

Design of Passive Damping Systems

Conor D. Johnson
President
CSA Engineering, Inc.
2850 West Bayshore Road
Palo Alto, CA 94303-3843

This paper presents a brief review of techniques for designed-in passive damping for vibration control. Designed-in passive damping for structures is usually based on one of four damping technologies: viscoelastic materials, viscous fluids, magnetics, or passive piezoelectrics. These methods are discussed and compared. The technology of using viscoelastic materials for passive damping is discussed in more detail than the other methods since it is presently the most widely used type of damping technology. Testing and characterization of viscoelastic materials and design methods for passive damping are discussed. An example showing the benefits of a passive damping treatment applied to a stiffened panel under an acoustic load is presented.

1 Introduction

Vibration and noise suppression are becoming more important in our society. Noise suppression of office machines, home appliances, aircraft, and automobiles makes for a more pleasant environment. Vibration suppression allows more precise medical instruments, faster and more compact disk drives, more precise images in ground- and space-based telescopes, safer buildings in the event of an earthquake, and lower stresses in products, generally leading to longer life and lighter weight. Passive damping is now the major means of suppressing unwanted vibrations. The primary effect of increased damping in a structure is a reduction of vibration amplitudes at resonances, with corresponding decreases in stresses, displacements, fatigue, and sound radiation. However, damping is one of the more difficult issues to deal with in structural dynamics.

Passive damping may be broken into two classes: inherent and designed-in. Inherent damping is damping that exists in a structure due to friction in joints, material damping, rubbing of cables, etc. The level of inherent damping in a structure is usually less than 2 per-

cent structural.¹ Designed-in damping refers to passive damping that is added to a structure by design. This damping supplements inherent damping, and it can increase the passive damping of a structure by substantial amounts.

To achieve a substantial increase in passive damping, a structural dynamist must have a good working knowledge of many factors: passive damping technologies, materials, concepts and implementations, in addition to design, analysis and predictions methods for passive damping systems. This paper will discuss each of these factors as related to the design of passive damping systems.

Much work has been done in the area of passive damping and this paper will not attempt to cite the many contributions made by a host of qualified individuals and companies. However, it is hoped that this paper will give the dynamist a basic understanding of passive damping technology and encourage him or her

¹Structural damping g is equal to twice the viscous damping ratio ζ or the inverse of the quality factor, Q , in the case of harmonic excitation.

	TYPE OF DAMPING MECHANISM			
	Viscoelastic Materials	Viscous Devices	Magnetic Devices	Passive Piezoelectrics
Types of Treatments	All	Struts & TMDs	Struts & TMDs	Strut Dampers
Temperature Sensitivity	High	Moderate	Low	Low
Temperature Range	Moderate	Moderate	Wide	Wide
Loss Factor	Moderate	High	Low	Low
Frequency Range	Wide	Moderate	Moderate	Moderate
Weight	Low	Moderate	High	Moderate

Table 1: Primary passive damping mechanisms and related information

to use passive damping as one more design option in seeking solutions to structural dynamic problems.

In the last ten years, there have been many papers published in the area of passive damping. The heaviest concentration has been in the “Passive Damping” conferences beginning in 1984. Since 1994, these conferences have become a conference within the annual *North American Smart Structures and Materials Conference*. The reader is directed to the proceedings of these conferences (Air Force, 1984, 1986, 1989, 1991, 1993); (Johnson, 1994) for many examples of passive damping applications.

2 Passive Damping Mechanisms

Designed-in passive damping for structures is usually based on one of four damping mechanisms: viscoelastic materials, viscous fluids, magnetics, or passive piezoelectrics. Each of these damping mechanisms must be understood in order to select the most appropriate type of damping treatment. The sections below describe each mechanism, and Table 1 presents a comparison.

The author believes that approximately 85 percent of the passive damping treatments in actual applications are based on viscoelastic materials, with viscous devices the second most actively used (the use of viscous devices is greater for isolation and shock). This paper will therefore concentrate on passive damping design using viscoelastic materials. Damping using viscous and magnetic technology usually requires either that the devices be purchased for the application or a large effort be spent in the mechanical design. This paper will therefore discuss only the applications of such devices, not their internal design. Passive piezoelectrics will be discussed in broad terms since this technology has only found limited applications to date.

