Design Consideration in Material Selection: Damping based

INTRODUCTION

The word design means many different things to different people. Here, design is used to denote an educated method of choosing and adjusting the physical parameters of a vibrating system in order to obtain a more favorable response. One of the most effective methods to solve noise and vibration problems is to dampen the system. Damping can be divided into two types: active and passive damping. Active damping requires external means to dissipate energy such as electronically controlled actuator. Passive damping requires add-on solutions such as shock absorber, isolator, structural member's internal damping, etc to dissipate energy. The term "*damping*" here refers to the energy-dissipation properties of a material or system under cyclic stress, but excludes energy-transfer devices such as dynamic absorbers. With this understanding of the meaning of the word, energy must be dissipated within the vibrating system. Damping occurs when vibrational energies are converted into some other energy form, usually heat. The kinds of vibration that people in the composites field are most often concerned with are either mechanical vibrations or acoustic vibrations (sound or noise). At the fundamental level, both of these vibrations are really the same and can be treated in similar fashion. In most cases a conversion of mechanical energy to heat occurs. For convenience, damping is classified here as material and system damping. Also, there are two common mechanisms for material energy dissipation is viscous damping and plastic deformation.

It has been a challenge to reduce unwanted vibration in rotating equipment. The impact of this vibration can range from premature failure of the machine to consumer's perception of poor product quality. Many designers and product engineers from all industries have been working diligently to solve noise and vibration problems in their products. Without a source of external energy, no real mechanical system maintains undiminished amplitude of vibration. 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 macro-continuous (solid) matter into heat. The amount of damping of a part depends on: (a) the materials out of which the part is made, and (b) the design of the part (geometry, mass, etc.). However, it is rare that a part would exist in total isolation, (c) any elements that might be added to the part, thus creating a system (which could include specific damping treatments but could also include additions whose effect on damping is more complicated). In this chapter, the material selection criteria for vibration control due to material damping will be discussed. The damping capacity of materials is a significant property in the design of structures and mechanical devices.

DAMPING PROPERTIES OF MATERIALS:

The specific damping energy dissipated in a material exposed to cyclic stress is affected by many factors as;

1. Condition of the material

a. In virgin state: chemical composition; constitution (or structure) due to thermal and mechanical treatment; in-homogeneity effects

b. During and after exposure to pretreatment, test, or service condition: Effect of stress and temperature histories on aging, precipitation, and other metallurgical solid-state transformations

2. State of internal stress

a. Initially, due to surface-finishing operations (shot peening, rolling, case hardening)

b. Changes caused by stress and temperature histories during test or service

3. Stress imposed by test or service conditions

a. Type of stress (tension, compression, bending, shear, torsion)

b. State of stress (uniaxial, biaxial, or triaxial)

c. Stress-magnitude parameters, including mean stress and alternating components; loading spectrum if stress amplitude is not constant

d. Characteristics of stress variations including frequency and waveform

e. Environmental conditions: temperature (magnitude and variation) and the surrounding medium and its (corrosive, erosive, and chemical) effects

THE NEED FOR MATERIAL SELECTION:

During the last decades many new materials and material types have been developed. This has been possible due to the development of the materials but also due to the appearance of new production methods. As a consequence of this rapid development many material types can be used for a given component. This also applies to situations where one previously only employed one material for example cast iron in cylinder heads where cast aluminum alloys are also used now. Another example is body panels in cars where low carbon mild sheet steel is still the dominating material but many other materials like high strength sheet steels, aluminum alloys, sheet molding compounds (SMC), thermoplastics, thermoplastic elastomers and expanded plastics are used. In fact it is quite a common case that many entirely different materials can be used to a given part. As a consequence material selection becomes quite a complex task.

LINEAR VISCOELASTIC MODEL:

As discussed, material damping is related to diffusion of atoms or molecules or internal friction of the material and system damping is related to energy dissipation in the total structure, which includes material damping and energy dissipation due to joints, interfaces, and fasteners. Here, the viscoelasticity in the material damping is discussed as the property of materials that exhibit both viscous and elastic characteristics when undergoing deformation. Viscoelastic material is characterized by possessing both viscous and elastic behavior. A purely elastic material is material which stores all the energy during loading and returns it when the load is removed (unloading). So there is no energy loss during loading and unloading for purely elastic material. Viscous materials, like honey, resist shear flow and strain linearly with time when astress is applied. Elastic materials strain instantaneously when stretched and just as quickly return to their original state once the stress is removed. Viscoelastic materials have elements of both of these properties and, as such, exhibit time dependent strain. Whereas elasticity is usually the result of bond stretching along crystallographic planes in an ordered solid, viscosity is the result of the diffusion of atoms or molecules inside an amorphous material. Let’s take an example of a slab of concrete with a thickness of T and cross section area A. When it is subjected to cyclic loading, F(t), the concrete will expand and contract, given by displacement function x(t). The stress is given by dividing the load by the cross section area; the corresponding strain on the material can be found by dividing the displacement by the thickness. For elastic material, Hooke’s law is obeyed; the modulus, E can be related to stress σ(t) and strain ε(t) as:

