PASSIVE DAMPING DESIGN METHODS USING NASTRAN

Michael J. Matla

Eastman Kodak Company
Federal Systems Division
Rochester, New York 14653-9527
(716) 726-0577

Abstract:

Improved finite element methods for the design and analysis of passive damping treatments in complex structures have been developed. These methods account for the non-linear properties of the viscoelastic materials commonly used in damping treatments and were implemented in the design of a constrained layer damping treatment for a large ring-and-truss structure. Modal damping for this structure was analytically predicted and then experimentally measured to verify the accuracy of these methods.

Specifically, a simplified method for modeling constrained layer damping treatments in finite element analysis was developed. This reduces the number of degrees-of-freedom required permitting faster and less costly analysis while accurately modeling the dynamic effects of the treatment.

A Damping design program was also developed which calculates modal damping for any number of modes simultaneously. The program accounts for the non-linear properties of viscoelastic materials and allows quick evaluations of designs. It is based on the modal strain energy method and utilizes Nastran strain energy calculations from complete dynamic math models.

Lastly, an enhanced Nastran-based modal frequency response analysis method for damped structures was developed. The analysis generates response curves using a non-orthogonal eigenstructure created by the above-mentioned design program to account for the non-linear properties of the viscoelastic materials used. This model is more representative of the damped structure than that of a conventional, constant stiffness model.

Table of Contents

Introduction

Large complex structures are typically designed and analyzed using finite element analysis (FEA). Visco-elastic materials (VEM) are being used to passively damp their vibration for improved performance, and to increase stability in active control systems. Analysis of a damping treatment design must involve the structure so it is only logical that it too be done in FEA. The modal strain energy method (MSE method) is a manner of calculating damping values which works well with FEA. It was therefore employed in the damping design program discussed in this paper. Implementation of FEA to damping design presents one problem. Conventional FEA solutions assume linear materials and utilize a constant stiffness matrix. This is inaccurate when non-linear VEM constitute part of the model. To overcome this, the damping design program employs the use of a family of FEA models which differ only in their VEM modulus values. The values cover the analysis frequency range. The program then interpolates the modal properties of not only modal damping and eigenvalues, but also the eigenvectors from among the discrete states presented by this family of finite element models.

With the total eigenstructure now defined, it is possible to create an enhanced frequency response (FR) analysis solution to utilize this eigenstructure, which better represents the true dynamics of the structure with VEM damping.

A simplified method to model constrained layer (CL) damping in FEA was also developed which reduces the number of degrees-of-freedom (DoF), and hence time and cost, for coverage on beam members.

All these methods were developed for use with MSC/Nastran.

1.0 Simplified Finite Element Modeling of Constrained Layer Damping

The MSE method requires accurate modeling of the damping materials in the structure to be analyzed. Previous to this study, CL damping treatments were modeled in FEA using solid elements for the damping material, and plate elements for the CL. This required at least two grid points across the width. Since the damping material thicknesses are typically quite small compared to other structural dimensions, aspect ratio concerns dictated the need for many elements along the length to model the treatment. If the structure was curved, even more elements were required. This can lead to very large models which are time consuming and expensive. The objective was to develop a simpler method of modeling the treatment in FEA by requiring fewer DoF.

The method developed proved to greatly reduce analysis time while sacrificing little in the way of accuracy. It utilizes spring elements and concentrated masses to model the damping material's stiffness and mass. Grid points from the structure are rigidly attached to one set of coincident-pair grid points placed at the damping material's center of thickness. Three concentrated springs connect the coincident-pair grid points' three translational DoF. Spring rates are based on area percentages of the damping material in the vicinity of each. The CL is modeled by bar elements whose end DoFs are offset to the other grid point of the spring pair. By modeling the damping material this way, only one DoF through the treatment width is necessary versus two for the solid element method. Figure 1 illustrates this method of modeling.

Results from a Nastran model of a cantilevered beam with a constrained layer damping treatment showed that modeling the damping material with springs results in frequencies for the bending modes within 2% of those calculated using solid elements (see table 1). Strain energy calculations for the first three bending modes were within 2%. Error in strain energy for higher modes increased rapidly, due to coarseness of the model for these higher modes. Computer run time, and therefore costs, were 55 percent lower for the spring modeling versus the solid modeling due to the fact that only one grid point through the treatment width is necessary for the springs. Table 1 compares frequency and strain energy calculations using solid elements and the spring elements in Nastran.

