system that puts the primary operating frequencies of the system in the isolation region. Many systems must operate at a number of primary frequencies or must frequently go through a startup or slowdown as part of the operation cycle. For these systems, damping in the mount becomes increasingly important when it must function at or near resonance.

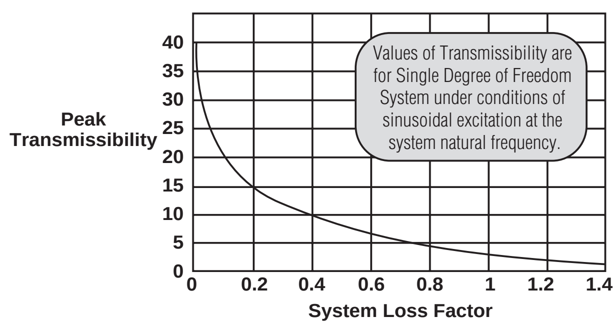


Figure 3

As frequency continues to increase above the crossover frequency, the level of isolation, or the *isolation efficiency,* increases. Figure 4 shows this relationship. Designers must know the isolation efficiency of the mounting system when transferred energy must be below a specified level, in devices such as CD-ROM or hard disk drives.

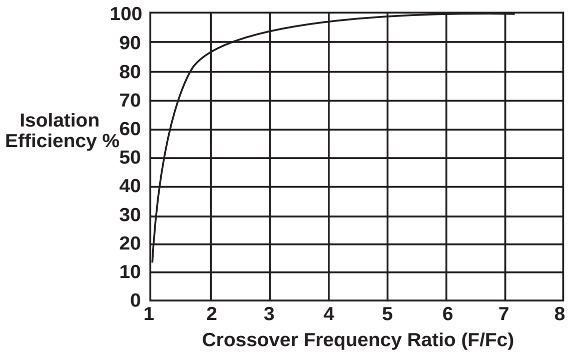


Figure 4

Mount Design Using Elastomeric Materials

There are many material options for producing resilient elastomeric mounts. Thermoplastic materials (ones that can be melted and formed), such as many vinyl and rubber elastomers, can be injection-molded into cost efficient and detailed parts. Thermosets (materials that react in the mold and cannot be remelted) offer another option, and must be molded through other methods, commonly compression or transfer molding. Due to longer cycle times, thermoset piece prices often exceed that of thermoplastics, but the materials can also offer chemical and strength properties that can not be met by thermoplastics. Damping and stiffness can also vary greatly with different materials. (E-A-R Specialty Composites manufactures several isolation materials with a variety of stiffnesses and damping properties. Information on these materials can be found on the company's Website or by contacting E-A-R. See back page.)

The following section outlines several key points to be considered when designing isolation systems.

Design Guide

Here are guidelines that will assist in the design of axially loaded isolation mounts and pads from sheet materials.

1. **Optimize load**. Proper performance depends on proper loading. Referring to the natural frequency equation, $f_n = 3.13\sqrt{K/W}$, if the mass of the load is very small for the stiffness of the selected mount, the natural frequency of the system will be high, reducing the isolation performance. An overloaded mount, can compress completely, or *bottom out*, increasing the effective stiffness of the mount. This also increases the natural frequency. Overloading an elastomeric mount can also cause internal stresses that can reduce the useful life of the mount. Generally, a 5% static compression of the mount is appropriate for most materials, although static compressions of up to 15% may provide adequate isolation and part life. For homogeneous elastomers with a durometer (hardness) of around 50-60 shore A, ideal loading is generally around 50 pounds per square inch (psi), although loading of anywhere from 10 - 100 psi may still be effective. Softer elastomers should be loaded less than stiffer elastomers.

2. **Shape factor (S) of 0.5 to 1.0.** Solid elastomers act as incompressible solids, and therefore must have room to bulge in order to deflect. Therefore, the *shape factor*, or bulge factor, should be optimized to achieve the expected stiffness. Shape factor (S) is defined as

$$S = \frac{\text{Area under load.}}{\text{Area free to bulge}}$$

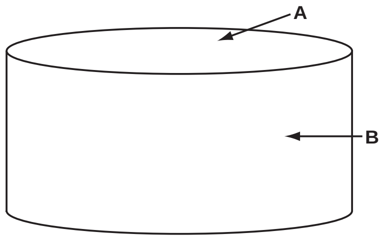


Figure 5

Example: Consider a mount of the shape in Figure 5. Surface A, the loaded surface, has an area of 2 square inches. Surface B, free to bulge on all sides of the mount, has a total area of 3 square inches. The shape factor is S = 2/3 = 0.66. A high shape factor produces a stiff mount. With a low shape factor, the mount may buckle and be unstable. A shape factor of 0.5 to 1.0 proves appropriate for most materials. Changing the thickness of the mount or changing the cross section of the mount changes the shape factor. Rings, strips, or other shapes can be useful in creating the proper shape factor.

