DESIGN, ANALYSIS, AND TESTING OF THE PACOSS D-STRUT TRUSS

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ABSTRACT

Future space systems may be large, lightweight, and flexible. Such systems will often include trusswork due to the high specific strength and stiffness typical of trusses. Damping of these structures will minimize detrimental vibration, which otherwise may reduce system performance to unacceptable levels.

A damping device suitable for application to trusses, which is designated the D-Strut™,.has been developed by Honeywell. This paper discusses the further development of the damping member, the derivation of analytic procedures required for efficient integration of these members into truss structures, and the results of testing of a structure which incorporates these damping members.

Following design, prototyping, fabrication, and impedance testing of D-Strut members, a truss structure which includes these members was assembled and subjected to modal testing. Comparisons of the finite element model of the truss with the experimental modal test data show excellent agreement for the first seven modes, and verify damping levels in the fundamental modes of nearly 10% critical.

The D-Struts were compared with viscoelastic extensional damping members designed to produce similar damping levels. These comparisons included weight, temperature stability, strength, etc. Results of the comparisons currently favor the viscoelastic members; however, advances in the design of the viscous device will allow the D-Strut to provide an efficient damping treatment for truss structures of the future.

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1.0 Introduction

A goal of the Passive and Active Control of Space Structures (PACOSS) Program is the development and verification of passive damping treatments for application to flexible space structures. An examination of the performance of the Honeywell viscous fluid damping struts (D-Struts) for use in truss structures was completed during the second phase of the PACOSS Program.

At the onset of the investigation, there were considered to be four potential advantages of the D-Strut over other types of damping treatments for truss structures, such as viscoelastic damping members. These potential benefits were reduced temperature dependency, decreased susceptibility to outgassing, higher static loading capability, and the potential for decreased weight as compared to viscoelastic dampers. The objectives of the research were to develop the analytic tools necessary to efficiently design and analyze truss structures including D-Struts, to design and build a structure consistent with goals for the test truss, and to confirm the performance of the members through unit testing and a modal test on the structure with the damping members installed.

To meet these objectives, a truss structure which contains the D-Strut members was designed, fabricated, and tested (Fig. 1). The truss structure was designed and fabricated by Martin Marietta, while the D-Strut members themselves were built by Honeywell. The structure consists of eight bays which are each 34-in. square, with damping members as the longerons for the lower three bays. During the design process, design techniques which allow for the efficient application of D-Strut damping treatments to structures was developed.

Following fabrication of the D-Struts, unit testing of the members, and their integration into the structure, a modal test was performed. The resulting modal parameters were compared with analytic predictions to determine model accuracy and D-Strut performance. Finally, the D-Strut damping members were compared with their viscoelastic counterparts for important properties. This paper discusses the design methodology and analysis techniques which were developed, the results of the D-Strut member design and fabrication, the structure modal test results, and the results of the damping member comparison.

2.0 The D-Strut Member and Viscous Fluid Damping Device

The D-Strut is comprised of a viscous fluid damping device, structural tubing, and end fittings. The tubing attaches to the joints of the structure and supplies the static stiffness of the element, while the damping element is used to attenuate vibration of the structure. A schematic of the damping device as used in the PACOSS D-Struts is provided as Figure 2.

The working elements of the damping device consist of a titanium diaphragm, a small orifice, and a bellows which contains a viscous silicone fluid. When a dynamic load is applied to the member, a portion of the load is transmitted through the inner tube, and a portion is transmitted through the outer tube. The force applied by the inner tube to the damping device bends the circular diaphragm which is connected to

