Influence of Temperature on Structural Joints with Designed-In Damping

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An innovative means to enhance the inherent damping in structures is provided by the designed-in incorporation of viscoelastic materials in joints. The joints, as envisioned, are double-lap shear joints that dissipate energy when worked in an axial direction. The damping and stiffness characteristics of such joints have been evaluated experimentally at various temperatures and frequencies that are expected to be representative for large space structures. A new, nonresonant experimental technique has been utilized for this investigation. It provides the complex stiffness of the test specimen at extremely low strain levels, and accounts for elastic deformations in the test setup by means of carefully measured calibration factors. The temperature and frequency variations of the overall joint properties follow, in general, the same trends as the corresponding properties of the particular viscoelastic material used in the joint. The test data show that properly selected viscoelastic materials and design configurations can reduce the dependence of the joint properties on temperature and frequency variations. Significant damping benefits are possible without unacceptable stiffness penalties.

Nomenclature

A = amplitude of exciter displacement in unloaded condition
B = amplitude of force signal from load cell
C = equivalent compliance of test setup, defined in Eq. (6)
K = axial stiffness of test specimen
K_1 = internal stiffness of piezoelectric exciter
K_2 = stiffness of load cell
K_3 = test fixture stiffness in the loading direction
K_{ref} = reference stiffness ($K_{\text{ref}} = 225{,}000$ lb/in.)
$P(t)$ = axial load applied on the test specimen
t = time variable
w = circular frequency
$x_u(t)$ = output motion of piezoelectric exciter in unloaded condition
$x_L(t)$ = output motion of piezoelectric exciter in loaded condition
Δ = parameter defined in Eq. (5)
η = loss factor of test specimen
θ = phase angle between displacement and load signals

Introduction

MISSION requirements for future space structures dictate extremely large dimensions by today's standards. Several important missions, such as defence systems, orbiting telescopes, or large antennae require very tight orientation tolerances, short settling times, and low vibration levels. The stringent controllability requirements of large space structures have spawned intensive research and development efforts in the vibration control area. They are generally divided in two basic approaches: active modal control and passive damping enhancement. A balanced approach, which properly combines active and passive means, is widely accepted today as the most realistic and practical solution to motion control of large space structures. High passive damping not only limits vibration amplitudes and shortens transient decay times, but also has favorable synergistic effects when it is combined with active controls.[1] Consequently, intensive research efforts have been directed recently toward the development of effective means for artificial damping enhancement.[2-4]

An innovative means to enhance the inherent damping in structures is provided by the designed-in incorporation of viscoelastic materials in joints. It combines the well-known damping capability of viscoelastic materials[5] with the predominant influence that joints and supports have on the overall damping of most structures.[6] As opposed to the commonly used add-on approach for damping treatments, the designed-in approach provides a promising opportunity to maximize the damping benefit, while minimizing the associated penalities in other structural properties.

Preliminary theoretical and experimental research for the development of passively damped joining concepts has been a recent cooperative undertaking of McDonnell Douglas Astronautics Co. and Georgia Institute of Technology. Two new experimental techniques developed during the initial phase of the program for dynamic characterization of such joints have been presented at the "Vibration Damping Workshop I."[7] They are referred to as the "simplified steady state" and the "sine pulse propagation" methods. The first yields the complex stiffness by using a nonresonant forced vibration approach, whereas the second provides the energy dissipated by damping, by using a stress wave propagation approach. Their application to room temperature testing of passively damped joint specimens and the corresponding results have been described in Ref. 8. A detailed description of the test specimens, their design criteria, assumptions and methodologies, and their fabrication procedures appears in Ref. 9. Two new analytical models have been developed during this research program for design analy-

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ses of passively damped joints of double-lap configuration. The first is a fully elastic model presented in Refs. 7 and 9, whereas the second is a quasistatic model that employs the complex modulus concept to describe the viscoelastic behavior of the adhesive layers.[10] Good correlation has been obtained between the two models as well as between analytical predictions and experimental results.[10] A detailed presentation of the room temperature tests, the analytical modeling, and parametric studies is given in Ref. 11. A complete overview of the entire program, including low- and high-temperature tests, is reported in Ref. 12. The concept of passively damped joints for vibration control has been extended lately from the initial double-lap configuration to alternate configurations, such as single lap[13] or rhombic shape.[14]

This paper deals only with the low- and high-temperature tests on representative specimens of passively damped joints. Its objective is to complete the reporting of results generated by the research program mentioned above. It is associated primarily with Ref. 8, since it shows how some of the room temperature damping and stiffness properties presented in that reference may change when measured either at low or high temperatures. The environmental temperature tests have been conducted on eight specimens at temperatures of −50, 25, 75, and 200 °F, at four of the test frequencies investigated in Ref. 8, namely 0.25, 1.0, 25.0, and 100.0 Hz. Since damping enhancement in large space structures was the primary motivation of this program, the tests have been confined to low-strain levels and frequency and temperature ranges considered to be representative for such structures.[1,8]

Test Specimens

The passively damped joint specimens selected for low- and high-temperature tests are listed in Table 1. Six different types of viscoelastic materials, which are considered to be suitable for space applications, have been used in these specimens. Their density, manufacturer, trade names, and chemical base are given in Table 1.

