

METHODS FOR IMPROVING DAMPING. Part 1: an introduction and guide to practical methods of increasing structural damping

1. NOTATION

A, B	parameters defined by Equations (7.2) and (7.3) respectively		
a, b	lengths of plate in x- and y-directions respectively	m	in
C	geometric parameter defined by Equation (7.12)		
c	speed of sound in air	m/s	in/s
D_0	flexural rigidity of untreated plate	N m	lbf in
D_s	energy dissipation per cycle [see Equation (7.15)]	N m	lbf in
E	real part of Young's modulus*	N/m ²	lbf/in ²
E_1, E_2, E_3	real part of Young's modulus of individual layers	N/m ²	lbf/in ²
f	frequency	Hz	Hz
f_m	natural frequency of m^{th} beam mode	Hz	Hz
f_{mn}	natural frequency of mn^{th} plate mode	Hz	Hz
f_s	resonant frequency of treated structure	Hz	Hz
G	real part of shear modulus*	N/m ²	lbf/in ²
G_2	real part of shear modulus of damping layer	N/m ²	lbf/in ²
g	shear parameter [see Equation (7.14)]		
H_1, H_2, H_3	layer thickness	m	in
K	non-dimensional parameter in Equation (4.1)		
k	real part of complex spring stiffness	N/m	lbf/in
L	length of structure or beam	m	in
l	optimum length of constrained layer damping treatment [see Equation (9.1)]	m	in
M	tuned damper mass	kg	lb s ² /in
m, n	mode numbers in x- and y-directions respectively		

For footnotes see end of Notation section

Issued August 1991
With Amendment A, September 1994

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N	number of layers in multiple constrained-layer damping treatment		
s	number of fasteners per inch	in ⁻¹	in ⁻¹
T	temperature	K	K
T_o	reference temperature	K	K
$\left. \begin{matrix} T_{-2}, T_{-1} \\ T_1, T_2 \end{matrix} \right\}$	temperature below and above reference temperature respectively	K	K
t	time	s	s
W	plate lateral displacement [see Equation (7.9)]	m	in
W_{mn}	amplitude in mn^{th} normal mode	m	in
W_o	amplitude of structural displacement [see Equation (7.15)]	m	in
x, y	orthogonal coordinate axis system		
α_{mn}	damping coverage parameter defined by Equation (7.7)		
α_T	shift factor on reduced-temperature nomogram		
β	shear loss factor		
β_{mn}	stiffness parameter defined by Equation (7.8)		
γ	shear strain		
δ	viscous damping ratio		
ϵ	direct strain		
η	extensional loss factor		
η_s	loss factor of treated structure		
ν	Poisson's ratio of treated plate with damping layer		
ξ_{mn}	eigenvalue of mn^{th} mode of untreated plate [see Equation (7.10)]		
ρ	density of structural material	kg/m ³	†lb/in ³
ρ_a	plate mass per unit area	kg/m ²	†lb/in ²
ρ_{air}	density of air	kg/m ³	lb/in ³

For footnotes see end of Notation section

ρ_D	density of damping material	kg/m ³	lb/in ³
σ	direct stress	N/m ²	lbf/in ²
τ	direct strain	N/m ²	lbf/in ²
$\phi_{mn}(x,y)$	normal mode of untreated plate		
ψ	specific damping capacity		
ω	circular natural frequency	rad/s	rad/s
ω_D	tuned damper resonant frequency	rad/s	rad/s
ω_{mn}	plate circular natural frequency in mn^{th} mode	rad/s	rad/s

* E and G are often referred to as storage moduli.

† See footnote to Section 7.2.2.

2. INTRODUCTION

Damping refers to the dissipation of energy in a vibrating system. In Item No. 73011* typical values of damping are given for the types of structure used in aerospace vehicles. Those data are all for structures that have no added damping treatment. This Item is one of three Items providing information on, and design data for, improving damping in structures. This Item provides an introduction to damping mechanisms and the materials for damping treatments and outlines the basic methods of increasing damping. Additionally, points to be considered in the practical application of damping treatment and good design practices are discussed. Companion Item No. 91013 provides computer programs for the analysis of free-layer and constrained-layer damping treatments based on the procedures discussed in this Item. The third Item in this trilogy, Item No. 92001, provides typical damping material data required for the analysis of layered damping treatments using Item No. 91013. These Items are directed principally to aerospace applications where, without special damping treatments, damping is likely to be very low.