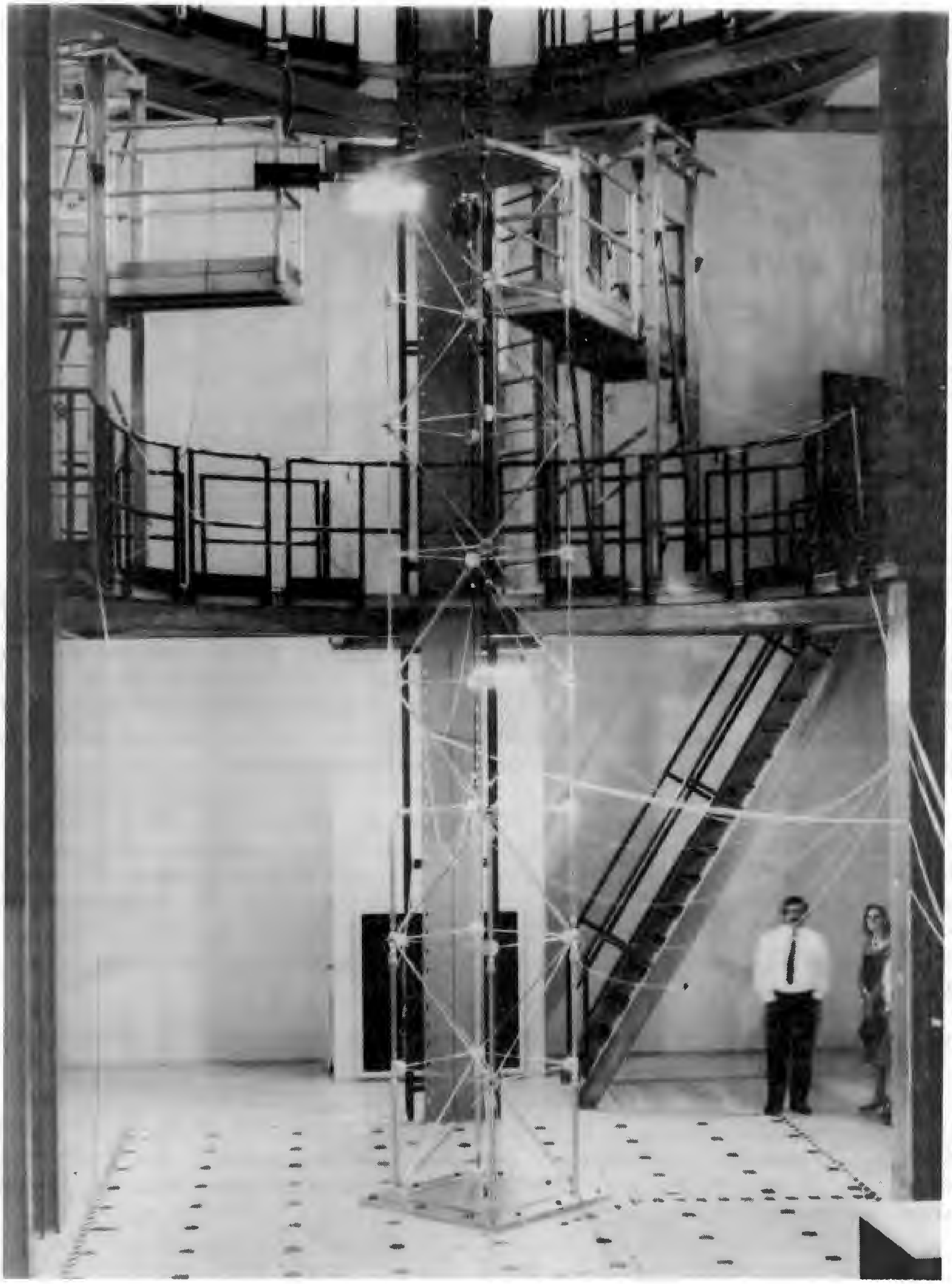


Figure 1 - The D-Strut Test Truss in the Modal Test Configuration

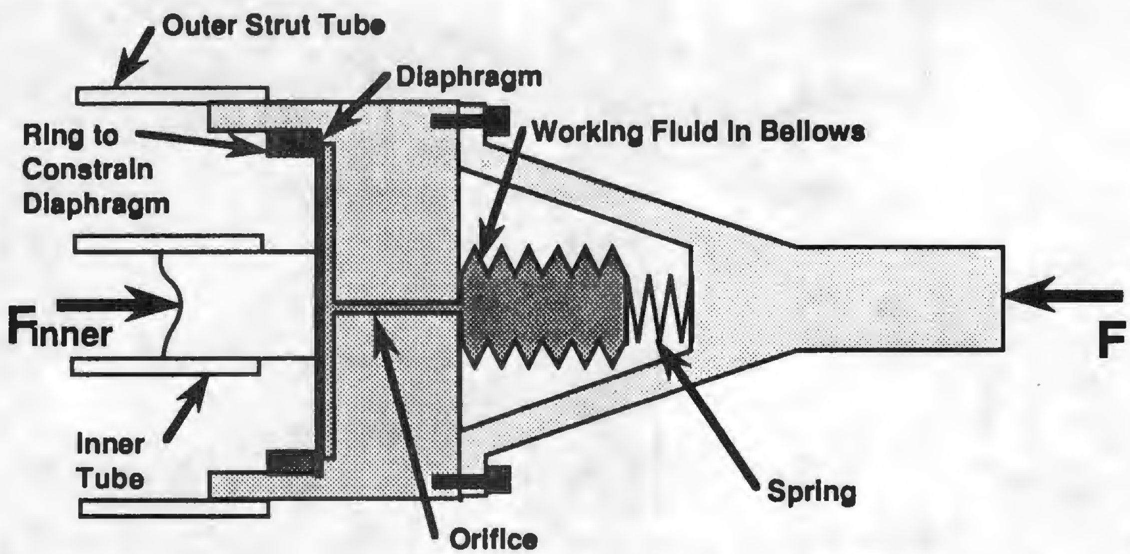


Figure 2 - Schematic of the Honeywell Viscous Fluid Damping Device

a fitting on the inner diameter, and is constrained to move with the damper housing on the outer diameter. The deformation of the diaphragm pressurizes the fluid behind it and forces the fluid through the orifice. The resistance of the fluid flow due to its viscosity creates a damping force which is applied to the structure at the joints. A spring is used to apply static pressure to the fluid to eliminate cavitation of the fluid for dynamic tension loads applied to the damping device. The spring and bellows also allow for expansion and contraction of the fluid with temperature.

Two D-Strut concepts were developed by Honeywell which have been designated the SD and the D1 strut. The SD strut incorporates two tubing members: an outer tube member which connects directly across the span element, and an inner tube which connects to the damping device. The D1 strut is similar. However, a single tube which connects to the end fitting on one side and to the damping device on the other is used. The potential application of both designs was examined during the PACOSS effort.

3.0 D-Strut Modeling Using the Spring / Dashpot Model and Impedance Methods

Design and analysis of truss structures which incorporate D-Struts necessitates models of the damping members. Two methods of modeling the D-Strut were investigated. These methods include modeling of the strut using a spring / dashpot network and modeling using impedance methods.

The D-Strut can be modeled using a series of springs and a dashpot [1], using a network which was previously formulated by Honeywell. The network model of the SD strut is included as Figure 3.

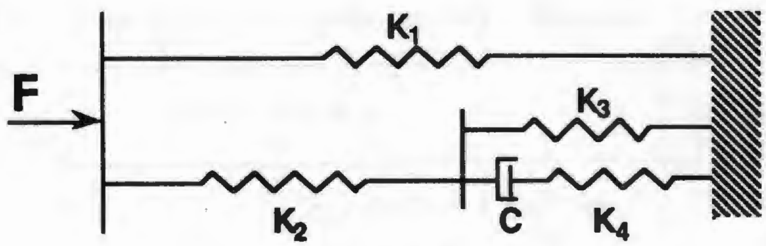


Figure 3 - Spring / Dashpot Model of the SD Strut

The correlation between hardware effects and network parameters is given in Table 1. The 5-parameter spring / dashpot model effectively simulates the dynamic response of the D-Strut, when appropriate values of the model elements are chosen. This analytic model can be directly incorporated into truss models and the matrix equations solved to provide the complex eigenvalues and eigenvectors of the damped structure. However, this method does not provide insight into the best selection of locations for the dampers or effective design of the damping members. Alternative modeling methods can be used to determine the effect of the strut on the dynamics of a structure, and to provide insight into the proper selection of the various strut parameters.

To efficiently model the D-Strut behavior, the impedance of the strut was developed. The impedance of the strut is a frequency-dependent complex number which provides the ratio of the applied force to the resulting displacement across the strut, as well as the phase relationship between them. The strut impedance can be determined by transforming a dynamic load and the resulting dynamic displacement across the member to the frequency domain using the Laplace transform:

$$X(s) = \mathcal{L}(x(t)) \qquad F(s) = \mathcal{L}(f(t)) \tag{1}$$

where:

$x(t)$ = the dynamic displacement across the damper
$f(t)$ = the dynamic force applied

The strut impedance then relates the frequency-dependent force and displacement. However, it is a function of the Laplace variable:

$$F(s) = Z(s)\,X(s) \tag{2}$$

where:

$Z(s)$ = member impedance function

The impedance can be written in many alternative forms which have utility in different applications. A representation of the impedance which is useful for the analysis of dynamic systems and for optimization of the D-Strut is the complex stiffness representation. The complex stiffness representation can be determined by evaluating the real part of the impedance, and defining the loss factor as the ratio of the imaginary part of the impedance to the real part of the impedance:

Table 1 - Hardware Effects Modeled by Various Springs

Model Parameter	Hardware Effect
K_1	Outer Tube Axial Stiffness
K_2	Inner Tube Axial Stiffness
K_3	Diaphragm Bending Stiffness
K_4	Fluid Compressibility and Chamber Compliance
C	Orifice Fluid Flow Restriction

$$K_{eq}(s) = \mathrm{Real}(Z(s)) \qquad \eta(s) = \mathrm{Imag}(Z(s)) / \mathrm{Real}(Z(s)) \tag{3}$$

When represented in this manner, the frequency-dependent impedance is identical to the "complex stiffness" which is typically used to model viscoelastic damping treatments. Impedance (complex stiffness) models of viscoelastic damping struts were previously used for the design and analysis of the PACOSS Dynamic Test Article (DTA) and were shown to provide a useful representation of the damping elements which allowed the determination of the dynamic properties of the damped structure [2].

To show how the impedance is determined for a simple spring / dashpot network, consider the network in Figure 4. The network consists of a spring which is in parallel with a spring and dashpot in series.

