LEVEL

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Maryland 20084

AD A091284

COMPRESSIONAL DAMPING IN THREE-LAYER BEAMS
INCORPORATING NEARLY INCOMPRESSIBLE
VISCOELASTIC CORES

by

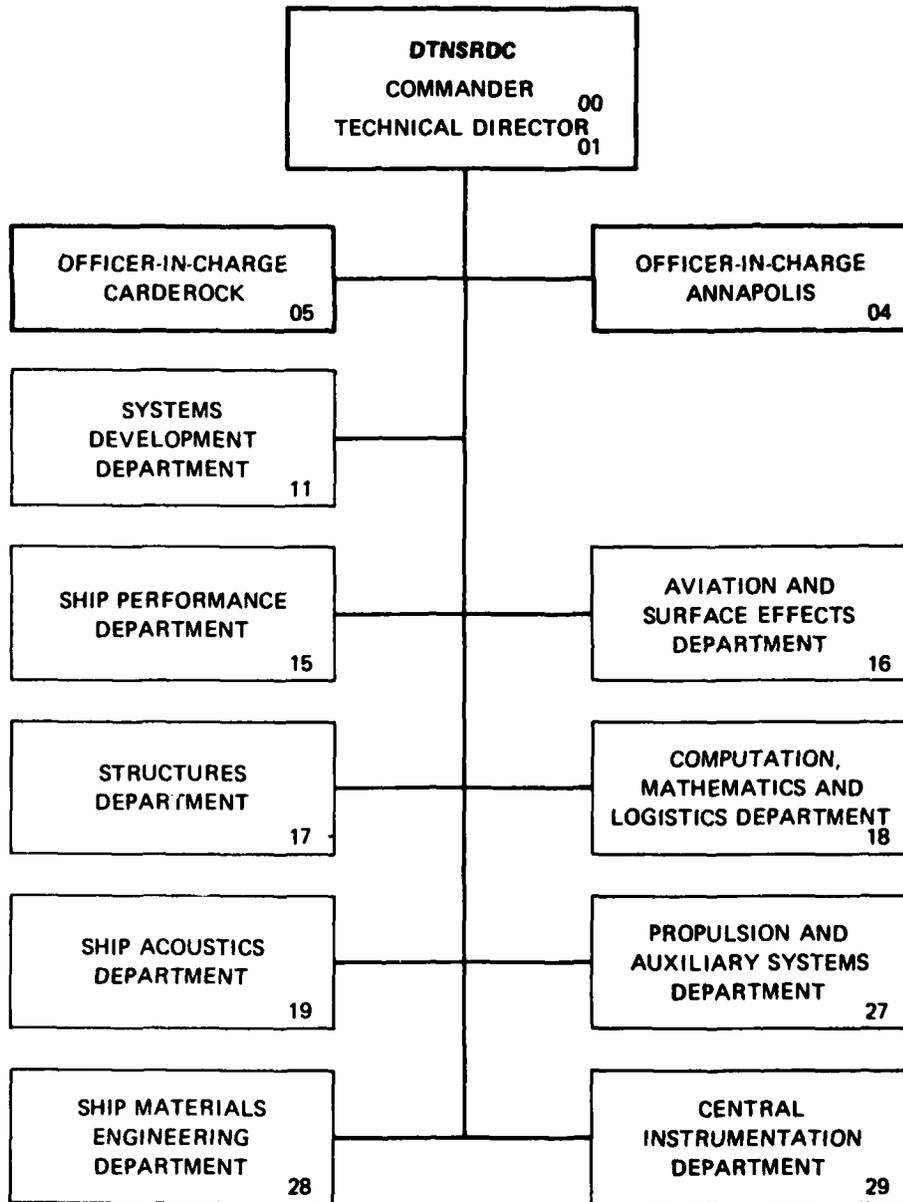Bruce Edward Douglas

DTIC

NOV 6 1980

PROPULSION AND AUXILIARY SYSTEMS DEPARTMENT
RESEARCH AND DEVELOPMENT REPORT

DDC FILE COPY

October 1980   DTNSRDC/PAS-80/26

80 10 30 043

MAJOR DTNSRDC ORGANIZATIONAL COMPONENTS

```
                    ┌─────────────────────────┐
                    │        DTNSRDC          │
                    │      COMMANDER      00  │
                    │  TECHNICAL DIRECTOR 01  │
                    └─────────────────────────┘
```

OFFICER-IN-CHARGE CARDEROCK 05	OFFICER-IN-CHARGE ANNAPOLIS 04
SYSTEMS DEVELOPMENT DEPARTMENT 11	
SHIP PERFORMANCE DEPARTMENT 15	AVIATION AND SURFACE EFFECTS DEPARTMENT 16
STRUCTURES DEPARTMENT 17	COMPUTATION, MATHEMATICS AND LOGISTICS DEPARTMENT 18
SHIP ACOUSTICS DEPARTMENT 19	PROPULSION AND AUXILIARY SYSTEMS DEPARTMENT 27
SHIP MATERIALS ENGINEERING DEPARTMENT 28	CENTRAL INSTRUMENTATION DEPARTMENT 29