Aerospace structures, and sensitive equipment mounted on those structures, are often required to function through a wide range of dynamic loads. When structural resonances are excited, excessive levels of noise or vibration may be produced that can lead to structural failure such as acoustic fatigue. Additionally, excessive vibration may lead to the malfunction of equipment having, for example, electronic or optical components. Controlling vibration can reduce the likelihood of structural failure and increase the reliability of sensitive equipment.

Ideally the possibility of problems arising through structural resonances should be considered at the design stage when the options available for damping treatments may be better than when the structure has been built and subsequent tests indicate the need for additional damping. Introducing damping at the design stage may also lead to lighter and cheaper designs than those resulting from later design changes.

The design of a damping treatment should not be directed exclusively towards maximising the damping. Such a course neglects the fact that other parameters change when a damping treatment is applied. Damping

* Damping in acoustically excited structures.

treatments will usually increase both the mass and stiffness of the structure and it is necessary to consider all three parameters (damping, mass and stiffness) in order to optimise the damping treatment to give minimum structural response to the dynamic input. More consideration of the optimisation of damping treatments is given in Section 8.

In order to design a damping treatment it is first necessary to understand the detailed dynamic behaviour of the system under all operating conditions. Such information will allow, for example, the identification of regions of maximum response, and the dominant response frequencies and their spacing. Knowledge of all these factors is an essential prerequisite for the design of suitable damping treatments.

In this Item the points to be considered when assessing a damping problem are discussed. Some measures of damping are defined, the principal damping mechanisms are described and information is given on materials commonly used in damping treatments. Basic methods for increasing damping are then considered and analytical methods for estimating damping from layered treatments are given. These methods are provided for making initial assessments of a damping treatment on idealised structures and are not intended to replace the full dynamic analysis that may be required to predict the behaviour of practical damped structures. For the full analysis of complex structures with damping treatments the finite element method offers an accurate means of prediction. Information on finite element methods of analysis are given in References 33, 34 and 35.

3. ASSESSMENT OF THE PROBLEM

A number of factors should be considered when assessing a structure to decide what action should be taken to alleviate excessive noise or vibration response. If damping treatments are to be effective it is necessary to ensure that the response to be attenuated is due to resonant vibration. The addition of damping to a system has little effect upon forced vibrations apart from those occurring at resonance. It should be stressed that narrow band response is not necessarily an indicator of resonance since some periodic forced vibrations produce that type of response. For lightly damped structures, typical of those used in the aerospace industry, the natural and resonant frequencies are approximately equal.

To understand the damping problem it is necessary to evaluate, under operating conditions, the resonance frequencies, mode shapes and damping of the untreated structure. For structures where the resonance and natural frequencies may be considered equal, the natural frequencies and mode shapes for a number of structural elements are given in other Data Items. (See Item No. 74037* for information on the structural elements for which modal data are available.) Guidance on the damping of structural elements under acoustic loading is given in Item No. 73011. It is the assessment of how much damping there is in the original structure that is most difficult, partly due to the number of damping mechanisms that exist (see Section 4). However, the definition of the untreated structural damping is most important since it is on that value that the required damping increase must be based.

For panels without additional damping treatments the range of values of viscous damping ratio, δ , based on response measurements on aircraft structural elements is tabulated below. The damping values given are typical of those for the lower frequency vibration modes including the fundamental mode. It should be noted that all the tabulated data were measured in air; in-vacuo damping values may be significantly reduced. For example, the in-vacuo damping for the first mode of vibration of a carbon fibre-reinforced, laminated free-free beam in flexure was approximately 70 per cent of that measured in air (Derivation 15). It is not possible to quantify exactly the increase in damping that may be achieved with damping treatments since this is dependent on the nature of the exciting force, the type of structure, the specific damping treatment and the mode of response. However, the increases in damping due to layered damping treatments on a number of panels under broad-band excitation are illustrated in the examples in Item No. 91013.

* Introduction and guide to ESDU data on acoustic fatigue.