All the specimens are based on the symmetric double-lap configuration of bonded joints in which the conventional elastic adhesive is replaced by viscoelastic material. The damping enhancement is achieved by shear deformation of the viscoelastic adhesive layers when the joint members are loaded in their axial direction (Fig. 1). The use of this joining concept is based on the hypothesis that it is possible to obtain a favorable tradeoff when structural stiffness is exchanged for increased structural damping. The need to accept a stiffness penalty arises from the requirement to make the joints somewhat flexible so that a reasonable amount of strain energy is resident in the joints. This strain energy is then available for dissipation by the viscoelastic material selected expressly for this purpose.

A slightly different configuration of a double-lap joint has also been investigated as a possible way to achieve better tradeoffs between the damping benefit and the associated stiffness penalty. It includes a direct glass-fiber connection between the members of the joint, as shown in Fig. 2. In addition to higher axial stiffness, this elastic link provides structural redundancy at elevated temperatures, and in the case of viscoelastic materials with poor creep resistance. Two of the specimens listed in Table 1 include such elastic connection elements. They

Table 1 Specimens for environmental temperature tests

		Viscoelastic material		
Specimen no.	Density, lb/in.^{3}	Manufacturer	Name	Type
1	0.035	3M	ISD 110	Acrylic
3	0.046	3M	EC 2216	Epoxy
5	0.035	Soundcoat	DYAD 606	Polyurethane
9	0.029	GE	SMRD 100F90	Epoxy
11	0.046	GE	RTV 630	Silicone
15[a]	0.046	3M	EC 2216	Epoxy
18[a]	0.035	Soundcoat	DYAD 606	Polyurethane
21	—	MDAC	—	Rubber plastic

[a]Includes elastic connection elements.

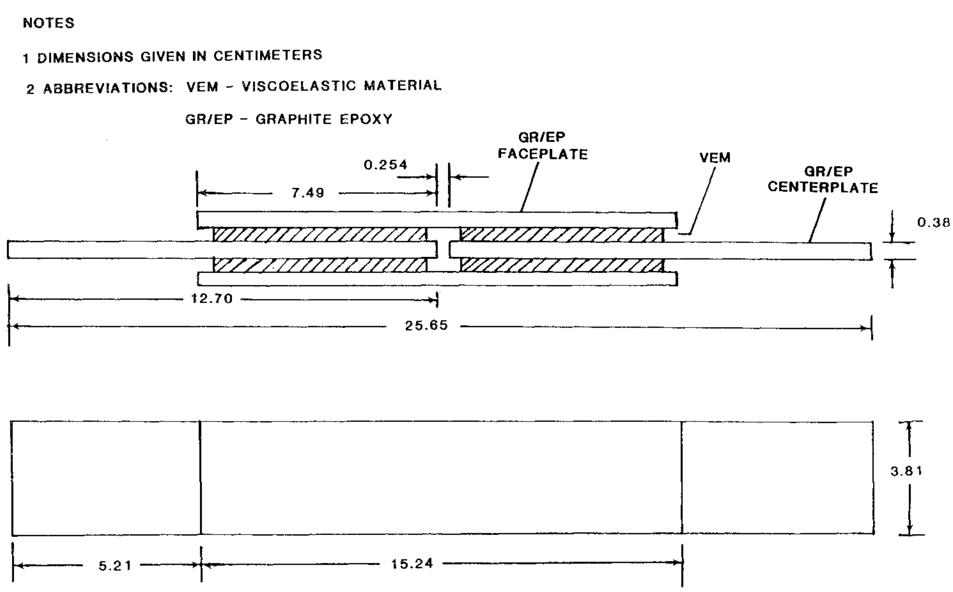

Fig. 1 Passively damped joint specimen without elastic link between members.

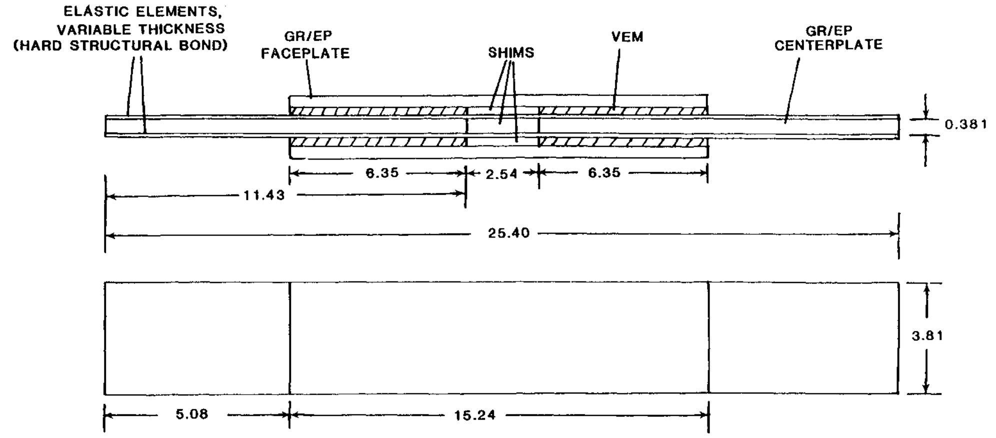

Fig. 2 Passively damped joint specimen with elastic elements.