16 F43452

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER DTNSRDC/PAS-80/26	2. GOVT ACCESSION NO. AD-A091284	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) COMPRESSIONAL DAMPING IN THREE-LAYER BEAMS INCORPORATING NEARLY INCOMPRESSIBLE VISCOELASTIC CORES.		5. TYPE OF REPORT & PERIOD COVERED Research and Development rept.
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Bruce Edward Douglas		8. CONTRACT OR GRANT NUMBER(s) SF43452702
9. PERFORMING ORGANIZATION NAME AND ADDRESS David W. Taylor Naval Ship R&D Center Bethesda, MD 20084		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS (See reverse side)
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Sea Systems Command (SEA 05H) Washington, DC 20362		12. REPORT DATE October 1980
		13. NUMBER OF PAGES 24
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Structural Damping
Mechanical Impedance
Vibrations
Acoustics

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The transverse free mechanical impedance response of an elastic-viscoelastic-elastic beam incorporating the compressional damping mechanism is considered. The work of Douglas and Yang is extended to include shear deformation and rotary inertia in the elastic layers. The effects of nearly incompressible viscoelastic damping cores on the compressional frequency, and hence on the spectral range of damping effectiveness for the composite,

(Continued on reverse side)

DD FORM 1473 EDITION OF 1 NOV 65 IS OBSOLETE
1 JAN 73
S/N 0102-LF-014-6601

408615

(Block 10)

     Program Element 62543N
     Task Area SF-43-452-702
     Task 18182
     Work Unit 2740-112

(Block 20 continued)

is also discussed. Results of the analysis are shown to compare
favorably with experimental results for a damped three-layer beam
which was optimized for compressional damping and in which the influ-
ence of the shear damping mechanism was intentionally minimized.

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

b	Beam width
E_i	Elastic (Young's) modulus of ith layer
E_v	Elastic storage modulus of viscoelastic core
$E^*(\omega)$	$E_v(1 + i\delta)$ = complex dynamic elastic modulus of viscoelastic core
G_i	Shear modulus of ith layer
I_i	Moment of inertia of the ith layer
i	$\sqrt{-1}$
k^*	Compressional spring constant per unit length
P_o	Applied force
r_i	Radius of gyration for the ith layer
t	Time
t_i	Thickness of ith layer
t_v	Thickness of viscoelastic core
$u(x)$	Longitudinal displacement
$w_i(x)$	Transverse displacement of ith layer
$Z(X,\omega)$	Transverse driving point mechanical impedance of beam at location x
δ	Elastic loss tangent of viscoelastic core
ε_z	Normal unit elongation (strain) component
κ	Timoshenko shear constant = 5/6
ν_v^*	Poisson's ratio of viscoelastic core
ρ_i	Mass per unit length of the ith layer

σ_z Normal stress component

ω Radial frequency

ω_c Compressional composite frequency

ℓ Beam length

ABSTRACT

The transverse free mechanical impedance response of an
elastic-viscoelastic-elastic beam incorporating the compres-
sional damping mechanism is considered. The work of Douglas
and Yang is extended to include shear deformation and rotary
inertia in the elastic layers. The effects of nearly in-
compressible viscoelastic damping cores on the compressional
frequency, and hence on the spectral range of damping effec-
tiveness for the composite, are also discussed. Results of
the analysis are shown to compare favorably with experi-
mental results for a damped three-layer beam which was
optimized for compressional damping and in which the influ-
ence of the shear damping mechanism was intentionally
minimized.

INTRODUCTION

The need to control the resonance response of naval structures and, as a re-
sult, radiated noise has been a prime motivation for research into the dynamic
response of inherently damped structures. To date, much of this research has been
concentrated on constrained layer damping due to the high composite loss factors
attainable. These studies[1-5]* have, for the most part, emphasized the shear damping
mechanism inherent in the composite due to the broad spectral band in which this
mechanism could provide high damping performance. Recently, Douglas and Yang[6]
examined the compressional damping mechanism inherent in elastic-viscoelastic-
elastic beams in the context of the shear damping mechanism. The Mead and Markus
model was shown to provide excellent agreement with experimental results for several
beams optimized to enhance shear damping. However, this paper also concluded that
transverse compressional damping can produce significant dissipation over a rela-
tively narrow spectral band for beams designed to enhance this mechanism.

---

*A complete listing of references is given on page 17.

This report extends the analysis presented by Douglas and Yang[6] to include rotary inertia and shear deformation in the elastic layers. Also, the effects of a nearly incompressible viscoelastic damping core are briefly discussed. The compressional damping mechanism was isolated in this study so that some understanding could be obtained without the complicating influence of other damping mechanisms inherent in the composite.