3. **Determine the dynamic modulus of the elastomeric (E).**

The dynamic modulus of the material can be determined using a *reduced frequency nomogram*. The dynamic modulus of a highly damped material will be affected by temperature and frequency. A nomogram can provide the dynamic modulus and loss factor information over a range of temperatures and frequencies. It may be useful to convert dynes/cm^2. To convert to psi, multiply by 1.45 x 10^{-5}. To convert to N/m^2, multiply by 0.10).

Calculate the effect of the shape factor using the following equations.

Disk Shape	Block Shape
$E_{Corrected} = E\ (1+2S^2)$	$E_{Corrected} = \frac{4}{3}\ E\ (1+S^2)$

4. **Calculate stiffness (K)**.

Calculate stiffness using the following formulas.

Disk

$$K = \frac{E_{Corrected}\ \pi a^2}{t}$$

Ring

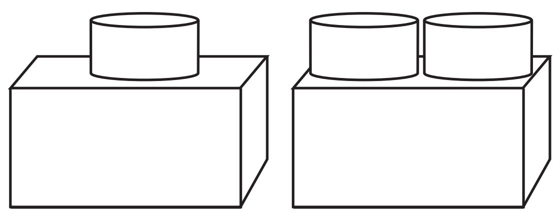
$$K = \frac{E_{Corrected}\ \pi(a_0^2 - a_i^2)}{t}$$

Block

$$K = \frac{E_{Corrected}\ lw}{t}$$

a= Disk radius
a_o= Outer ring radius
l= Block length
t= Thickness
a_i= Inner ring radius
w= Block width

Combine shapes to determine the stiffness of complex parts. Many parts can be considered a combination of two blocks, a block and a disk, or any other combination of geometry. Determine the stiffness of each section as outlined in 4a. Then determine if the parts are in series or parallel.



Example A
Block and disk in series

Example B
Disks in parallel
Block and disks in series

Figure 6

Figure 6 shows some examples. Example A shows a block in series with a disk (they are stacked on top of each other). Example B shows two disks in parallel (they are next to each other). These two disks are also in series with the block. To determine to overall stiffness, use the equations below to combine the stiffnesses of the individual shapes.

Shapes in series:

$$1/K_{Overall}=1/K_1+1/K_2+1/K_3+\ldots+1/K_n$$

Shapes in parallel:

$$K_{Overall}=K_1+K_2+K_3+\ldots+K_n$$

Assume the blocks in Figure 6 have a stiffness of 20 lb/in and the disks have a stiffness of 10 lb/in. The total stiffness of Example A would be

$$1/K_A=1/K_{Block}+1/K_{Disk}=1/20+1/10=3/20$$

$$K_A=20/3=6.66\ lb/in$$

The total stiffness of Example B would be

Stiffness of disks in parallel,

$$K_{Disks}=K_{Disk\ 1}+K_{Disk\ 2}=10+10=20\ lb/in.$$

Stiffness of block and disks in series,

$$1/K_B=1/K_{Block}+1/K_{Disks}=1/20+1/20=1/10$$

$$K_B=10\ lb/in$$

5. **Calculate the natural frequency.** Use the following relationship. W is the overall weight load in pounds. M is the overall mass load in kilograms.

English units: $f_n=3.13\sqrt{K_{overall}/W_{overall}}$
Metric units: $f_n=.16\sqrt{K_{overall}/M_{overall}}$

Remember, vibration isolation in the system will occur above $\sqrt{2}\ f_n$. Most systems have a certain frequency of concern from which they must be isolated. This may be the rotational speed of a motor, the blade passing frequency of a fan, and so forth. As a rule of thumb, the natural frequency of the mounting system should be one-third of the frequency of concern.

E-A-R Specialty Composites

E-A-R Specialty Composites offers a wide range of standard molded grommets, bushings and other isolators molded from ISODAMP C-1000 Series vinyl thermoplastic, ISOLOSS® HD urethane, and VersaDamp™ TPE. These have been designed with the appropriate geometry and load specifications in mind, and have been used in many kinds of products. Please consult E-A-R's "Designing with Isolators" booklet for more information, including a worksheet to determine the natural frequency when using the molded parts.

E-A-R also manufactures several other propriety materials that can be used to solve various vibration and shock isolation problems. These include

ISOLOSS VL Low Modulus Urethane Elastomer

ISOLOSS LS High Density Urethane Foams

CONFOR® Ergonomic Urethane Foams

Acknowledgements

Gardner, Ross. "The Reduction of High Frequency Vibration and Noise by Using E-A-R ISODAMP® Isolation Materials." Indianapolis: E-A-R Division, Cabot Corporation, 1976.

Inmann Daniel. *Engineering Vibration*. Englewood Cliffs, N.J.: Prentice-Hall, Inc., 1994.

Lord, Harold, William Gatley and Harold Evensen. *Noise Control for Engineers*. New York: McGraw Hill Book Company, 1980.

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