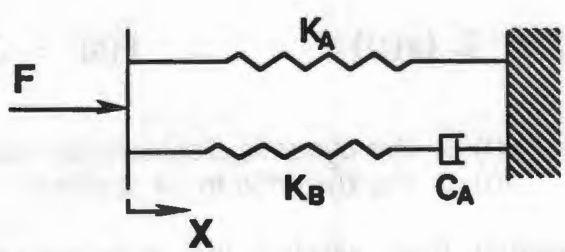


Figure 4 - Simple Spring / Dashpot Network

The impedance of any network can easily be determined using network simplification. The impedance of this network is:

$$Z(s) = K_A + \frac{C_A s K_B}{C_A s + K_B} \tag{4}$$

The strut impedance for sinusoidal inputs as a function of frequency is determined by evaluating along the imaginary axis in the Laplace domain (at $s = i\omega$):

$$K_{eq}(i\omega) = \mathrm{Real}(Z(i\omega)) = \frac{-K_B^3}{C_A^2\omega^2 + K_B^2} + K_A + K_B$$

$$\eta(i\omega) = \frac{\mathrm{Imag}(Z(i\omega))}{\mathrm{Real}(Z(i\omega))} = \frac{C_A K_B^2 \omega}{C_A^2\omega^2(K_A + K_B) + K_A K_B^2} \tag{5}$$

The complex stiffness representation can be plotted versus frequency to show the shape of the impedance function. Figures 5 and 6 provide representative impedance plots for the simple network. The values used to generate the impedance plots were 50,000 lb/in., 100,000 lb/in., and 2,000 lb-s/in. for K_A, K_B, and C_A, respectively. The equivalent stiffness asymptotically approaches the sum of K_A and K_B, while the loss factor displays a distinct peak. The low-frequency stiffness of the network is the stiffness of the shunt spring K_A.

A high loss factor is desirable to provide high damping ratios to the modes of flexible structures [3]. To determine the maximum loss factor of the spring / dashpot network and the frequency of the peak as a function of the network parameters, the derivative of the loss factor equation with respect to frequency is taken, and the frequency ω_{max} is determined:

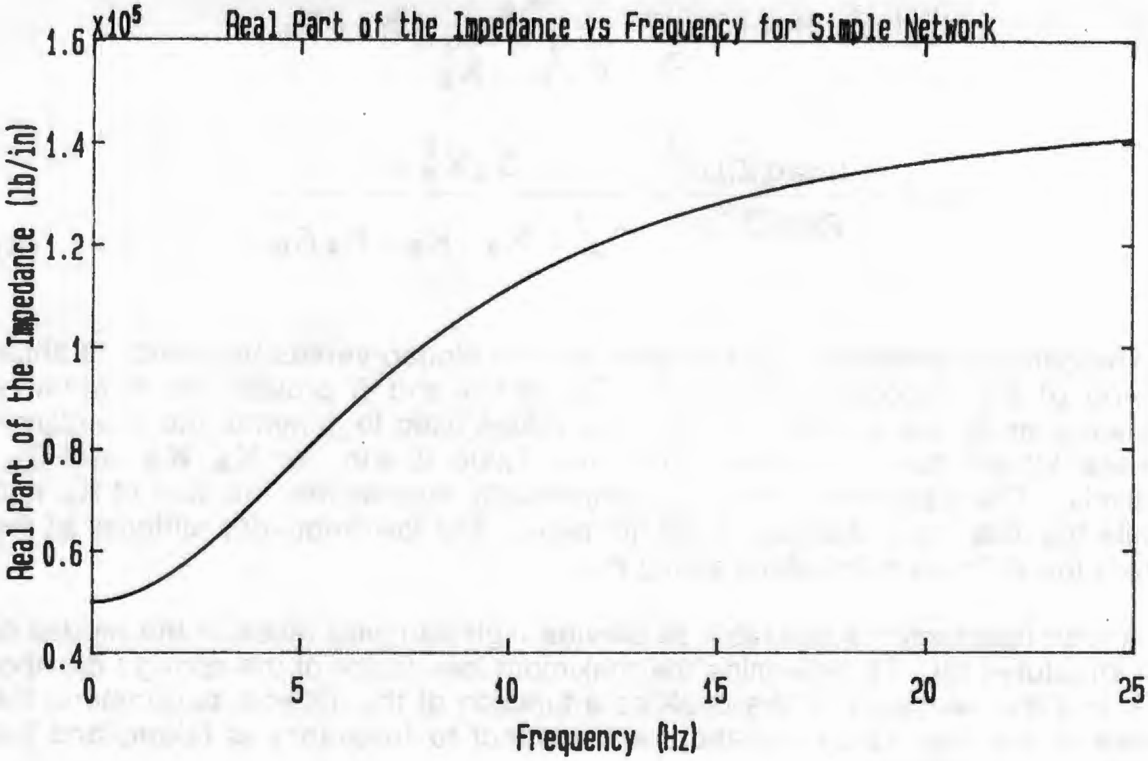
$$\omega_{max} = \frac{K_B}{C_A\sqrt{\dfrac{K_A + K_B}{K_A}}} \tag{6}$$

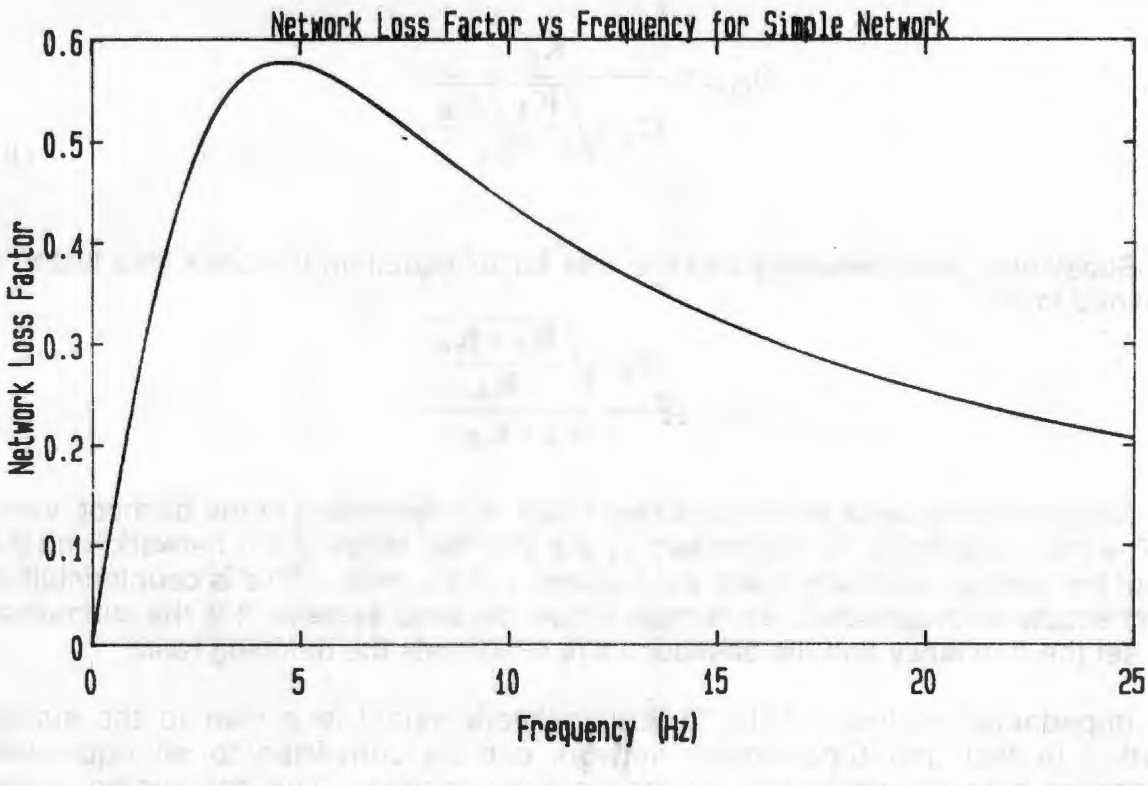
Substituting this frequency into the loss factor equation, the peak loss factor is determined to be:

$$\eta_{max} = \frac{K_B\sqrt{\dfrac{K_A + K_B}{K_A}}}{2(K_A + K_B)} \tag{7}$$

Notice that the value of the peak loss factor is independent of the dashpot value C_A. The peak loss factor is determined by the stiffness ratios in the network, and the value of the dashpot coefficient sets the frequency of the peak. This is counterintuitive to most structural dynamicists, as in most simple dynamic systems it is the stiffnesses which set the frequency and the dashpot value which sets the damping ratio.

Impedance analysis of the D-Strut network model is similar to the simple network. In fact, the 5-parameter network can be converted to an equivalent 3-parameter network which has an identical impedance. The conversion of the 5-parameter model to the 3-parameter model is useful to provide insight into the proper selection of the spring stiffnesses and dashpot values for the more complex





Figures 5 and 6 - Impedance Characteristics of the 3-Parameter Network

network. Solving for the values of K_A, K_B, and C_A of the 3-parameter model in terms of the parameters K_1, K_2, K_3, K_4, and C of the 5-parameter model yields:

$$K_A = \frac{K_1 K_2 + K_1 K_3 + K_2 K_3}{K_2 + K_3}$$

$$K_B = \frac{K_2^2 K_4}{K_2^2 + 2 K_2 K_3 + K_3^2 + K_4 K_2 + K_4 K_3}$$

$$C_A = \left[\frac{K_2}{K_2 + K_3}\right]^2 C \qquad (8)$$

The 3-parameter network is entirely equivalent to the 5-parameter network with this selection of the parameters, with any internal effects of inertias neglected.

The behavior of the spring / dashpot impedance as a function of frequency is very similar to the impedance characteristics of viscoelastic materials (VEMs). The network stiffness increases monotonically, while the loss factor shows a distinct maximum value. The maximum loss factor occurs in the transition region between soft and stiff behavior. The strut properties, as the viscous dashpot coefficient is altered, can be written in terms of a reduced frequency, which depends on both the forcing frequency and the dashpot coefficient. The similar characteristics of the spring / dashpot network and viscoelastics has previously been used to model viscoelastics materials as networks, as in Maxwell's and other models of damping material behavior.

The impedance representation of the strut properties provides insight into the design and efficient use of D-Strut members. The incorporation of these models into truss structures and evaluation of alternative structural modeling techniques was evaluated for use in the design and analysis of the PACOSS truss.

4.0 Modeling Techniques and Design Methodology for Structures Incorporating D-Struts

For D-Strut members to be used efficiently to provide damping to flexible truss structures, a coherent design methodology is required to provide structural designs which meet requirements with minimal additional weight and system impact. Modeling methods were developed which allow simple calculations to estimate the effects of incorporating D-Struts on system natural frequencies, damping ratios, and mode shapes. These methods were shown to be accurate and allowed the development of a simple design methodology. When applied to a structure, the methodology will provide efficient damping designs without the high cost associated with the solution of large complex eigenproblems.

A preliminary concept which must be developed is the conversion of a viscous system to one with complex stiffnesses (impedances). The concept can be extended to arbitrary systems with many networks and many degrees of freedom. In general, the Laplace transform of the free vibration equations of motion of an arbitrary system with viscous damping are:

$$\left[\mathbf{M}s^2 + \mathbf{C}s + \mathbf{K}\right]\mathbf{X} = 0 \tag{9}$$

where:
- $\mathbf{M}$ = system mass matrix
- $\mathbf{C}$ = system viscous damping matrix
- $\mathbf{K}$ = system stiffness matrix
- $\mathbf{X}$ = vector of transformed global displacements

By choosing an initial estimate to the eigenvalue, the transformation from a system with viscous damping to a system with a complex impedance can be obtained:

$$\left[\mathbf{M}s^2 + \mathbf{C}\sigma + \mathbf{K}\right]\mathbf{X} = 0 \tag{10}$$

define: $\mathbf{C}\sigma + \mathbf{K} = \mathbf{Z_R} + \mathbf{i} \cdot \mathbf{Z_I}$
where: σ is an initial eigenvalue estimate

The impedance matrix can be obtained as shown above, or by the assembly of elemental impedances:

$$\mathbf{Z}(s) = \sum_{i=1}^{NE} \mathbf{Z}_{\mathbf{R}_i}(s) + \mathbf{i} \cdot \mathbf{Z}_{\mathbf{I}_i}(s) \tag{11}$$

This conversion will typically be performed on an element level prior to assembly of the impedance matrix. The utility of the method is in noting that the impedance matrix has the form of a complex stiffness matrix. If iterations are performed to calculate an eigenvalue and eigenvector from the initial estimate, and the impedance is updated using the result as a new initial eigenvalue guess, the procedure will converge to an exact eigenvalue and eigenvector of the viscously damped system.

The transformation transfers the 2 N-size complex eigenproblem (solution of the 2 N real matrix eigenproblem in state form) to a 1 N-size eigenproblem with a complex matrix which must be solved iteratively. The iteration method can be used efficiently in conjunction with matrix iteration methods such as the inverse power method, since this method converges to a single eigenvalue and eigenvector at a time. In fact, the inverse power iterations performed in MSC/NASTRAN to calculate the complex modes of a viscously damped system are performed essentially by making this substitution and iterating with a shifted dynamical matrix. The real value of the transformation of the eigenproblem, however, is that approximate methods can be used to obtain modes, natural frequencies, and damping ratios without the solution of a complex eigenproblem.

The modal strain energy (MSE) method is a well known method for approximating the solution of a system with complex stiffness using only real eigenproblems. The MSE method is typically used to approximate the behavior of systems with viscoelastic damping treatments, and was used in previous PACOSS efforts in the design and analysis of the PACOSS DTA [4]. Similar to the D-Strut network, VEMs exhibit frequency-dependent complex stiffness behavior (frequency-dependent shear modulus and loss factor). Therefore, visco-elastic damping problems must also be solved iteratively by supplying an initial estimate to the natural frequency, substituting this value to determine the material complex modulus, and then solving a real eigenproblem to provide approximate mode shapes, frequencies, and damping ratios for modes "nearby" the initial frequency estimate.