2.1 Viscoelastic Materials

Viscoelastic materials (VEMs) are widely used for passive damping in both commercial and aerospace applications. Viscoelastic materials are elastomeric ma-

terials whose long-chain molecules cause them to convert mechanical energy into heat when they are deformed. For a detailed discussion of viscoelastic materials, see Aklonis and MacKnight (1983) or Ferry (1980). VEM properties are commonly described in terms of a frequency- and temperature-dependent complex modulus (G^*). Complex arithmetic provides a convenient means for keeping track of the phase angle by which an imposed cyclic stress leads the resulting cyclic strain. The complex shear modulus is usually expressed in the form

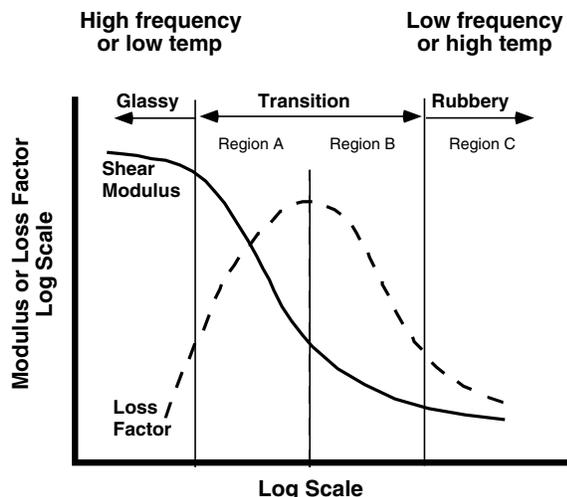
$$G^*(\omega, T) = G_0(\omega, T)[1 + i\eta(\omega, T)] \quad (1)$$

The real and imaginary parts of the modulus, which are commonly called the storage or shear modulus and loss modulus, are given by $G_0(\omega, T) = G_R$ and $G_0(\omega, T)\eta(\omega, T) = G_I$, respectively. The loss factor (η) is a measure of the energy dissipation capacity of the material, and the storage modulus is a measure of the stiffness of the material. The shear modulus is important in determining how much energy gets into the viscoelastic material in a design, and the loss factor determines how much energy is dissipated. Although both are temperature and frequency dependent, temperature has a greater effect on damping performance in typical applications. Figure 1 shows the typical variation of material properties as a function of temperature and frequency. This figure also shows the optimum regions of VEMs for various types of damping devices. Region A is optimum for free-layer treatments, having a high modulus and high loss factor. Region B is optimum for constrained-layer treatments, having a low modulus and high loss factor. Region C is optimum for tuned-mass dampers, having a low modulus and low loss factor, while regions A and B are optimum for most other types of discrete dampers.

In order to design accurately passive damping systems using VEMs, one must know their material properties accurately. Since viscoelastic materials are temperature and frequency dependent, they must be tested over both temperature and frequency ranges to characterize the material. VEM test methods fall into two broad classes: resonant and nonresonant.

Resonant tests infer VEM properties from measured normal mode properties of some simple structure that includes the viscoelastic material, such as a sandwich beam. Resonant tests have the advantage of being relatively insensitive to both gain and phase errors in the transducing systems. However, a major disadvantage is that the measurement is indirect; material properties are inferred from modal properties by working backwards through some theoretical solution. Also, material properties are obtained only at discrete frequencies.

Nonresonant tests, often called complex stiffness



Increasing temperature at constant frequency →
 or
 Decreasing frequency at constant temperature →

Figure 1: Temperature- and frequency-dependence of VEMs

tests, utilize a VEM sample connected to a rigid fixture and loaded dynamically, usually in shear. The force transmitted through the specimen and the resulting deformation across it are transduced directly. Damping is determined from the phase angle by which the displacement lags the force. Stiffness, or storage modulus, is determined from the ratio of in-phase force to displacement. Stiffness and loss factor are obtained as almost continuous functions of frequency and at discrete temperatures.