<i>Type of structure</i>	<i>Range of equivalent viscous damping ratio</i>
Riveted aluminium alloy skin/stringer panels	0.005 to 0.030
Aluminium alloy integrally machined panels	0.003 to 0.012
Laminated carbon fibre-reinforced panels (epoxy resin matrix)	0.004 to 0.026
Honeycomb sandwich panels with aluminium alloy face plates	0.010 to 0.030
Honeycomb sandwich panels with carbon fibre-reinforced face plates (epoxy resin matrix)	0.005 to 0.023
Honeycomb sandwich panel with aramid fabric face plates	0.013 to 0.028

Having estimated the response of the untreated structure, it is then necessary to evaluate the required increase in damping to reduce the critical response parameter to an acceptable level. The selection of a damping treatment to achieve the required damping is discussed in Section 8.

4. DAMPING MECHANISMS

An understanding of the damping mechanisms within a particular structural configuration is necessary before an assessment can be made of the damping in any mode of response. Because of the wide range and complexity of damping mechanisms reliable estimation of damping in a resonant response is very difficult. Damping mechanisms are of two main types: firstly the energy loss that occurs through interaction with an adjacent fluid which is termed acoustic damping, and secondly the energy loss that occurs within the structural configuration. The main sources of these two types of damping mechanisms are considered in the following sub-sections.

For stiffened plate structures the relative magnitude of the structural and acoustic damping depends on the mode of vibration. For example, considering a multispan panel supported on its long sides by stiff frames and periodically stiffened by stringers normal to the frames, the relative structural and acoustic damping for a particular mode depends on the relative phase of motion of adjacent panels which is reflected in the nature of the stringer motion. In a mode of vibration in which the stringer motion is predominantly twisting and adjacent panels are out-of-phase, acoustic damping is likely to be negligible compared with the structural damping. However, in a mode in which the stringers are predominantly bending and adjacent panels are in-phase, acoustic damping is likely to be considerably larger than the structural damping.

4.1 Acoustic Damping

The dominant source of acoustic damping derives from the radiation of sound energy into the fluid adjacent to the vibrating structure. The damping arises from the acoustic pressure that is in counter phase with and proportional to the plate velocity. Since acoustic damping involves interaction between a structure and its surrounding medium there is no acoustic damping in space vehicles in their in-orbit service environments.

Within a panel array, the magnitude of the acoustic damping is dependent on the phase relationship between the motion of adjacent panels. Maximum acoustic damping occurs when adjacent panels are vibrating in-phase. An approximate expression for acoustic radiation damping has been developed (Derivations 5

and 23) in terms of the viscous damping ratio.

$$\delta = \frac{64}{\pi^4} \cdot \frac{\rho_{air}}{c} \cdot \frac{f_{mn}}{\rho_a} \cdot \frac{ab}{m^2 n^2} \cdot \frac{1}{K} \quad (4.1)$$

where parameter K is unity when all panels vibrate in-phase and K is equal to the product of the number of bays in the panel array in the x - and y -directions when adjacent panels are out-of-phase. Equation (4.1) is applicable to panel arrays in which the individual plates may be considered to be simply-supported. Also it is assumed that the acoustic wavelength is much greater than the panel dimensions.

4.2 Material Damping

Because materials are not perfectly elastic, energy is dissipated under cyclic stress. This damping within the material is often referred to as intrinsic damping. For structural metallic materials, hysteretic damping is usually small. However, in some circumstances material damping may provide a significant contribution to the overall damping: for example, in structures where structural damping is low (see below) and viscoelastic materials may not be safely incorporated. Values of material damping are not currently derivable by theory and so are usually found by mechanical testing.

4.3 Joint Damping

For structures without damping treatments, structural damping arises from the friction at joints due to relative motion of adjoining elements. For riveted or bolted structures, structural damping is usually much higher than material damping. However, for structures that are fabricated from welded, bonded or integrally machined sub-structures, such that joint slippage either does not occur or is much reduced, structural damping is likely to be low.

At very small displacement amplitudes little or no joint slippage is likely to occur so that only hysteretic damping will contribute to structural damping. As the vibration amplitude increases a condition is reached at which the onset of slippage occurs and friction damping results. The damping force is assumed to be equal to the product of the coefficient of sliding friction and the normal force between the jointing components. Experimental evidence suggests that the coefficient of sliding friction is independent of frequency and maximum velocity but dependent on the normal force. Further tests on riveted structural elements have shown that the structural damping loss factor appears to be independent of the load amplitude at the fastener line. Based on these experimental results it is concluded (Derivation 23) that the friction damping at the fastener line of stiffened panels is a constant, dependent only on the number of fasteners along the panel periphery. Further assuming that the fasteners at the corners of the panel, and those at the nodal lines that cross the panel, do not contribute to the friction damping, it is suggested (Derivation 23) that an approximate expression for the equivalent viscous damping ratio due to friction damping at a dry jointed fastener line is

$$\delta = 0.0253 \frac{\left\{ s(a+b) - \frac{n}{s} - \frac{m}{s} \right\}}{ab}, \quad (4.2)$$

where s is the number of fasteners per inch*. It should be noted that Equation (4.2) is based on limited test data and may not apply generally. It is recommended that the relationship be used only as a guide to dry jointed friction damping.