Note that the member impedances should be calculated at the system pole (eigenvalue) for the analysis to provide the best approximation. For D-Struts, the impedance can readily be calculated at the system poles using the spring / dashpot network. Computation of the impedance at the system eigenvalue provides a more accurate approximation to the system behavior when used with the MSE method.

Due to the similarity of the D-Strut behavior and the behavior of viscoelastic damping struts, the methods previously developed for viscoelastic damping struts to select damping member locations and provide approximate system behavior can be directly used for the design of D-Strut damping treatments [4]. A simple methodology to be used when designing a truss structure with D-Strut members as a damping treatment is provided in Table 2.

This method allows the designer to achieve a satisfactory design for strength, natural frequencies, and damping ratios using only real eigenproblems, except for complex analyses at the end of the design cycle. It is apparent, when using this method, that the optimum locations for the dampers are areas of high strain energy, and that the sizing of the selected members for damping should be such that the maximum possible strain energy is contained in these members within the constraints of the problem. It is also evident that the damping members should have high loss factors at frequencies of the modes in which they have high strain energy to provide the highest system damping ratios.

A final benefit of the design method is that it allows the number or locations of dampers to be readily changed in the finite element model, since the only difference between the damper modeling and undamped member modeling is the member axial stiffness and member weight. Using this method, the only input data alteration required to change an undamped member to a damping strut is a property designation. With network modeling, a significant effort is required to add and/or remove additional nodes and element connectivities, when a damping element location is changed.

The D-Strut design procedure was exercised on several sample problems using the MSE method, in order to determine its accuracy and applicability for

Table 2 - Design Methodology for Incorporating D-Struts

1) Create a finite element model of the undamped structure.

2) Determine system natural frequencies and the required modal damping levels through simulation.

3) Select favorable damping locations for struts from MSE and loading considerations.

4) Size members such that high percentage of MSE is obtained in damping locations without causing an excessive shift in frequencies. Revise damping requirements based on altered mode shapes obtained in this step, if changes in the mode shapes significantly alter the performance.

5) Specify maximum additional weight over undamped members for the D-Strut members. Or, alternatively, specify the required D-Strut loss factors, since there is a direct relationship between strut weight and maximum achievable loss factor.

6) Specify the minimum static stiffness and strength for the damping struts.

7) Design a D-Strut such that the maximum weight is not violated, the designed stiffness and maximum loss factor are achieved at the frequency of highest D-Strut participation, and strength requirements are met. It is possible that all the constraints cannot be met, while simultaneously achieving the desired loss factor. The maximum weight constraint will then have to be relaxed, lower damping levels may be required, or additional damping locations must be selected.

8) Estimate the damped eigenvalues using the MSE method.

9) Calculate the strut properties at the damped eigenvalues.

10) Insert the damper properties into the model as an equivalent bar element; and recalculate system modes, natural frequencies, MSE. (Several runs may be necessary for several frequency and/or damping ratio values.)

11) Check the frequency and damping values to be sure that the eigenvalues have not changed significantly and, therefore, damper properties are accurate. Iteration may be required on the strut loss factors and system damping.

12) Determine if performance requirements have been met. If not, return to step 3 and select additional damping strut locations or alter D-Strut designs for higher loss factors.

13) When the design requirements have been met for all modes, model the dampers as spring / dashpot networks with K_1 and K_2 implemented as bar elements, and solve the complex eigenproblem. This will check the results and allow final complex modes to be used in simulations. Alternatively, use accelerated complex subspace iteration with the dampers modeled as frequency-dependent complex stiffness elements [5].

problems of this type. From these sample problems, several conclusions were drawn. Most importantly, it was shown that for light damping levels the procedure is accurate and effective. For modest damping levels, however, the MSE method often produces approximations which have relatively high error. Therefore, an alternative method of determining the properties of damped structures using real eigenproblems, termed the absolute value modal strain energy method [6], was developed which improves the accuracy of the solution for higher damping levels .

5.0 Preliminary Design of the D-Strut Truss

To ensure the success of the PACOSS D-Strut truss as a verification testbed while providing an economic validation tool, several goals for the structural design were established. The basic configuration selected for the truss consisted of eight bays with a 34-in. bay size.

The fundamental goal was to achieve high modal damping of major structural modes in the frequency range characteristic of future Large Space Structures (LSS). Modal damping levels of 10% critical viscous damping in the fundamental truss modes were selected as a goal for the truss design. This level of damping was to be achieved using 12 D-Strut members in locations selected to achieve the highest damping possible in the fundamental modes.

To provide data in the frequency range characteristic of future LSS, a fundamental frequency of 5.0 Hz was selected for the truss. In addition, a frequency separation of 0.5 Hz for the two fundamental bending modes (bending in two orthogonal planes) was desired, to allow the separation of the modes during the modal test and simplify the modal test data reduction problem.

Hardware design issues, such as joint design and member integrity, were addressed during the preliminary design process. Bonded joints similar to those used on other PACOSS structures were selected [4], primarily since they have been shown to provide strong connections with negligible inherent damping. Aluminum members were selected for the basic truss to provide the greatest economy.

A finite element model of the truss was created to allow for member sizing and preliminary design. A back-to-back K-diagonal pattern was selected to provide a structure with two planes of symmetry. This diagonal pattern results in two separated modes which have their primary motion along 45° axes with respect to the sides of the truss bays. The members were sized so that the major strain energy portions were in the longerons of the structure, as the damper locations were selected to be the lower longeron members due to the high modal strain energy content.

A plate located at the top of the truss was adjusted to achieve the desired 5.0-Hz frequencies; and the desired frequency separation between the fundamental modes of the truss were obtained by replacing the aluminum longerons in two of the vertical sides of the truss with steel members. This nearly achieved the desired 0.5-Hz frequency separation. Member offsets at the joints were included in the model to model the effective free length of the tubes when bonded in place.

An investigation of the effect of various tip weights was also undertaken for the truss. Various tip weights were incorporated into the truss model and analyzed to determine the natural frequencies and damping ratios. With tip weights which were twice the nominal, one-fourth, and without a tip weight, the fundamental modes of the truss could be altered within a range of 3.5 to 11.0 Hz. These values of the tip weight were selected for use on the truss so that the D-Strut members could be validated over a greater frequency range.

A summary of the truss properties following the preliminary design is included in Table 3. Notice that the 5.0-Hz frequencies with 0.5-Hz separation and high strain energy in the dampers were achieved with stock aluminum and steel sections for the members. This design provided an economically-manufactured structure which allows extraction of the two major modes due to the frequency separation and inherent structural symmetry.

Preliminary design for the damping members were generated based on the impedance equations for the strut network. The parameters of the network were selected to provide high damping levels with minimum strut weight. The hydraulic stiffness in the damping device was selected to be as high as was thought to be achievable, and the sizes of the tubing members were selected to provide the required loss factor with minimum weight. The dashpot coefficient was selected to place the peak loss factor at the frequency of the fundamental frequency of 5.0 Hz and, therefore, to supply the peak possible damping. Using the parameters of the preliminary D-Strut design, the damping in the truss was calculated using the MSE method for both the SD and D1 strut types. The damping ratios using these members are given in Table 4.