It was determined by Williams et al. (1955) and Ferry et al. (1952) that viscoelastic material test data could be shifted in temperature and frequency such that a relationship could be developed that characterizes the material at all combinations of temperature and frequency. This process is referred to as characterization and much work has been performed in this area (see, for example, Fowler, 1989 and Rogers, 1989). A layman's view is to determine a functional relationship (the temperature shift function α_T) between temperature and frequency such that both the storage modulus and loss factor at any temperature and frequency can be determined. Incorrect characterization processes can lead to major errors in property data. The end result of characterization is a viscoelastic material nomogram (also called the international plot (Jones, 1977)). Figure 2 gives an example and its use for Soundcoat Dyad 601. These types of plots are the preferred method of presenting VEM data, and most damping material manufacturers present their material data in this form. It

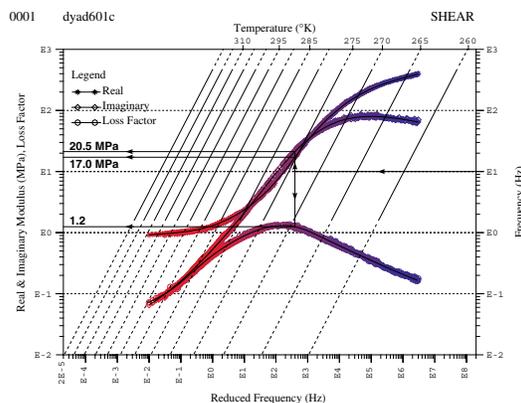


Figure 2: Reduced-temperature nomogram

is therefore important that designers know how to use these plots.

To get modulus and loss factor values corresponding to 10 Hz and 273°K, for example, read the 10 Hz frequency on the right-hand scale and proceed horizontally to the 273°K temperature line. Then proceed vertically to intersect the curves along a line of reduced frequency. Finally, proceed horizontally from these intersections to the left-hand scale to read the values of 20.5 MPa for the real (shear) modulus, 17 MPa for the imaginary modulus, and 1.20 for the loss factor.

Once a material has been characterized accurately, its parameters may be placed into a database. The designer may then perform searches for materials that meet specific engineering criteria in a method exactly analogous to reading the international plot. Since these searches are based solely on the characterization parameters, the importance of quality data and characterization methods should not be underestimated.

2.2 Viscous Devices

These devices dissipate energy via a true velocity-dependent mechanism, typically by forcing a fluid through a precision orifice. Although the actual viscous damping coefficient is usually not frequency dependent, the viscous damping force under periodic loading ($c\omega$) is obviously frequency dependent. Viscous dampers are most effective for axial deformations. The levels of loss obtainable by a viscous device are higher than those obtainable with VEM-based struts, but a price is paid in the “bandwidth” of effectiveness. That is, a viscous damper is usually effective at damping only modes in a relatively narrow frequency range because the damper is usually “tuned” to a frequency range. As with VEM damping treatments, the effectiveness of viscous dampers is affected by changes in temperature, but

to a lesser degree. This change is due to the viscosity of the fluid changing.

Viscous damping mechanisms have been adapted to address bending deformations, but this is not the most direct or efficient use of the technology. This approach is thus not attractive for situations dominated by panel bending, such as acoustics-driven problems.

2.3 Magnetic Devices

With advancements in the production of powerful magnets, magnetic (eddy current) damping is proving to be a viable solution to problems where temperature extremes are a factor. The power and effectiveness of the magnetics are relatively unaffected by changes in temperatures. As with fluid-based systems, this technology produces a true, velocity-dependent viscous damping force. However, the damping coefficients of magnetic devices are usually less than viscous devices per unit weight.

This is another technology that is not well suited for most bending problems. However, magnetic tuned-mass dampers (TMDs) have been shown to be effective in harsh environments where neither viscoelastic or viscous damping mechanisms are possible.

