Modeling of Constrained Layer Damping in Trusses

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ABSTRACT

This work reexamines the use of constrained layer damping for controlling the lower modes of vibration of a large flexible truss. The device used to suppress vibrations is a "viscoelastic strut". The design, modeling and experimental verification of the viscoelastic strut as well as the three longeron truss with and without the constrained layer treatment is presented. A complete finite element analysis (FEA) and experimental verification will be presented. The experimental results are represented by examining transfer functions and modal damping ratios as well as a video presentation of the effectiveness of the viscoelastic approach for vibration suppression.

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INTRODUCTION

Viscoelastic damping of large space structures shows promise of providing high loss factors with low cost in additional weight and no moving parts. Although many methods have been derived which are capable of determining the damping matrix of a structure after collecting response data, prediction of non viscous damping is still quite elusive. For large space structures which cannot be ground tested, such as the space station, accurate prediction of damping is required before optimal vibration control may be implemented. The modal strain energy technique is often used to predict damping in lightly damped structures with real modes, however, the introduction of highly damped struts may make the assumption of real modes improper. The Golla-Hughes-McTavish (GHM) model for viscoelastic damping is a finite element based method which models the frequency dependant complex modulus of materials and is fully compatible with the usual linear second-order equations of motion most commonly used to model structure dynamics. In order to test this theory, a highly damped viscoelastic strut has been constructed and tested. Loss factors of the viscoelastic strut are also determined using the concept of modal strain energy. Finally, the GHM method is used to predict the equivalent modal damping ratios of a test bed and a comparison is made between experimental and predicted response when the viscoelastic strut is placed in the structure.

STRUT DESIGN

The viscoelastic strut design goals were to provide a strut which would not significantly change the natural frequencies of the test structure (i.e.. the real part of the complex stiffness should be of the same order as the undamped strut's stiffness between 10Hz and 100Hz) and to create a strut which had the typical viscoelastic characteristic of frequency dependant complex stiffness. Since proving a strut with a high stiffness could still significantly increase the damping in the structure and modeling this effect were the primary concerns, creep was not considered in the design. Tests on the actual structure showed that creep was not significant over a period of more than one week while our dynamic tests lasted only minutes, verifying our assumption that for our purposes creep was not an issue.

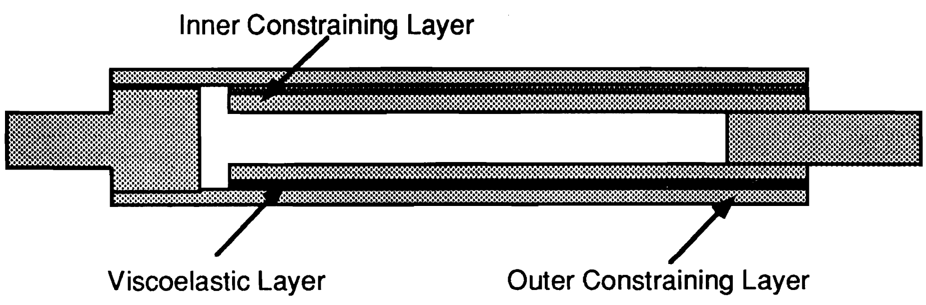

FIGURE 1: A schematic of the Viscoelastic Strut design illustrating the constrained layer configuration relative to the strut.