6.0 Detail Design, Fabrication, and Impedance Testing of the D-Struts

The preliminary D-Strut parameter selection information was provided to Honeywell to allow them to perform the detail design and fabrication of a prototype D-Strut member [7]. The detail design of the strut members was performed using the member design equations developed by Honeywell in prior internal research efforts.

Table 3 - Summary of D-Strut Truss Natural Frequencies and Strain Energy Distribution Following Preliminary Design

Mode Description	Frequency (Hz)	% SE in Dampers
1st Bending Mode (Plane 1)	4.98	61.7
1st Bending Mode (Plane 2)	5.44	66.0
1st Torsion Mode	14.26	4.0
2nd Bending Mode (Plane 1)	28.97	2.6
2nd Bending Mode (Plane 2)	30.84	1.7
Total Weight Including Joints and Tip Weight	267 lb	
Tip Plate Weight	113 lb	

Table 4 - Modal Damping Levels Using SD and D1 Strut Preliminary Designs

MODE DESCRIPTION	SD STRUT DAMPING	D1 STRUT DAMPING
Plane 1 1st Bending	13.1	19.1
Plane 2 1st Bending	14.0	20.4

From the detail design effort, it was apparent that the stresses within the viscous damping device and the achievable values for the device hydraulic stiffness were the major considerations in the design of the damping members. These constraints on the design eliminated the D1 strut from consideration, due to the very high damping element stresses for this configuration.

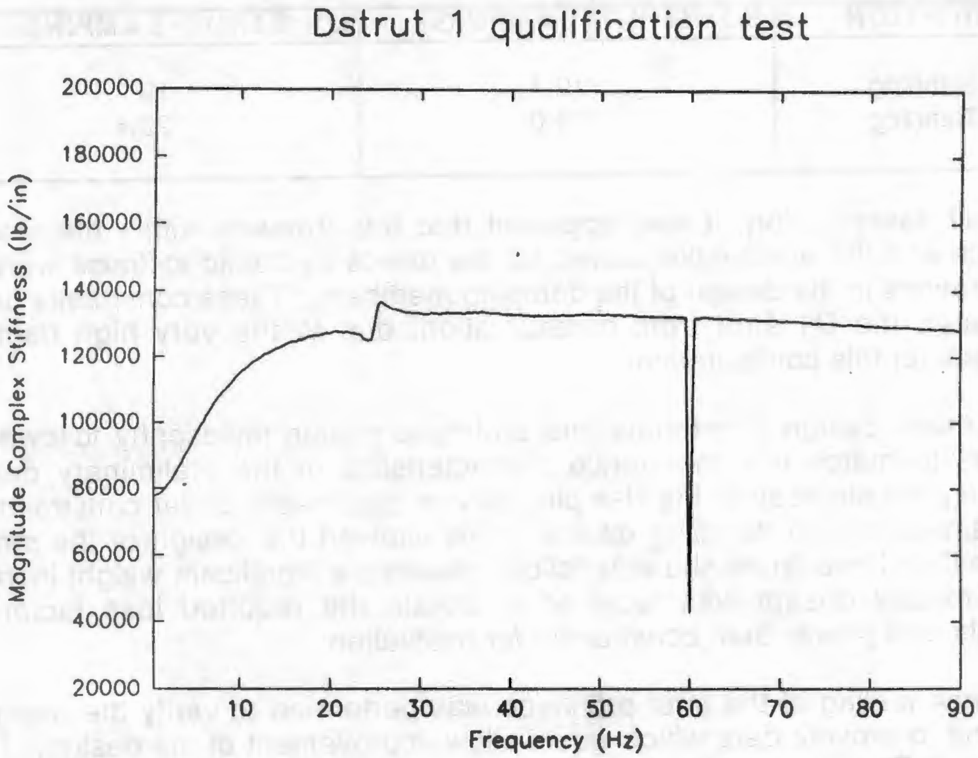
Due to these design constraints, the prototype design philosophy followed by Honeywell was to match the impedance characteristics of the preliminary design, while minimizing the stresses in the damping device diaphragm under constraints on the overall diameter of the damping device. This allowed the design of the strut to achieve the desired impedance characteristics, however, a significant weight increase over the preliminary design was required to obtain the required loss factor. A prototype of this design was then constructed for evaluation.

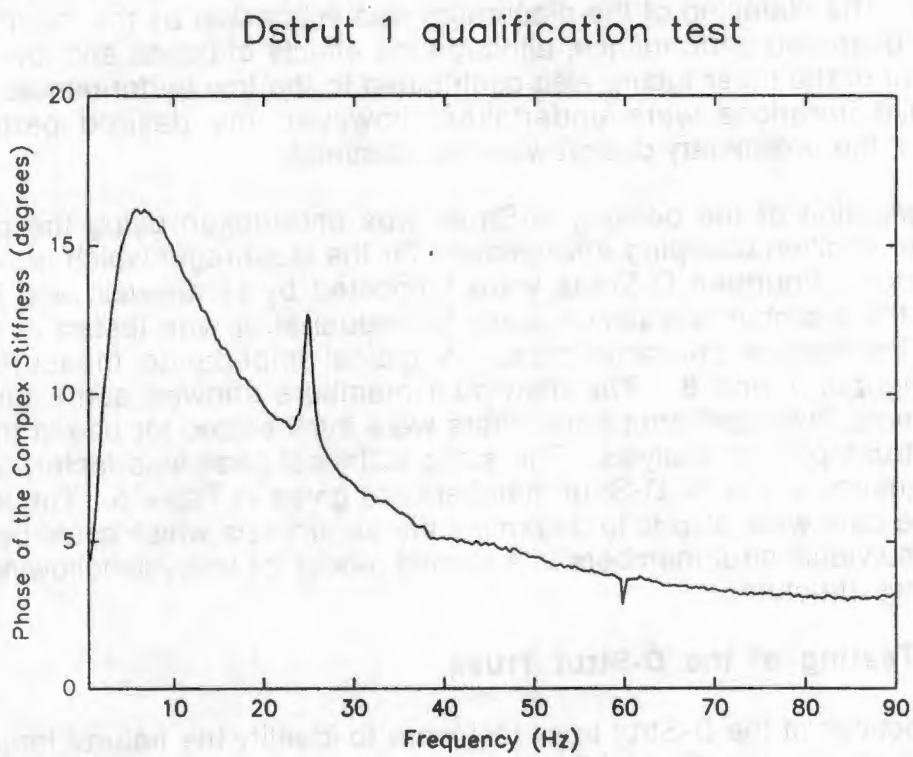
Impedance testing of the strut prototype was performed to verify the design of the member and to provide data which would allow improvement of the design. Tests of the prototype D-Strut showed that the strut possessed good linearity and provided a peak loss factor at the desired frequency. The peak loss factor, however, was lower than expected. The clamping of the diaphragm was suspected as the most probable reason for the degraded performance, although the effects of bonds and low modulus of the aluminum of the inner tubing also contributed to the low performance. Several design/test/build iterations were undertaken, however, the desired performance specifications of the preliminary design were not obtained.