ANALYTICAL DEVELOPMENT

The effects of shear deformation and rotary inertia on the dynamic response of compressionally damped elastic-viscoelastic-elastic beams are significant in the consideration of high order vibrational modes and for low length to thickness ratio beams. For these cases, the equations developed by Douglas and Yang[6] for the fully constrained damped three-layer laminate must be modified by incorporating these terms into the dynamic response of the elastic face layers. By replacing Bernoulli-Euler beam theory with Timoshenko beam theory, the equations of motion for this system can be expressed for steady-state harmonic response as

$$\frac{\partial^4 w_1}{\partial x^4} - \frac{\rho_1}{E_1}\left(\frac{1}{\kappa}\frac{E_1}{G_1} + 1\right)\frac{\partial^4 w_1}{\partial x^2 \partial t^2} + \frac{\rho_1^2}{\kappa E_1 G_1}\frac{\partial^4 w_1}{\partial t^4}$$

$$+ \frac{\rho_1}{E_1 r_1^2}\frac{\partial^2 w_1}{\partial t^2} = \frac{k^*}{E_1 I_1}(w_3 - w_1)$$

$$\frac{\partial^4 w_3}{\partial x^4} - \frac{\rho_3}{E_3}\left(\frac{1}{\kappa}\frac{E_3}{G_3} + 1\right)\frac{\partial^4 w_3}{\partial x^2 \partial t^2} + \frac{\rho_3^2}{\kappa E_3 G_3}\frac{\partial^4 w_3}{\partial t^4}$$

$$+ \frac{\rho_3}{E_3 r_3^2}\frac{\partial^2 w_3}{\partial t^2} = \frac{k^*}{E_3 I_3}(w_1 - w_3) \tag{1}$$

where the viscoelastic layer is modeled as a distributed complex compression spring. Assuming a separable solution of the form $w_1(x,t) = w_1(x)e^{i\omega t}$ and $w_3(x,t) = w_3(x)e^{i\omega t}$, Equations (1) can be combined into a single eighth-order differential equation of motion expressed in terms of the response of layer 1

$$\frac{d^8 w_1}{dx^8} + (a_1 + b_1) \frac{d^6 w_1}{dx^6} + (a_2 + b_1 a_1 + b_2) \frac{d^4 w_1}{dx^4}$$

$$+ (a_2 b_1 + b_2 a_1) \frac{d^2 w_1}{dx^2} + \left(b_2 a_2 - \frac{k*^2}{E_1 I_1 E_3 I_3} \right) w_1 = 0 \tag{2}$$

where $a_1 = n_1^4 r_1^2 \left(\dfrac{E_1}{\kappa G_1} + 1 \right)$

$a_2 = \dfrac{k*}{E_1 I_1} + n_1^8 r_1^4 \dfrac{E_1}{\kappa G_1} - n_1^4$

$b_1 = n_3^4 r_3^2 \left(\dfrac{E_3}{\kappa G_3} + 1 \right)$

$b_2 = \dfrac{k*}{E_3 I_3} + n_3^8 r_3^4 \dfrac{E_3}{\kappa G_3} - n_3^4$

$n_i^2 = \dfrac{\omega}{r_i} \sqrt{\dfrac{\rho_i}{E_i}}$

3

Employing progressive wave methods, a solution can be written in the form

$$w_1(x) = H_1 e^{\zeta_1 x} + H_2 e^{-\zeta_1 x} + H_3 e^{\zeta_2 x} + H_4 e^{-\zeta_2 x}$$

$$+ H_5 e^{\zeta_3 x} + H_6 e^{-\zeta_3 x} + H_7 e^{\zeta_4 x} + H_8 e^{-\zeta_4 x}$$

and

$$w_3(x) = K_1 \left(H_1 e^{\zeta_1 x} + H_2 e^{-\zeta_1 x} \right) + K_2 \left(H_3 e^{\zeta_2 x} + H_4 e^{-\zeta_2 x} \right)$$

$$+ K_3 \left(H_5 e^{\zeta_3 x} + H_6 e^{-\zeta_3 x} \right) + K_4 \left(H_7 e^{\zeta_4 x} + H_8 e^{-\zeta_4 x} \right) \tag{3}$$

where $\quad K_1 = \dfrac{E_1 I_1}{k*} \left(\zeta_1^{\,4} + a_1 \zeta_1^{\,2} + a_2 \right)$

$$K_2 = \dfrac{E_1 I_1}{k*} \left(\zeta_2^{\,4} + a_1 \zeta_2^{\,2} + a_2 \right)$$

$$K_3 = \dfrac{E_1 I_1}{k*} \left(\zeta_3^{\,4} + a_1 \zeta_3^{\,2} + a_2 \right)$$

4

and

$$K_4 = \frac{E_1 I_1}{k^*} \left(\zeta_4{}^4 + a_1 \zeta_4{}^2 + a_2 \right)$$

The complex flexural wave numbers ζ_1, ζ_2, ζ_3, and ζ_4 can be determined from differential operator methods. Such methods reduce Equation (2) to solving the equivalent general quartic (biquadratic) algebraic equation. A solution to the general quartic equation following Lagrange's method of symmetric functions yields the desired complex flexural wave numbers in closed form

$$\zeta_1 = \left[\left(-a_1 - b_1 + \theta_1 + \theta_2 + \theta_3 \right) /4 \right]^{1/2}$$

$$\zeta_2 = \left[\left(-a_1 - b_1 + \theta_1 - \theta_2 - \theta_3 \right) /4 \right]^{1/2}$$

$$\zeta_3 = \left[\left(-a_1 - b_1 - \theta_1 + \theta_2 - \theta_3 \right) /4 \right]^{1/2}$$

and

$$\zeta_4 = \left[\left(-a_1 - b_1 - \theta_1 - \theta_2 + \theta_3 \right) /4 \right]^{1/2}$$

