AN INVESTIGATION INTO THE DAMPING CAPACITY OF BARS OF DIFFERENT CROSS-SECTIONS UNDER TORSIONAL VIBRATION

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بسم الله الرحمن الرحيم

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SUMMARY

Material damping is one of the most useful sources of damping. Sometimes it is the only source of damping readily available, as in crankshaft and milling machine arbors.

In the design of a machine element subjected to mechanical vibration, consideration is usually given to (1) the peak stress reached anywhere across the element to avoid failure and, (2) the rigidity with the idea not only to minimise deformations due to static forces but also to avoid resonance. A variety of cross-sectional shapes may attain the same values of rigidity and peak stress, and hence the question is which shape should be chosen to attain optimum damping capacity, and the most suitable ratio of the damped energy to the strain energy.

In this investigation the effect of the cross-sectional shape of bars under torsional vibration on the damping capacity is discussed. A variety sections with the same torsional rigidity are considered, including the solid circular, the hollow circulars with inner to outer diameter ratios of 0.7 and 0.9, square and rectangular with width to length ratios of 1:3 and 1:6. Two identically shaped groups of specimens, one from C.I.

and the other from CK-15, are tested.

An apparatus to apply torsional pulses on the specimens is redesigned based on the original design of PERTZ[2]. A testing program is carried out using an optical means of measurement to record the applied torsional pulses and their decaying amplitudes, i.e. the free decay curve.

The apparatus self damping capacity being evaluated by using a test specimen of negligible material damping capacity, a group of curves between the logarithmic decrement (a direct measure of the damping capacity) and the peak stress are obtained for each specimen.

A theoretical investigation is also carried out to verify the experimental results.

On the light of the theoretical and experimental investigation, the following may be derived when the magnetoelastic mechanism of damping domains:-

1) For regions of stresses where the damping capacity is directly proportional to the stress amplitude:

   a) The thinner the section the higher is the damping capacity for same peak stress, for circular section. This agrees with the torsional rigidity principle.

b) If rectangular sections are free from any axial constraints, then the thinner the section the higher is the damping capacity for the same peak stress. If they are constrained to remain planes at the ends, then the thicker the section the higher is the damping capacity for the same peak stress.

c) The solid circular (which has the lowest damping capacity in circular sections) has higher values of damping capacity than the most thin rectangle for the same peak stress.

2- For regions of stresses where the damping capacity is inversely proportional to the stress amplitude:-

a) For circular sections, the thicker the section the higher is the damping capacity for same peak stress.

b) For rectangular sections, the thicker the section the higher is the damping capacity for same peak stress, whether they are constrained at the ends or not.

In these regions the sqaure section has the highest values of damping capacity with respect to all investigated sections.

I - INTRODUCTION

Energy dissipation due to material damping is often one of the most effective means of minimizing dangerous vibrations under resonance operating conditions and, in some cases the only type of damping readily available is the internal damping of the material. The effect of various test variables on the damping properties of a certain machine element is therefore of considerable significance.

One of ~~the~~ these test variables is the cross sectional shape of the machine element subjected to mechanical vibrations. In the design of a machine element the usual procedure is to consider the maximum values of the induced stresses reached and the rigidity of the part as the only two significant factors. A variety of cross-sectional shapes may attain the same values of maximum stress and rigidity, but from the point of view of the energy dissipated in the machine element operating at -or near- resonance conditions, it is strongly desired to know what cross sectional shape should be chosen in order to increase the damping capacity of the member.

In the case of the milling machine arbor, for example one of the most important requirements is to minimize the amplitude of vibration resulting from the cutting forces that act on the milling cutter in order to improve the surface roughness quality. Then for the same material and torsional rigidity (since the design is based on rigidity considerations) what cross sectional shape should be chosen?

In this investigation the effect on damping of varying the cross sectional shape of prismatical bars made of the same material and having the same torsional rigidity is studied.

The chosen sections are the solid circular, the hollow circular with different ratios of inner to outer diameters, the square and the rectangular with different ratios of length to width. Two identically shaped groups of specimens (one from CK-15 and the other from C.I.) are tested.

Although a considerable number of theoretical and experimental investigation has been carried out in the field of mechanical vibrations and damping capacity, yet relatively little attention has so far been given to the problem of studying the effect of cross sectional shapes on the damping properties[1].

The method adopted for measuring the damping capacity in this investigation is the free torsional oscillations method. The instrument used was originally designed by Pertz[2] and modified later by several investigators[3,4,5] and finally redesigned and modified by the author.

A theoretical investigation is also carried out to verify the experimental results.

II - THEORETICAL REVIEW

a) Methods of Measuring Damping Properties in Torsion

1. Stress-strain hysteresis loop: (6,7,8)

By plotting the hysteresis loop point by point, then measuring the enclosed area, which directly represents the energy dissipated in the specimen per cycle.

Due to the very small width of the loop obtained[1], this method necessitates the use of extremely high-sensitivity transducers and recording devises to attain sufficient accuracy for measuring such strains especially at low stresses.

2. Heat measurement in the steady state: (9,10,11)

When a specimen is vibrated under a certain cyclic stress, the energy dissipated in the material will be converted to heat, and the specimen temperature will increase until the rate of energy dissipated becomes equal to the rate of heat lost to the environment by radiation and conduction, and a steady state condition is reached.

