Damping and Erosion Coatings

COATINGS FOR ENHANCED PASSIVE DAMPING

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ABSTRACT

The amplitude of vibration in a structure undergoing resonant vibration is governed by the total damping of the system. As the inherent damping of materials suitable for use in the fabrication of structures and machine components is often quite low, increasing the total system damping by including a dissipative material or mechanism can often provide significant reductions in the peak values of response (stress, strain, and displacement), enabling more efficient designs and enhanced performance. Available methodologies include active dampers and such passive techniques as friction and impact dampers and constrained layer treatments. It has also been found that metals and ceramics applied as free-layer hard coatings by plasma spray or electron beam physical vapor deposition add significant damping to vibrating members. In order to incorporate the influence of a damping coating in a prediction of system response during a preliminary design, it is essential that properties of the coating be known. The relevant damping characteristic is a measure of the energy dissipated by a homogeneous unit volume of material undergoing a completely reversed cycle of oscillation. A useful metric for this is the loss modulus. As all of these materials are inherently non-linear, as evidenced by amplitude-dependent measures of damping, determinations of properties must be made at the levels of strain appropriate to the application. Methods for determining the damping properties of materials are discussed, and comparisons are made of the damping of various classes of materials with those of ceramic coatings deposited by plasma spray or electron beam physical vapor deposition.

I. THE NEED FOR DAMPING

Although the assumption of a perfectly elastic material is very convenient for use in the analysis of structures, and adequate in most cases, no structural materials are truly elastic. A system given an initial perturbation will eventually come to rest unless the dissipation is offset by the addition of energy. Cyclic motion of a structure can be sustained at constant amplitude only if the energy lost through dissipation is offset by work done on the system. Energy dissipation, or damping, can be advantageous to the performance of a system as it governs the maximum amplitude achieved under resonance and the rate at which a perturbed system progresses to a satisfactorily quiescent state. In addition to lowering the probability of failure due to fatigue, a reduction of amplitude can have other benefits such as reducing a visible vibration, reducing the sound transmitted from a valve cover, or reducing the signature of a vibrating submarine propeller. Damping, however, can also be disadvantageous, contributing as it does to such unwanted phenomena as shaft whirl, instrument hysteresis, and temperature increases due to self-heating.

The term damping (not dampening) refers to the dissipation of energy in a material or structure under cyclic stress or strain through a process of converting mechanical energy (strain and kinetic) to heat. When such dissipation occurs locally within the material, the process is referred to as material damping, taken as inclusive of the dissipative mechanisms variously referred to as mechanical hysteresis, anelesticity, or internal friction.

The distinction between material damping and system damping should be observed. Material damping, the inherent ability of a substance to dissipate energy under cyclic stress or strain, is a material property and can be expressed in absolute units of energy dissipated per unit volume per cycle or as a dimensionless measure formed from a ratio of the energy dissipated to the energy stored. System damping, on the other hand, is a measure of the influence of the total dissipation of all the components of the system on the overall response of a structure or structural component.

At the present time, there is a particularly high level of interest in reducing the amplitude of vibration of the blades in gas turbine engines. As each blade in a rotating component passes a blade (stator) in the static component, a pressure pulse is generated. As the blades rotate at a high cyclic rate (~10,000 RPM) and there may well be several dozen stators, excitation rates of several thousand impulses per second are common. As a response at around 3K Hz gives rise to one million cycles of vibration in only 10 engine hours, and as the maximum velocity of structural motion is proportional to maximum strain, it is evident that excitation at a resonant frequency can lead to failure of the blade by fatigue. This phenomenon is known as high cycle fatigue (HCF) and is to be distinguished from the blade damage known as low cycle fatigue (LCF) that results from the much less frequently occurring perturbations due to major changes in engine RPM and temperature.

As it is the amount of damping present in the system that governs the magnitude of the response at resonance, it would be highly advantageous to be able to apply a passive damping treatment to rotating turbine blades. However, a successful treatment must survive not only the high vibratory stresses for the service life of the engine but also the environmental challenges of high temperature, erosion, corrosion, foreign object damage, and centrifugal forces due to the high rotation rates. Further, a successful treatment must not degrade the performance of the engine by adding excessive weight, by detrimental changes in the shapes of aerodynamic surfaces, or by inducing cracks in the blade material.

A number of concepts for the reduction of vibratory response are being explored. These include the use of friction dampers, impact dampers, energy absorbers, and the inclusion of dissipative materials in constrained layer damping treatments. Of a special interest is the use of free layer coatings in the form of thin layers of high damping metals and ceramics applied through plasma spray and electron beam physical vapor deposition. The use of such ceramics as alumina, magnesium aluminate spinel, and yttria stabilized zirconia appear to have received the greatest attention.

II. DAMPING AS A MATERIAL PROPERTY

A. Measures of Material Damping

The most fundamental measure of the dissipative capability of a material is the specific damping energy or unit damping, defined as the energy dissipated in a unit volume of material at homogeneous strain and temperature while undergoing a fully reversed cycle of cyclic stress or strain. The specific damping energy, D, has dimension of energy per unit volume, per cycle, and is, in general, a function of the amplitude and history of stress or strain, temperature, and frequency. For some materials, the unit damping is also dependent on the mean (static) stress or is influenced by magnetic fields. The unit damping is customarily given in terms of the amplitude of a uniaxial tensile or shear stress or strain. Multiaxial states of stress are characterized by an equivalent uniaxial stress[1].