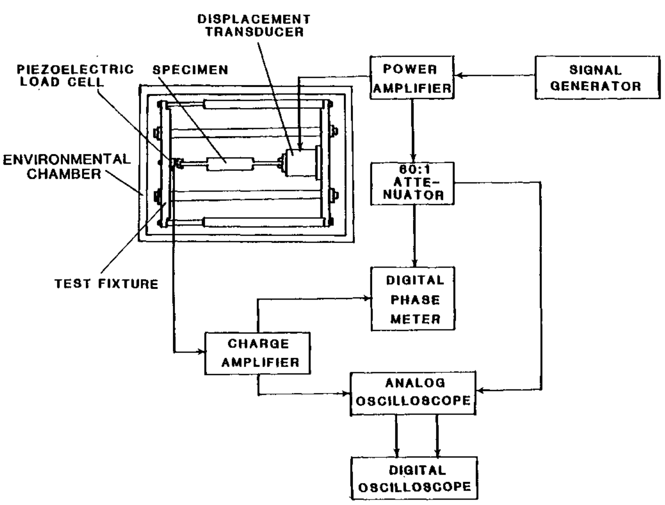

Fig. 3 Experimental setup for the simplified steady-state approach.

are specimens 15 and 18, which use the same viscoelastic materials as the regular double-lap specimens 3 and 5, respectively. The analytical models developed for passively damped joints (Refs. 9 and 10) account for the effect of these elastic links on the overall properties of the joint.

The dimensions shown in Figs. 1 and 2 are typical of all the test specimens. The only geometric parameter that varies significantly from one specimen to another is the thickness of the adhesive layers. Its value is about 0.007 in. for specimens 1 and 11, about 0.05 in. for specimens 5, 9, 15, and 21, and about 0.125 in. for specimen 3. The axial stiffness of certain specimens could have been increased by using thinner adhesive layers when feasible from the manufacturing standpoint.

Summary of Room Temperature Experiments

The damping and stiffness characteristics of selected joint specimens have been measured at room temperature (75 °F) and eight different frequencies (−0.1, 0.25, 0.5, 1.0, 10, 25, 50, and 100 Hz). Because the primary candidates for large space structures are repetitive lattice trusses, the main loading direction considered in the design and testing of these specimens was the axial one. The main experimental procedures and results are presented in Ref. 8.

Three different experimental methods have been used for the room temperature tests. Two methods are new and have been developed during this research program. They are referred to as the simplified steady state and the sine pulse propagation techniques, respectively. In addition, the classical hysteresis-loop approach has been applied on all the specimens in order to enhance the data base at very low frequencies. New and innovative data acquisition and reduction procedures, based on advanced digital instrumentation, have been utilized for all the three testing techniques. Good correlation has been obtained among the data generated by different methods in the same testing conditions, as shown in Ref. 8.

The results reported in Ref. 8 show that favorable tradeoffs between the damping benefit and the associated stiffness penalty can be achieved if the designed-in approach is adopted. The damping characteristics of soft joints, like specimen 1, are dominated by those of the adhesive materials, but as these materials become stiffer the joint loss factor is reduced from the damping material loss factor, since the viscoelastic layers share less of the total strain energy stored in the joint.[10] The stiffness increase with frequency, as indicated by the test data in Ref. 8, may yield, therefore, a reduction in the damping performance of the joint at higher frequencies, even for a viscoelastic material with a relatively flat loss factor-frequency behavior.

Low- and High-Temperature Tests

The environmental temperature tests have been performed on all the specimens listed in Table 1 at four different temperatures within the range expected for space structures: −50, 25, 75, and 200 °F. These tests have been confined to four of the eight frequencies used in the room temperature tests, namely 0.25, 1.0, 25.0, and 100 Hz.

Only the simplified steady-state technique has been utilized for the enviromental temperature tests, since it is expedient and correlates well with the other two methods applied in the room temperature experiments. The hysteresis-loop approach has been excluded, since it is applicable only for very low frequencies, usually below 0.5 Hz. The sine-pulse propagation approach is preferable in the case of multiple-branch joints, or when accurate absolute values of damping in transient phenomena are needed. It does not require a custom-made, piezoelectric motion transducer, as needed for the simplified steady-state method. However, the fact that all the equipment required for the simplified steady-state method was already available from the room-temperature tests, has facilitated its selection for low- and high-temperature tests. In addition, this approach is more closely related to conventional measurement and interpretation concepts of damping data, and it permits a faster data reduction procedure than the sine-pulse propagation method.