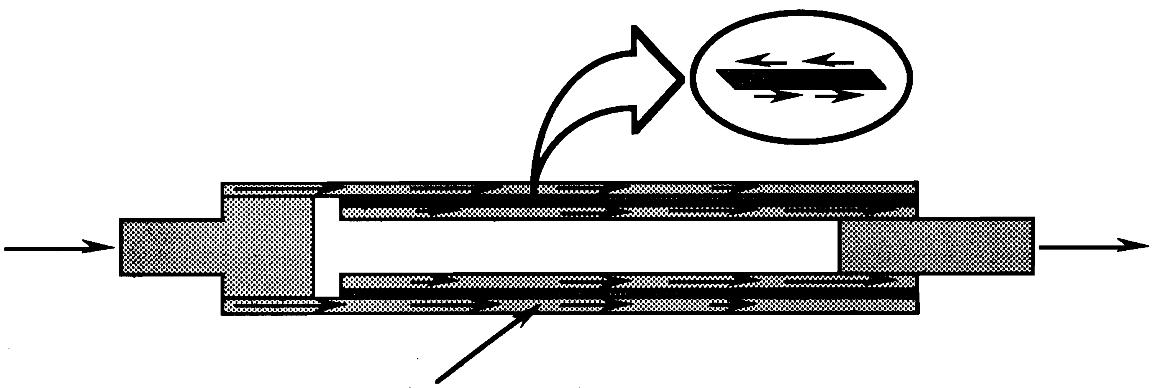

FIGURE 2: A schematic of the Viscoelastic Strut indicating the stress flow path.

The concept of the viscoelastic strut is simple in that an outer and inner layer act to shear a constrained viscoelastic layer when the strut is placed in tension or compression (Figure 1). The constraining layers are T-6061 aluminum on the order of .05 inches while the viscoelastic layer is .005 inch thick Scotchdamp™ SJ-2015X Type 112 viscoelastic material. Due to the fact that the viscoelastic layer is so thin the majority of the deflection takes place in the constraining layers. However, the high loss factor of the viscoelastic material maintains a high overall loss factor for the strut in the frequency range of interest.

STRUT "LOCAL" MODEL

The strut was modeled using discretized extensional springs for the constraining layer and discretized shear springs for the viscoelastic layer. With one end of the strut in a clamped condition, the model used was a 29 degree of freedom model similar to the one shown in Figure 3. The variables K_i and K_o are the discretized inner and outer stiffnesses and K_v is the discretized stiffness of the viscoelastic layer. Note that this model neglects shear deformation in the constraining layers as well as other less significant effects. The modulus $G'(\omega)$ for the viscoelastic layer was taken from manufacturer produced charts at room temperature. Due to the dependance of the complex shear modulus of the viscoelastic layer on frequency, K_v is also frequency dependant. At selected intervals, the stiffness of the strut and the percent strain energy in the viscoelastic layer relative to the strain energy in the entire strut were found using the shear modulus of the viscoelastic layer at that frequency. Using the principle of modal strain energy, the loss factor of the strut at a given frequency is given by

$$\eta(\omega)=\eta_v(\omega)\,\frac{V_v(\omega)}{V(\omega)}$$

where $\frac{V_v(\omega)}{V(\omega)}$ is the fraction of elastic strain energy in the viscoelastic layer at the frequency ω and $\eta_v(\omega)$ is the loss factor of the viscoelastic material.

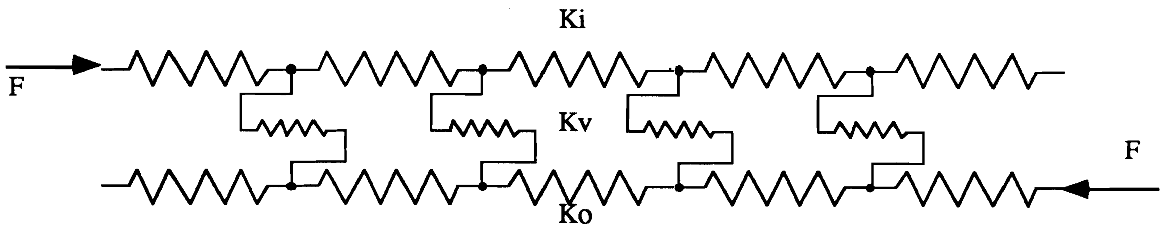

FIGURE 3: A schematic of the discretized spring model of Viscoelastic Strut of Figures 1 and 2.

These calculations give a complete characterization of the complex stiffness of the viscoelastic strut over the frequency range of 1-100 Hz.