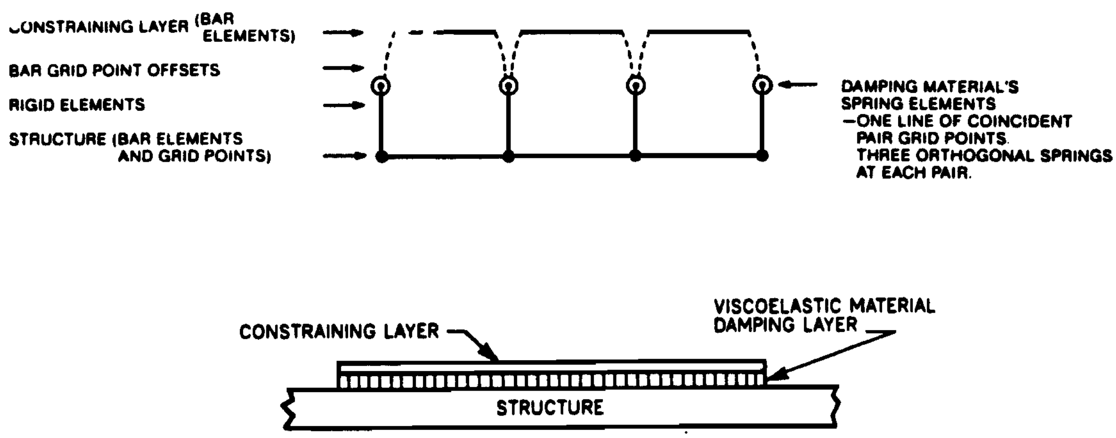

FIGURE 1: Modeling of Constrained Layer Damping Using Spring Elements.

TABLE 1

Accuracy of Simplified Modeling Method

Cantilevered Beam with Constrained Layer Damping

		Solid Elements	Springs
Run Time (sec)		308	132
Bending Mode			
1	Hz.	16.6	16.4
	%SE	17.8	17.8
2		107.6	106.6
		23.1	23.1
3		269	266
		14.9	15.2
4		503	496
		8.4	9.2
5		814	799
		5.1	6.1

2.0 Damping Design Program

A damping design program has been written which calculates both modal damping values, and a corrected eigenstructure for use in dynamic analysis of structures damped by VEM. The term eigenstructure refers to both the eigenvalues and eigenvectors of a structure's math model.

The program is based on the MSE method * , whose basic principle is:

$$\eta^{(r)} = \sum_{j=1}^{M} \eta_j \frac{SE_j^{(r)}}{SE_{Total}^{(r)}}$$

where

$\eta^{(r)}$ = damping loss factor for mode (r).

η_j = material loss factor for material region, j .

$SE_j^{(r)}$ = strain energy in material region, j, for mode (r).

$SE_{Total}^{(r)}$ = total strain energy for mode (r).

The design program consists of four segments, which are run in series to complete an entire design and analysis.

The program uses Nastran real eigenvalue finite element solutions to calculate the structure eigenvalues, eigenvectors, and strain energy in each material for each mode of interest. Since the moduli and loss factors for VEMs are strong functions of frequency, a single analysis would not result in accurate frequencies and mode shapes over a wide frequency range. This is overcome by performing a series of real eigenvalue analyses on the same structure, which differ only in the value of the VEM's modulus. The modulus values correspond to frequencies which are logarithmically spread over the analysis bandwidth.

* Johnson, Conor D. and David A. Kienholz. Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers. AIAA Journal, Vol. 20, No. 9, Sept. 1982, pp. 1284-1290.

The F06 output file from each Nastran run is read into the first segment, named GET_SD for " get strain data" , where mode frequencies, and percentage of mode strain energy for all user defined material groups, are extracted for each user identified mode of interest. The loss factor for all material groups other than the VEM group are also entered. This segment is done for each Nastran run, creating a separate output file for each. Although the damping design program can only work on one VEM type at a time, the identification and use of multiple material groups allows for the computation of damping contribution from frequency independent mechanisms, such as frictional and structural damping.

The second segment, named SORT_SD for " sort strain data", reads in all the output files from GET_SD and organizes the data into two matrices: 1) modal frequency versus VEM modulus for each mode, and 2) percentage of modal strain energy in each material region versus VEM modulus for each mode. This data contains two very important functions of the structure: 1) modal frequency versus VEM modulus, for each mode, and 2) percentage strain energy in the VEM versus it's modulus, for each mode. Third order and fourth order polynomials are fitted to this data, respectively, to simplify it's handling and further calculation operations. The equations are included in SORT_SD's output file along with plots of these two functions. Figures 2 and 3 are examples of these plots.

The third segment, named MSEM, is where the calculation of modal damping and the interpolation of eigenvalues occur. Two functions of the VEM under consideration are supplied to MSEM, namely: 1) VEM modulus versus frequency, and 2) VEM loss factor versus frequency. Third order polynomial functions are fitted to this data, and plots are created so that the user can verify the accuracy of the curve fitting (see Figures 4 and 5 for examples). The program then calculates modal damping by first finding the intersection points of the VEM's modulus versus frequency function (Figure 4) with that of the structures (Figure 2), for each mode. Figure 6 illustrates this operation. This gives the frequency at which each mode of the structure with this VEM damping treatment will vibrate, and the modulus that the VEM will have for each mode. With the modal frequencies now determined, the loss factor of the VEM can be found for each mode from the polynomial for VEM loss factor versus frequency (Figure 5). Likewise, with the VEM modulus known for each mode, the percentage strain energy in the VEM can be found from the structure functions of strain energy versus modulus of the VEM material (Figure 3). The