$\sigma(t) = E\,\varepsilon(t)$ (6.1)

A purely viscous material does not return any of the energy stored during loading. All energy is lost once the load is removed. In this case, the stress is proportional to the rate of the strain, and the ratio of stress to strain is known as viscosity, η. In most cases, linear (Newtonian) viscosity is considered to be the principal form of energy dissipation. These materials have only damping component. Viscoelastic materials (e.g. rubbers, plastics) resemble the pure viscous and elastic materials. Some of the energy stored in a viscoelastic material is recovered upon removal of the load, and the remainder is dissipated in the form of heat. The stress-strain relationship of viscoelastic material is presented by:

$$\sigma(t) = E\,\varepsilon(t) + ¦\,\frac{d\varepsilon}{dt} \quad (6.2)$$

The equation above contains elastic and viscous components; where viscous component contains viscosity of material, η multiplied by time derivative of strain. This term is related to material damping; the ability of material to dissipate energy or absorb vibration.

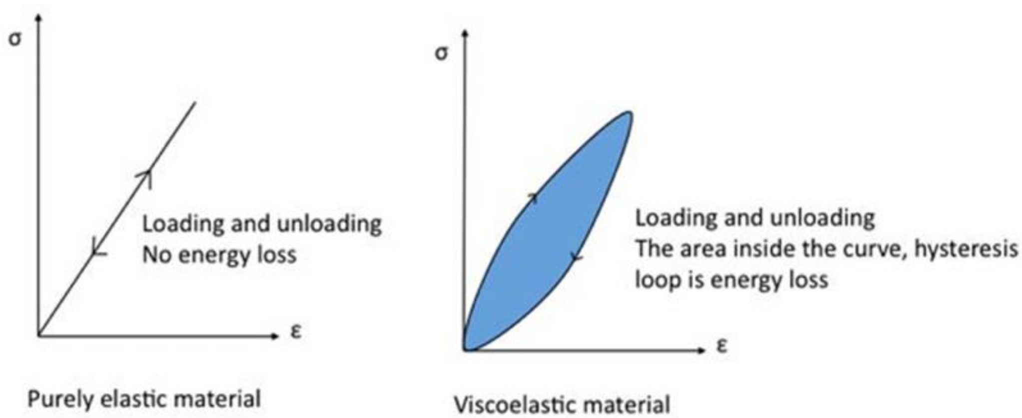

Fig. 6.1 Elastic behavior versus viscoelastic behavior

Stress-Strain Curves for a purely elastic material (a) and a viscoelastic material (b). The blue area is a hysteresis loop and shows the amount of energy lost (as heat) in a loading and unloading cycle. It is equal to $\oint \sigma \, d\varepsilon$, where σ is stress and ε is strain. Unlike purely elastic substances, a viscoelastic substance has an elastic component and a viscous component. The viscosity of a viscoelastic substance gives the substance a strain rate dependent on time. Purely elastic materials do not dissipate energy (heat) when a load is applied, then removed. However, a viscoelastic substance loses energy when a load is applied, then removed. Hysteresis is observed in the stress-strain curve, with the area of the loop being equal to the energy lost during the loading cycle. Since viscosity is the resistance to thermally activated plastic deformation, a viscous material will lose energy through a loading cycle. Plastic deformation results in lost energy, which is uncharacteristic of a purely elastic material's reaction to a loading cycle. Specifically, viscoelasticity is a molecular rearrangement. When a stress is applied to a viscoelastic material such as a polymer, parts of the long polymer chain change position. This movement or rearrangement is called Creep. Polymers remain a solid material even when these parts of their chains are rearranging in order to accompany the stress, and as this occurs, it creates a back stress in the material. When the back stress is the same magnitude as the applied stress, the material no longer creeps. When the original stress is taken away, the accumulated back stresses will cause the polymer to return to its original form. The material creeps, which gives the prefix visco-, and the material fully recovers and also gives the suffix -elasticity.