Material damping may be categorized as being linear or non-linear. In the first class, the energy dissipated per cycle is dependent on the square of the amplitude of cyclic stress or strain. As the strain energy density is also normally proportional to the square of amplitude, the ratio of dissipated and stored energies, as well as other dimensionless measures of damping, are then independent of amplitude. Materials displaying these attributes are said to display linear damping. In the second class of materials, the energy dissipated per cycle varies as amplitude of cyclic stress to some power other than two. If the strain energy density varies as, or nearly as, the square of amplitude, the ratio of dissipated to stored energy is then a function of the amplitude of stress or strain. Such materials are said to display nonlinear damping.

While some important mechanisms of damping, such as viscoelastic and thermoelastic, are essentially linear, many others are not. In the case of structural materials for which the predominant damping mechanism is plastic deformation on a scale which leaves the material macroscopically linear, the specific damping energy has been found[1] to depend on the amplitude of cyclic stress as

$$D = J\sigma_d^n \tag{1}$$

with n ≈ 2.4 for cyclic stresses below about 70% of the endurance limit. For metals, the parameters J and n are generally independent of frequency, but dependent on temperature. At higher stress, the same functional form may be applied, but the damping typically increases more rapidly with stress. The parameter n is then much greater and may increase or decrease with the number of cycles.

While rooted in the concept of a linear viscoelastic material, the concept of the complex modulus may be adapted to characterize the dissipation of other materials undergoing cyclic loading. In the case of a nonlinear material, we may define amplitude-dependent effective values of a storage and a loss modulus, E_1 and E_2, by

$$E_1(\omega, T, \varepsilon_d) \equiv \frac{2U(\omega, T, \varepsilon_d)}{\varepsilon_d^2} \quad \text{and} \quad E_2(\omega, T, \varepsilon_d) \equiv \frac{D(\omega, T, \varepsilon_d)}{\pi\varepsilon_d^2} \tag{2a, b}$$

where U is the stored (strain) energy in the unit volume, D is the specific damping energy, and ε_d is the amplitude of cyclic strain. For structural materials, the values of storage and loss modulus are typically independent of frequency, but vary with amplitude and temperature. In the case of viscoelastic materials, the moduli are typically independent of amplitude, but vary strongly with both frequency and temperature. A material loss factor may also be defined as the ratio of energy dissipated in the unit volume per radian of oscillation to the peak energy stored.

$$\eta \equiv \frac{D}{2\pi U} = \frac{1}{2\pi}\frac{\pi E_2(\omega, T, \varepsilon_d)}{E_1(\omega, T, \varepsilon_d)/2} = \frac{E_2(\omega, T, \varepsilon_d)}{E_1(\omega, T, \varepsilon_d)} \tag{3}$$

If either or both of the components of the modulus are dependent on amplitude, then the material loss factor is also dependent on amplitude. Note that the use of the loss factor, defined in terms of energy dissipated per radian, is to be preferred over the sometimes-used specific damping capacity or damping index, computed from the energy dissipated per cycle by $\Psi = D/U$. This is because the unit of the radian is more truly a dimensionless quantity than is the cycle.

The complex modulus is defined as the ratio of the Fourier transform of stress to that of strain,

$$E^* = \sigma^*(\omega)/\varepsilon^*(\omega) = E_1 + jE_2 = E_1(1 + j\eta) \tag{4}$$

and is particularly convenient for analyzing the response of time-dependent materials to sinusoidal forcing functions. The storage modulus, E_1, is the customary Young's or elastic modulus; the loss modulus, E_2, is the product of the storage modulus and the loss factor. The use of a complex quantity to describe a real, physical material is troubling to some. The proper interpretation is not that the material behaves in a mathematically complex manner, but rather that the presence of dissipation is associated with an out-of-phase relationship between stress and strain. The loss factor quantifies the degree to which they are out of phase. The angle by which the strain lags the stress is given by $\tan\phi = E_2(\omega, T, \varepsilon_d)/E_1(\omega, T, \varepsilon_d)$ and is referred to as the loss tangent.

B. High Damping Materials

1. HIDAMETS

The specific damping energy of most structural materials falls in a fairly narrow range, particularly at stresses of design interest, i.e. well below the endurance limit. The mean curve for a wide range[1] of structural metals may be represented by $D = 14(\sigma_d / \sigma_E)^{2.4}$ Joule/m^3 for applied stresses, σ_d, up to about 70% of the endurance limit, σ_E. There are, however, a few exceptions, as such magnetoelastic materials as 403 alloy and the alloys of nickel and cobalt; alloys of manganese and copper; and the shape memory (SM) alloys. The high damping of magnetoelastic materials arises because the application of stress induces rotations of the magnetic domains similar to that produced by the application of a magnetic field, a process that dissipates energy. The high damping of Mn-Cu and the shape memory alloys is associated with transitions from face centered cubic to face centered tetragonal structures. Attempts to exploit these mechanisms so as to obtain high inherent material damping while meeting the other requirements of a material for primary structures have not, to date, been successful. These, and other high damping metals, may yet find application, perhaps as coatings.

2. Viscoelastic Materials

Another class of materials finding extensive application in structural additions for enhanced damping includes the elastomers and polymers. These are characterized by stress-strain relationships incorporating rate dependence. In the classical form, the constitutive relationship is taken to be a linear operator on stress proportional to a linear operator on strain. While this provides an adequate description of observed material properties, it has been found that many terms (necessitating many experimentally determined parameters) are required. More recently, it has been shown[2] that modifying the form by replacing the integer derivatives in the linear operators with fractional derivatives enables an excellent characterization over as many as 11 orders of magnitude in frequency. A characteristic of viscoelastic materials is that, in addition to having a strong dependence on frequency, they display a strong dependence on temperature. Fortunately, these are related. The concept of the reduced frequency has been accepted as a means of jointly describing both effects. A complete discussion is available elsewhere.[3]

As the ability to dissipate energy is dependent on the loss modulus, Eq. 2b, the loss modulus is an appropriate metric for identifying high damping materials. The solid line of Figure 1 represents a constant value of loss modulus (product of loss factor and storage modulus), with 1 GPa used as a reference value. Structural metals are seen to generally fall below this line; materials considered for treatments to enhance damping, such as the high damping alloys and magnetoelastic materials, fall on or above this line.

