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CONSTRAINED LAYER DAMPING - SOME RECENT WORK

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INTRODUCTION

The ever growing requirement for a reduction in noise from all kinds of machinery and from vehicles has renewed interest in sandwich type construction with a viscoelastic damping layer. In the constrained layer construction the damping layer is bonded to the principal structural member and a stiff constraining layer is applied to the outside surface of the damping material. Upon bending, shear strains are induced into the damping layer by the outer constraining layer. As these shear strains are distributed more or less uniformly throughout the damping layer the energy dissipated per cycle of oscillation is usually much greater than for an unconstrained single layer in which there is only a rather non-uniformly distributed extensional strain. It is important to have the constraining layer as rigid as possible by using a material with a higher modulus than the main member.

With a 3-layer construction the optimum effectiveness of the damping is achieved only over a limited frequency range[1,2]. A greater effectiveness can be obtained by incorporating several damping layers each possessing very different properties. This could be a double constrained layer construction which together with the structural member would comprise five layers. A simplification is to eliminate the inner constraining layer to form a 4-layer construction and this can be very effective over a wide range of frequency when the ratio of the shear moduli of the two damping layers is 500 or greater[3].

The viscoelastic damping material should have a high loss factor + 1. To possess a high shear stiffness is usually in conflict with a high loss factor, except when a composite material is used in which embedded fibres[4] or particles[5] act as stiffening elements in a viscoelastic matrix.

DESCRIPTION OF A VISCOELASTIC MATERIAL

Interest in describing precisely the dynamic behaviour of a system with structural damping was revived during the early 1970's[6-9] but it is now generally accepted that viscoelastic damping can be accounted for by introducing complex moduli, in plane strain $E^* = E' + i E''$ and in shear $G^* = G' + i G''$. For most viscoelastic materials the Poisson's ratio is real and hence the ratio of the loss modulus (E'' or G'') to the in-phase component (E' or G') will be the same and equal to the loss factor. However, all the

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moduli and the loss factor are usually dependent upon frequency; for most materials there is an increase in stiffness with frequency. The actual amount of damping inherent in a structure when vibrating at resonance in a certain mode can be characterized by an overall loss factor - η - based on an energy ratio:- $\eta = \Delta E / 2\pi V$
where ΔE = energy dissipated per cycle
V = the maximum elastic strain energy for the mode.

BEAMS

Many parameters have to be considered in the design of a constrained layer beam but, nevertheless, it has been found feasible to give guidelines and design formulae for multi-layer symmetric beams[1] and for certain types of asymmetric constructions[3,10]. Mead[10] has given criteria for assessing the damping effectiveness in relation to the increase in stiffness and in mass. A comprehensive parameter study[11] has been based on the early analysis by Ross et al[12] but it has been pointed out more recently[13] that considerable errors (more than 100%) can be introduced when using the early analysis for thick and stiff damping materials, in particular for the lower modes, because the extensional strains within the damping layer and the effect upon the shear deformation had been neglected.

PLATES

Even more parameters than for beams are required to describe the dynamic response of a sandwich plate. The equations of motion and limited parameter studies, mainly for simply supported symmetric plates, have been given[14-17]. More general solutions, based on finite element methods, have been derived recently[18,19].

SHELLS, RINGS AND TUBES

An early solution[20] for the forced response of a cylindrical sandwich shell was based on the modal analysis approach by Mead[10]. Triangular finite elements have been used by Ioannides[18] for shallow shells. The point impedance of a 3-layer ring has been calculated and verified by experiment[21]. A full analysis for tubes has yet to be published.

PARTIAL CONSTRAINED LAYER COVERAGE

Most of the early work[22-26] was confined to the partial coverage of beams, using a damping tape and with the assumption that the mode shape was as for an undamped beam. Several parameter studies were included[23,25,26]. Kolarik[27] carried out detailed experiments which confirmed in part the analysis for damping tapes[25].

Recently Hamidieh[19] and Sainsbury[28] have applied a finite element technique to compute the dynamic response of beams and plates with partial coverage without any restrictions on the thickness of the layers. Two examples of the parameter study[19] are given; the subscripts refer to a layer, H is thickness, L is length. Table 1 is for the third mode of a simply supported beam. Coverage with three separate parts gives a higher loss factor than with a single piece when the shear modulus (G_2) of the damping layer is high, but not otherwise. The reverse is the case for the first mode. The loss factor for the first mode of a plate with free edges and with varying % coverage is given in Table 2. The best result is with 60% coverage and a high value for G_2. An explanation for some of these effects can be offered and design rules can be given for the advantageous placing of a partial coverage.