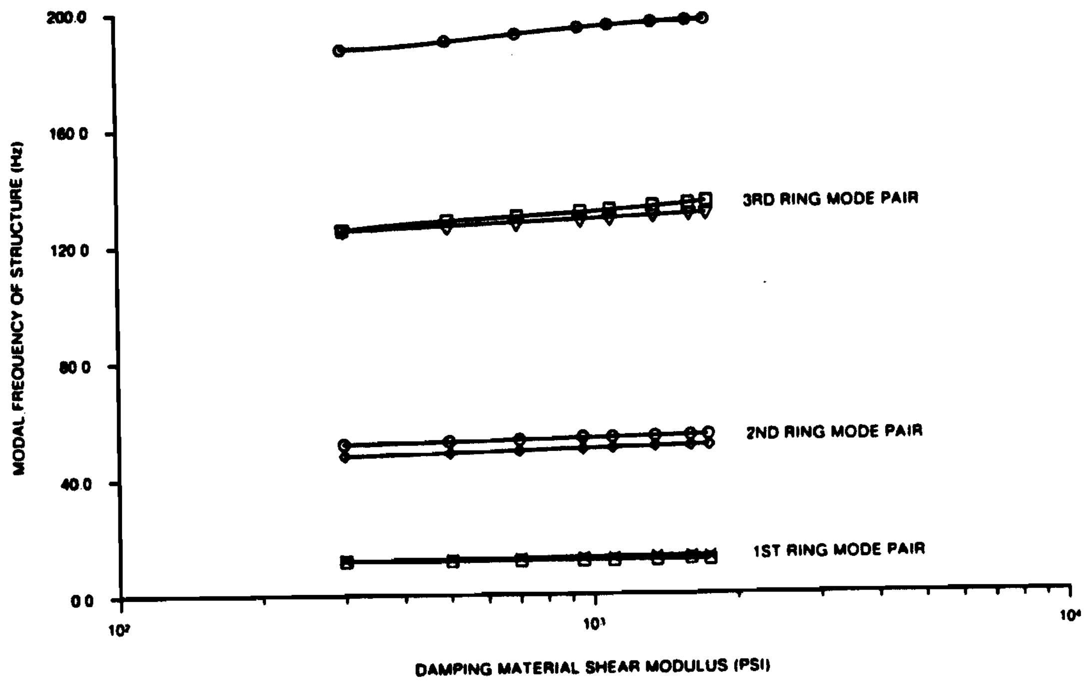

FIGURE 2: Modal Frequency of Structure versus Damping Shear Modulus, SORT-SD Program; Model: Ring/Truss Structure.

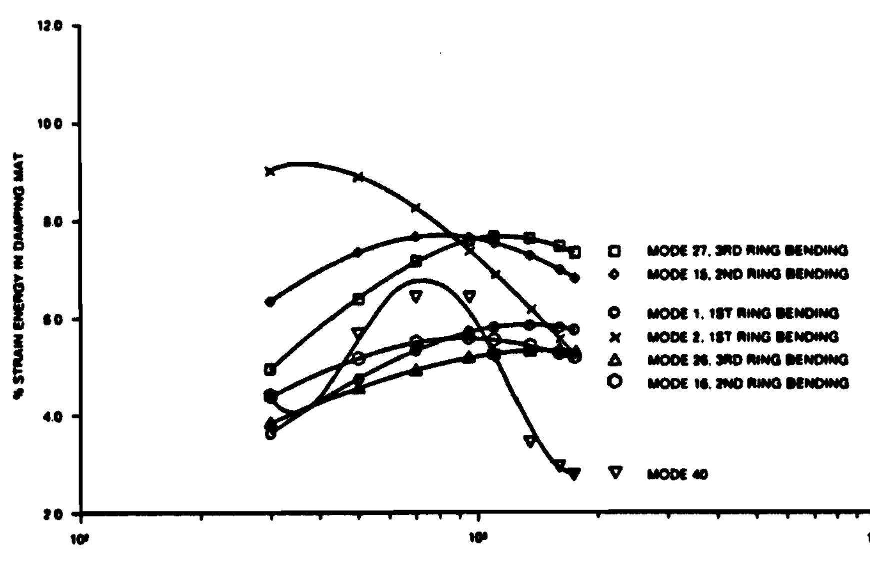

FIGURE 3: Percentage Strain Energy in Damping Material verus Damping Shear Modulus, SORT-SD Program; Model: Ring/Truss Structure.

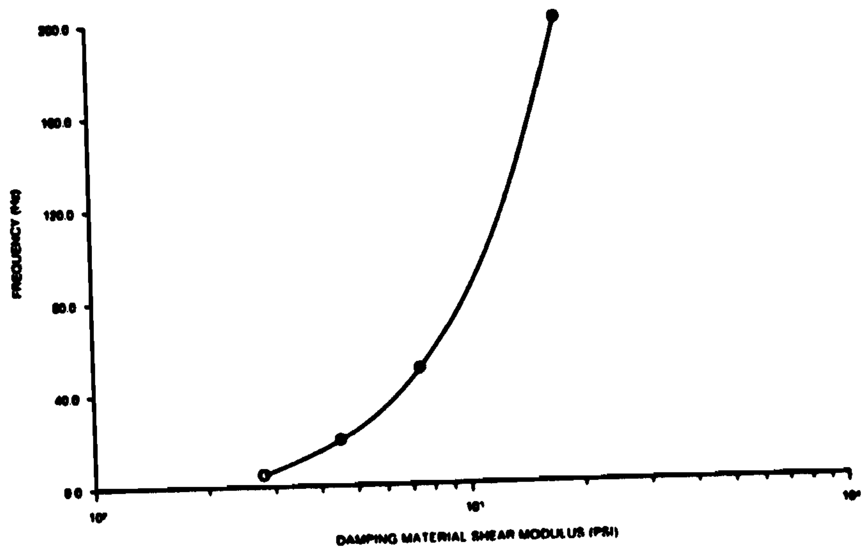

FIGURE 4: Damping Material Shear Modulus as a Function of Frequency, MSEM Program. Damping Material: 3M-4945.