THE GHM VISCOELASTIC MODEL FOR A MASSLESS ROD

The GHM viscoelastic model is a linear transfer function method for modeling frequency dependant complex modulus. Curve fitting the GHM transfer function to complex material data over a frequency range of interest creates a linear model which is compatible with standard finite elements. Using transformations described by McTavish[2] viscoelastic finite elements can be easily derived. The GHM transfer function is

$$K(s) = \frac{F(s)}{X(s)} = \hat{K}^0 \left[1 + \sum_{n=1}^{k} \hat{\alpha}_n \frac{s^2+2\hat{\zeta}_n\hat{\omega}_n s}{s^2+2\hat{\zeta}_n\hat{\omega}_n s+\hat{\omega}^2_n} \right]$$

where the hatted variables $\hat{\alpha}$, $\hat{\zeta}$, $\hat{\omega}$, and $\hat{K}^0$ are free variables for curve fitting (The transfer function is shown in terms of the rod stiffness. The references show the more general case for a material modulus). The number of terms in the expansion is dependant on the accuracy of the curve fitting desired, the frequency range size, and the degree of frequency dependance of the viscoelastic material. For a viscoelastic material which does not exhibit creep, $\hat{K}^0$ would be the static stiffness of the strut. However, since no static stiffness is assumed, this variable is also free for curve fitting. The linear second order matrix realization of this transfer function for a rod with k=1 is

$$\tilde{K} = \hat{K}^0 \begin{bmatrix} 1+\hat{\alpha} & -(1+\hat{\alpha}) & \hat{\alpha}\sqrt{2} \\ -(1+\hat{\alpha}) & 1+\hat{\alpha} & -\hat{\alpha}\sqrt{2} \\ \hat{\alpha}\sqrt{2} & -\hat{\alpha}\sqrt{2} & \hat{\alpha}\,2 \end{bmatrix} \qquad \tilde{D} = \hat{K}^0 \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{4\,\hat{\alpha}\hat{\zeta}}{\hat{\omega}} \end{bmatrix} \qquad \tilde{M} = \hat{K}^0 \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{2\,\hat{\alpha}}{\hat{\omega}^2} \end{bmatrix}$$

Setting $\hat{\alpha} = 0$ in these matrices reduces them to the ordinary finite element rod matrices except for the third coordinate. The third coordinate represents a dissipation coordinate which has no physical significance except perhaps as a state estimator. The static properties of the strut element are not effected by this coordinate. These additional coordinates add overdamped 'false' modes to the model which are easily identifiable, as demonstrated in references 1 and 2.The curve fitting was done on the complex stiffness data found from the discretized spring model. The results of the curve fitting are shown in Figure 4.