REFERENCES

1. J.A. AGBASIERE & P. GROOTENHUIS 1968 JMES 10(3) 269-281.
2. P. GROOTENHUIS 1970 J.Sound Vib. 11 (4) 421-433.
3. B.C. NAKRA & P. GROOTENHUIS 1972 Trans.ASME, J.Eng.Ind. 94 81-86.
4 & 5. S.A. PAIPETIS & P. GROOTENHUIS 1979 Two papers to be publ. in Fibre Sci. & Tech.
6. S.H. CRANDALL 1970 J. Sound Vib. 11 (1) 3-18.
7. R.D. MILNE 1970 J. Sound Vib. 12 (3) 377.
8. D.J. MEAD 1970 Symp. Struct. Dyn., Loughborough 1 paper C5.
9. P. GROOTENHUIS 1972 Proc. Symp. ISVR "Appl. Exp. & Theor. Struct. Dyn."
10. D.J. MEAD 1961 Noise Control 7 (3) 27-38.
11. T.F. DERBY & J.E. RUZICKA 1969 NASA Rep. C.R. 1269.
12. D. ROSS, E.E. UNGAR & E.M. KERWIN 1959 Struct. Damping ASME 49-87.
13. B.C. NAKRA & P. GROOTENHUIS 1974 Aero. Quart. 25 (3) 225-231.
14. S.N. TATANI 1970 Ph.D. thesis Un. Lon. (Imp. Col.).
15. D.J. MEAD 1972 J. Sound Vib. 24 (3) 275-295.
16. Y.V.K.S. RAO & B.C. NAKRA 1974 J. Sound Vib. 34 (3) 309-326.
17. A.K. MUKHOPADHYAY & H.B. KINGSBURY 1976 J. Sound Vib. 47 (3) 347-358.
18. E. IOANNIDES 1975 Ph.D. thesis Un. Lon. (Imp. Col.).
19. Y. HAMIDIEH 1978 Ph.D. thesis Un. Lon. (Imp. Col.).
20. N.P. BIENJEK & A.M. FREUDENTHAL 1962 J. Ae. Sci. 29 (2) 180-184.
21. Y.P. LU, B.E. DOUGLAS & E.V. THOMAS 1973 AIAA Jnl. 11 (3) 300-304.
22. L.J. PULGRANO 1962 64th Meet. ASA, Seattle.
23. E.M. KERWIN, Jr. 1965 ASTM Publ. 378.
24. T.R. YOOS, Jr. & F.C. NELSON 1967 ASME paper 67-vibr-62.
25. P.S. NOKES & F.C. NELSON 1968 Shock Vib. Bul. 38 (3).
26. R. PLUNKETT & C.T. LEE 1970 JASA 48 (1) 150-161.
27. J. KOLARIK 1972 M.Sc. thesis Un. Lon. (Imp. Col.).
28. M.C. SAINSBURY 1976 Ph.D. thesis Un. Lon. (Imp. Col.).

Table 1

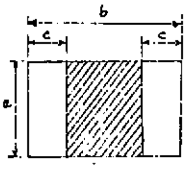
		50% Coverage			75% Coverage		
	G_2'/E_1 =	1.5×10^{-3}	5×10^{-4}	6×10^{-5}	1.5×10^{-3}	5×10^{-4}	6×10^{-5}
	η =	0.09	0.046	0.007	0.134	0.09	0.017
	η =	0.059	0.052	0.013	0.111	0.079	0.019

Third modes of S-S beams excited at the middle.

Layers 1 and 3 are both steel.

$\eta_2 = 0.5$, $H_2/H_1 = 0.5$, $H_3/H_1 = 0.2$, $L/H_1 = 60$.

Table 2

	G_2'/E_1	η (30% Coverage)	η (60% Coverage)	η (90% Coverage)
	5×10^{-4}	0.063	0.128	0.100
	5×10^{-5}	0.013	0.056	0.097

First mode of a F-F-F-F plate excited at the centre. Layers 1 & 2 are steel.

$\eta_2 = 0.5$, $H_2/H_1 = 0.5$, $H_3/H_1 = 0.2$, $H_1 = 10$ mm, $b/a = 2$, $a = 300$ mm.