The simplified steady-state method is described in Refs. 7, 8, and 11. The only difference between the experimental setup of the room temperature tests and that of the environmental temperature tests is the placement of the test fixture inside an environmental chamber, as shown in Fig. 3. This is a BLUE-M LN-270C-1 constant-temperature environmental chamber with mechanical convection horizontal airflow. It uses liquid nitrogen as the cryogenic agent. Because of the highly corrosive nature of this cooling medium, the test fixture and the dynamic response pickups had to be made of stainless steel. The chamber can provide a temperature range from -180 to 550°F, which has not been fully utilized in these tests in order to avoid any potential irreversible damage to the joint specimens.

When a harmonic voltage is applied to the piezoelectric exciter, it generates a small axial harmonic motion that is transmitted through the test specimen and results in an axial harmonic force measured by the piezoelectric load cell. If the displacement output of the exciter in unloaded condition, i.e., with no specimen attached to it, is expressed as

$$x_u(t) = A \cos wt \tag{1}$$

where w is the circular frequency (rad/s), then the force signal measured by the load cell is

$$P(t) = B \cos(wt + \theta) \tag{2}$$

where the phase lag θ is caused by the viscoelastic behavior of the test specimen. By using the complex stiffness concept of viscoelasticity,[15] the following equations are developed for the real and imaginary stiffness components of the specimen.[7]

$$\mathrm{Re}(K) = \frac{1}{\Delta}\left(\frac{B}{A}\cos\theta - \frac{B^2}{A^2}C\right) \tag{3}$$

$$\mathrm{Im}(K) = \frac{1}{\Delta}\frac{B}{A}\sin\theta \tag{4}$$

where

$$\Delta = 1 - 2C\frac{B}{A}\cos\theta + C^2\left(\frac{B}{A}\right)^2 \tag{5}$$

$$C = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} \tag{6}$$

In the loaded condition, the displacement amplitude of the exciter departs from A because its internal stiffness K_1 is not infinite. The actual axial motion can be determined from the equilibrium condition between the external reaction force and the internal force generated by the piezoelectric effect. Its expression is

$$x_L(t) = P(t)\left(\frac{1}{K} + \frac{1}{K_2} + \frac{1}{K_3}\right) \tag{7}$$

which is derived by assuming that the total external force is the resultant of three elastic restoring elements connected in series, namely the specimen, the load cell, and the test fixture whose stiffness parameters are K, K_2, and K_3, respectively. The equivalent compliance C in the above equations is a correction term that accounts for the stiffness contributions of the test fixture, load cell, and displacement transducer. It can be neglected when the stiffness of the specimen is much lower than the combined stiffness of these elements.

The elastic stiffness of the test specimen is calculated from Eq. (3), whereas its overall loss factor is given by

$$\eta = \frac{\mathrm{Im}(K)}{\mathrm{Re}(K)} \tag{8}$$

The parameters A, B, and θ are determined from the two harmonic signals acquired in each test—the input voltage applied to the piezoelectric exciter, and the response of the load cell. These signals are averaged, stored, and analyzed on separate channels of a NICOLET 4094A digital oscilloscope. The load phase lag θ, with respect to the applied displacement, is determined by the difference between the zero crossings of the corresponding time waves. After possible "dc" components are eliminated from the two signals, the displacement amplitude A and the force amplitude B are calculated by using the appropriate calibration factors. The sensitivity of the PCB piezoelectric load cell was about 17 pC/lb, whereas the sensitivity of the piezoelectric exciter in unloaded condition was about 0.2μ in./V. The latter was measured in preliminary calibration tests at each of the selected test frequencies, and the exact value corresponding to each frequency has been used in calculating A.[11] Since the amplitude of the input voltage to the motion transducer was about 300 V in all tests, its maximum stroke was about 60μ in., with the exact value varying slightly from one frequency to another.[11] A schematic of the data analysis procedure in the simplified steady-state method is shown in Fig. 4.

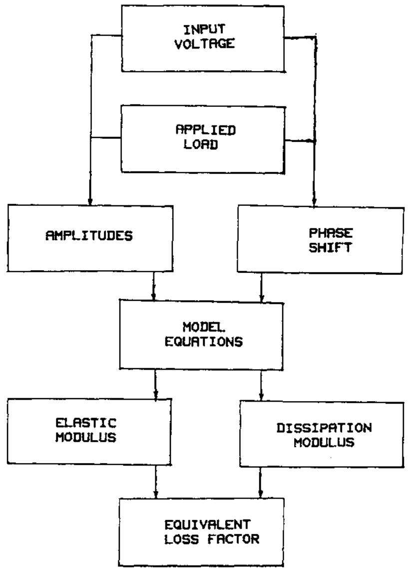

Fig. 4 Data analysis diagram for the simplified steady-state approach.