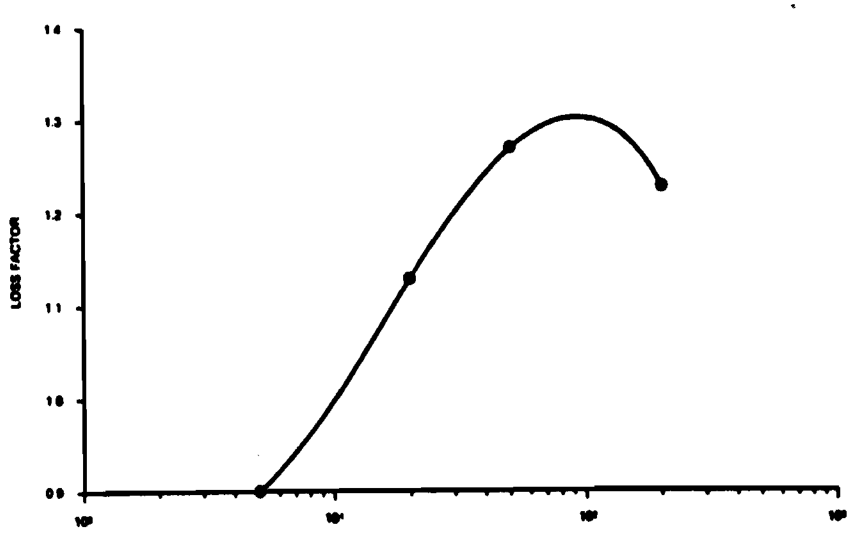

FIGURE 5: Damping Material Loss Factor versus Frequency, MSEM Program Damping Material: 3M-4945.

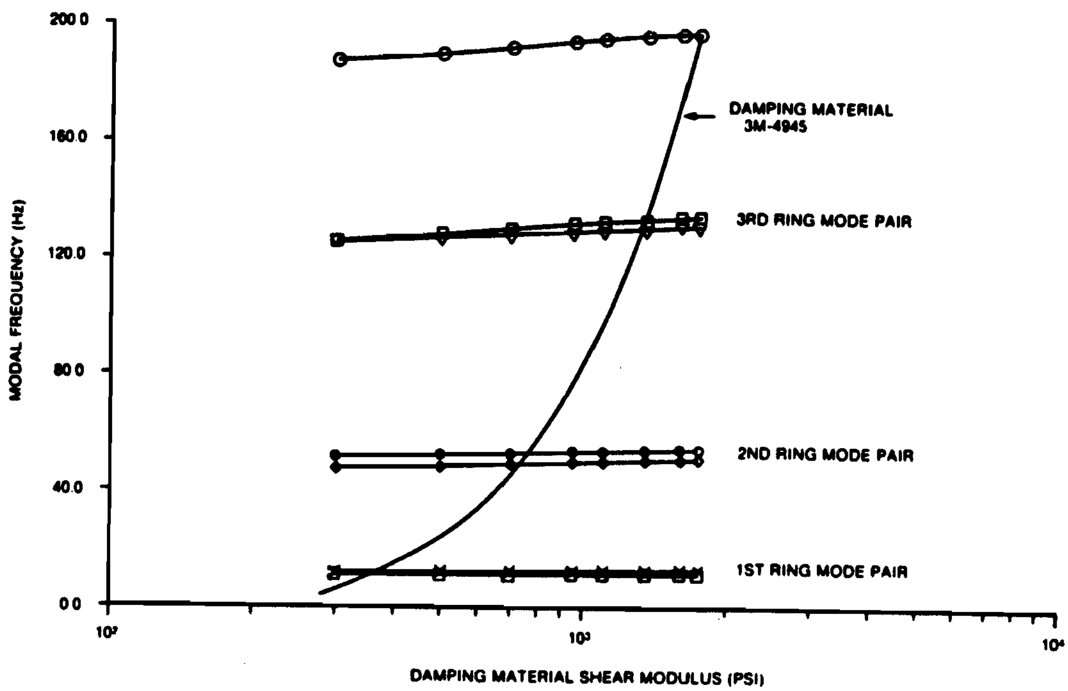

FIGURE 6: Modal Frequency versus Damping Shear Modulus, Damping Design Program. Model: Ring/Truss Structure; Damping Material: 3M-4945.

percentage strain energy in each of the other material groups is found by linearly interpolating the data from SORT_SD with the known VEM modulus for each mode. With loss factors and percentage strain energy now calculated for each material group, for each mode, the modal damping values are calculated using the MSE method equation. The eigenvalues are determined from their simple relationship with modal frequency.

The fourth and final segment of the design program, named "INTERP_EIGEN", calulates the eigenvectors which correspond to the eigenvalues found earlier. This is accomplished by retrieving output information from the third and first segments, MSEM and GET_SD, respectively. The VEM modulus for each mode is used to select the two Nastran F06 files containing the eigenvectors between which the interpolation will occur for a given mode. The interpolation point is determined by the relationship of the calculated eigenvalue to the eigenvalues corresponding to the two retrieved eigenvectors. The validity of determinig eigenvectors by interpolation is discussed in the following section on enhanced FR analysis.