5

where $\theta_1 = \left[y_1 + y_2 - \dfrac{f}{3} \right]^{1/2}$; $\qquad \theta_2 = \left[\dfrac{y_1 + y_2}{2} + i\sqrt{3}\, \dfrac{y_1 - y_2}{2} - \dfrac{f}{3} \right]^{1/2}$

$$\theta_3 = \left[\dfrac{y_1 + y_2}{2} - i\sqrt{3}\, \dfrac{y_1 - y_2}{2} - \dfrac{f}{3} \right]^{1/2}$$

$$y_1 = \left[-\dfrac{p}{2} + \left(\dfrac{p^2}{4} - \dfrac{q^3}{27} \right)^{1/2} \right]^{1/3} ; \qquad y_2 = (-q/3y_1)$$

$$p = \dfrac{1}{27} \left[+ 2f^3 - 9fh - 27k \right]; \qquad q = h - \dfrac{f^2}{3}$$

$$f = 8c_2 - 3c_1^{\,2}$$

$$h = 3c_1^{\,4} - 16c_1^{\,2}c_2 + 16c_1c_3 + 16c_2^{\,2} - 64c_4$$

$$k = \left(c_1^{\,3} - 4c_1c_2 + 8c_3 \right)^2$$

$$c_1 = a_1 + b_1; \qquad c_2 = a_2 + a_1b_1 + b_2$$

$$c_3 = b_1a_2 + b_2a_1; \qquad c_4 = a_2b_2 - \dfrac{k^{*2}}{E_1I_1E_3I_3}$$

6

To facilitate comparison with experiment, cantilever end conditions were selected. The equations of constraint for a cantilever beam driven on the free end require zero bending moment at the free end of elastic face layers or

$$\chi_1 = -E_1 I_1 \left(\frac{d^2 w_1}{dx^2} + \frac{\omega^2 \rho_1}{\kappa G_1} w_1 \right) = 0 \tag{4}$$

and a shear force S equal to the applied concentrated sinusoidal force or

$$S_1 = \kappa b t_1 G_1 \left[(1 - \sigma_1) \frac{\partial w_1}{\partial x} + \psi_1 \frac{\partial^3 w_1}{\partial x^3} \right] = P_o e^{i\omega t} \tag{5}$$

where

$$\sigma_1 = \frac{\left(\dfrac{r_1^2 E_1}{\kappa G_1} \right)^2 (n_1)^4 + 1}{\left[1 - \dfrac{n_1^4 r_1^4 E_1}{\kappa G_1} \right]}$$

and

$$\psi_1 = \frac{r_1^2 E_1}{\kappa G \left[1 - \dfrac{n_1^4 r_1^4 E_1}{\kappa G_1} \right]}$$

7

The boundary conditions on layer 1 at the fixed end constrain motion such that

$$w_1(\ell) = 0 \quad \text{and} \quad \left(\frac{dw_1}{dx}\right)_{x=\ell} = 0 \tag{6}$$

The free-free boundary conditions on the constraining layer (layer 3) required zero bending moment and zero shear force at $x = 0$ and $x = \ell$ so that

$$\left. \begin{array}{ll} \chi_3(0) = 0 & \chi_3(\ell) = 0 \\ \\ S_3(0) = 0 & S_3(\ell) = 0 \end{array} \right\} \tag{7}$$

These eight constraint equations can be placed in matrix form, Equation (9), and the wave amplitude coefficients H_i can be obtained from inverting the constraint matrix and multiplying by the loading matrix. The quantities P_i and Q_i in the constraint matrix are defined by the relations

$$P_n = \kappa b t_1 G_1 \left[(1 - \sigma_1) \zeta_i + \psi_1 \zeta_i^{\,3} \right]$$

$$Q_n = (- E_1 I_1) \left(\zeta_i^{\,2} + \frac{\omega^2 \rho_1}{\kappa G_1} \right)$$

for n = 1 to 4

and

8

$$P_n = \kappa b t_3 G_3 \left[(1 - \sigma_3)\, \zeta_1 + \psi_3 \zeta_1^{\,3}\right]$$

$$Q_n = (-\,E_3 I_3) \left(\zeta_1^{\,2} + \frac{\omega^2 \rho_3}{\kappa G_3}\right)$$

for n = 5 to 8 (8)