The fabrication of the delivery D-Struts was undertaken using the prototype design with the modified clamping arrangement for the diaphragm which provided the best performance. Fourteen D-Struts were fabricated by Honeywell, and following completion of the member fabrication, each individual strut was tested to verify its strength and impedance characteristics. A typical impedance measurement is provided as Figures 7 and 8. The individual members showed some unit-to-unit variation, however, "average" strut parameters were synthesized for use in the model of the D-Strut truss pretest analysis. The static stiffness, peak loss factor, and peak loss factor frequency of the 14 D-Strut members are given in Table 5. The individual strut impedance data were also fit to determine the parameters which could be used to represent the individual strut members in a refined model for analysis following testing of the actual truss structure.

7.0 Modal Testing of the D-Strut Truss

The objectives of the D-Strut truss test were to identify the natural frequencies, mode shapes, and damping ratios of the test truss, and to allow for correlation with the truss modal analysis. The undamped truss assembly was first tested to validate the





Figures 7 and 8 - Measured D-Strut Impedance Function

Table 5 - Static Stiffness, Peak Loss Factor, and Peak Loss Frequency as Read from Impedance Plots

Strut No.	Static Stiffness	Peak Loss Factor	Frequency of Peak Loss
1	78,000	0.29	6.2
2	80,000	0.23	4.5
3	83,000	0.26	4.0
4	80,000	0.26	4.5
5	78,000	0.28	6.2
6	80,000	0.20	6.4
7	80,000	0.23	5.2
8	78,000	0.28	4.8
9	80,000	0.30	6.2
10	82,000	0.28	5.2
11	80,000	0.29	6.0
12	77,000	0.28	6.3
13	78,000	0.27	5.8
14	77,000	0.28	6.0

test fixturing and setup and allow the verification of the structural modeling without additional damping treatments. The nominal tip weight was used in the undamped truss test, which caused the fundamental frequencies of the truss to be close to the damped truss fundamental frequency of 5.0 Hz. A full modal survey of the undamped structure was performed to determine the structure natural frequencies, damping ratios, and mode shapes below 45 Hz.

The damped truss assembly was then tested with the nominal tip weight to determine the properties of the damped assembly with the D-Struts as the longerons in the lower three bays. Again, a full modal survey was performed on the truss to determine the modal properties of the damped assembly.

Finally, various tip weights were added to the truss to determine the truss properties over a significant frequency range. Sufficient data were acquired in each configuration to permit the identification of the natural frequencies, damping ratios, and mode shapes of the two lowest frequency structural modes, which are most greatly affected by the D-Strut members.

8.0 Analytic / Test Correlation of the Truss Modal Data

Following completion of the D-Strut truss modal test, correlation with the analytic model was evaluated. The undamped truss test results were initially compared with the analysis, and minor revisions to the model were made to improve the correlation. The model was compared with the test results in terms of the natural frequencies and mode shapes, with excellent agreement found for the modes below 45 Hz. Table 6 provides the analytic / test comparison for the modes of the undamped structure. Note the light damping ratios which were determined by parameter

Table 6 - Undamped Truss Tuned Analysis/Test Comparison

Measured Mode No.	Analytic Frequency (Hz)	Measured Frequency (Hz)	Measured Damping Ratio (%)
1	4.62	4.61	0.07
2	4.78	4.79	0.08
3	13.64	13.72	0.08
4	26.46	27.10	0.08
5	29.32	28.84	0.08
6	31.86	31.55	0.08
7	43.24	43.74	0.11

estimation. Decay measurements of the structure without cabling show damping of 0.01% critical in the fundamental mode.

Following correlation of the undamped structure, the model of the damped assembly was modified to reflect the changes in the undamped portion of the model. The truss properties with the nominal tip weight and each strut modeled as the "average" strut were then computed and compared with the test data. Excellent correlation was found for natural frequencies and mode shapes, and good agreement was found for the damping ratios. One final model modification was made which involved the incorporation of the properties of each individual strut in the proper strut locations, with the individual strut parameters determined from member impedance data. Table 7 provides the comparison of the damped structure modal analytic and test results. With the individual members incorporated, the agreement with the test data showed some improvement over the "average" strut model, although the previous model showed good agreement. Modal orthogonality checks were performed between the analytic and test modes using the analytic mass matrix. Table 8 provides the orthogonality results, which show outstanding agreement between the experimental and analytic mode shapes.

Finally, the refined model of the damped assembly was altered to reflect the various tip weights and reanalyzed to determine the truss properties. Again, these results agreed well with the measured data and verified the performance of the damping members at frequencies other than the peak loss factor location. Table 9 provides the frequency and damping correlation for the various tip weights.

The model correlation showed that accurate modeling of truss structures incorporating D-Struts was achieved using the 5-parameter D-Strut model. However, determination of the model parameters from impedance measurements on the individual struts was required. The test/model correlation of both the damped and undamped structures are excellent, particularly if the highly damped nature of the structure with the D-Strut incorporated is considered, and the difficulty in parameter estimation associated with these high damping ratios is recognized.

Table 7 - Nominal Tip Mass Tuned Analytic/Test Comparison with the D-Struts Modeled Using Parameter Fits

	Analytic		Measured	
Mode No.	Frequency (Hz)	Damping Ratio (%)	Frequency (Hz)	Damping Ratio (%)
1	4.98	7.23	5.00	6.59
2	5.10	9.62	5.25	9.43
3	13.61	<0.1	13.68	0.11
4	26.20	0.72	26.79	0.79
5	28.42	0.28	28.16	0.39
6	30.77	0.49	30.67	0.52
7	40.70	0.22	40.99	0.26

Table 8 - Cross-Orthogonality Matrix for Tuned Analysis of Damped Truss

Analytic Frequency

	4.98	5.10	13.61	26.62	28.42	30.77	40.70
5.00	1.00	0.00	0.00	0.00	-0.01	0.03	0.01
5.25	0.00	0.99	0.00	0.00	0.01	0.00	0.00
13.68	0.00	0.00	1.00	0.00	0.00	0.00	0.00
26.79	0.00	0.00	0.00	1.00	-0.03	0.00	0.00
28.61	0.00	0.00	0.00	0.03	1.00	0.02	0.00
30.67	0.00	-0.01	0.00	0.00	-0.02	1.00	-0.01
40.99	0.00	0.00	-0.01	0.00	0.02	0.00	1.00

Experimental Frequency

Table 9 - Tuned Analysis/Test Comparison with Various Tip Weights

	Measured Mode No.	Analytic Frequency (Hz)	Analytic Damping Ratio (%)	Measured Frequency (Hz)	Measured Damping Ratio (%)
Twice Nominal Weight	1 2	3.62 3.70	7.17 8.83	3.68 3.87	6.59 9.39
One-Fourth Nominal	1 2	7.85 7.98	6.17 8.82	7.84 8.18	6.70 8.67
No Tip Weight	1 2 3	10.46 10.45 10.68	7.40 5.08 4.27	10.27 10.87 10.97	7.23 5.91 5.52

9.0 Comparison of Viscoelastic Extensional Shear Dampers and D-Struts for Truss Damping Applications

The PACOSS DTA [2,4] includes damping strut members which were designed and fabricated using VEMs. For treatment of the box truss and equipment platform, viscoelastic extensional shear dampers (VESDs) were designed and incorporated into these truss structures. As the D-Strut is also designed to be used as a damping element for truss structures, a comparison of the state-of-the-art hardware for the two methods of damping treatment is beneficial.