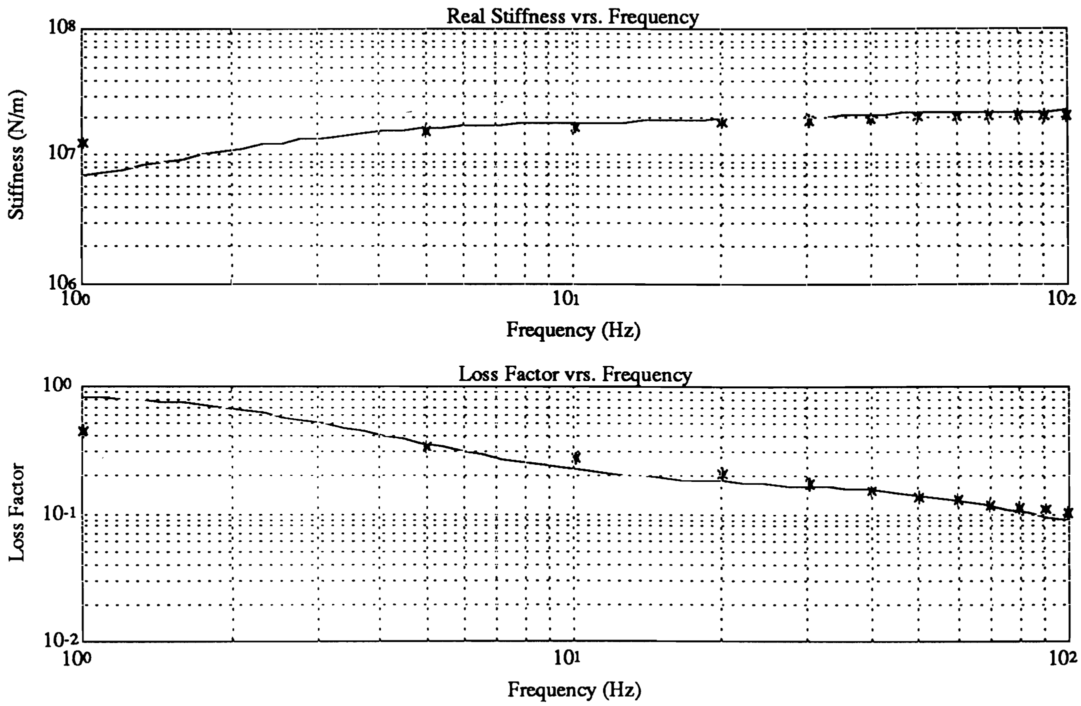

FIGURE 4: Strut model data points and GHM approximation. The line represents the curve fit of the GHM transfer functions to the data points represented by the '*'.

TEST BED

The test bed (Shown in Figure 5) is an eight bay triangular meroform truss cantilevered off of a 2000 lb steel and concrete monolith. It exhibits 5 modes below 100 Hz. The modes targeted for damping were the first and fourth modes, which were the first two modes in the vertical plane similar to those of a cantilevered beam. The second and fifth modes are horizontal bending modes similar to cantilevered beam modes while the third mode is a torsional mode.

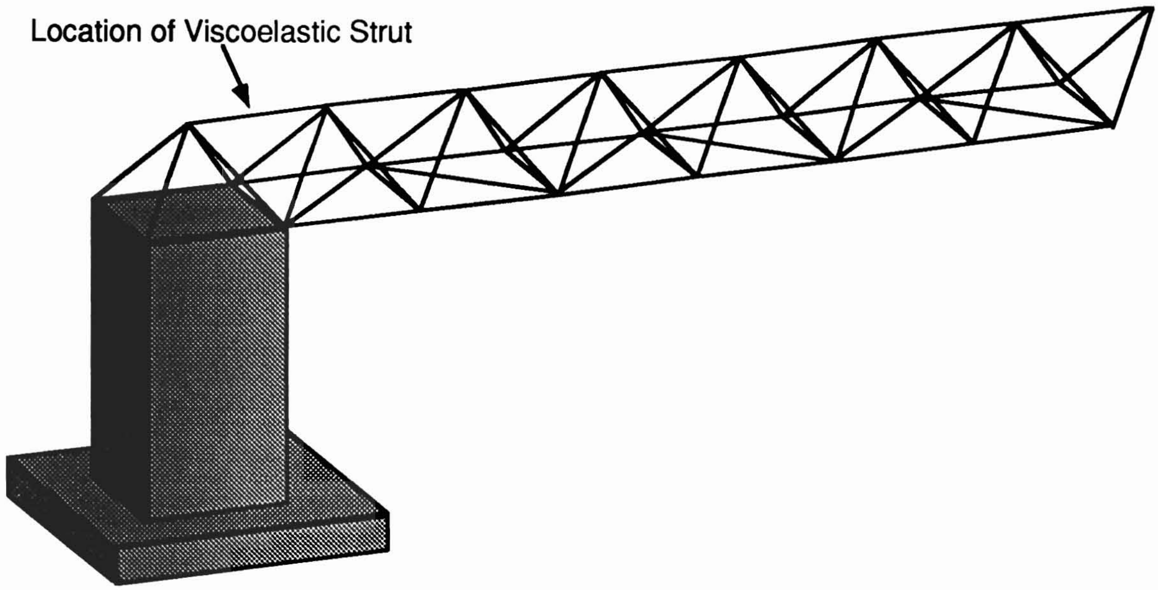

FIGURE 5: The test bed arrangement illustrating the monolith with the eight bay truss cantilevered off the top.