$$
\begin{bmatrix}
P_1 & -P_1 & P_2 & -P_2 & P_3 & -P_3 & P_4 & -P_4 \\
Q_1 & Q_1 & Q_2 & Q_2 & Q_3 & Q_3 & Q_4 & Q_4 \\
e^{\zeta_1 \ell} & e^{-\zeta_1 \ell} & e^{\zeta_2 \ell} & e^{-\zeta_2 \ell} & e^{\zeta_3 \ell} & e^{-\zeta_3 \ell} & e^{\zeta_4 \ell} & e^{-\zeta_4 \ell} \\
\zeta_1 e^{\zeta_1 \ell} & -\zeta_1 e^{-\zeta_1 \ell} & \zeta_2 e^{\zeta_2 \ell} & -\zeta_2 e^{-\zeta_2 \ell} & \zeta_3 e^{\zeta_3 \ell} & -\zeta_3 e^{-\zeta_3 \ell} & \zeta_4 e^{\zeta_4 \ell} & -\zeta_4 e^{-\zeta_4 \ell} \\
P_5 K_1 & -P_5 K_1 & P_6 K_2 & -P_6 K_2 & P_7 K_3 & -P_7 K_3 & P_8 K_4 & -P_8 K_4 \\
Q_5 K_1 & Q_5 K_1 & Q_6 K_2 & Q_6 K_2 & Q_7 K_3 & Q_7 K_3 & Q_8 K_4 & Q_8 K_4 \\
Q_5 K_1 e^{\zeta_1 \ell} & Q_5 K_1 e^{-\zeta_1 \ell} & Q_6 K_2 e^{\zeta_2 \ell} & Q_6 K_2 e^{-\zeta_2 \ell} & Q_7 K_3 e^{\zeta_3 \ell} & Q_7 K_3 e^{-\zeta_3 \ell} & Q_8 K_4 e^{\zeta_4 \ell} & Q_8 K_4 e^{-\zeta_4 \ell} \\
P_5 K_1 e^{\zeta_1 \ell} & -P_5 K_1 e^{-\zeta_1 \ell} & P_6 K_2 e^{\zeta_2 \ell} & -P_6 K_2 e^{-\zeta_2 \ell} & P_7 K_3 e^{\zeta_3 \ell} & -P_7 K_3 e^{-\zeta_3 \ell} & P_8 K_4 e^{\zeta_4 \ell} & -P_8 K_4 e^{-\zeta_4 \ell}
\end{bmatrix}
\begin{bmatrix}
H_1 \\ H_2 \\ H_3 \\ H_4 \\ H_5 \\ H_6 \\ H_7 \\ H_8
\end{bmatrix}
=
\begin{bmatrix}
P_o \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0
\end{bmatrix}
\qquad (9)
$$

The expressions for the mechanical impedance of this system can be calculated from the expressions

$$Z_1(x,\omega) = \frac{P_o}{i\omega} \left[H_1 e^{\zeta_1 x} + H_2 e^{-\zeta_1 x} + H_3 e^{\zeta_2 x} \right.$$

$$\left. + H_4 e^{-\zeta_2 x} + H_5 e^{\zeta_3 x} + H_6 e^{-\zeta_3 x} + H_7 e^{\zeta_4 x} + H_8 e^{-\zeta_4 x} \right]^{-1}$$

$$Z_3(x,\omega) = \frac{P_o}{i\omega} \left[K_1 \left(H_1 r^{\zeta_1 x} + H_2 e^{-\zeta_1 x} \right) \right.$$

$$+ K_2 \left(H_3 e^{\zeta_2 x} + H_4 e^{-\zeta_2 x} \right)$$

$$+ K_3 \left(H_5 e^{\zeta_3 x} + H_6 e^{-\zeta_3 x} \right)$$

$$\left. + K_4 \left(H_7 e^{\zeta_4 x} + H_8 e^{-\zeta_4 x} \right) \right]^{-1} \tag{10}$$

Intuitively, one would expect that the composite beam modes with eigenfrequencies in the spectral vicinity of the delamination frequency of the composite would experience the highest compressional damping since the viscoelastic layer receives the greatest dynamic compression strains. Thus, if the elastic layers are treated

as lumped masses and the viscoelastic layer as a distributed complex spring, then this model gives rise to a compressional frequency working to delaminate the face layers at

$$\omega_c^* = \left[\left(\frac{E_v^*}{t_v} \right) \left(\frac{1}{\rho_1 t_1} + \frac{1}{\rho_3 t_3} \right) \right]^{1/2} = \omega_c' \, (1 + i\delta)^{1/2} \tag{11}$$

From this result, compressional damping can be characterized as a narrowband phenomenon spectrally centered near ω_c with a bandwidth dependent upon the elastic loss tangent of the viscoelastic layer.

NEARLY INCOMPRESSIBLE VISCOELASTIC DAMPING CORES

For the case where the viscoelastic damping core is nearly incompressible (i.e., $\nu_v^* \to 1/2$), the expression for the compressional frequency of the composite must be suitably modified to account for shape and the complex Poisson's ratio of the core. This modification can take the form of replacing the complex modulus of elasticity $E_v^*(\omega)$ with the concept of the apparent complex modulus of elasticity $\tilde{E}_v^*(\omega)$. An expression for $\tilde{E}_v^*(\omega)$ can be easily derived from the principle of correspondence for harmonic motion and consideration of Hooke's law applied to an elastic layer sandwiched between two rigid plates such that lateral strain is prevented.