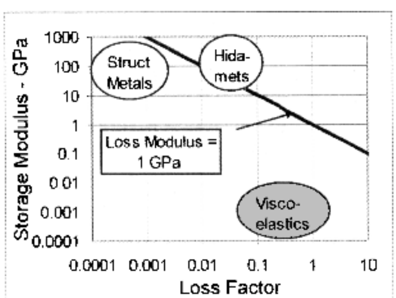


Figure 1. Categories of Damping Materials

Dissipation depends not only on the loss modulus, but also on the square of strain. As many of the viscoelastic materials can safely withstand repetitive strain on the order of unity, they most truly earn their reputation as high damping materials when used in a damping treatment with configuration designed so as induce strains in the viscoelastic material that are 2-3 orders of magnitude more than that of the base structure.

III. DAMPING AS A SYSTEM PROPERTY

The damping of systems containing one or more dissipating elements is of interest for two reasons. In order to incorporate damping into predictions of the system performance during the preliminary design process, a mathematical model of the system, including damping, is necessary. Secondly, as it is typically very difficult to perform materials testing on samples of a homogeneous material in a homogenous state of strain, as is necessary for the determination of a true material property, material testing is typically done on a system containing the sample with material properties then extracted from the system response.

Although continuous systems have an infinity of vibratory modes, unless the system is highly nonlinear or the modes very closely spaced, the response near a resonant or natural frequency is quite well represented by the single mode dominant at that frequency. And, although the systems of interest may not be truly linear nor is the damping likely to be viscous, we base the analysis of system damping on the well-known responses of the classical one-degree-of-freedom system consisting of a mass, a linear spring, and a viscous damping element.

Our objective is to relate observable measures of system response to the damping present in the system and then to use, with caution, these same measures in the characterization of systems for which the origin of damping is of a more general nature.

A. The Prototype Damped System

The familiar linear oscillator with linear spring, linear Newtonian dashpot, and mass has a displacement x(t) in response to a time dependent force F(t) that is given by:

$$M\frac{d^2x(t)}{dt^2}+C\frac{dx(t)}{dt}+Kx(t)=F(t) \tag{5}$$

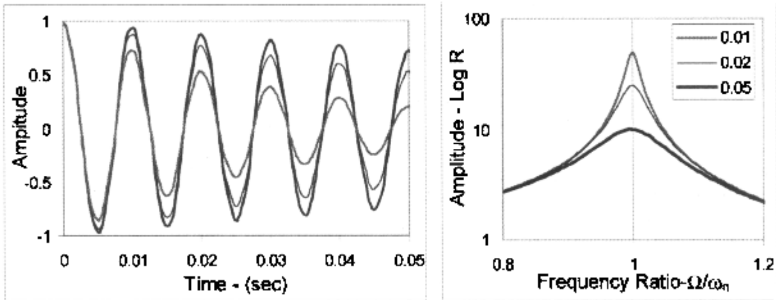
M, C, and K are presumed to be real constants, independent of frequency or amplitude. The solution may be written in terms of two parameters: a natural frequency, $\omega_n = \sqrt{K/M}$, and a fraction of critical damping (damping ratio), $\xi = C/(2\sqrt{KM})$. The homogeneous solution (free vibration)

$$x(t)=C\exp(-\xi\omega_n t)\cos(\omega_d t-\phi_0) \tag{6}$$

is a decaying sinusoid, shown in Figure 2a for several values of the fraction of critical damping.

The frequency response function, i.e., the magnitude of the response to a harmonic excitation at frequency Ω, is shown in Figure 2b. The displacement is normalized by the response in the limit as forcing frequency goes to zero (static displacement). Values for three levels of damping ratio, ξ – 1%, 2%, and 5%, are given as functions of a dimensionless frequency ratio, $f=\Omega/\omega_n$.

$$R=\left|\frac{x(t)}{x_{st}}\right|=\frac{1}{\sqrt{(1-f^2)^2+4\xi^2 f^2}} \tag{7}$$



a. Free Response b. Forced Response

Figure 2. Response of Linear 1DOF System at various Levels of Damping

B. Logarithmic Decrement

The logarithmic decrement is defined in terms of the ratio of the amplitude at an arbitrary cycle, n, to the amplitude after an arbitrary number, N, of additional cycles.

$$\delta = \frac{1}{N}\ln\frac{x_n}{x_{n+N}} = \frac{2\pi\xi}{\sqrt{1-\xi^2}} \tag{8}$$

Because the displacement peaks of Figure 2a would fall on a straight line if plotted semi-logarithmically, observations at any pair of cycles, n and n + N, lead to the same result for a linear system. But the presence of a curvature in the time history of the logarithm of amplitude is indicative of a dependence of the decrement on amplitude, i.e., nonlinear damping.