This completes the damping design program. In summary, it enables the calculation of modal damping values, eigenvalues, and eigenvectors for a structure with damping treatment. This eigenstructure better describes the true resonant frequencies and mode shapes of the actual structure compared to standard FEA modeling means.

3.0 Enhanced Frequency Response Analysis Method

The damping design program calculates modal damping values for use in dynamic modal analyses, such as frequency response (FR). However, Nastran assumes a constant stiffness matrix which is not true for structures with VEMs. For an approximate FR analysis, a modulus of the VEM can be used which corresponds to a frequency in the middle of the analysis range. Resonant frequencies and amplitudes will be accurate in the vicinity of this assumed frequency, but will be less accurate for higher and lower resonances.

An improvement in the modal FR can be made if, instead of using the orthogonal eigenstructure of the middle-frequency structure, it is replaced by an eigenstructure consisting of the interpolated modes found in the damping design program. This results in an eigenstructure which is no longer orthogonal, but orthogonal eigenstructures are only true for structures with linear material properties. This non-orthogonal eigenstructure of interpolated eigenvalues and eigenvectors better desribes the true resonant frequencies and mode shapes of the actual structure with VEM damping by accounting for the frequency dependency of VEM properties.

The approach is based on the ability to accurately interpolate modal parameters. This was demonstrated in an excercise where interpolated parameters errored less than 0.2 percent from actual values. Table 2 shows one set of interpolated parameters from a structure whose damping material modulus was varied over a significant range.

TABLE 2
Interpolation of Modal Parameters

	Nastran Solution	Interpolated Values	% Error
Damping Material Shear Modulus	350 psi	350 psi	
Generalized Mass	1.450 E-3	1.4528 E-3	0.15
Generalized Stiffness	16.84	16.81	0.17
Eigenvector Component	2.203	2.206	0.12
Eigenvalue	17.14 Hz.	17.12 Hz.	0.16

An enhanced FR solution method was created which makes use of this non-orthogonal, interpolated eigenstructure. The method uses a standard finite element method FR solution sequence, with commands added to enable the substitution of the interpolated eigenstructure from the damping design program. The tedious writing of these commands has been automated by another program.

The other quantities required to perform a standard modal frequency response analysis are:

Generalized stiffness matrix

Generalized mass matrix

Generalized force vector

Modal damping matrix

The generalized stiffness matrix is a diagonal matrix with components equal to the corresponding eigenvalue, squared. In the enhanced FR analysis the eignvalues are replaced as mentioned early and therefore cause the proper adjustment to the stiffness matrix. If the eigensolution normalizes the eigenvector to unit modal mass, the generalized mass matrix is equal to the identity matrix and hence remains unchanged for the enhanced method. The modal forces for the enhanced method are calculated in the same manner as those in a standard FR solution. The modal damping values are calculated by the damping design programs as mentioned earlier, and are included in the solution sequence.

4.0 Application of Methods

The methods described above were implemented in the design and analysis of a CL damping treatment for a structure of moderate complexity to demonstrate the ability and accuracy of these methods.

The structure choosen was a ring-and-truss shown in figure 7. It consists of an aluminum ring 100" in diameter with a box cross section, and a fixture kinematically supported above the ring by six composite tubes approximately 70" in length. For test purposes, the structure was secured to a large, isolated granite surface plate by means of 12 rod flexures and 3 plate flexures.

4.1 Damping Treatment Design

The objective of this task was to design and test a CL damping treatment for the ring/truss structure using the simplified modeling method and the damping design program. The goal was to significantly damp the first three ring modes of the structure with an inexpensive, easily constructed and applied damping treatment.

3M Company's acrylic core foam tape, Y-4945, was chosen for the design for two reasons: 1) ease of application due to high compliance, 2) excellent loss factor at room temperature and 1-200 Hz. frequency range (the analysis bandwidth).

The first segment of the design program, GET_SD, was used to design the damping treatment. Five conclusions were drawn from this work:

- Increasing extensional stiffness of the constraining layer increases the strain energy in the damping material.

- Increasing bending stiffness of the CL increases the strain energy in the damping material.

- All the strain energy in the damping material is due to shear strain. Tension and compression contribute an insignificant amount.