$$\frac{\sigma_z}{\varepsilon_z} \equiv \tilde{E}_v^*(\omega) = E_v^* \left[1 - \nu_v^* \right] \Big/ \left[\left(1 - 2\nu_v^* \right) \left(1 + \nu_v^* \right) \right] \tag{12}$$

From this expression, it is apparent that the compressional frequency of a three-layer damped beam with a thin relatively incompressible viscoelastic layer and low thickness to width ratio will be significantly affected by Poisson's ratio of the core. Figure 1 illustrates the dependence of the compressional frequency of the composite as well as the apparent modulus of elasticity on Poisson's ratio of the
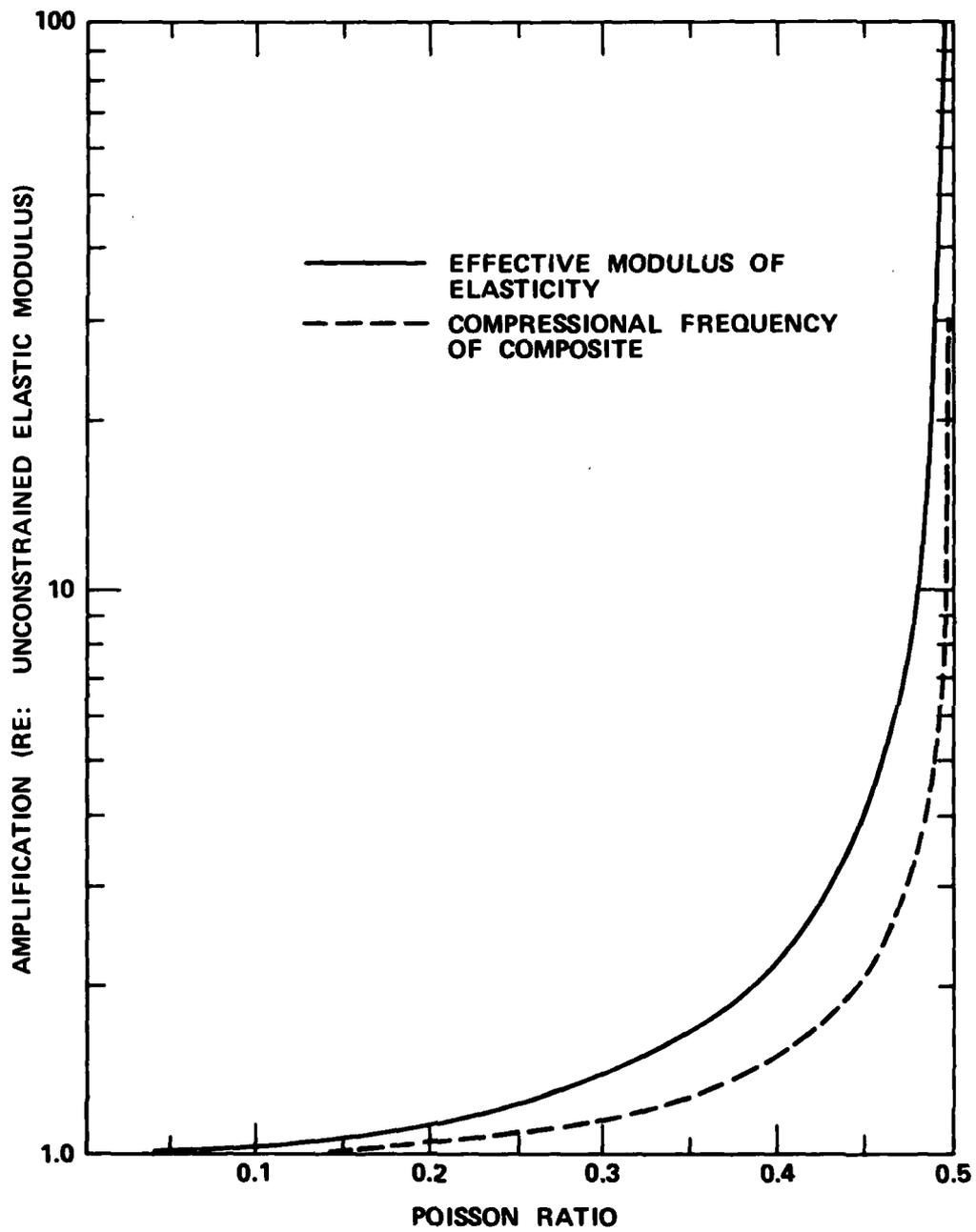
11

Figure 1 - Effective Modulus of Elasticity as a Function
of Poisson's Ratio of the Viscoelastic Core

core assuming that Young's loss tangent is equal to the shear loss tangent (i.e., a real Poisson's ratio). Examination of Figure 1 indicates that the compressional frequency of such a composite is highly sensitive to the value of Poisson's ratio of the core in the range from 0.45 to 0.5 to a degree that such a composite may provide a method of evaluation for Poisson's ratio of viscoelastic materials.

For damped three-layer beams with relatively thick rubberlike damping cores, the apparent complex modulus of elasticity can be written[7-9] for rectangular beams as

$$\tilde{E}^*_v(\omega) = E^*_v(\omega) \left[\left(4 + \frac{\beta b^2}{2t_v^2} \right) / 3 \right] \tag{13}$$

where β is a material parameter (e.g., $\beta = 2$ for rubber cores unfilled by carbon black).