2.4 Passive Piezoelectrics

Piezoelectric ceramic materials have the unique ability to produce a strain when subjected to an electrical charge, and, conversely, they produce a charge when strained mechanically. This property has made them popular as actuators and sensors in active vibration control systems. This dual transformation ability also makes them useful as passive structural dampers (Forward, 1979). In passive energy dissipation applications, the electrodes of the piezoelectric are shunted with a passive electric circuit. The electrical circuit is designed to dissipate the electrical energy that has been converted from mechanical energy by the piezoelectric. Two major types of shunted circuits exist: a resistor alone and a resistor in series with an inductor. Other circuits can be visualized and have been reported elsewhere (Hagood and von Flotow, 1991).

A resistor shunt provides a means of energy dissipation on the electrical side and thus increases the total piezoelectric loss factor above that of the unshunted piezoelectric. With a shunted resistor, the ceramic behaves like a standard first-order viscoelastic material. The material properties of the resistive shunted piezoelectric can be represented as a complex modulus as is typically done for viscoelastic materials, $\bar{E}^{eff}(\omega) = \bar{E}(\omega)(1 + i\eta(\omega))$, where \bar{E} is the ratio of shunted stiffness to open circuit stiffness of the piezoelectric and η is the material loss factor. The nondi-

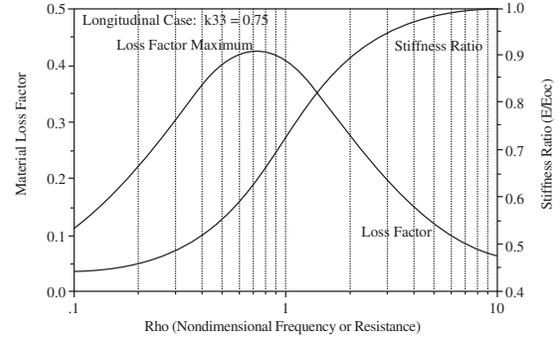


Figure 3: Effective material properties of a resistively shunted piezoelectric assuming strain in the polarization (longitudinal) direction

mensional expressions for η and \bar{E} are

$$\eta(\omega) = \frac{\rho k^2}{(1 - k^2) + \rho^2}, \quad \bar{E} = 1 - \frac{k^2}{1 + \rho^2}, \quad \rho = RC^s \omega, \quad (2)$$

where ρ is the nondimensional frequency ratio, k is the electromechanical coupling coefficient, R is the shunting resistance, and C^s is the clamped piezoelectric capacitance. These relations have been plotted versus ρ , the nondimensional frequency (or the nondimensional resistance) in Figure 3 for a typical value of k for longitudinal strain. As illustrated in the figure, for a given resistance the modulus of the piezoelectric changes from its short circuit value at low frequencies (about that of aluminum) to its open-circuit value at high frequencies. The transition occurs at the frequency RC^s . The material loss factor peaks at this transition frequency at a value of 44% for longitudinal or shear strain and 8% for transverse use. The point of maximum loss factor can be assigned to the desired frequency by the appropriate choice of resistor.

While the loss factor levels are not as high as those for viscoelastic materials, the high stiffness of the shunted piezoelectric materials (typically a ceramic) allows them to store many times the strain energy of a viscoelastic for a given strain. The piezoelectric material properties are also relatively temperature independent. The coupling coefficient for several common ceramic compositions vary by only $\pm 10\%$ over a temperature range from $-200^\circ C$ to $+200^\circ C$. The piezoelectric material density ($\sim 7500 \text{ kg/m}^3$) is much higher than that of viscoelastic materials, however.

Shunting with a resistor and inductor, along with the inherent capacitance of the piezoceramic, creates a resonant LRC circuit that is analogous to a mechanical tuned-mass damper, except that it counters vibrational strain energy instead of kinetic energy. High loss fac-

tors are possible. However, heavy shunt inductors are required for typically sized piezoceramics. More compact active inductors have been built, but this defeats the passivity of the system.

3 Passive Damping Concepts

Although passive damping is often attributed to friction or other such “accidental” mechanisms, designed-in damping using high-loss materials and techniques can yield energy dissipation that is orders of magnitude higher and much more predictable. All passive damping treatments share a common goal: absorb significant amounts of strain energy in the modes of interest and dissipate this energy through some energy-dissipation mechanism. The effectiveness of all passive damping methods varies with frequency and temperature, though some more than others. For each of the basic passive damping *mechanisms*, there are several choices for implementation which can be divided into two major categories: discrete and distributed. Table 2 summarizes the primary passive damping concepts along with their typical uses. For a description of many types of damping devices, see Nashif, Jones, and Henderson (1985).