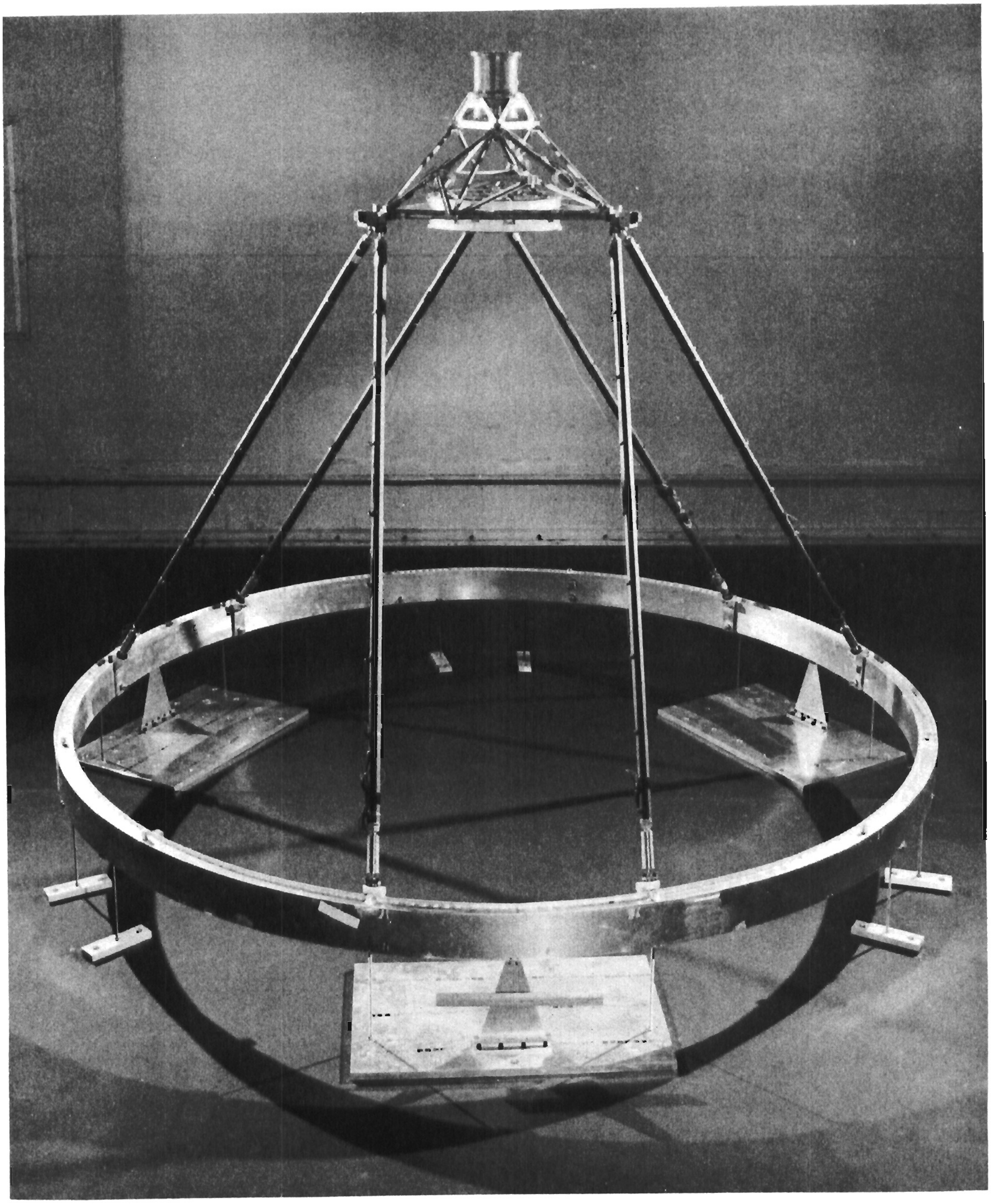

FIGURE 7: Ring/Truss Test Structure

-An optimum shear stiffness of the damping material exists for a given structure and CL.

- The effect of segmenting the constraining layer is mode specific. For treatments covering the entire circumference, segmenting it in four equal sections, causes a decrease in strain energy > 10 %.

The design chosen consisted of the 3M tape two inches wide, doubled to a thickness of 0.090 inches, and constrained by a steel band three inches wide, 0.125 inches thick and covering the entire circumference with four equal segments rolled to the outer curvature of the structure's ring.

The CL was made of steel because it is inexpensive and has a high modulus. Extensional and bending stiffnesses were limited to that afforded by the 3.0 inch by 0.125 inch dimensions because of handling, application, and removal limits. Doubling the 3M tape thickness to 0.090 inches gave the design the optimal damping material shear stiffness to damp the first three rings modes.

Modal damping values were calculated for this design and are shown in Table 3. Significant levels of structural damping were achieved by this design (damping loss factor > .10). Figures 2 - 6 were produced by the damping design program. Figure 2 shows how modal frequencies for the structure and this damping design change with modulus of the VEM. It is important to note that these numbers are based only on the asumed dimensions of the damping material and the range of modulus values used. The numbers are not based on a particular damping material. Figure 3 shows how strain energy in the VEM changes with VEM modulus for each mode. Note the strong maximums for many modes. Figures 4 and 5 show the material properties of the VEM used in this design.

4.2 Damping Test Measurement

The objective was to measure the actual damping levels of the first three ring modes of the structure to verify the passive damping values predicted by the damping design program with the simplified modeling method. This was accomplished by a modal survey test.

Test data was first acquired for the structure without the CL damping treatment to check the accuracy of the math model, and to measure damping values before addition of the treatment. Results show that the math model calculated ring bending mode frequencies to within 1 - 5 % of the test measurements (see Table 3). This is very good considering the ring is an assembly of plate sections welded to form a box cross-section. Structural damping values are also shown for the ring/truss before the damping treatment was added.

Modal properties of the ring/truss with the CL damping are also shown in Table 3. Comparing analytical to test values, the modal frequencies again agree to within 2 - 4.5 % even though the frequencies changed significantly after the damping treatment was added (compare test values before and after the damping treatment was added). This confirms the simplified modeling method's and the damping design program's ability to accurately represent the mass and stiffness of the CL treatment.