* When SI units are used, s must be corrected to fasteners per inch before substituting in Equation (4.2) (1 in = 25.4 mm).

Friction damping may be reduced when jointing compound is used to avoid problems associated with fretting and corrosion. However, the jointing compound may also act as a viscoelastic insert that can increase the damping due to shearing distortions within the compound layer.

4.4 Damping of Composite Materials

When using fibre-reinforced composite materials, joints are usually bonded so that slippage does not occur at the joints and material damping is important for the reduction of dynamic response. The main sources of material damping in a composite arise from microplastic or viscoelastic phenomena associated with the matrix and relative slipping at the interface between the matrix and the reinforcement.

The following factors influence the material damping in composites.

- (i) The properties and relative proportions of the matrix and reinforcement materials within the composite.
- (ii) The size of the inclusions (*e.g.* long or short fibres, and fibre diameter).
- (iii) The orientation of the reinforcing material to the axis of loading.
- (iv) The surface treatment of the reinforcement.

The effect on the structural damping of some of the above factors is discussed in Section 7.1.1.

Item No. 85012* provides a means of estimating the material damping of multilayered composite laminated plates. From a knowledge of the specific damping capacities of the individual layers, the damping of balanced symmetric laminates may be evaluated using that Item.

Additionally, damping may be increased in laminated composites by any cracks or debonding that occur in either the manufacturing process or in the operating loading cycle.

5. COMMON MEASURES OF DAMPING

The commonly used measures of damping in companion Data Items are specific damping capacity, viscous damping ratio and loss factor. These damping measures are defined as follows.

The specific damping capacity, ψ , is the ratio of energy dissipated in a unit volume per stress cycle to the maximum strain energy per unit volume in the course of the cycle.

The viscous damping ratio, δ , is the ratio of viscous damping to critical damping. Critical damping is the lowest value of viscous damping that leads to non-oscillatory motion following an initial disturbance.

When the damping is non-viscous or hysteretic, it is commonly quantified by the loss factor which (for computational purposes) can be identified with twice the equivalent viscous damping ratio. A better definition of loss factor is

$$\text{Loss factor} = \left(\frac{1}{2\pi} \right) \frac{\text{Energy dissipated per cycle of displacement}}{\text{Maximum energy stored in the course of the cycle}}.$$

The loss factors, η and β , are defined in terms of the complex moduli that relate the complex cyclic stresses

* Estimation of damping in laminated and fibre-reinforced plates.

and strains. The direct stress, σ , and shear stress, τ , are given by

$$\sigma = E(1 + i\eta)\epsilon, \quad (5.1)$$

and
$$\tau = G(1 + i\beta)\gamma, \quad (5.2)$$

where ϵ and γ are the direct and shear strains and η and β are the loss factors in direct and shear strains respectively.

Throughout this series of Items on methods of improving damping, a distinction has been made between extensional loss factors and shear loss factors. However, because of scatter in measured loss factor data, it is not possible to distinguish between the measurements of these two values. Therefore, when using measured loss factors, it is reasonable to assume that the extensional and shear loss factors are equal. Both loss factors are often denoted by η .

The relationship between a number of the most common measures of damping is given in Item No. 73011 and more details are given in Reference 31. The relationship between the three measures of damping considered in this Item, for harmonic vibrations and light damping, is given by

$$\delta = \frac{\Psi}{4\pi} = \frac{\eta}{2}. \quad (5.3)$$

6. MATERIALS FOR DAMPING TREATMENTS

The common materials that are used to improve damping in lightweight structures are elastomers, polymers and vitreous enamels. The damping of these substances varies with environmental factors: an understanding of the variation is essential for the design of effective treatments for noise and vibration control. The major environmental factors to be considered are temperature, frequency and dynamic load.