The primary drawback of the simplified steady-state approach utilized in the experiments is the tedious preliminary calibration tests that are required for measuring the equivalent compliance C. They include two different sets of experiments: one in unloaded configuration of the piezoelectric motion transducer, and the other in its loaded configuration. A reference aluminum specimen, whose axial stiffness is 225,000 lb/in., is mounted in the test fixture when the loaded sensitivity of the exciter is measured. A piezoelectric ENDEVCO accelerometer with high sensitivity (about 500 pC/g) is used to measure the displacement output of the motion transducer in the calibration tests. The data analysis for these tests is performed in the frequency domain by using a HP 5423A Dynamic Analyzer, as described in detail in Ref. 11.

The equivalent compliance C is calculated from the equation

$$C = \frac{A}{B} - \frac{1}{K_{\mathrm{ref}}} \tag{9}$$

where K_{ref} is the stiffness of the reference specimen ($K_{\mathrm{ref}} = 225{,}000$ lb/in., in the present case). Equation (9) is obtained from Eqs. (3) and (5) by substituting Re(K) by K_{ref} and assuming that the reference specimen has no damping, i.e., $\theta = 0$. Since repeated measurements of C revealed a yet unexplained sensitivity to both frequency and temperature, its value has been determined separately for each combination of frequency and temperature selected for the tests. These values are shown in Table 2. For each test, only the corresponding value of C from Table 2 has been used in the data analysis according to Eqs. (3)–(6). One may notice that the stiffness contributions included in the parameter C, Eq. (6), become more significant as the temperature increases or the frequency decreases. This behavior can possibly be related to improper mounting and preloading of the stack of piezoelectric crystals in the motion transducer. However, a thorough analysis of the exciter is needed before the problem can be clearly identified and corrected.

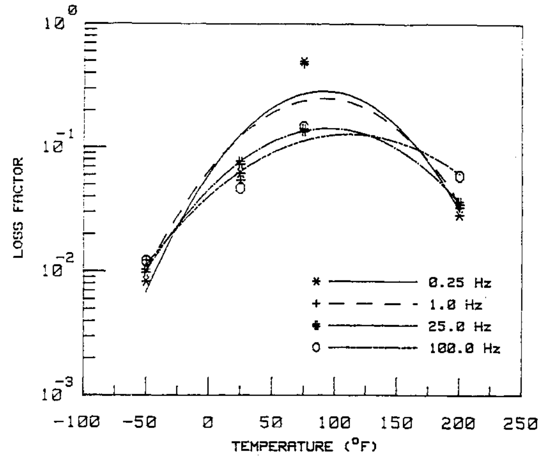

Fig. 5 Temperature effect on loss factor of specimen no. 3.

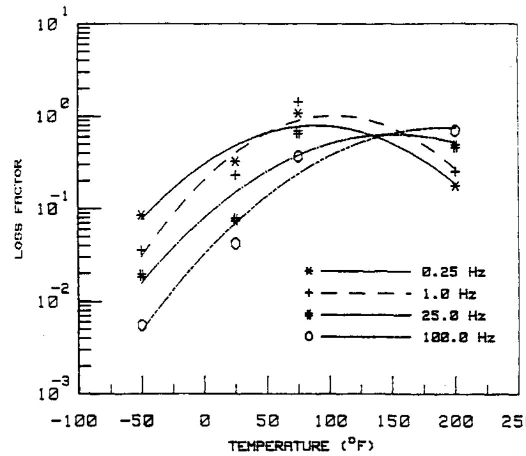

Fig. 7 Temperature effect on loss factor of specimen no. 1.

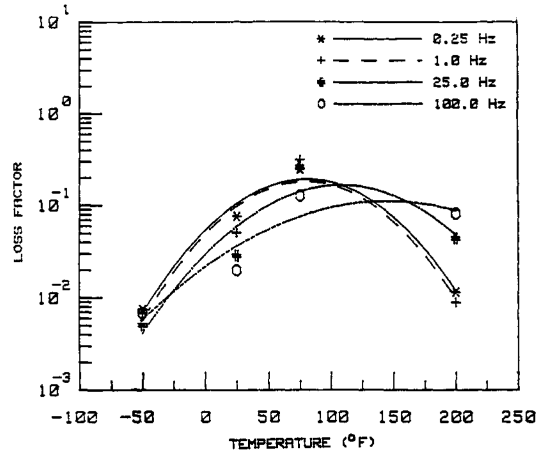

Fig. 6 Temperature effect on loss factor of specimen no. 9.

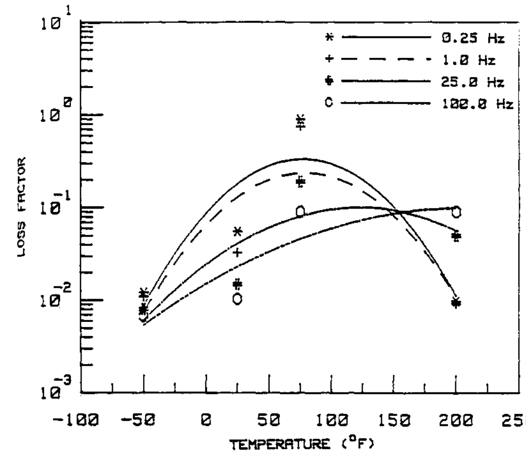

Fig. 8 Temperature effect on loss factor of specimen no. 5.