DISCUSSION AND CONCLUSIONS

Using the experimental results reported by Douglas and Yang,[6] a comparison of the analysis presented in this paper with both experiment and a compressional damping model incorporating a Bernoulli-Euler model for the face layers was made in a free mechanical impedance format (Figure 2). The damping core used for this specimen was an acrylic base viscoelastic material whose complex elastic modulus could be functionally approximated at the test temperature (22°C) by the expression

$$E_v = 4.26 \times 10^5 \; \exp[0.494 \; \ln \; (\omega/2\pi)] N/m^2 \tag{14}$$
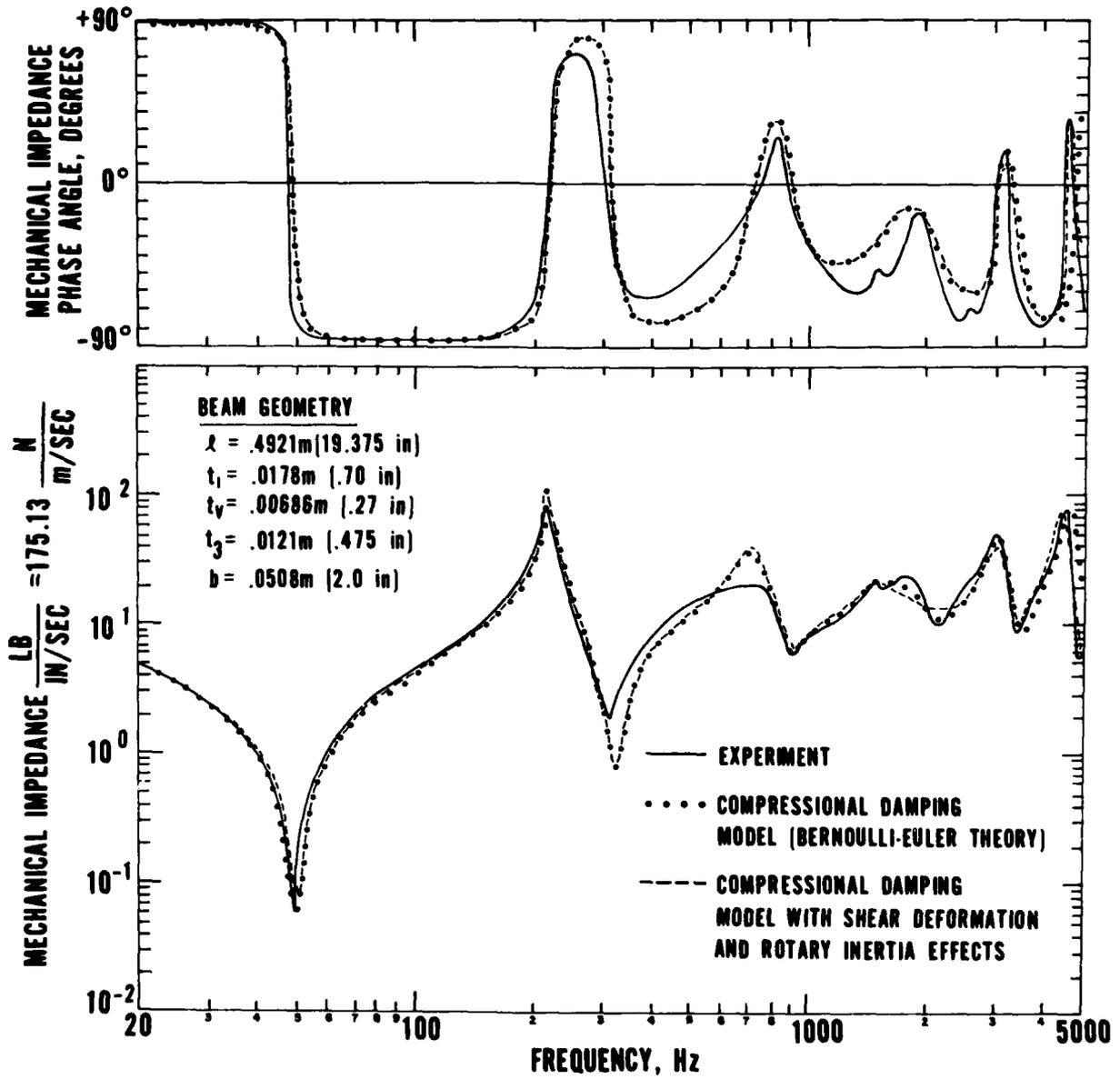
$$\delta = 1.46$$

Figure 2 – Transverse Driving Point Free Mechanical Impedance
and Phase Angle Spectrum for an Elastic-
Viscoelastic-Elastic Beam

Excellent agreement was observed between both analytical models and the experimental results throughout the spectral range of interest. The shear damping model showed a good prediction of the resonance and antiresonance frequencies of the composite but also indicated composite loss factors ranging from less than 0.01 to 0.02, thus indicating that this mechanism is not dominant in the spectral range reported for this specimen. It should be noted that this specimen was intentionally designed to optimize the compressional damping mechanism near 1 kHz and that such a design results in low shear damping.