C. Resonant Amplification (or Quality) Factor

The maximum amplitude may be found from the frequency response function, Eq. 7.

$$R_{max} = \frac{1}{2\xi\sqrt{1-\xi^2}} \equiv Q \tag{9}$$

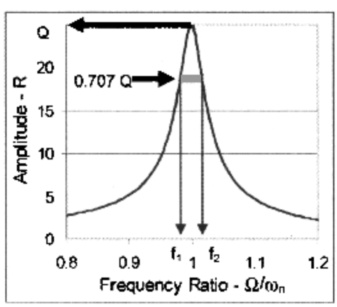


Figure 3. Response near Resonance for a One-DOF Viscous System

The response function shown in Figure 3 is for $\xi = 0.02$. As the maximum depends only on the damping, it provides an observable measure of Q for the system. However, unless the ordinate can be properly scaled by the static displacement, this measure is useful only for comparing the influence of small changes made to the damping of the same system. For small values of damping, $Q \approx 1/2\xi$.

D. Bandwidth for Half-Power Frequencies

Another attribute of the resonant response shown in Figure 3 has come to be the "industry standard" for the determination of the damping of a system. Not only is the peak response determined by the damping, so is the relative width of the response function. Since the energy of a linear system is proportional to the square of the amplitude, the system is said to be at "half-power" at those frequencies for which the amplitude is $1/\sqrt{2}$ of the peak value. There will be two such frequencies, one above and one below the frequency of maximum amplitude, as may be seen in Figure 3. These frequencies may be determined by setting the expression for the magnitude of the response, Eq. 7, equal to $Q/\sqrt{2}$ and solving for the two frequencies, f_1 and f_2, at which that amplitude occurs. For small values of viscous damping, i.e., $\xi << 1$, we have $B \approx 2\xi$, since

$$B = f_2 - f_1 = \frac{2\xi\sqrt{1-\xi^2}}{[(f_1 + f_2)/2]} \cong 2\xi(1 + \frac{5}{2}\xi^2 + ...) \tag{10}$$

E. Evaluation of Dissipation from the Hysteresis Loop

The hysteresis loop, the trajectory of force vs. displacement, is an ellipse for a linear system with viscous damping, and the enclosed area is the energy dissipated, D. In any dissipative system, the maintenance of a sinusoidal displacement-time history requires application of a force-time history out of phase with the displacement. While the hysteresis loop will not be an ellipse in a non-linear and/or inviscid system, the area remains a valid measure of the dissipation. The area under the right triangle connecting the origin to the point of maximum displacement is the peak stored energy in the system, U_S, enabling the specific damping capacity for any system to be evaluated from the ratio of these two measures of energy, $\Psi = D/U_S$. Also, the ratio of the minor to major axes of the hysteresis loop provides an approximate indication of the fraction of critical damping for a general system.

F. Relationships between Measures of Damping

The fraction of critical damping (or damping ratio) has been used as the fundamental measure of system damping. While it is not an observable quantity, we have seen that it may be related to a several observable measures: the logarithmic decrement, the quality factor, the bandwidth, and the specific damping capacity of the system. The first three are implicitly defined at (essentially) the same specific frequency, i.e., at a natural or resonant frequency of the system. Thus, for the same system, they may be readily linked. The system specific damping capacity differs in this regard, however, in that it is applicable at any frequency.

A system loss factor may be defined from the ratio of energy dissipated per cycle to the peak stored energy and then evaluated from any of the observable measures. Because of the likelihood of amplitude dependence, the system loss factors obtained by this means must be regarded as (potentially) dependent on both amplitude and frequency. The relationships are particularly simple for light damping, i.e. $\xi^2 << 1$.

$$\eta_S(\omega, X) \equiv \frac{\Delta W(\omega, X)}{2\pi U_S} = \frac{\Psi_S(\omega, X)}{2\pi} = \frac{\delta(\omega_n, X)}{\pi} = \frac{1}{Q(\omega_n, X)} = \frac{1}{B(\omega_n, X)} \tag{11}$$

And finally, the system loss factor must not be confused with a material loss factor.

G. Adaptation to Other Damping Mechanisms

These same measures of damping may be used with caution as representations of the damping of mildly non-linear systems. Three issues are of particular concern. First, in the presence of non-linear stiffness the period of oscillation of free vibration will vary throughout the decay. In the case of a forced vibration, the frequency response function, Figures 2b and 3, will not be symmetric about the resonant frequency. For sufficiently strong stiffness nonlinearities, the resulting instability may preclude the determination of one of the bandwidth frequencies. Second, in the presence of non-linear (amplitude dependent) damping, the logarithmic decrement will change as the decay progresses and the amplitude diminishes. In consequence, it must be evaluated over a narrow and moving window of cyclic amplitudes. In the case of a forced response, the damping is different at all amplitudes of the frequency response function. In consequence, the bandwidth will not provide a true measure of the system loss factor and must be appropriately adjusted.[4] Finally, a measure of damping given for a nonlinear system is essentially meaningless unless the corresponding amplitude is also provided.

IV. APPLICATIONS FOR ENHANCED DAMPING

Although other means of increasing the passive damping of vibrating systems have been developed (tuned mass dampers, impact dampers, and frictional devices), interest here is in those damping treatments which seek to exploit the inherent damping of a thin layer of a high damping material added to the surface of a component. These fall into two classes, free and constrained layer treatments. The essential distinction is whether the added material is subjected to (essentially) the same strain as the base structure, or if some mechanism for strain amplification is provided.

A. Free Layer Damping Treatments[5]

Free layer damping treatments are constructed by coating some or all of one or both sides of a member in a vibrating structure with a material capable of dissipating energy when subjected to a cyclic strain. A segment of a beam, coated on one side only, is shown in Figure 4. For a coating on one side only, the neutral plane is shifted a distance δ from the neutral plane of the substrate.