The truss was modeled using Euler-Bernoulli beam/rods. The damping matrix of the original undamped structure was constructed using equivalent modal damping factors from tests of the structure. The reason for assuming an original damping matrix is that the objective of this experiment is to model the change in equivalent modal damping due to the introduction of viscoelastic damping to the structure. Homogeneous, clean (without joint damping) aluminum structures tend to have damping ratios on the order of .17%. It will be shown this was the case for this truss as well, therefore predicting the truss internal damping effect is not of concern. Compliance in the monolith was accounted for by using spring to ground instead of clamped boundary conditions at the base of the truss. The viscoelastic strut was situated in the horizontal position on the top of the truss nearest the base. This provided maximum damping effect in the first mode. The Euler-Bernoulli beam/rod element in this position was replaced with the GHM rod element to model the installed viscoelastic strut. The results of the models and tests are shown in Figure 6,7 and 8.

Mode	Original Structure		FEM of Original Structure		Structure With Visco-Strut		FEM with GHM model	
	ω_d (Hz)	ζ (%)	ω_d (Hz)	ζ (%)	ω_d (Hz)	ζ (%)	ω_d (Hz)	ζ (%)
1	13.22	.18	12.88	.18	13.37	2.00	13.29	1.37
2	19.25	.17	18.82	.17	19.25	.17	18.32	.17
3	49.25	.17	50.38	.17	49.25	.17	50.30	.17
4	76.75	.17	77.82	.17	77.58	.57	77.22	.25
5	95.00	.17	97.83	.17	95.00	.17	97.75	.17
6	136.0	.17	134.8	.17	136.5	.36	136.0	.34

FIGURE 6: Experimental verses theoretical results indicating the damped natural frequencies and equivalent modal damping ratios.

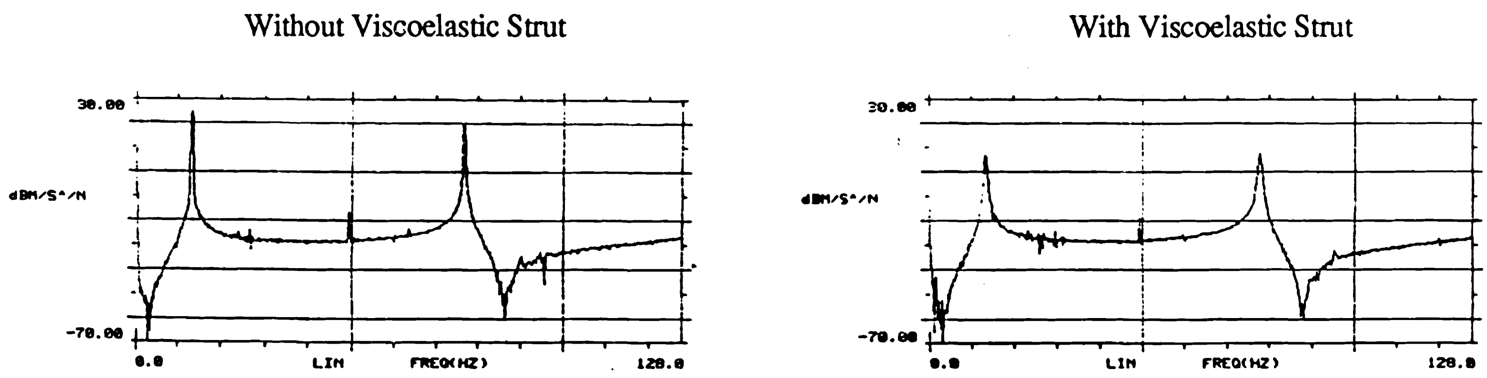

FIGURE 7: Experimental frequency response of structure with and without viscoelastic strut.

