

FORCED VIBRATION OF A STEEL CANTILEVER BEAM WITH THICK VISCOELASTIC DAMPING LAYER

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Introduction

The control of radiated noise is important for naval applications. DREA has been conducting research on the application of elastomeric materials to anechoic, decoupling and vibration damping tiles for ship hulls and machinery vibration isolation systems. As well, over the past twenty years DREA has developed, in-house and through contract, the general purpose finite element (FE) code VAST [1] for vibration and strength analysis of complicated structures. Recently a direct frequency response method was incorporated in VAST [2, 3] to allow modelling of frequency dependent dynamic mechanical properties. DREA has also developed methods for measuring the dynamic mechanical properties of elastomeric materials in the frequency domain [4].

This paper considers the vibration of a cantilever beam with a thick layer of viscoelastic damping material bonded to one surface. The measured forced response is compared to numerical results obtained using the VAST direct frequency response method in conjunction with measured dynamic mechanical properties for the damping material. Predictions of the composite system loss factor, made using VAST and independently using a code PREDC, are also compared to the experimental data. PREDC is a computer program obtained from University of Dayton, Ohio which employs analytical equations for free and constrained damping treatments of beams and rectangular plates.

The VAST Direct Frequency Response Method

The VAST direct frequency response method assumes a steady state harmonic forcing function and results in the following system of equations for the displacement of nodes in the FE model

$$([K] + j[K'] - \omega^2[M])\{\delta\} = \{F(\omega)\} \quad (1)$$

where $[K]$ and $[K']$ are the real and imaginary parts of the global stiffness matrix (generally frequency dependent), $[M]$ is the mass matrix, $\{F(\omega)\}$ is the complex load vector, $\{\delta\}$ is the complex nodal displacement vector, ω is the forcing frequency and j is $\sqrt{-1}$. This system of equations was solved at each specified forcing frequency to obtain the amplitude and phase for each component of the nodal displacement vector $\{\delta\}$.

The dynamic mechanical properties for groups of elements in the VAST FE model can be represented using a complex Young's modulus $E^*(\omega) = E(\omega)(1 + j\eta(\omega))$ where η is the material damping or loss factor. The frequency dependence of $E(\omega)$ and $\eta(\omega)$ can be specified using tables of frequency weighting values with linear interpolation between values or by using quadratic polynomials over a specified frequency range.

The Cantilever Beam Description

The cantilever beam considered is shown in Figure 1. The steel beam was 9.5 mm thick and clamped between steel blocks at one end. A 27 mm thick layer of EAR Isodamp C-1002 viscoelastic damping material was bonded to the upper surface of the beam. The assumed properties of the steel were Young's

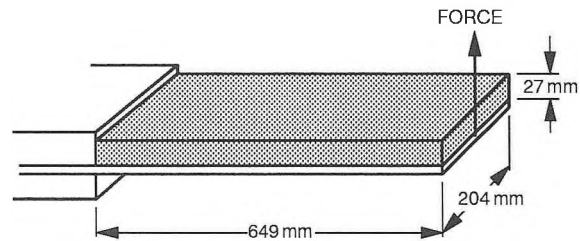


Figure 1: Steel cantilever with 27 mm thick damping layer

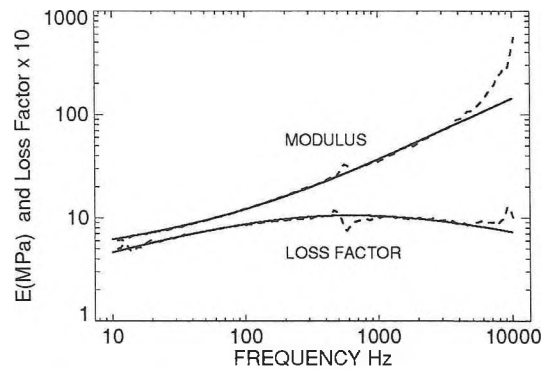


Figure 2: C-1002 properties (— Fitted, --- Measured)

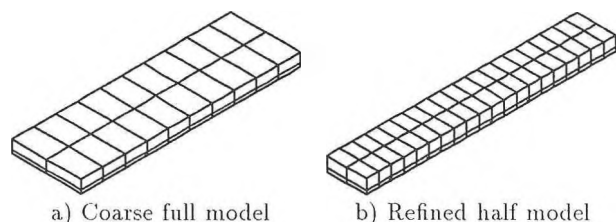
modulus $E = 2.07 \times 10^5$ MPa, Poisson's ratio $\nu = 0.3$ and density of 7870 kg/m^3 . The Young's modulus E and loss factor η for the damping material, shown in Figure 2, were determined using a direct stiffness method [4]. The fitted curves, required for the PREDC program, were also used to generate tables of frequency weighting values for the FE analysis. The density of the damping material was 1280 kg/m^3 .