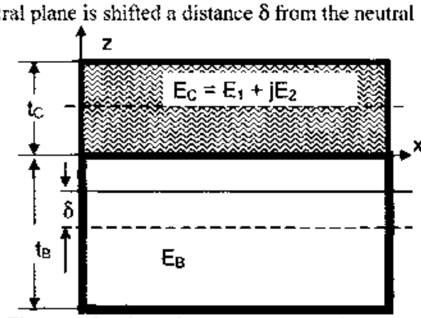


Figure 4. Section of Free-Layer Damping Treatment

Although the coating need not be of a uniform thickness, it will here be treated as such. The development will be outlined for the case of a beam, but the same result is obtained in the case of a plate if the Poisson's ratio for the substrate and coating are the same. It is assumed that the dissipative properties of the coating may be represented by a complex modulus, with the provision that if the modulus is frequency dependent the parameters of the modulus are to be evaluated at the frequency of vibration for the coated structure, which will differ from that of the uncoated structure.

After incorporating the beam and coating moduli, E_B and E_C, and thicknesses, t_B and t_C, into a stiffness ratio and a thickness ratio, $S \equiv E_C t_C / E_B t_B$ and $t \equiv t_C / t_B$, an effective stiffness for the section shown in Figure 4 may be found. For a complex value of coating modulus, the system stiffness becomes complex; a loss factor may be evaluated[5] from it as:

$$\eta_{SYS} = \frac{\Im[(1+St^2)+3S(1+t)^2(1+S)^{-1}]}{\Re[(1+St^2)+3S(1+t)^2(1+S)^{-1}]} \tag{12}$$

While this general result is not conveniently written in as a simple expression in terms of the loss factor of the coating, the result is easily evaluated numerically. Some numerical results are given in Figure 5 for various values of the ratio E_1 / E_B and the thickness ratio t_C/t_B.

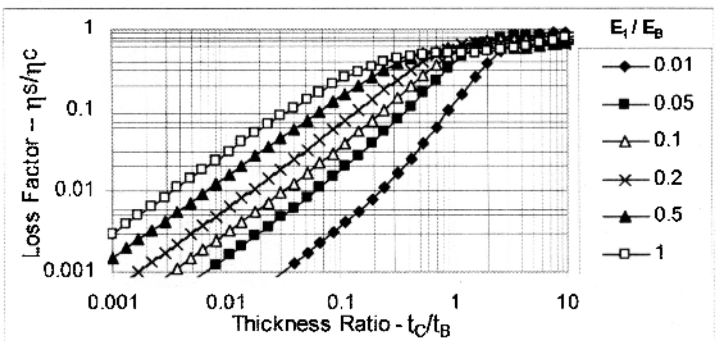


Figure 5. Loss Factors (Normalized) for Free Layer Treatments - Various E_1/E_b

These computations were performed for a coating loss factor of $\eta_C = 1.0$ with the results then normalized by division by that loss factor. The normalized ratio for any other value of material loss factor, η_C, is indistinguishable up to a thickness ratio of 10% and a modulus ratio of unity. The results of Figure 5 are applied by multiplying the value read from the ordinate by the material loss factor of interest.

For small ratios of moduli (as at $E_1 / E_B = 0.01$ of Figure 5) and thickness ratios on the order of unity, as might be characteristic of automotive undercoats, the loss factor is found to vary with the square of the thickness ratio. But for the thin, stiff coatings, as in the left half of Figure 5, a simple result is found:

$$\eta_{SYS} \cong 3\eta_c \frac{E_1 t_c}{E_b t_b} = 3\frac{E_2 t_c}{E_b t_b} \tag{13}$$

B. Constrained Layer Damping Treatments[6]

A constrained layer damping treatment (Figure 6) is formed by bonding a thin elastic sheet to a thin layer of a dissipative material adhering to a structural member. Under a bending deformation, an elastic constraining layer with free ends elongates less than does the surface of the structure and induces a shear strain in the damping layer. By making this layer very thin, the strain is amplified. As many viscoelastic materials can withstand a shear strain of unity, the potential dissipation is very high.

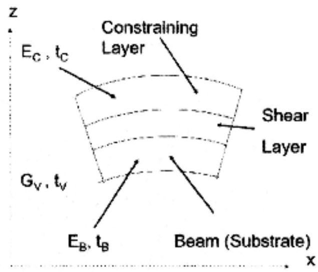


Figure 6. Constrained Layer Configuration

Several means of evaluating the resulting dissipation have been provided. Ross, Ungar, and Kerwin computed[7] the effective stiffness of the three layer configuration, and from that, the system loss factor for beams of infinite length, or with simple supports. A review of the successive advances enabling the treatment of general boundary conditions of beams is available,[8] as is a method for estimating the loss factors for rectangular plates with various boundary conditions.[9]

The results of the RUK analysis for a constrained layer treatment of length L covering a beam of thickness t_B and modulus E_B vibrating in the n^{th} simply supported mode can be expressed as

$$\eta_{SYS} = \frac{Y\,\Im\{g^*/(1+g^*\}}{1+Y\Re\{g^*/(1+g^*\}} \qquad (14)$$

$$g^*|_{Beam} = \frac{G^*}{t_V(n\pi/L)^2}\left(\frac{1}{E_C t_C}+\frac{1}{E_B t_B}\right) \quad \text{and} \quad Y = \frac{3(2+t_S+t_C)^2}{(E_C t_C^2+E_B t_B^2)}\left(\frac{1}{E_C t_C}+\frac{1}{E_B t_B}\right)^{-1} \qquad (15a, b)$$

where G* is the complex shear modulus of the damping layer of thickness t_V and the constraining layer is of thickness t_C and modulus E_C. Stretching of the constraining layer reduces the shear strain in the damping layer, thereby reducing the dissipation. In order to maximize the system loss factor, it is necessary that the parameters of the shear and constraining layers satisfy the relationship:

$$\left|\frac{|G^*|L^2}{E_C t_C t_V}\right|_{opt} \cong (n\pi)^2 \qquad (16)$$

From this it is seen that optimum damping can not be simultaneously achieved for several modes.