Significant structural damping values were measured, ranging from 2.8 - 6.8 %. This compares with values for the bare ring/truss ranging from 0.2 - 1.4 %. Figure 8 illustrates the increased damping for these modes. Comparing test damping values to analytical for the structure with the CL treatment, shows test values ranging from 42 - 89% of those predicted. Although it was hoped that these values would have agreed better, these first-attempt results are not discouraging. The design program and simplified modeling method enabled the conscientious design of a CL damping treatment which significantly increased modal damping by as much as 18 times. The agreement between test and analytical values is significant when the following sources of error are considered:

- The MSE method calculates approximate damping values for a nonlinear structure using linear analyses.

- Error in Nastran's calculation of modal strain energy is difficult to quantify by experimental testing.

- VEM properties are complex and less consistent than that of elastic materials, making their measurement more difficult.

- The extraction of damping values from test data of highly damped, closely spaced resonances is difficult and prone to errors. Modal analysis software is least accurate in it's determination of damping. It is most accurate and more commonly used to determine resonant frequencies and mode shapes.

TABLE 3
Ring-Truss Structure Modal Properties.
Analytical versus Test Values.

Structure without constrained layer damping:

Ring Bending Mode Pair	Frequency (Hz) NASTRAN	Test	Structural Damping Test
1st	11.474 11.478	11.304 11.372	.0021 .0037
2rd	49.1 53.5	51.40 56.20	.0038 .0086
3rd	141.5 142.1	137.6 138.1	.0120 .0116

Structure with constrained layer damping:

Ring Bending Mode Pair	Frequency (Hz) NASTRAN	Test	Structural Damping MSE	Test
1st	11.11 11.77	10.88 11.50	.036 .095	.032 .068
2nd	48.1 51.1	49.5 53.40	.106 .067	.056 .028
3rd	128.8 131.2	125.5 127.3	.078 .107	.037 .065

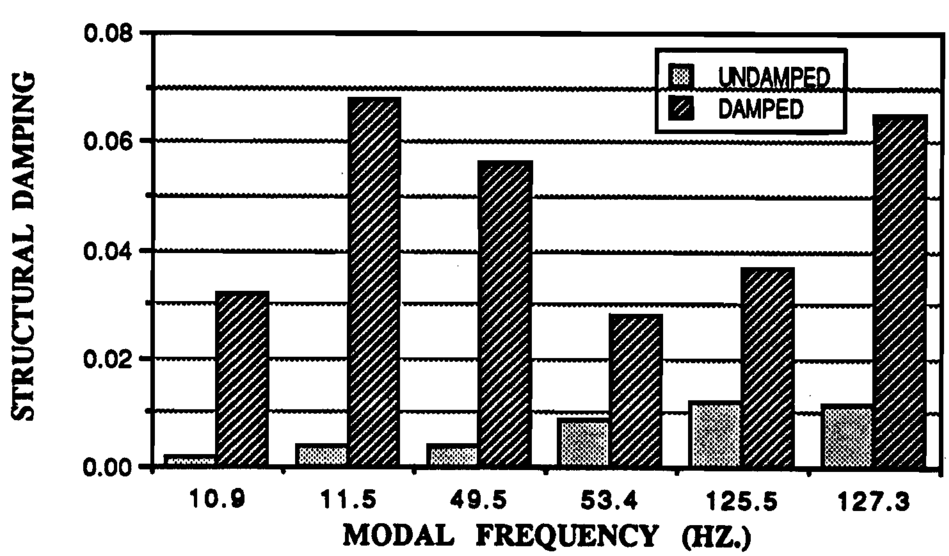

FIGURE 8: Ring/Truss Passive Test Results

- The results reported are for one treatment designed and one modal test at one shaker location. Given sufficient time, multiple designs should be tested numerous ways to better account for the origin and extent of errors.

Effort will continue to increase the match-up of analytical damping predictions to test measurements. Possible errors will be pursued in both the analytical calculation, and the measurement techniques.

4.3) Enhanced Frequency Response of Test Structure

An enhanced FR analysis was performed on the ring/truss test structure with it's constrained layer damping treatment as described earlier in section 4.1 .

Figure 9 is a plot of the enhanced FR analysis for a point on the ring of the damped ring/truss structure. Acceleration is plotted verses frequency (Hz.). The first three ring bending modes (six modes total since each bending shape has a twin) are marked. The damping treatment damped only the ring modes appreciably and hence eigenvalues and eigenvectors for only these modes were calculated by the damping design program. All other modes are as normally calculated for the structure with a damping material shear modulus corresponding to a frequency in the middle of the analysis bandwidth.

A standard Nastran FR analysis for the damped structure (using damping material shear modulus corresponding to a frequency in the middle of the analysis bandwidth) was also performed. Table 4 shows differences of up to 5% in eigenvalue, and 12% in response amplitude at resonance. These are small, but may be greater for different treatments and other structures.

