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Material damping is a name for the complex physical effects that convert kinetic and strain energy in a vibrating mechanical system consisting of a volume of macrocontinuous (solid) matter into heat. Studies of material damping are employed in solid-state physics as guides to the internal structure of solids. The damping capacity of materials is also a significant property in the design of structures and mechanical devices; for example, in problems involving mechanical resonance and fatigue, shaft whirl, instrument hysteresis, and heating under cyclic stress. Three types of material that have been studied in detail are:

- Viscoelastic materials. The idealized linear behavior generally assumed for this class of materials is amenable to the laws of superposition and other conventional rheological treatments including model analog analysis. In most cases linear (Newtonian) viscosity is considered to be the principal form of energy dissipation. Many polymeric materials, as well as some other types of materials, may be treated under this heading.
- Structural metals and nonmetals. The linear dissipation functions generally assumed for the analysis of viscoelastic materials are not, as a rule, appropriate for structural materials. Significant nonlinearity characterizes structural materials, particularly at high levels of stress. A further complication arises from the fact that the stress and temperature histories may affect the material damping properties markedly; therefore, the concept of a stable material assumed in viscoelastic treatments may not be realistic for structural materials.
- Surface coatings. The application of coatings to flat and curved surfaces to enhance energy dissipation by increasing the losses associated with fluid flow is a common device in acoustic noise control. These coatings also take advantage of material and interface damping through their bond with a structural material.

When an engineering structure is subjected to a harmonic exciting force $F_g \sin t$, an induced force $F_d \sin (t - \phi)$ appears at the support. The ratio of the amplitudes, F_d/F_g , is a function of the exciting frequency ω . It is known as the vibration amplification factor. At resonance, when $\phi = 90^\circ$, this ratio becomes the resonance amplification factor A_r :

$$A_r = \frac{F_d}{F_g} \quad (6.25)$$

This condition is pictured schematically in Fig. 6.3 for low, intermediate, and high damping (curves 1, 2, 3, respectively). The magnitude of the resonance amplification factor varies over a wide range in engineering practice. In actual engineering parts under high stress, a range of 500 to 10 is reasonably inclusive. These limits are exemplified by an airplane propeller, cyclically stressed in the fatigue range, which displayed a resonance amplification factor of 490, and a double leaf spring with optimum interface slip damping which was observed to have a resonance amplification factor of 10. Because of the wide range of possible values of A_r , each case must be considered individually.

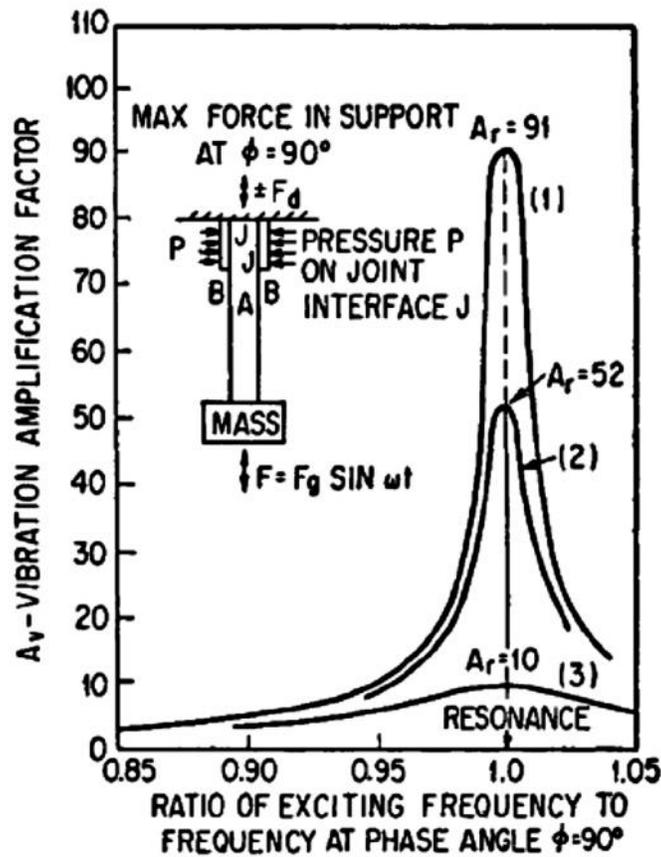


Fig. 6.3 Effect of material and slip damping on vibration amplification, Curve (1) illustrates case of small material and slip damping; (2) one damping is large while other is small; (3) both material and slip damping are large [2]