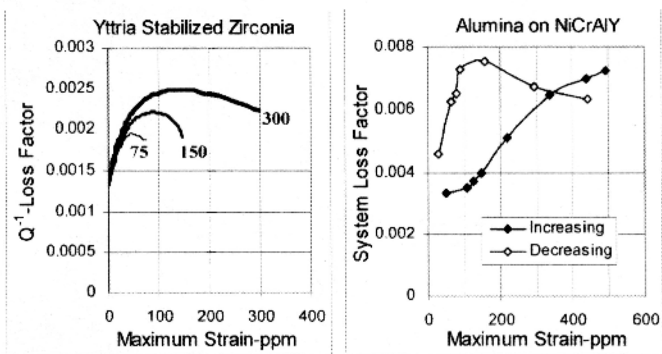
In the usual application, the constraining layer is constructed of a thin metal foil. However, the successful bonding of such a foil to a layer of damping material on a surface with compound curvature, such as an airfoil, is problematic. To address this problem, Rongong, *et. al.*, successfully constructed a constraining layer by plasma-spraying a ceramic layer onto the damping layer.[10]

V. DAMPING OF CERAMIC COATINGS

A. Techniques for Determining Coating Properties

A distinctive feature of sprayed ceramic coatings is that the damping depends on history and amplitude of vibration. System loss factors (η = 1/Q) are shown in Figure 7a for a 1.59 mm super alloy cantilever beam coated on both sides over about 20% of the length in a high strain region with 0.12 mm of 8% yttria stabilized zirconia. System damping was determined[11] from the decay of free vibrations in the second mode from initial amplitudes of 75, 150, and 300 ppm, successively, and adjusted by subtracting the bare-beam loss factor of 0.00033. It is noteworthy that the loss factors observed for all decays do not fall on the same line. A decay initiated at higher amplitude led to higher damping throughout the decay than did a decay initiated at lower amplitude.

A dependence on history was also noted[12] in the determination of system damping in the second cantilever mode of a 152mm long, 2.3 mm thick titanium beam fully covered on both sides with plasma sprayed alumina (0.25mm) on a NiCrAlY bond coat. Values of system loss factors determined by bandwidth at successively increasing amplitudes led to the increasing values of loss factor shown in Figure 7b as solid points. But when retested with values of maximum amplitude monotonically decreasing from the maximum, the measurements denoted with open circles resulted.



a.8% Yttria Stabilized Zirconia (Patsias and Williams data[11])

b. Alumina on NiCrAlY (Torvik, Willson, Hansel data[12])

Figure 7. Influence of History on System Damping of Beams with Plasma Sprayed Coatings

In order to extract material properties of a non-linear coating material from data for fully covered beams, such as in Figure 7b, account must be taken of the variation in strain throughout the coating. The observed system damping is the integrated contribution of coating dissipation at all strains from zero to the maximum value. For a thin coating, the variation in strain throughout the coating volume is nearly the same as the variation at the beam-coating interface. The uniformity of the strain distribution may be characterized by determining from the mode shapes the fraction of the coating at or below a given fraction of maximum strain, i.e., $A(\varepsilon < \varepsilon_{max})/A_0 = g(\varepsilon/\varepsilon_{max})$. This function varies with boundary condition and mode number but for the first several modes of cantilever, free-free, and simply supported beams it is roughly proportional to the fraction: $g(\varepsilon/\varepsilon_{max}) \approx \varepsilon/\varepsilon_{max}$.

The extraction of the unit damping function requires the solution of an integral equation with the weighting function being the derivative of the area-strain relationship. Given a series of measurements at various levels of maximum strain of the total dissipation in a thin coating of thickness t_c fully covering a beam area A_o, the total energy dissipated is related to specific damping energy by

$$D_{total}(\varepsilon_{max}) = \int_{volume} D(\varepsilon)dV = t_c A_0 \int D(\varepsilon)\frac{d(A/A_o)}{d(\varepsilon/\varepsilon_{max})}d(\varepsilon/\varepsilon_{max}) \quad (17)$$

One can assume a functional form for the dependence of D (ε) on local strain, and then use the observed values of total dissipation with Eq. 17 to find the parameters of that assumed form. An alternative methodology is to coat only a small area of the beam in a region of nearly uniform strain and then neglect the resulting modest variations in strain. The specific damping energy of the material is then approximated by averaging the total dissipation over the coating volume, or

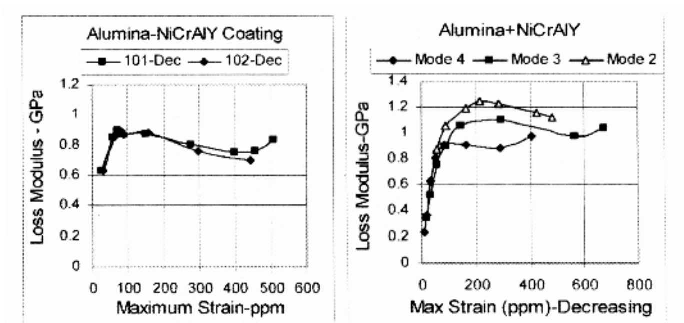
$$D(\varepsilon) \cong D_{total}(\varepsilon_{max})/(A_c t_c) \quad (18)$$

B. Survey of Coating Properties

A review was recently prepared of the damping properties of hard coatings as found in the open literature. In some cases, material properties were given in the source; in others, material properties were deduced from system measurements. Data sources, original system level data and the methodologies used to extract material properties are given elsewhere.[12,13] Some of the findings are summarized below. Reported data, in general, did not give indications of the history of loading.