Table 2 Equivalent compliance values of the test setup (C) for test frequencies and temperatures (in./lb × 10^6).

	Frequency, Hz			
Temp., °F	0.25	1.0	25.0	100.0
−50	1.84	1.77	0.98	0.69
25	1.97	1.97	1.22	0.91
75	2.53	2.57	1.80	1.45
200	4.49	4.87	3.94	3.79

Results and Discussion

The low- and high-temperature test results are presented here in the same format as the room temperature test results in Refs. 8 and 11. The conventional format of reduced frequency nomograms[16] is not necessary in this case because of the low number of data points. Although the number of data points is not sufficient to determine exact numerical values over the whole test temperature and frequency ranges, general trends can be established. The "best fit" of a given set of data points is obtained, as in the room temperature tests, by using a least-square fitting routine, and is shown as a line in the following figures.

Figures 5–8 show that the damping performance of regular double-lap joints with EC 2216, SMRD 100 F90, ISD 110, and DYAD 606 viscoelastic materials, respectively, varies strongly with temperature. In most cases the loss factor peaks at room temperature, approximately, but drops sharply at low and high temperatures. Such a behavior is expected because the viscoelastic materials used in the joints are usually designed to perform best at room temperature. It should be emphasized that the damping data shown in Figs. 5–8 are expressed in terms of

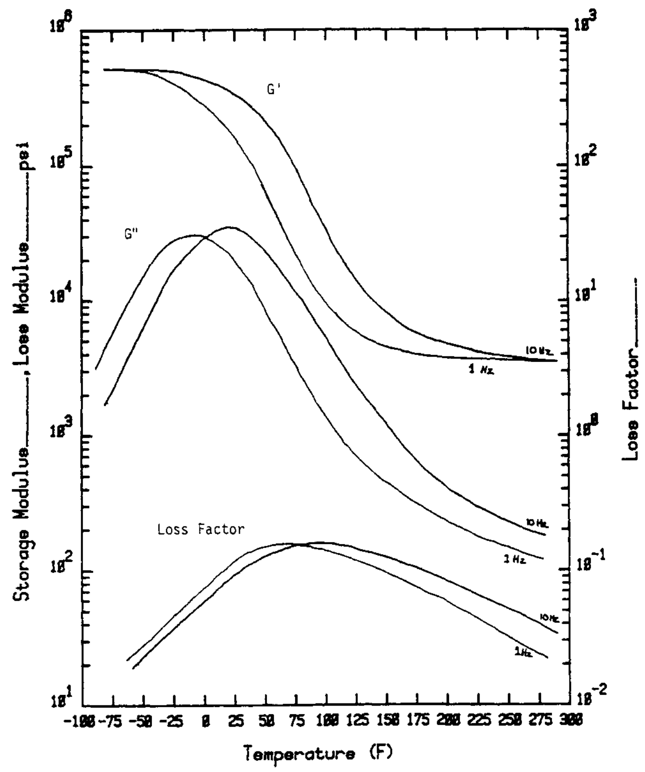

Fig. 9 Damping properties of EC 2216 (Anatrol, Inc.).

the overall loss factor of the joint system rather than the material loss factor of its adhesive. The analytical model described in Ref. 10 provides a uniquely defined relationship between these two loss factors. Material properties of the EC 2216 and the GE SMRD viscoelastic adhesives are illustrated in Figs. 9 and 10, respectively. One may notice that the joint damping shown in Fig. 5 follows the same trends as the corresponding material damping in Fig. 9. A similar conclusion can be drawn by comparing Fig. 6 with Fig. 10.

For all of the specimens shown in Figs. 5–8, the loss factor dependence on frequency is much weaker than its sensitivity to temperature changes. This observation agrees with Figs. 9 and 10 and with the room temperature test results, as shown in Ref. 8. At low temperatures, the loss factor tends to decrease as the frequency increases. At high temperatures, however, the loss factor increases with frequency. These trends are illustrated in Fig. 11 for specimen 3.

Because the design tradeoffs between potential damping benefits and the stiffness penalties associated with them are an important topic of this research, the stiffness characteristics of specimen 3 are shown in Fig. 12. If Fig. 12 is analyzed in conjunction with Fig. 11, one may identify a certain temperature range for the corresponding viscoelastic material within which a favorable tradeoff between damping and stiffness may be achieved. Figure 12 shows a significant stiffness reduction with increasing temperature, but a much lower sensitivity to frequency changes. The "relative stiffness" parameter used in this figure is the ratio between the axial stiffness of the actual joint specimen and that of a continuous reference specimen whose stiffness is 585,000 lb/in.

Two of the viscoelastic materials listed in Table 1 seem to be more suitable to applications in which good damping performance is required over a wide temperature range. One is RTV 630 used in specimen 11, and the other is a new rubber plastic material manufactured by McDonnell Douglas Astronautics Company[12] and used in specimen 21. The foregoing conclusion is based upon the relatively flat behavior displayed by these specimens in Figs. 13 and 14 over the test temperature range. Small peaks can still be observed in the room temperature region.