To allow a direct comparison of the properties of these damping members, a viscoelastic damper was designed, which had an identical stiffness and loss factor at the 5.0-Hz frequency of the fundamental modes. Similar elements were designed, built, and tested under the PACOSS Program, and the analytic design equations for these members have been adequately verified. The pertinent properties of the VESD were then compared to those of the D-Strut.

The results of the comparison show that the D-Strut hardware which was developed for the PACOSS truss structure has one advantage over viscoelastic members, and several disadvantages. A comparison of the important characteristics of these two damping members is provided in Table 7. The primary advantage of the D-Strut is its reduced temperature dependency. The D-Strut has a ±40°F temperature range for a 10% variation in the impedance, while a similar viscoelastic member has only a ±5°F range.

Table 8 - Comparison of Important VESD and D-Strut Properties

	D-Strut	VESD
Peak Loss Factor/Frequency	0.275 / 6.0 Hz	0.285 / 4.0 Hz
Loss Factor at 5.0 Hz	0.270	0.280
Equivalent Stiffness at 5.0 Hz	96,000 lb/in.	94,000 lb/in.
Static Stiffness	78,000 lb/in.	54,000 lb/in.
Static Strength	600 lb	5,700 lb
Damped Element Weight	2.71 lb	1.74 lb
Added Weight / Undamped Weight	2.31	1.13
Required Temperature Control	± 40°F	± 5°F

The viscoelastic member, however, has a much higher load carrying capacity and adds less weight to produce a similar damping ratio. The added weight of the D-Strut member was roughly twice the added weight of a viscoelastic member. This added weight is primarily due to the low hydraulic stiffness (K_4) of the current damping device. The load carrying capacity of the D-Strut is currently very low, due to stress constraints within the damping device. These stresses cause the maximum load capacity of the D-Strut to be approximately a factor of 10 lower than its viscoelastic counterpart.

While, at this time, the comparison favors the viscoelastic member, future advances in the design of viscous damping members should exceed the capabilities of viscoelastic struts. The success of an alternative damping device which can sustain high deflections and has a much larger hydraulic stiffness can cause this comparison to favor the D-Strut. Damping devices which show almost no temperature dependence may be developed, which will eliminate the need for temperature control of the strut members.

10.0 Conclusions

From the PACOSS investigation of the D-Strut, it was determined that efficient MSE techniques in conjunction with member impedances can be used to design both the damped structure and to optimize the characteristics of the damping members. Final analysis of a damped structure can then be performed by incorporating a spring / dashpot model of the damping struts directly into the finite element model.

The D-Strut members which were fabricated and tested qualitatively agreed with the 5-parameter spring / dashpot model developed by Honeywell. Quantitatively, however, the parameters of the model derived from the design equations for the member did not accurately predict the performance of the viscous fluid damping device. In particular, the predicted hydraulic stiffness and the diaphragm stiffness were significantly different from the values predicted by the model. The lower hydraulic stiffness caused the damping of the truss test structure to be approximately 35% lower than anticipated during preliminary design.

Following fabrication and testing of a strut prototype, each individual D-Strut member was dynamically tested to determine its impedance characteristics. Testing of the individual strut members provided data for the determination of the parameters of the strut model which would provide an impedance consistent with the measured data for each unit. The model of the truss was modified to reflect these data.

An undamped lower section of the structure was also fabricated to determine the accuracy of the model of the undamped section and verify that very light inadvertent damping was present. Decay testing of the undamped truss verified an extremely low damping ratio of 0.01% critical, which demonstrates the light damping ratios expected for precision truss structures without damping augmentation. With the D-Strut members incorporated into the truss, the damping ratios of the damped structure were approximately 7.5% and 10.0% critical for the first two structural modes. This represents a 4-order-of-magnitude increase in damping over the undamped structure. Four tip weights were used on the truss, which allowed a significant variation in the fundamental frequencies and verified the performance of the damping members over a significant frequency range.

When the modal data as synthesized from measurements are compared with the analytic data, the correlation in excellent. The first seven natural frequencies of the tuned model of the undamped structure agree with the test data to within 2%, and other measures of model accuracy show good agreement. The model of the damped structure also has similar accuracy in terms of natural frequencies, and all damping ratio predictions agree to within 9% relative error. This agreement verifies that the modal properties of damped structures which incorporate D-Strut members can be accurately predicted with finite element models and the Honeywell 5-parameter model, if the appropriate parameters of the model are determined from member impedance test data.

Finally, a comparison of the D-Strut to the PACOSS VESD was made to determine whether the potential advantages of the D-Strut were achieved. From the comparison, it was concluded that the only current advantage of the D-Strut is its larger temperature range. The current D-Strut design was shown to be heavier than a VESD which was designed to produce identical frequencies and damping ratios when incorporated into the truss. The D-Strut was shown to have only a 600-lb strength (due to the stress limitations of the damping element), while the VESD design had a strength of 5,700 lb. This lower strength of the current D-Strut design may preclude its use in situations where high loads are present.

While the current design of the D-Strut has some deficiencies, alternative designs of the viscous fluid damping device may reduce or eliminate these shortcomings. In particular, if the viscous damping element can be designed to have a much higher hydraulic stiffness than the current device, the weight of the damper for a given performance will be significantly decreased such that it will be lighter than its viscoelastic counterpart. Similarly, if the damping element can be designed to sustain high deflections, the strength of the element will be comparable to the VESD. Future work should be undertaken to investigate alternative designs of the damping element.

The PACOSS investigation has shown that the viscous fluid damping strut can be successful in producing high damping ratios for truss structures. The device can be modeled accurately from impedance test data, and properties of structures which incorporate these devices are predictable. Refinement of the design of the damping element will make the D-Strut concept extremely successful and attractive for incorporation into damped trusses for space applications.

References

1) Wilson, J. F., and Davis, L. P, "Very High Damping in Large Space Structures," The Role of Damping in Vibration and Noise Control, ASME Publication, 1987.

2) Gehling, R. N., "Large Space Structure Damping Treatment Performance - Analysis and Test Results," The Role of Damping in Vibration and Noise Control, ASME Publication, 1987.

3) Johnson, C. D., and Kienholz, D. A., "Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers," AIAA Journal, Vol 20, No 9, pp 1284-1290, 1982.

4) Morgenthaler, D. R., "Design and Analysis of Passively Damped Large Space Structures," The Role of Damping in Vibration and Noise Control, ASME Publication, 1987.

5) Morgenthaler, D. R., "Practical Design and Analysis of Systems with Fractional Derivative Materials and Active Controls," Presented at Damping '91, West Palm Beach, Florida, 12-15 May 1991.

6) Morgenthaler, D. R., "The Absolute Value Modal Strain Energy Method," Presented at Damping '91, West Palm Beach, Florida, 12-15 May 1991.

7) Cunningham, D. C., "Development of the PACOSS D-Strut," Presented at Damping '91, West Palm Beach, Florida, 12-15 May 1991.