1. Alumina

Material properties were extracted from two data sets obtained with beams fully covered with plasma-sprayed alumina on NiCrAlY. Loss moduli, extracted from tests conducted at decreasing amplitudes of excitation, are given below as Figures 8a and 8b.

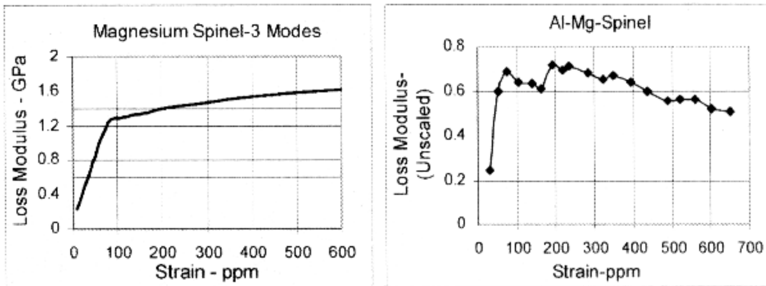


a. Tested on Titanium b. Tested on Hastelloy X

Figure 8. Loss Modulus of Plasma Sprayed Alumina-NiCrAlY Coating System
(Data taken by UTC staff using AFRL Facility)

2. Magnesium Aluminate Spinel

Values of the loss modulus for magnesium alumina spinel, extracted from the response at increasing amplitudes of fully covered beams vibrating in the 2nd, 3rd and 4th modes, are shown in Figure 9a. The properties of magnesium spinel were also extracted from vibration decay tests at Sheffield University on partially coated beams of a C263 super-alloy vibrating in the 2nd cantilever mode. Material properties (loss factor and storage modulus) were treated as unknowns at each of 20 values of strain. Trial values of each were then adjusted through iteration so as to produce satisfactory agreement between observed and predicted values of system loss factors and natural frequencies. The loss modulus in Figure 9b was formed from the product of their (unscaled) results. As the system loss factors for ceramic coated beams depend on load history (see Figure 7), material properties extracted from measurements at decreasing levels of excitation can be expected to display the same trends.



a. From Frequency Response Functions (Data taken using AFRL Facility)

b. From Vibration Decay Records (Using results from Patsias and Williams)

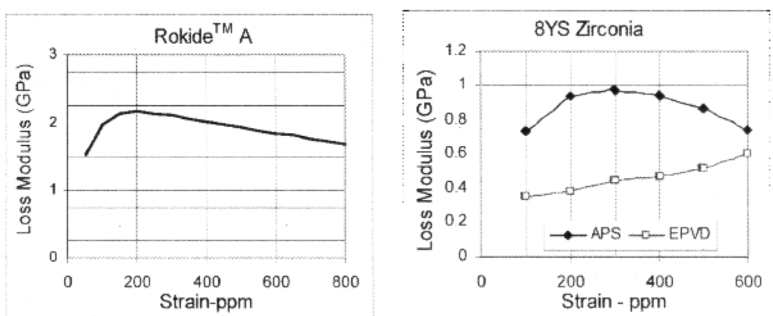
Figure 9. Loss Modulus of Plasma Sprayed Magnesium Aluminate Spinel

3. Rokide® A

Volume-averaged values of the loss factors and storage modulus have also been determined from vibration decay data for fully covered beams. The storage modulus ranged from 30-40 GPa. The strain dependent loss modulus shown in Fig. 10a was constructed from reported results. Beams were coated with Rokide® A aluminum oxide, combustion sprayed at room temperature.

4. Yttria Stabilized Zirconia

Properties of 8 wt% yttria stabilized zirconia (YSZ) deposited by air plasma spray (APS) and electron beam physical vapor deposition (EB-PVD) methods have been compared. In this case, the coating was applied on both surfaces of the beam, but only over 13% of beam length, at the root. No bond coat was used with air plasma sprayed specimens; coatings applied by electron beam physical vapor deposition were on top of a NiCoCrAlY bond coat applied by air plasma spray. Values of the material loss factors and storage modulus were reported; the values of loss modulus shown in Figure 10b were constructed from these.



a. Rokide® (Data by Patsias, *et. al.*)

b. 8wt% YSZ (Data by Tassini, *et. al.*)

Figure 10. Loss Modulus of Sprayed Ceramics

Damping properties of sprayed ceramics and other materials are compared in Table I.

Table I Damping Properties of Materials

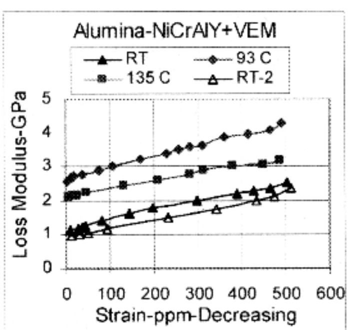
Material	Storage Modulus	Loss Factor	Amplitude Dependence	Temperature Dependence	Other Factors
Structural Metals (typical)	70-200 GPa	0.0002-0.005	Mild	Moderate	None
Magnetoelastic Materials	15-200 GPa	0.01-0.10	Strong	Modest-well below Curie	Mean Stress Dependent
Mn-Cu and SM Alloys	70-80 GPa	0.03-0.08	Mild	High	Processing Dependent
Polymers and Elastomers	0.1-10 MPa	0.5-1.0	Linear	High	Frequency Dependent
Plasma Sprayed Ceramics	30-55 GPa*	0.01-0.06	Strong	Minimal	None

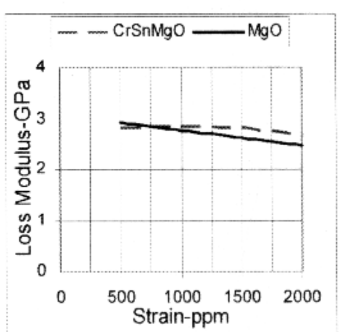
*Values of the coating storage modulus deduced from the natural frequencies of coated beams.