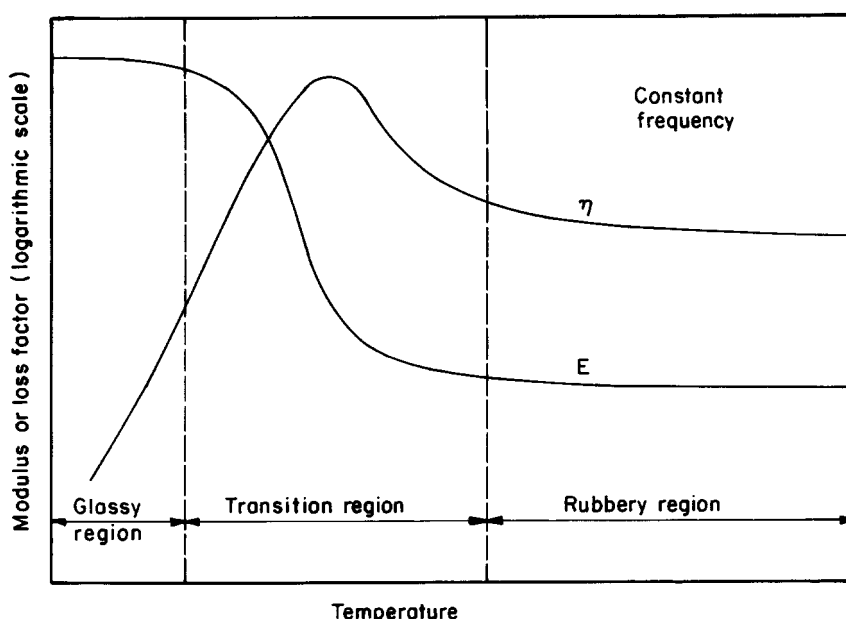
Using complex modulus notation, the parameters that are needed to define the properties of a damping material are the real part of Young's modulus, or shear modulus, and the loss factor. The variations of these parameters with environmental factors are considered below.

6.1 Effect of Temperature

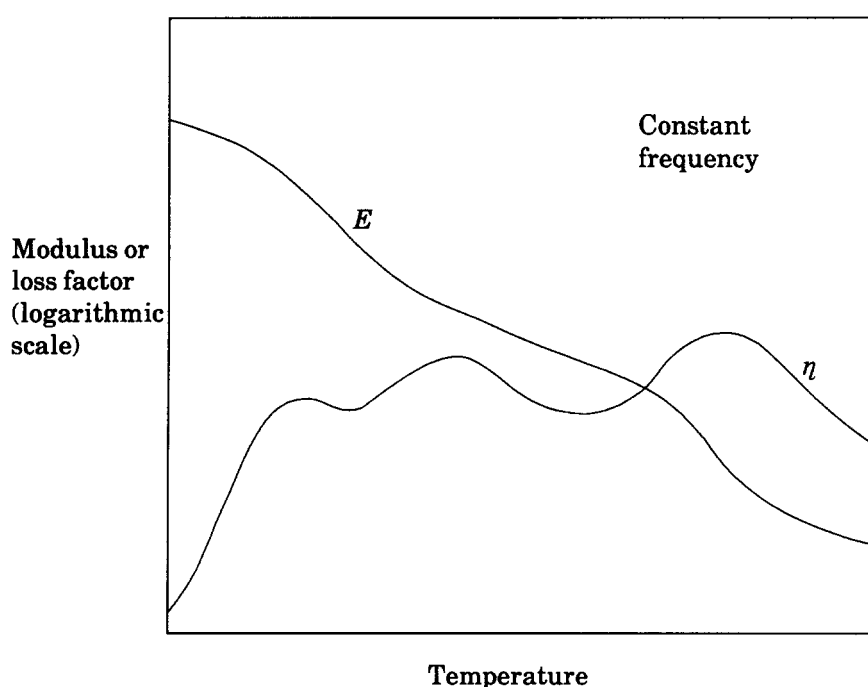
A feature of the properties of elastomers and plastic damping materials is their extreme sensitivity to changes in temperature. The variation of Young's modulus, E , or shear modulus, G , and loss factor, η , with temperature, at constant frequency and low cyclic strain amplitude, is typically of the form shown in Sketch 6.1. This sketch shows three distinct regions. In the lowest temperature region, referred to as the glassy region, the material modulus has its maximum value and the damping is low. In the transition region, which includes the transition temperature at which the damping peaks, the modulus varies rapidly with temperature and the damping is high. In the third region, referred to as the rubbery region, the modulus varies slowly with temperature and the damping is lower than in the transition region but may still be significant. For some materials such as vitreous enamels and thermoplastics, a fourth region can be identified at the highest temperatures. In that region, as temperature increases the modulus rapidly decreases and the damping rapidly increases. This region, in which the material melts, is referred to as the flow region but usually has no application in the design of damping treatments.