Linear viscoelasticity is when the function is separable in both creep response and load. All linear viscoelastic models can be represented by a Volterra equation connecting stress and strain:

$$\epsilon(t) = \frac{\sigma(t)}{E_{\text{inst,creep}}} + \int_0^t K(t-t')\dot{\sigma}(t')dt' \qquad (6.3)$$

or

$$\sigma(t) = E_{\text{inst,relax}}\epsilon(t) + \int_0^t F(t-t')\dot{\epsilon}(t')dt' \qquad (6.4)$$

where t is time, $\sigma(t)$ is stress and $\epsilon(t)$ is strain, $E_{\text{inst,creep}}$ and $E_{\text{inst,relax}}$ are instantaneous elastic moduli for creep and relaxation K(t) is the creep function, F(t) is the relaxation function

Linear viscoelasticity is usually applicable only for small deformations.

Nonlinear viscoelasticity is when the function is not separable. It usually happens when the deformations are large or if the material changes its properties under deformations.

An **anelastic** material is a special case of a viscoelastic material: an anelastic material will fully recover to its original state on the removal of load.

Dynamic modulus: Viscoelasticity is studied using dynamic mechanical analysis, applying a small oscillatory stress and measuring the resulting strain.

- Purely elastic materials have stress and strain in phase, so that the response of one caused by the other is immediate.
- In purely viscous materials, strain lags stress by a 90 degree phase lag.
- Viscoelastic materials exhibit behavior somewhere in the middle of these two types of material, exhibiting some lag in strain.

Complex Dynamic modulus (G) can be used to represent the relations between the oscillating stress and strain:

$$G = G' + iG'' \tag{6.5}$$

where $i^2 = -1$; G' is the storage modulus and G'' is the loss modulus:

$$G' = \frac{\sigma_0}{\varepsilon_0} \cos \delta$$

$$G'' = \frac{\sigma_0}{\varepsilon_0} \sin \delta \tag{6.6}$$

where σ_0 and ε_0 are the amplitudes of stress and strain and δ is the phase shift between them.

(a) DAMPING DESIGN

In the design of systems, damping is introduced to achieve a reduced level of vibrations, or to perform vibration suppression. Noise and vibration problems are often generated by the vibrating surfaces of sheet metal components that exhibit low levels of inherent material damping. Source excitation transmitted via an airborne or structure-borne path act to excite dynamic instabilities, or resonances, of these components, often amplifying levels to an excessive amount. This lightly damped, resonant behavior may cause actual high cycle fatigue and failure of critical components or simply result in the presence of unwanted vibration or radiated noise detected by the consumer. When mechanical systems vibrate in a fluid medium such as air, gas, water and oil, the resistance offered by the fluid to the moving body causes energy dissipate. The amount of dissipated energy depends on many factors, such as the size and shape of the vibrating body, the viscosity of the fluid, the frequency of vibration, and the velocity of the vibrating body. In viscous damping, the damping force is proportional to the velocity of the vibrating body.

One method of solution is to increase the level of damping to these sheet metal components by manufacturing them from damped laminate material instead of plain sheet metal. The laminate metal consists of two layers of sheet metal sandwiched together with a thin, energy absorbing viscoelastic core. The reduced vibration in the product is achieved by converting the vibration energy into heat energy dissipated by the viscoelastic material as the part is subject to cyclic oscillation. A typical laminate sheet consists of two rigid metal layers sandwiched together by a polymer, or viscoelastic core adhesive. As the structure is subject to cyclic vibratory stresses, the individual metal layers move relative to each other, resulting in a shearing action imparted to the core material. This polymer core does more than just bond the laminate together. It also provides the mechanism that creates the damping effect, as these shear strains are converted to heat energy within viscoelastic material.

Secondly, the parameters of main vibrating systems can be varied to control the excitation level. Consider a symmetric system of the form

$$M\ddot{x} + D\dot{x} + kx = 0$$

where M, D, and K are the usual symmetric, positive definite mass, damping, and stiffness matrices, to be adjusted so that the modal damping ratios, ξi , have desired values. This in turn provides insight into how to adjust or design the individual elements mi, Di, and ki such that the desired damping ratios are achieved.