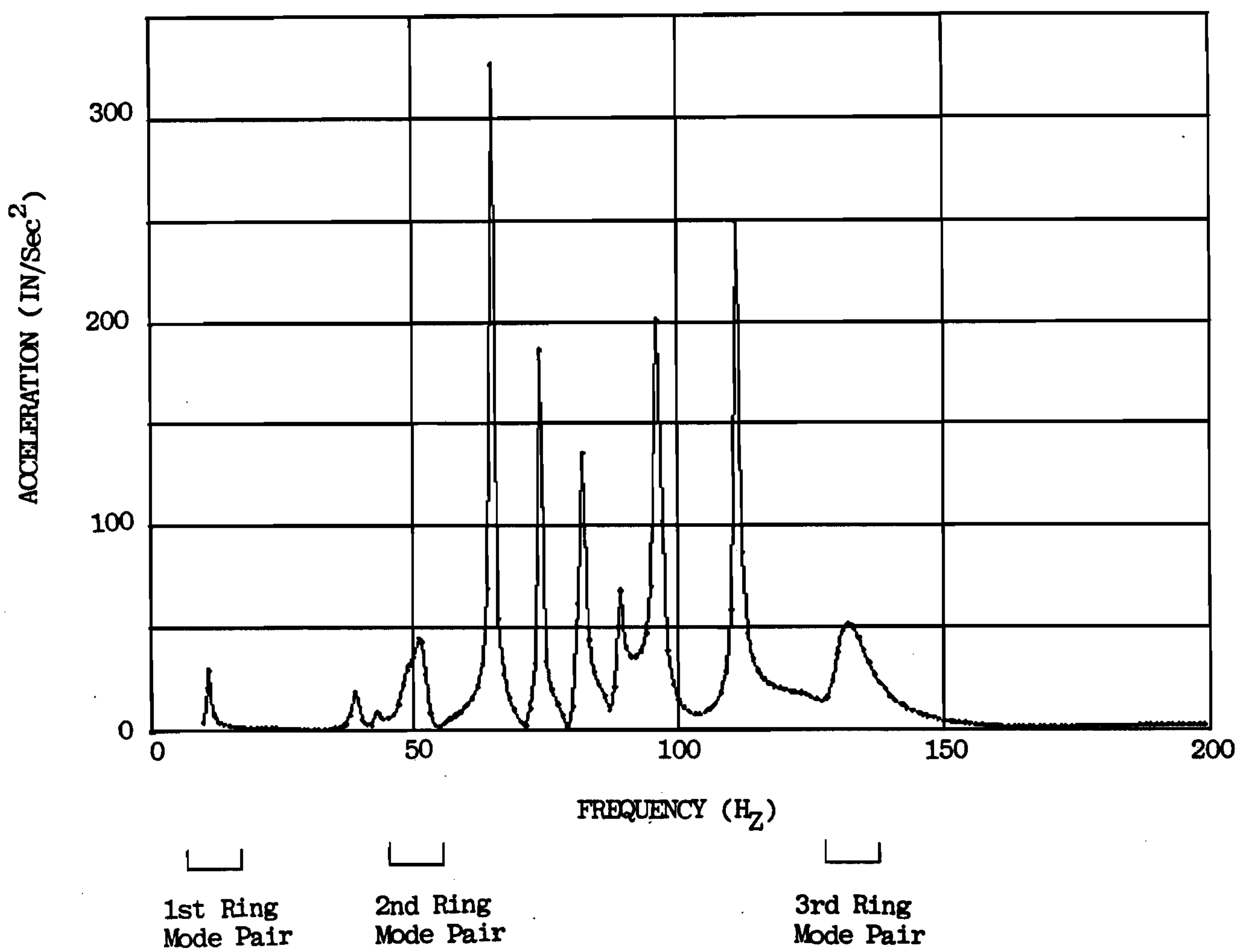

FIGURE 9: Enhanced Frequency Response of a Point on the Ring/Truss Structure.

TABLE 4
Comparison of Resonant Frequency and Amplitude between Enhanced and Standard FR for the Ring/Truss Structure

Ring Bending Mode Pair	Resonant Frequency Enhanced	Resonant Frequency Standard	Resonant Frequency %Diff.	Peak Amplitude Enhanced	Peak Amplitude Standard	Peak Amplitude % Diff.
1	11.18	11.60	4%	29.7	29.2	2%
	11.94	12.57	5%	7.5	6.2	21%
2	48.7	49.6	2%	29.2	28.4	3%
	51.5	52.0	1%	43.0	42.4	1.5%
3	129.9	129.3	.5%	36.2	40.4	12%
	132.8	131.8	.75%	50.0	52.6	5%

Summary

Improved finite element methods for the design and analysis of passive damping treatments in complex structures have been developed. These methods account for the non-linear properties of the viscoelastic materials commonly used in damping treatments and were implemented in the design of a constrained layer damping treatment for a large ring-and-truss structure. Modal damping for this structure was analytically predicted and then experimentally measured to verify the accuracy of these methods.

Bibliography

1) Frater, N. K. . Implementation of Modal Strain Energy Method Using MSC/Nastran and Post-processing Utility Programs. Damping 1986 Proceedings, AFWAL-TR-86-3059, Vol. 1, May 1986.

2) Johnson, C. D., and D. A. Kienholz. Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers. AIAA Journal, Vol. 20, No. 9, Sept1982.

3) Johnson, C. D., D. A. kienholz, E. M. Austin, and M. E. Schneider. Design and Analysis of Damped Structures Using Finite Element Techniques. ASME Journal, 85-DET-131.

4) Nashif, A. D., D. I. G. Jones, and J. P. Henderson. Vibration Damping. John Wiley and Sons, New York, 1985.