Discrete dampers can be very effective and may be easy to design and implement. Damped struts or links are commonly used in truss structures, though they can also be used to damp structures where two or more parts of the structure are moving relative to each other. Depending on the design constraints, the damping material may be in series or parallel with other structural members. Because of creep problems, one should not require damping materials to carry high static loads, but should provide alternate static load paths. A tuned-mass damper (TMD) is a discrete damping device attached to the structure at or near an antinode of a troublesome mode of vibration. These devices transfer energy at a particular resonance to two new system resonances, each highly damped. TMDs are in general the most weight-efficient damping devices for single mode damping.

One of the simplest passive damping methods, but the least effective, is the unconstrained or free-layer damping treatment. In this treatment, a high-modulus, high-loss-factor material is applied to a surface of the vibrating structure. Free layer treatments must be fairly thick in order to absorb sufficient amounts of strain energy, and are therefore not weight efficient. Constrained layer treatments are surface treatments where the damping material is sandwiched between the base structure and a constraining layer. The constraining layer causes shear in the damping material as the structure deforms. This type of damping treatment is most

commonly used to damp bending modes of surfaces (shell-type modes). A small increase in damping may be achieved by placing damping materials in joints. The advantage of this type of damping is that it requires very little added weight. Embedded dampers have damping material embedded into a structural member (mainly composites) during manufacture.

4 Passive Damping Design Methods

The frequency and temperature dependencies of passive damping mechanisms must be taken into account during the design. Damping design is not just the selection of a high loss mechanism (material, device) for the temperature range of interest; it is an integrated structural and materials design process. To achieve damping, two conditions must be met: significant strain energy must be directed into the high loss mechanism for all modes of interest, and the energy in the mechanism must be dissipated. The first condition requires most of the design effort and is dependent on structural properties, location, mode shapes, stiffness, wave lengths, thickness of material, etc. The second condition is met by selecting the mechanism with the proper loss factor that matches the designed stiffness.

Before the design of the passive damping treatment can begin, it is imperative that the true nature of the vibration problem be understood thoroughly. The designer must have in mind some figure of merit, which could be as simple as the response of a fundamental mode of a panel or as complicated as the RMS beam jitter of multiple optics in an optical system due to acoustic excitation. In any case, the engineer must determine whether the problem is a single mode or many modes over a broad frequency band. In the later case, the precise modes that are driving the figure of merit must be identified. Knowing all of this, the proper damping mechanism, analysis technique, and hardware can be chosen.

Passive damping treatments for complex structures are usually designed using finite element techniques. Methods for finite element analysis of damped structures can be classified as response-based or mode-based. Response-based methods use the bottomline dynamic response (e.g., RMS acceleration) to guide the design. Mode-based methods use a substitute metric which is easier to compute but which is known to influence the bottom line response significantly. Designs produced by mode-based methods should normally be verified by a final dynamic response computation.

The best known mode-based methods are modal strain energy (MSE) (Johnson and Kienholz, 1982) and complex eigenvalue analysis. Using the MSE method, the modal damping of a structure may be approximated

TYPE OF TREATMENT						
	Strut / Link Dampers	Constrained Layers	Tuned-Mass Dampers	Joint/Interface Dampers	Embedded Dampers	Unconstrained or Free Layers
Target Modes	Global	Member Bending and Extension	Narrow Frequency Range, Any Mode Shape	Local or Global	Member Bending and Extension	Member Extension and Bending
Primary Design Method	Modal Strain Energy	Modal Strain Energy	Complex Eigenvalues	Joint Test or Modal Strain Energy	Modal Strain Energy	Hand Calculations Modal Strain Energy
Special Features	Removable, Lightweight	Flexible, Wide Bandwidth	Low Cost, Low Weight	Low Weight Low Volume	Embedded, Low Outgassing	Low Cost Easy Design