This rate of heat loss equals the rate of cooling the specimen directly after stopping the test. Thus by measuring this rate, the energy dissipated in the specimen material at the stated stress per unit time, may be readily obtained.

This method becomes inaccurate with materials having their damping capacity affected by temperature rise, which is usually the case with engineering materials.

3. Measurement of energy input:[12,13,14]

By vibrating the specimen at certain cyclic stress and measuring the energy input to the vibrator per unit time. This energy input assumes a direct measure of the energy dissipated in the specimen at the stated stress when friction in the different parts of the apparatus is comparatively small.

4. Measurement of phase angle[15]:

By vibrating the specimen with sinisoidal loading ~~[illegible]~~, the phase angle (ϕ) by which strain lags stress is a good indication to the damping capacity, where,

$$\tan(\phi) = \Delta/\pi^{(1)} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(II-1)}$$

and Δ = the logarithmic decrement.

5. Measurement of the system resonance curve bandwidth at the half power point:[1,16,17,18]

By plotting the system resonance curve fig. (II-1) point by point noticing the necessity of obtaining steady state condition before measuring the vibration amplitude.

Thus,

$$\text{Logarithmic decrement} = \pi\{(f_1/f_n) - (f_2/f_n)\}^{(1)} \ldots \text{(II-2)}$$

This method fails when :

1. The damping properties of the system change with the stress amplitude,

2. The system behavior is nonlinear.

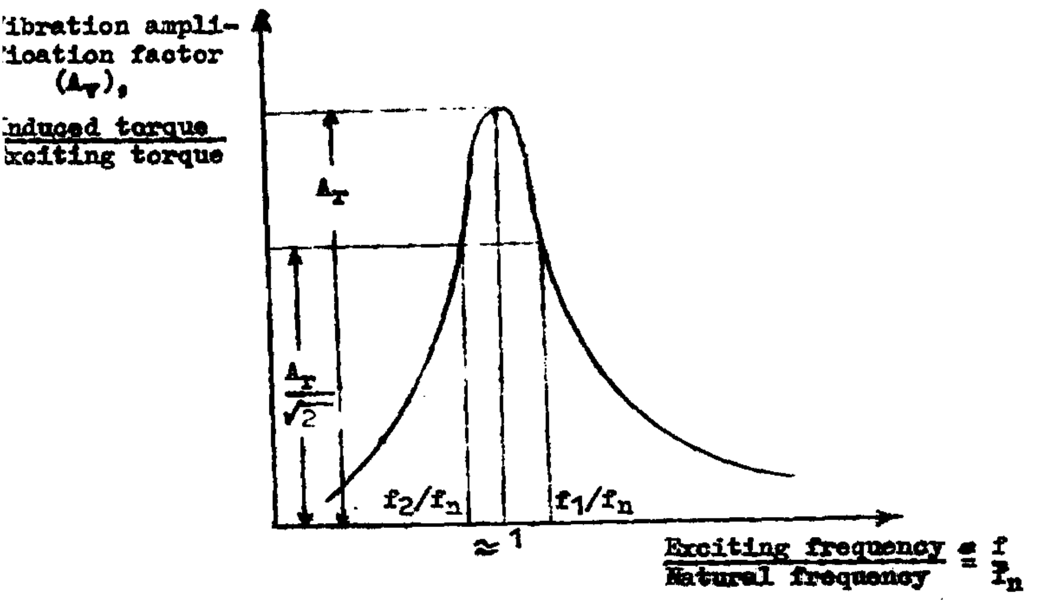

Fig. (II-1), A_r is the vibration amplification factor at resonance.

6. Damped free vibration :

By plotting the free decay curve and measuring the logarithmic decrement directly, which assumes a useful measure of the damping properties of the specimen as will be shown later.

b) Fundamental Relationships: [1]

Two general types of units [19] are used to specify the damping properties of structural materials: 1) The energy dissipated per cycle in a structural element or test speciman, and 2) The ratio of this energy to a reference strain energy or elastic energy. Absolute damping energy units are :

D_o = total damping energy dissipated by the specimen or structural element per cycle of vibration, m.Kg./cycle.

D_a = average damping energy, determined by dividing total damping energy D_o by the volume (V_o) of the specimen or structural element that dissipates energy, m.Kg./m^3/cycle.

D = Specific damping energy which is equal to the area within the stress-strain hystresis loop of the material, m.Kg./m^3/cycle.

Of these absolute damping energy units, the total energy D_o is usually of greatest interest to the engineer. The average damping energy D_a depends upon the shape of the specimen or structural element and upon the nature of the stress distribution in it, even though the specimens are made of the same material and have been subjected to the same stress level at the same temperature and frequency.

The specific damping energy D is the most fundamental of the three absolute units of damping since it depends only on the material in question and not on the shape, stress distribution or volume of the vibrating element. However, most of the methods previously discussed for measuring damping properties yield data on total damping energy D_o rather than on specific damping energy D. Therefore, the development of the relationships between these quantities is necessary.

Relationship between damping energies D_o,D_a and D :

If the specific damping energy is integrated throughout the stressed volume, then,

$$D_o = \int_0^{V_o} D \, dV \quad \text{.........................(III-3)}$$

This is a tripple integral; dV = dx dy dz and D is regarded