Although the variation of modulus and damping shown in Sketch 6.1 is typical of damping materials, different materials have specific properties that are characterised both by different levels of modulus and loss factor within the temperature regions and by variations of the region boundaries relative to temperature. By combining more than one material, polymer blends have been developed that give high damping over

a much wider temperature range than the individual constituent materials. Young's modulus and loss factor are shown plotted against temperature for a blended polymeric material in Sketch 6.2.



Sketch 6.1 Typical variation of modulus and loss factor with temperature

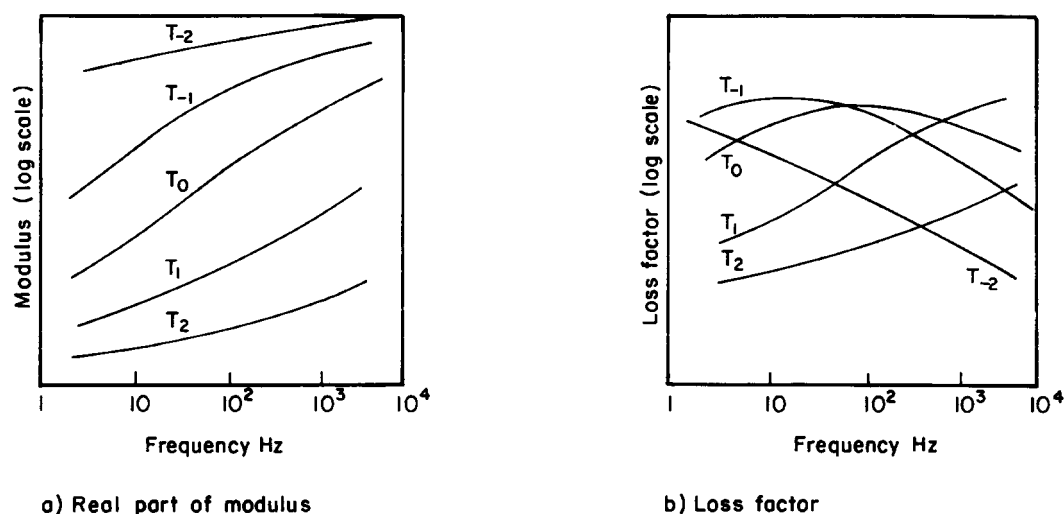


Sketch 6.2 Variation of modulus and loss factor for a polymer blend

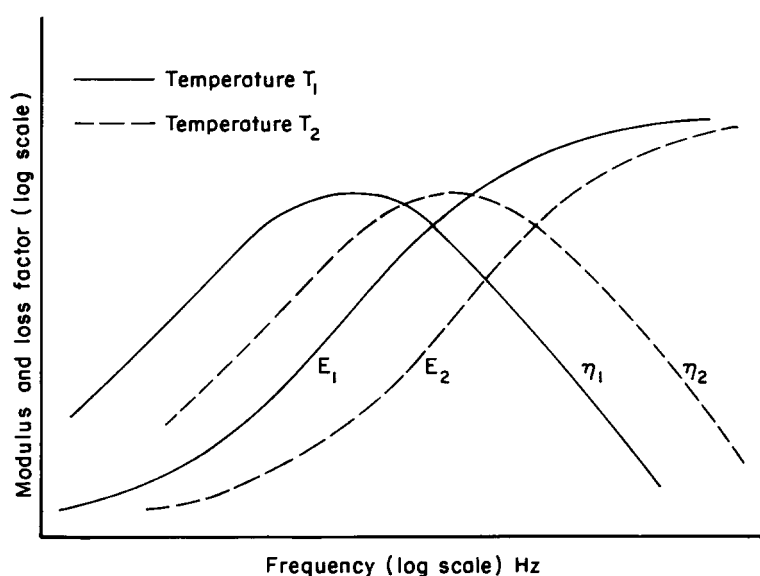
6.2 Effect of Frequency

The variation of damping parameters with frequency is illustrated in Sketch 6.3 for a typical material having high damping potential over a frequency range of approximately 10 Hz to 1 kHz. Data are shown on that sketch for a range of temperatures above and below a reference temperature T_0 . If data were available over a wider frequency range (which is usually difficult to achieve with normal testing procedures) the variation

of damping parameters with frequency would be as shown in Sketch 6.4.