For most problems where broadband damping is required, such as controlling piping or fan duct vibrations, shear damping remains as the most practical dissipation mechanism to utilize in constrained layer composites. However, for structural response problems where narrowband vibrational energy is the overriding consideration, transverse compressional damping can achieve composite loss factors approaching the elastic (Young's) loss factor of the damping core if suitable attention is paid to Poisson's ratio and shape factor of the core. Unfortunately for most commercially available viscoelastic materials, it is difficult to simultaneously optimize the composite for both shear and compressional damping, especially at low frequencies.

REFERENCES

1. Kerwin, E. M., Jr., "Damping of Flexural Waves by a Constrained Visco-Elastic Layer," Journal of the Acoustical Society of America, Vol. 31, No. 7 (Jul 1959).

2. DiTaranto, R. A., "Theory of Vibratory Bending for Elastic and Visco-elastic Layered Finite-Length Beams," Journal of Applied Mechanics, Vol. 32, Trans. ASME, Vol. 87, Series E (1965).

3. Mead, D. J. and S. Markus, "The Forced Vibration of a Three-Layer, Damped Sandwich Beam with Arbitrary Boundary Conditions," Journal of Sound and Vibration, Vol. 10, No. 2 (1969).

4. Mead, D. J., "Governing Equations for Vibrating Constrained-Layer Damping Sandwich Plates and Beams," Journal of Applied Mechanics, Vol. 40 (1973).

5. Lu, Y. P. and B. E. Douglas, "On the Forced Vibrations of Three-Layer Damped Sandwich Beams," Journal of Sound and Vibration, Vol. 32, No. 4 (1974).

6. Douglas, B. E. and J. C. S. Yang, "Transverse Compressional Damping in the Vibratory Response of Elastic-Viscoelastic-Elastic Beams," AIAA, Vol. 16 (Sep 1978).

7. Gent, A. N. and P. B. Lindley, Proc. Inst. Mech. Engrs., Vol. 173 (1959).

8. Snowdon, J. C., "Handbook of Vibration and Noise Control," Applied Research Lab Tech Memo 79-75 (Apr 1979).

9. Lindley, P. B., "Engineering Design with Natural Rubber," Natural Rubber Producers Research Assoc., 3rd Ed. (1970).

INITIAL DISTRIBUTION

Copies				CENTER DISTRIBUTION	

Copies			Copies	Code	Name
2	CHONR				
	1 Dr. Sykes				
	1 Dr. Perrone		1	19	W. Brown
1	U.S. Naval Academy/Library		2	1942	J. Chen/K. Lewis
1	NOSC/R. Gales		1	196	D. Feit
12	NAVSEA		1	2702	W. Levedahl
	1 SEA 05D				
	2 SEA 05H		2	274	L. J. Argiro
	1 SEA 08				
	1 SEA 322		1	2740	Y. F. Wang
	1 PMS 393				
	1 PMS 396		1	2742	D. E. Goldsmith
	2 SEA 921N				
	1 SEA 5213/R. Taddeo		7	2742	B. E. Douglas
	2 SEA 99612				
1	NAVSSES PHILADELPHIA PA/Lib		1	522.1	Unclassified Library
3	SUPSHIPS		2	5231	Office Services
	1 Groton				
	1 Newport News				
	1 Pascagoula				
12	DTIC				
1	Brown Univ.				
	School of Engineering				
1	UCLA				
	Mechanical Engineering Dept.				

DTNSRDC ISSUES THREE TYPES OF REPORTS

1. DTNSRDC REPORTS, A FORMAL SERIES, CONTAIN INFORMATION OF PERMANENT TECH-
NICAL VALUE. THEY CARRY A CONSECUTIVE NUMERICAL IDENTIFICATION REGARDLESS OF
THEIR CLASSIFICATION OR THE ORIGINATING DEPARTMENT.

2. DEPARTMENTAL REPORTS, A SEMIFORMAL SERIES, CONTAIN INFORMATION OF A PRELIM-
INARY, TEMPORARY, OR PROPRIETARY NATURE OR OF LIMITED INTEREST OR SIGNIFICANCE.
THEY CARRY A DEPARTMENTAL ALPHANUMERICAL IDENTIFICATION.

3. TECHNICAL MEMORANDA, AN INFORMAL SERIES, CONTAIN TECHNICAL DOCUMENTATION
OF LIMITED USE AND INTEREST. THEY ARE PRIMARILY WORKING PAPERS INTENDED FOR IN-
TERNAL USE. THEY CARRY AN IDENTIFYING NUMBER WHICH INDICATES THEIR TYPE AND THE
NUMERICAL CODE OF THE ORIGINATING DEPARTMENT. ANY DISTRIBUTION OUTSIDE DTNSRDC
MUST BE APPROVED BY THE HEAD OF THE ORIGINATING DEPARTMENT ON A CASE-BY-CASE
BASIS.

2-8