Table 2: Primary implementations of passive damping and associated design methods for structures

by the sum of the products of the loss factor of each material and the fraction of strain energy in that material for each mode. In the case of a multimaterial system, the system loss is given by

$$\eta^{(r)} = \sum_{j=1}^M \eta_j \frac{SE_j^{(r)}}{SE^{(r)}}, \quad (3)$$

where η_j = material loss factor for material j , $SE_j^{(r)}$ = strain energy in material j when the structure deforms in natural vibration mode r , and $SE^{(r)}$ = total strain energy in natural vibration mode r . In the MSE method, the material properties are real and a real eigenvalue analysis is performed. The underlying assumption of the MSE method is that the real eigenvalues are a good approximation to the complex eigenvalues. For high damping, this is not a good approximation. The more time-consuming complex eigenvalue method provides direct computations of $\eta^{(r)}$. The computational advantage of the MSE method is not as important as it once was, because computers are so much faster. However, MSE distributions are also valuable for deciding where to place damping materials or devices, determining optimum design parameters, or for general understanding of the character of a structure's modes. In both the MSE and complex eigenvalue method, the analyses are performed with constant material properties.

Response prediction can be carried out in either the time or frequency domain. Modal superposition may be used to advantage in either case, but in frequency response, one pays a price in addition to the approximation due to modal truncation: there is no way to account for frequency-dependent materials. This is because modes must be computed using properties corresponding to a single selected frequency. A direct frequency response formulation (no modal superposition) can account for this dependency but at substantial computational cost.

For each of the analysis and prediction methods discussed above, the damping device or material must be represented in the analytical model. This may be as simple as representing a damped strut with a spring element, or as complex as modeling an embedded treatment in very precise detail with many finite elements. For a viscoelastic material damping treatment, the shear deformation of the VEM must be accurately captured and this is best done by using solid elements to represent the VEM (see, for example, Johnson et al., 1985). Large aspect ratios of the elements modeling the VEM can be accommodated in many finite element codes. Many finite element codes allow grid point offsets, so that grid points at the corners of solid elements that model the VEM can also be used for the constraining and base layer plates, thereby saving many degrees of freedom in a model. For modeling purposes, Poisson's Ratio of VEMs is typically set to 0.4999. The amount of detail that is required in the finite element model is problem dependent.

The damping treatment design cycle for a VEM design using finite element techniques may be summarized as follows:

- Define the problem
 - Determine specifications, requirements, and constraints
 - Determine dynamics that cause high responses
- Perform preliminary design
 - Develop damping concepts
 - Perform analysis using reduced order modeling and appropriate analysis methods
 - Vary design variables to select good damping candidates
 - Perform response prediction analysis
- Perform final design
 - Develop detailed finite element model of damped structure

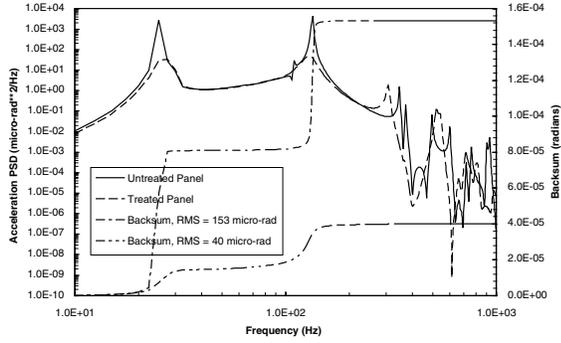


Figure 4: Displacement PSD for θ_x with backsum (undamped and damped)

- Vary design variables for best design
- Select viscoelastic material based on analysis results
- Perform response analysis using selected VEM properties
- Determine if all specifications and constraints have been met

Current work is now being performed by several researchers in the area of optimization of viscoelastic damping treatments. All of this work is based on performing sensitivity analysis of strain energy (see, for example, Gibson and Johnson, 1987).

5 Passive Damping Design Example Using VEM

For illustration purposes, consider the application of passive damping to a stiffened panel supporting a simulated component that is sensitive to its vibration environment. To simulate excitation by an acoustic field, a random pressure loading is applied over the surface of the panel. The chosen figure of merit is the RMS (10–1,000 Hz) values of normal displacements and their slopes. The analysis is performed with a finite element model using NASTRAN in which the panel and ribs are modeled with plate elements, and the component is a lumped mass.