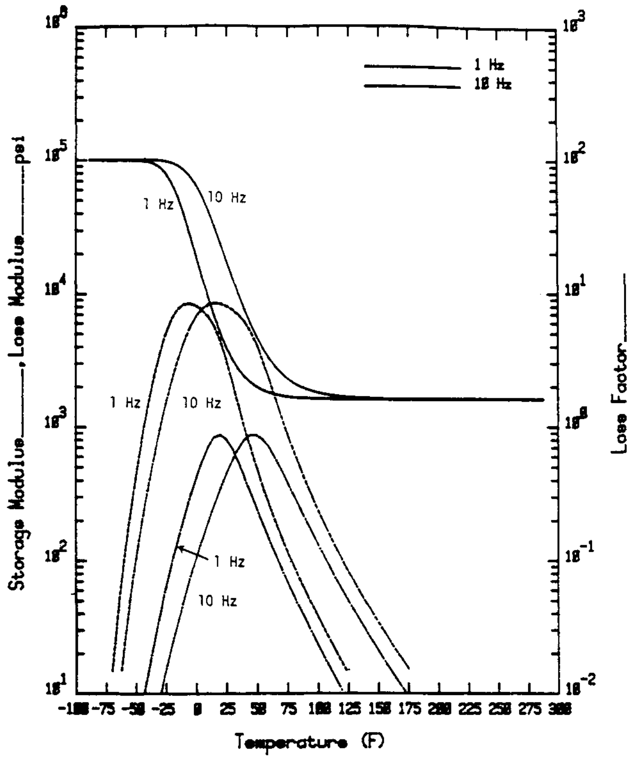

Fig. 10 Damping properties of SMRD 100F90 (Anatrol, Inc.).

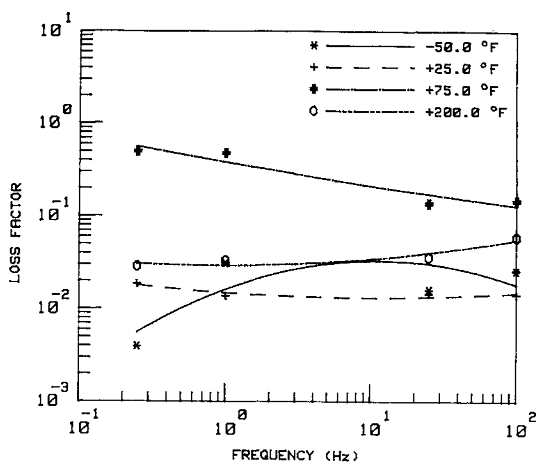

Fig. 11 Frequency effect on loss factor of specimen no. 3.

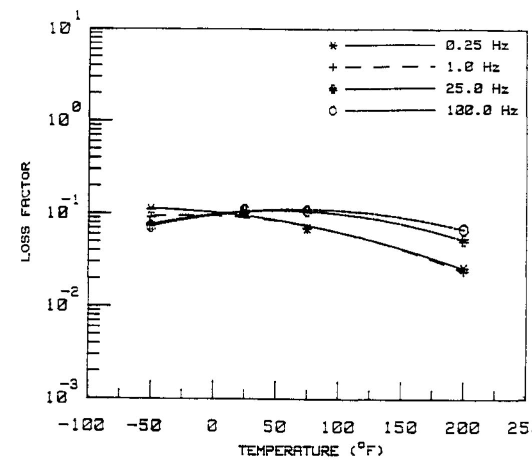

Fig. 13 Temperature effect on loss factor of specimen no. 11.

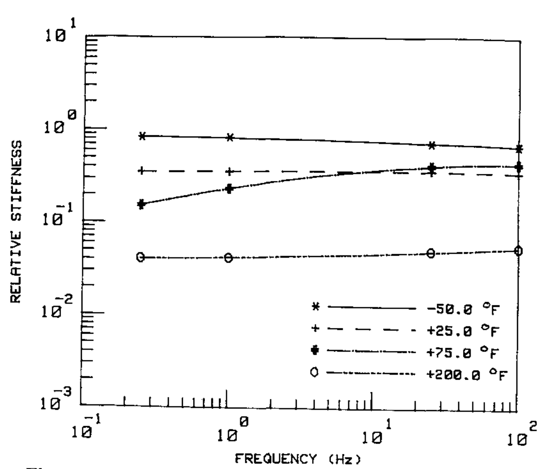

Fig. 12 Frequency effect on stiffness of specimen no. 3.