Viscoelastic damping materials typically have Poisson's ratios near 0.5 (incompressible) in the 'rubbery' region, decreasing through a transition region to a value of 0.3 in the 'glassy region' [6]. The PREDC code assumed a Poisson's ratio of 0.5. The VAST FE code presently cannot consider a truly incompressible material so that a Poisson's ratio of 0.47 was used in the FE analysis.

The Beam Forced Vibration Response

The measured forced response was reported in reference [5] both for the bare beam and the beam with damping layer. A vibration exciter was used to apply a load at the centre-line, 9 mm from the tip, normal to the bottom surface of the steel beam. The applied force was measured with a force transducer and the acceleration of the top surface of the beam measured with an accelerometer placed at several locations along the centre-line.

The forced response of the damped beam was predicted using the direct frequency response method in VAST Version 7.1. Loss factors extracted from the bare beam experiment were used for steel in the FE analysis of the damped beam.



a) Coarse full model b) Refined half model
Figure 3: Finite element meshes of cantilever

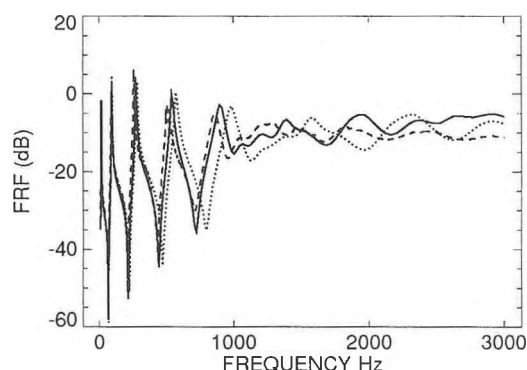


Figure 4: Frequency response at the centre of the tip; (— refined FE, ... coarse FE, - - - measurement)

The FE models were constructed using 20-noded isoparametric solid elements. Two meshes were used; a coarse model of the entire beam and a refined 'half' model taking advantage of symmetry (see Figure 3).

The predicted nodal displacement amplitudes δ (mm) were converted to acceleration amplitudes and reduced to a frequency response function FRF (magnitude in dB) using $FRF = 20 \log(\omega^2 \delta)$ at each forcing frequency ω (radians). The predicted FRF's for the coarse and refined meshes are compared to the measured FRF's in Figure 4. The phase information was also predicted but no experimental measurements were available for comparison. Below 1800 Hz there was reasonably good agreement between the refined FE model result and the experimental curve. The predicted frequency of the first resonant peak was ten percent higher than measured. The next four resonant peaks ranged from four to seven percent higher than measured, consistent with five percent differences obtained between the bare beam experiment and bare beam FE predictions. Over this frequency range the experimental and refined FE response levels differed by 3 dB or less. The predicted FRF levels over the frequency range 1800 to 3000 Hz were up to 6 dB higher than measured.

The composite loss factors predicted with the VAST direct frequency response module are compared to the measured loss factors and PREDC results in Figure 5. There was reasonable agreement between the VAST loss factor predictions and the measured points. Above 1000 Hz there was a significant frequency shift between the curves for the coarse and refined meshes, suggesting that a further refinement of the mesh should be used in this frequency range. The loss factors predicted by the PREDC code were significantly lower than both the measured points and the VAST curves at very low frequencies and at frequencies above 500 Hz. The PREDC code only considers the damping material loss factor. When the bare beam loss factors were neglected in the VAST damped beam analysis, the VAST results were close to the PREDC points in the lower frequency range.

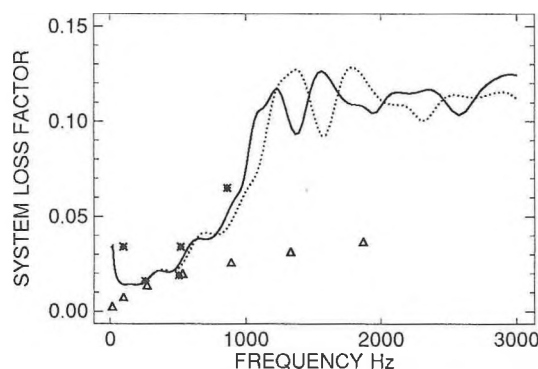


Figure 5: Composite loss factor; (— refined FE, ... coarse FE, * measurement, Δ PREDC)

Concluding Remarks

The VAST finite element predictions of the forced vibration response of the steel cantilever beam with a thick damping layer were in good agreement with experimental results up to 1800 Hz. Above this frequency the analysis indicated that a further refinement of the FE mesh should be considered. The FE analysis also showed that there was significant axial and transverse shear deformation in the damping layer at higher frequencies. Modification of the VAST code, to include frequency dependent values of Poisson's ratio, should improve modelling of the shear properties which may improve response predictions for higher frequencies.

The PREDC code considered only flexural bending. The VAST FE analysis predicted much more complicated wave motion in the damping layer at higher frequencies. This may explain why the measured loss factors and those predicted by VAST were significantly larger than obtained with the PREDC code at forcing frequencies above 500 Hz.

References

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