The first step is to determine which modes contribute the most to the figure of merit. For the displacement normal to the panel, it is easy to see that the fundamental bending mode dominates this response. However, for the rotations, the fundamental mode along with either the second (for θ_x) or third (for θ_y) modes are of equal importance. The displacement PSD (solid line) for the x rotation, along with its backsum (153 μrad), are shown in Figure 4.

If only the normal displacement were important, this problem would be a good candidate for a tuned-mass damper, since this displacement PSD is strongly dominated by just the fundamental mode. However, the

VEMT	VEMG	CL	% MSE in VEM		
(mm)	(MPa)	(mm)	Mode 1	Mode 2	Mode 3
0.254	1.73	1.27	5.41	6.57	5.43
0.254	1.73	2.54	10.0	11.43	8.93
0.254	1.73	3.81	13.65	15.02	11.49
use CLT = 2.54 mm as baseline, now vary VEMG					
0.254	0.35	2.54	5.61	4.93	3.15
0.254	13.79	2.54	7.18	9.44	10.05
0.254	6.89	2.54	9.09	11.82	11.42
use VEMG = 1.73 MPa for baseline, now vary VEMT					
0.051	1.73	2.54	8.05	10.53	10.44
0.127	1.73	2.54	9.77	12.14	10.53
0.102	1.73	2.54	9.49	12.01	10.77
0.152	1.73	2.54	9.93	12.11	10.22
Runs for final predictions of MSE in modes 1–3					
0.152	0.65*	2.54	8.88	n/a	n/a
0.152	1.93†	2.54	n/a	12.21	10.48

*3M Y-966 shear modulus at ~ 25 Hz
 †3M Y-966 shear modulus at ~ 130 Hz

Table 3: Summary of trade study for add-on constrained-layer damping treatment

importance of the second and third modes makes a constrained-layer treatment more appropriate for this case.

There are five basic design parameters for a constrained-layer treatment: thickness of the constraining layer, modulus of the constraining layer, thickness of the VEM, modulus of the VEM, and placement of the treatment. In most practical situations, some of these parameters are determined by outside factors such as constraints on weight, thermal expansion, clearance, etc. Where weight is a factor, it is usually advantageous to make the constraining layer from advanced materials, such as metal matrix or graphite-epoxy. For this example, the constraining layer is made from the same material as the base panel: aluminum. The entire top surface of the panel, including under the component, is covered by the constrained-layer damping treatment.

A brief trade study with the remaining parameters showed that the VEM should be approximately 0.152 mm thick and have a shear modulus near 1.73 MPa. This trade study is documented in Table 3. One material that fits this closely for the three modes of interest is 3M’s Y-966. The shear modulus and loss factor for this VEM at 25 and 130 Hz are approximately 0.65 MPa and 1.93 MPa, respectively. Two additional runs were then made with these values to get a better approximation for the MSE in the modes of interest. They are also reported in Table 3. These values of modal strain energy were subsequently multiplied by the loss factors for

	RMS		
	z (μm)	θ_x (μrad)	θ_y (μrad)
Untreated	12.3	153	191
With Damping	2.1	40	43

Table 4: Reductions in RMS values resulting from passive damping

their respective frequencies and used to predict the responses of the structure with the added damping treatment. The RMS values are given in Table 4, and the effects are shown clearly by the PSD in Figure 4 (dotted curve).

There are three variations on this concept that bear mentioning:

1. The damping treatment could be shrunk so that it covered a smaller portion of the base panel.
2. The panels themselves could be constructed from a sandwich of metal and VEM.
3. Instead of a constraining layer, the VEM could be sandwiched between the base structure and built-up sections (I-beams, C-channels, hat sections, etc.).

Each of these alternatives is likely to save weight, though they are not discussed in this paper.

6 Conclusions

Passive damping can provide substantial performance benefits in many kinds of structures and machines, often without significant weight or cost penalties. Full realization of these benefits depends on (1) properly characterized materials, (2) knowledge of the strengths and weaknesses of the various materials and mechanisms, and (3) appropriate design / analysis methods and software.

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