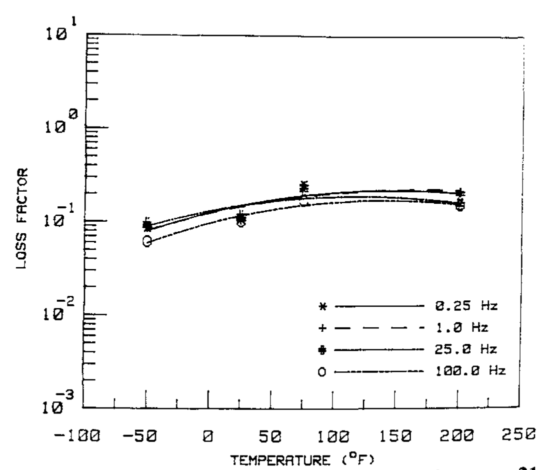

Fig. 14 Temperature effect on loss factor of specimen no. 21.

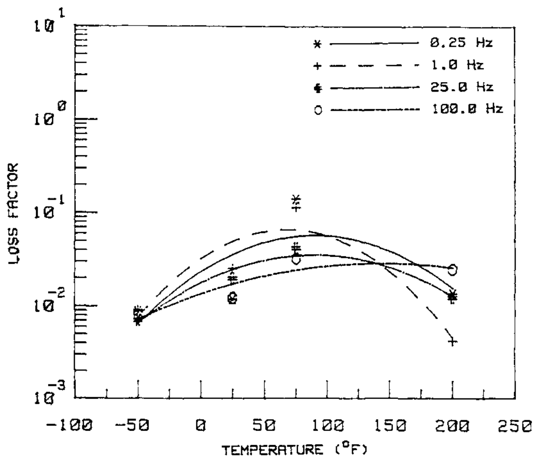

Fig. 15 Temperature effect on loss factor of specimen no. 15.

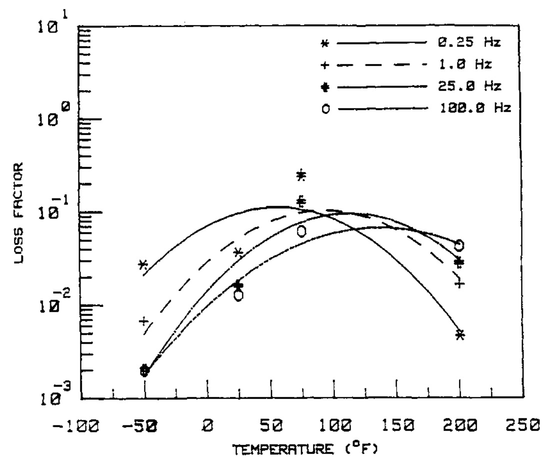

Fig. 16 Temperature effect on loss factor of specimen no. 18.

The effect of the elastic connection elements between the members of the joint (Fig. 2) upon its overall loss factor can be analyzed by comparing Fig. 15 to Fig. 5 for the EC 2216 viscoelastic material, or by comparing Fig. 16 to Fig. 8 for the DYAD 606 viscoelastic material. In both cases, as expected, the elastic links reduce the damping benefit, but they also reduce its variations with temperature and frequency changes. This is because the influence of the viscoelastic material properties on the overall properties of the joint is reduced with such a configuration, and the structural interactions among the joint constituents become more dominant.[10]

A good indication of the reliability of the test data generated by the simplified steady-state method can be obtained by comparing the results obtained on certain specimens during the environmental temperature tests with those measured on the same specimens at the same frequencies during the room temperature tests. Since the two series of tests have been performed at a time interval of about one year with no temperature measurement during the room temperature tests, an approximate agreement among the results should be considered satisfactory. Such an agreement is, indeed, observed by comparing the room temperature (75 °F) data reported in this paper with the corresponding results presented in Ref. 8.

Concluding Remarks

Passively damped joints provide a cheap, simple, and efficient means to enhance the inherent damping in space structures. However, careful consideration should be given to their design configuration and material selection in order to achieve favorable tradeoffs between the damping benefit and the associated stiffness penalty. Potential penalties that have not been analyzed in this research, such as strength, weight, and cost, must also be considered.

The test results show, in general, that the overall damping and stiffness characteristics of axially loaded double-lap joints are determined primarily by the material properties of their viscoelastic adhesive layers. For instance, the temperature variations of the joint properties follow the same qualitative trends as the corresponding properties of the particular viscoelastic material used in the joint. These variations can be reduced by tailoring the material selection to the temperature range within which the joint is expected to operate. When it is not possible to choose viscoelastic materials with relatively weak temperature dependence, such as RTV 630 or the rubber plastic alloy of McDonnell-Douglas Co., the use of thermal protection on the joints may be considered for wide temperature range applications.

The simplified steady-state technique provides a reliable characterization of passively damped joint specimens if sufficient care is taken in measuring and using the various calibration factors included in its equations. The accuracy of the test data generated by this method is extremely sensitive to possible errors in the calibration factors. Since some of these factors may change, not only with the test frequency and temperature, but also because of slight mounting inconsistencies of the specimens in the test fixture, the calibration factors should be rechecked any time a new specimen is mounted in the fixture. The calibration problem is greatly reduced in the sine-pulse propagation approach, which may be considered a good alternative to the simplfied steady-state method when sufficient time is available for data reduction.

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