In defining the various energy ratio units, it is important to distinguish between loss factor s of a specimen or part (having a variable stress distribution) and the loss factor for a material (having a uniform stress distribution). By definition the loss factor of a specimen (identified by subscript s) is:

$$y_s = \frac{D_0}{2\pi W_0} \quad (6.26)$$

where the total damping D_0 in the specimen is given by Eq. The total strain energy in the part is of the form:

$$W_0 = \int_0^{V_0} \frac{1}{2} \left(\frac{\sigma^2}{E} \right) dV = \frac{1}{2} \left(\frac{\sigma_d^2}{E} \right) V_0 \beta \quad (6.27)$$

where E denotes a modulus of elasticity and β is a dimensionless integral whose value depends upon the volume-stress function and the stress distribution:

$$\beta = \int_0^1 \left(\frac{\sigma}{\sigma_d} \right)^2 \frac{d(V/V_0)}{d(\sigma/\sigma_d)} d \left(\frac{\sigma}{\sigma_d} \right) \quad (6.28)$$

On substituting Eq., it follows that:

$$\eta_s = \frac{E}{\pi} \frac{D_d}{\sigma_d^2} \frac{\alpha}{\beta} \quad (6.29)$$

If the specimen has a uniform stress distribution, $\beta = 1$ and the specimen loss factor η_s becomes the material loss factor η ; in general, however,

$$\eta = \frac{ED_d}{\pi\sigma_d^2} = \eta_s \frac{\beta}{\alpha} \quad (6.30)$$

Other energy ratio (or relative energy) damping units in common use are defined below:

- For specimens with variable stress distribution:

$$\eta_s = (\tan \phi)_s = \frac{\Delta_s}{\pi} = \frac{\Psi_s}{\pi} = \left(\frac{\delta\omega}{\omega_n} \right)_s = \frac{1}{(A_r)_s} = \frac{1}{Q_s} = \frac{ED_d}{\pi\sigma_d^2} \left(\frac{\alpha}{\beta} \right) \quad (6.31)$$

- For materials or specimens with uniform stress distribution:

$$\eta = \tan \phi = \frac{\Delta}{\pi} = \frac{\Psi}{\pi} = \frac{\delta\omega}{\omega_n} = \frac{1}{A_r} = \frac{1}{Q} = Q^{-1} = \frac{ED_d}{\pi\sigma_d^2} \quad (6.32)$$

where η = loss factor of material = dissipation factor (high loss factor signifies high damping)

ϕ = loss angle, where ϕ is phase angle by which strain lags stress in sinusoidal loading

η_s = specific damping capacity

ω/ω_n = (bandwidth at half-power point)/(natural frequency)

A_r = resonance amplification factor

$Q = 1/\eta$ = measure of the sharpness of a resonance peak and amplification produced by resonance

The material properties are related to the specimen properties as follows:

$$\psi = \psi_s \frac{\beta}{\alpha} \quad \Delta = \Delta_s \frac{\beta}{\alpha} \quad A_r = (A_r)_s \frac{\alpha}{\beta}$$

Thus, the various energy ratio units, as conventionally expressed for specimens, depend not only on the basic material properties D and E but also on β/α . The ratio β/α depends on the form of the damping-stress function and the stress distribution in the specimen. As in the case of average damping energy, Da , the loss factor or the logarithmic decrement for specimens made from exactly the same material and exposed to the same stress range, frequency, temperature, and other test variables may vary significantly if the shape and stress distribution of the specimen are varied. Since data expressed as logarithmic decrement and similar energy ratio units reported in the technical literature have been obtained on a variety of specimen types and stress distributions, any comparison of such data must be considered carefully. The ratio β/α may vary for specimens of exactly the same shape if made from materials having different damping-stress functions.