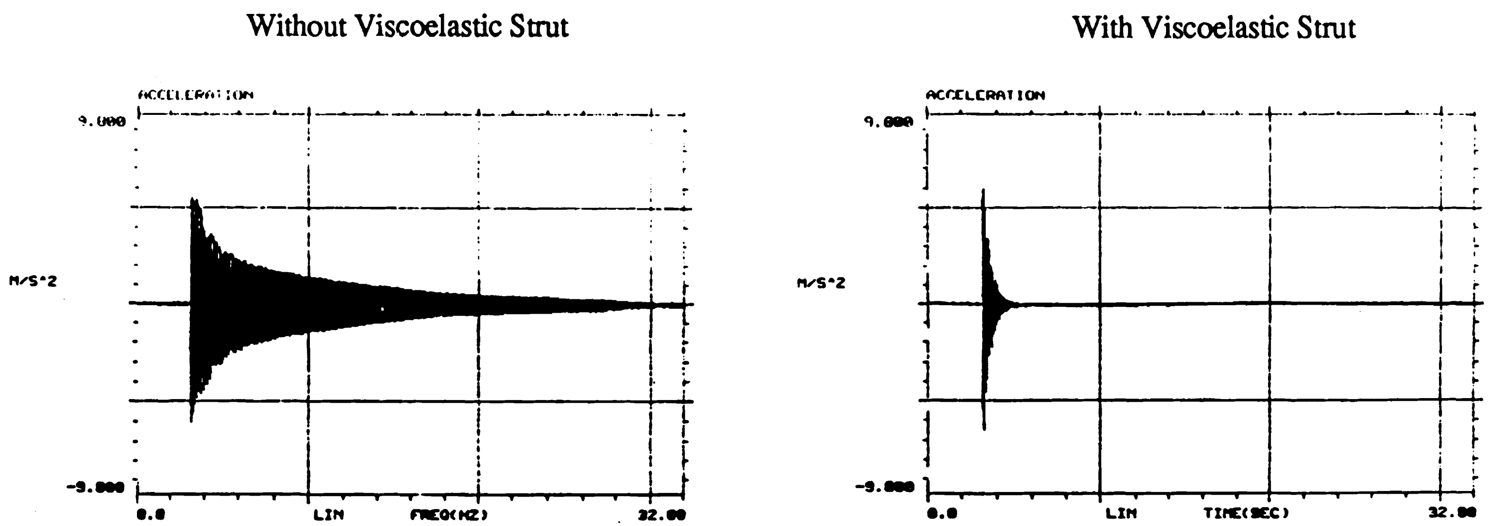

FIGURE 8: Experimental time response of structure with and without viscoelastic strut.

A number of results are presented in this table. First, note that the natural frequencies of the structure did not significantly change, although the viscoelastic strut has no real static strength. Secondly, the damping ratios of the targeted modes increased by a factor of ten in the first mode and a factor of 3.3 in the fourth. However, the prediction of the increase in damping was errant by 35% in the first mode and 75% in the second. Two effects may partially explain this. The viscoelastic material samples used in the viscoelastic strut had a thickness tolerance of ± 20% which can lead to obvious model error. The discretized spring model, although simple to use, is probably not sophisticated enough to properly model the deformation in the strut. Ideally, test data for the viscoelastic strut would be used in the GHM modeling instead of the data derived from any FEM or discretized spring model.

CONCLUSION

Viscoelastic struts introduced into flexible trusses can significantly increase the inherent damping of the structure without significantly changing the natural frequencies of the structure. The procedure for modeling the effect of viscoelastic struts using the GHM technique has been outlined. Accurate knowledge of the complex stiffness characteristics of the viscoelastic strut is vital and test data for the complex stiffness should be used if at all possible.

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