Sketch 6.3 Variation of damping parameters with frequency



Sketch 6.4 Variation of damping parameters over an extended frequency range

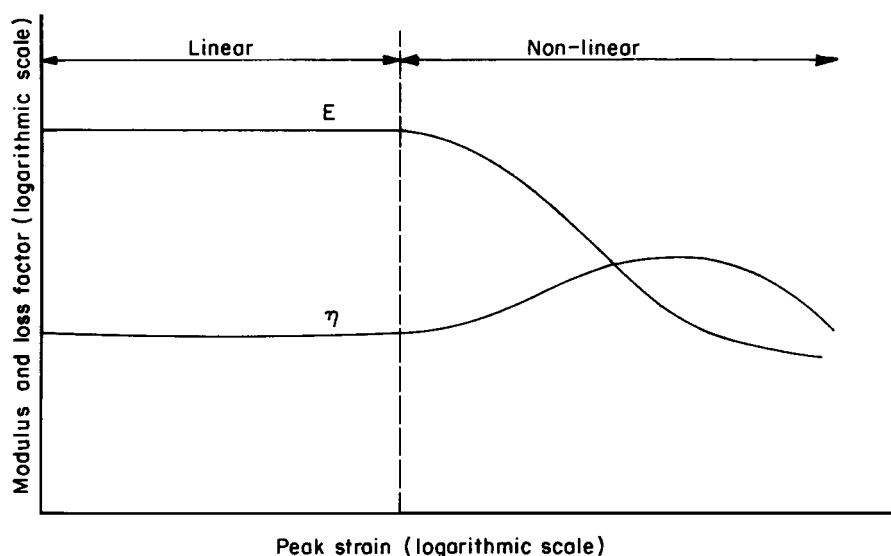
From Sketch 6.4 it is seen that the modulus always increases with frequency with a maximum rate of increase in the region of peak damping (*i.e.* the transition region). The general form of the variations of damping material modulus with frequency is the reverse of variations with temperature. However, it should be noted that a relatively large frequency variation is required to give the same change in damping parameters that is obtained with a relatively small change in temperature.

Sketch 6.4 illustrates a characteristic that is common to many damping materials. If the curves for temperature T_1 are shifted horizontally, at some particular frequency shift they approximately coincide with curves for temperature T_2 . This phenomenon is the basis for the “reduced temperature” nomogram for the presentation of damping material data (see Section 6.4).

6.3 Effect of Cyclic Dynamic Strain

Continuous high cyclic strain amplitude usually gives rise to high energy dissipation and consequent temperature increase in damping materials. Under those circumstances it is difficult to distinguish the effect of strain amplitude and temperature. Consequently the effects of strain amplitude are usually available only for the rubbery region where the variations of modulus and loss factor with temperature are relatively slow. Typical variations of modulus and loss factor with peak cyclic strain are shown in Sketch 6.5 for a damping material in the rubbery region.

The theoretical methods for investigating damping treatments require that the damping material properties are independent of strain amplitude (*i.e.* the material behaves linearly). The extent of the linear region is dependent on the molecular composition of the damping material.



Sketch 6.5 Variation of damping parameters with strain amplitude

6.4 Reduced-temperature Nomogram

To allow the effects of frequency and temperature on damping parameters to be considered simultaneously, a reduced-temperature nomogram has been developed (see, for example, Reference 32). This presentation makes use of the shift property mentioned in Section 6.2 to define a shift factor, α_T . An example of a reduced-temperature nomogram is shown in Sketch 6.6. The derivation of α_T and the construction of the nomogram is described in Reference 32.

To use the nomogram for particular values of f and T , a horizontal line is drawn through the value of f on the frequency scale to intersect the oblique temperature line for temperature T . The vertical line through that intersection point cuts the modulus and loss factor curves at the required values of E and η . The construction for using the nomogram is illustrated by the dotted lines on Sketch 6.6.