VISCOELASTIC MATERIALS

Some materials respond to load in a way that shows a pronounced influence of the rate of loading. Generally the strain is larger if the stress varies slowly than it is if the stress reaches its peak value swiftly. Among materials that exhibit this viscoelastic behavior are high polymers and metals at elevated temperatures, as well as many glasses, rubbers, and plastics. As might be expected, these materials usually also exhibit creep, an increasing deformation under constant applied load. When a sinusoidal exciting force is applied to a viscoelastic solid, the strain is observed to lag behind the stress. The phase angle between them, denoted by ϕ , is the loss angle. The stress may be separated into two components, one in phase with the strain and one leading it by a quarter cycles. The magnitudes of these components depend upon the material and upon the exciting frequency, ω . For a specimen subject to homogeneous shear ($\nu = 1$),

$$\tau = \tau_0 \sin \omega t \quad (6.33)$$

$$\sigma = \sigma_0 [G'(\omega) \sin \omega t + G''(\omega) \cos \omega t] \quad (6.34)$$

This is a linear viscoelastic stress-strain law. The theory of linear viscoelasticity is the most thoroughly developed of viscoelastic theories. In Eq., $G'(\omega)$ is known as the

“storage modulus in shear” and G'' is the “loss modulus in shear” (the symbols G_1 and G_2 are also widely used in the literature). The stiffness of the material depends on G' and the damping capacity on G'' . In terms of these quantities the loss angle $\phi = \tan^{-1}(G''/G')$. The *complex, or resultant, modulus in shear* is $G^* = G' + iG''$. In questions of stress analysis, complex moduli have the advantage that the form of Hooke’s law is the same as in the elastic case except that the elastic constants are replaced by the corresponding complex moduli. Then a correspondence principle often makes it possible to adapt an existing elastic solution to the viscoelastic case. The moduli of linear viscoelasticity are readily related to the specific damping energy D introduced previously. For a specimen in homogeneous shear of peak magnitude γ_0 , the energy dissipated per cycle and per unit volume is

$$D = \int_0^{2\pi/\omega} \sigma \left(\frac{d\gamma}{dt} \right) dt \quad (6.35)$$

Also,

$$\begin{aligned} D &= \int_0^{2\pi/\omega} \gamma_0^2 \omega (G' \sin \omega t + G'' \cos \omega t) \cos \omega t dt \\ &= \pi \gamma_0^2 G''(\omega) \end{aligned}$$

Controlling damping

Damping can be controlled by two major methods – **passive control and active control**. Passive control can involve several strategies for damping, all of which involve some mechanical characteristic of the system, either inherent or added, to control vibrations. Once the characteristic becomes part of the system, no further action is taken; hence, the system is passive. All of the treatments which have been implied in the previous examples, such as adding damping materials, weight or stiffness are passive. Passive control also includes the use of discrete devices, such as shock absorbers, and the addition of materials that have high inherent energy loss. Materials with highly mobile molecules, such as elastomers, have long been known as

highly damped materials. Therefore, damping control can be achieved by making the part out of an elastomer. A part could also be damped by adding elastomers to the normal material that the part is made of. In both of these cases, the internal molecular nature of the part furnishes the desired damping. Changing the shape of a vibrating system by joining system components together with elastomeric adhesives would also increase damping. Damping could also be achieved by mounting the vibrating part on an elastomer, such as would be done by using a damping pad for a motor. You might also wrap the part in an elastomer. These solutions reflect the general methods of damping that were discussed previously in the discussion of damping fundamentals.

Active damping is a much more recent development in damping engineering. This strategy involves the addition of elements to the part that sense the amount of vibration and trigger some remedial action to dampen the movement. The most common system of this type can be achieved by embedding sensors in a part to detect vibrations and piezo-electric devices which extend and retract in response to the sensor signals in such a way as to counteract the vibrations. This system requires that electric power be supplied to the actuators.

Active systems of the type described above have been used in aircraft. These systems drastically reduce the vibrations associated with flight, especially at times such as breaking the sound barrier. They are able to control these vibrations without the penalty of reducing the stiffness of the aircraft parts or changing their shape. Some advanced active systems also use the signals from piezo-electric devices to drive actuator motors which can make minor adjustments to the shape (geometry) of the airplane components. For instance, the wing shape can be changed during high turbulence to optimize flight control.

Another method of active control is through the use of embedded fiber optics. These fibers can sense gross vibrations much like electronic sensors. The fiber optics can be monitored for changes in cross-sectional area of the fiber which will cause a change in the light transmission and, therefore, indicate that vibrations are occurring. These changes might even be able to pinpoint the actual location of the vibration, thus giving tighter control than is usually possible with electronic sensors.

Source:

<http://nptel.ac.in/courses/112107088/21>