In the case of a beam or plate covered on both sides, the system loss factor is twice that of Eq. 13. Thus, if the total coating thickness is to be limited to 10% of the thickness of a titanium substrate (i.e., 10% on one side or 5% on each side), the required loss modulus to obtain an increase in system loss factor ($\eta - Q^{-1}$) of 0.01 must be on the order of 3.6 GPa.

This is a daunting requirement, as typical values of the loss modulus for plasma sprayed ceramics appear to be limited to the range of 1-2 GPa. Coating materials with notably higher values of loss modulus (product of storage modulus and loss factor) are necessary.

Impregnation of plasma sprayed ceramic with a viscoelastic component has been considered. It is conjectured that the viscoelastic constituent would then dissipate additional energy through being deformed by the relative motion of the platelets formed in the spray process. Patsias[14] reported that a doubling of system loss factors at 50-150 C resulted from impregnating alumina with polyurethane. In other work,[15] the surface infiltration of a viscoelastic material in alumina led to material loss moduli for the impregnated ceramic shown in Figure 11a and seen to be 300-400% of the values for unimpregnated alumina, Figure 8. While this approach shows promise of leading to the desired values of loss modulus, a frequency dependence not seen in uninfiltrated ceramics can be expected.





a. Alumina with Viscoelastic Infiltrate (Data by Torvik, *et. al.*)

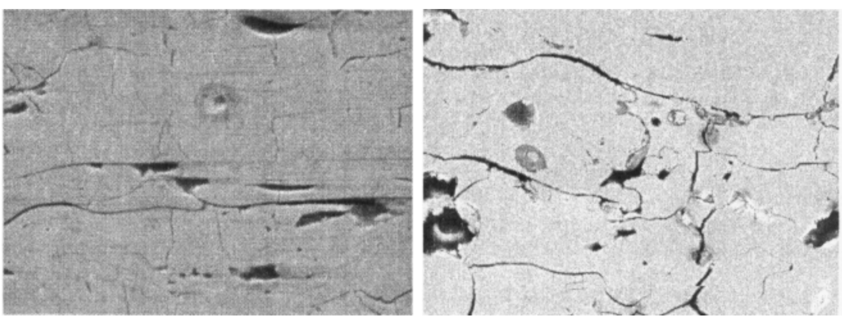
b. EBPVD Graded Coating at RT (Data by Ustinov, *et. al.*)

Figure 11. Enhancements to the Damping of Sprayed Ceramics

The use of electron beam physical vapor deposition to deposit a ceramic on top of a thin layer of a highly deformable metallic layer has also been considered.[16] A 40 μm graded coating was formed of 10 wt % Sn, 15 wt% Cr, and balance MgO. Extraction of material properties[13] from reported system data led to values shown in Figure 11b for the coating system, and for the ceramic alone.

C. ORIGIN OF DAMPING IN PLASMA SPRAYED CERAMICS

Shipton and Patsias[17] examined the defect structure of thermally sprayed MgAl spinel and found a system of horizontally aligned solidified splats with adjacent horizontal splat boundaries that are parallel and well aligned (Figure 12a). The population of vertical defects within splat boundaries subdivided the splats into an array of ordered, parallel-sided plates or 'mobile blocks.' As the vertical cracks were observed to open under strain, Shipton and Patsias inferred that the damping results from the friction on interfaces within and between the 'splats.' Experiments simulating a coated beam by a vibrating beam with segmented and overlapping cover plates showed a dependence of damping on amplitude similar to that seen in coatings. A computer simulation[18] employing springs and Coulomb sliders was also found to predict an amplitude dependent loss factor and storage modulus having the same characteristics as those seen in material properties extracted from experimental data. A system of generally horizontal and vertical defects was also found[17] in plasma sprayed yttria stabilized zirconia (YSZ), as is seen in Figure 12b. But the structure here (and also in plasma sprayed alumina) is notably less well ordered than in the case of MgSpinel coatings. And, as the damping of MgSpinel (Figure 9a) is greater than that of these other materials (Figures 8 and 10b) it is inferred that the higher degree of regularity enables a greater effectiveness of the frictional mechanism, leading to higher damping.



a. MgAl Spinel (x 10K) b. Typical PYSZ (x 5K)

Figure 12. Micro-structure of Typical Plasma Sprayed Ceramics (used with permission)

VI. SUMMARY

The role of damping in the control of vibrations was discussed. Definitions, damping measures, and the damping properties of several classes of materials were summarized and compared, with emphasis given to application as free-layer coatings for the reduction of vibratory amplitudes. The damping potentials of several sprayed ceramics were reviewed and found to be comparable to those of the metallic alloys known to have the highest damping capabilities. The justification for the hypothesis that friction at interfaces between and within the 'splats' resulting from the spray process was also reviewed. The damping capability of plasma sprayed ceramics appears to be less than desired for some applications, although techniques for increasing the damping are being developed. A significant challenge in the characterization of such materials remains, however, as experiments show that the loading history has a significant impact on the damping properties of sprayed ceramic coatings.

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