Often in the design of a mechanical part, the damping in the structure is specified in terms of either a value for the loss factor or a percentage of critical damping, i.e., the damping ratio. This is mainly true because these are easily understood concepts for a single- degree-of-freedom model of a system. However, in many cases, of course, the behavior of a given structure may not be satisfactorily modeled by a single modal parameter. An n-degree-of-freedom system has n damping ratios, ξı. Recall that, if the equations of motion decouple, then each mode has a damping ratio ξi defined by

$$\xi = \frac{\lambda_i(D)}{2\omega_i} \tag{6.7}$$

where ω_i is the ith undamped natural frequency of the system and $\lambda_i(D)$ denotes the ith eigen value of matrix D.

To formalize this definition and to examine the non-normal mode case ($DM^{-1}K = KM^{-1}D$), the damping ratio matrix, denoted by Z, is defined in terms of the critical damping matrix Dcr. The damping ratio matrix is defined by

$$z = D_{Cr}^{-\frac{1}{2}}\, D\, D_{Cr}^{-\frac{1}{2}} \tag{6.8}$$

where D is the mass normalized damping matrix of the structure. Furthermore, define the matrix Z to be the diagonal matrix of eigenvalues of matrix Z , i.e.,

$$Z = \mathrm{diag}|\lambda_i z| = \mathrm{diag}|\xi^*| \tag{6.9}$$

Here, the ξ^* are damping ratios in that, if $0 < \xi^* < 1$, the system is underdamped. Note, of course, that, if DM −1 K = KM −1 D, then Z = Z'.

By following the definitions of underdamped and critically damped systems, it can easily be shown that the definiteness of the matrix I – Z determines whether a given system oscillates.

Example 6.1 [1]

As an example, consider Equation with the following numerical values for the coefficient matrices:

$$M = \begin{bmatrix} 2 & 0 \\ 0 & 1 \end{bmatrix}, \quad B = \begin{bmatrix} 6 & -\sqrt{2} \\ -\sqrt{2} & 1 \end{bmatrix}, \quad k = -\begin{bmatrix} 10 & -\sqrt{2} \\ -\sqrt{2} & 1 \end{bmatrix}$$

In this case, Dcr is calculated to be

$$D_{Cr} = 2k^{\frac{1}{2}} = \begin{bmatrix} 4.4272 & -0.6325 \\ -0.6325 & 1.8974 \end{bmatrix}$$

where ˜K =M−1/2KM−1/2.

From Equation (6.21) the damping ratio matrix becomes

$$Z = \begin{bmatrix} 0.3592 & -0.2205 \\ -0.2205 & 0.4660 \end{bmatrix}$$

It is clear that the matrix [I − Z] is positive definite, so that each mode in this case should be

underdamped. That is

$$(I - Z) = \begin{bmatrix} 0.3592 & 0.2205 \\ 0.2205 & 0.5340 \end{bmatrix}$$

so that the principle minors become 03592 > 0 and det (I – Z) = 01432 > 0.

Hence, the matrix [I −Z] is positive definite.

Calculating the eigen values of the matrix Z yields $\lambda_1(z)$ = 07906 and $\lambda_2(z)$ = 03162, so that 0<$\lambda_1(z)$ <1 and 0< $\lambda_2(z)$ <1, again predicting that the system is underdamped in each mode, since each ξ^*i is between 0 and 1.

To illustrate the validity of these results for this example, the latent roots of the system can be calculated. They are

$\lambda_{1,2}(z) = -0337 \pm 08326\, j$ $\lambda_{3,4}(z) = -166 \pm 1481\, j$

Where $j = \sqrt{-1}$. Thus, each mode is, in fact, underdamped as predicted by both the damping ratio matrix Z and the modal damping ratio matrix Z'.

It would be a useful design technique to be able to use this defined damping ratio matrix to assign damping ratios to each mode and back-calculate from the matrix Z' to obtain the required damping matrix D. Unfortunately, although the eigenvalues of matrix Z' specify the qualitative behavior of the system, they do not correspond to the actual modal damping ratios unless the matrix DM−1K is symmetric. However, if the damping is proportional, then Equation can be used to calculate the desired damping matrix in terms of the specified damping ratios, i.e.,

$$D = D_{Cr}^{-\frac{1}{2}} z\, D_{Cr}^{-\frac{1}{2}} \tag{6.10}$$

This damping matrix would then yield a system with modal damping ratios exactly as specified.

This section is a prelude to active control where one specifies the desired eigenvalues of a system (i.e., damping ratios and natural frequencies) and then computes a control law to achieve these values. The hardware concerns for achieving the desired damping rates are discussed in Nashif, Jones, and Henderson (1985).

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