Constrained Layer Damping For A
Space-Based Optical System

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Abstract

Pointing requirements for present space-based optical systems dictate state of the art precision structural designs. Future space-based optical systems will require even tighter pointing budgets. A vibration control technique, such as constrained layer damping, is an attractive approach for reducing vibration induced pointing error in these systems. This paper presents the results of a study to integrate constrained layer damping in the design of a graphite/epoxy truss structure. This precision structure is representative of those designed for space-based laser communication systems. Several constrained layer damping treatments were analytically and experimentally evaluated for a component of the structure. The component results were used to analytically predict system performance. Analysis of the system, with constrained layer damping integrated in the design, predicts an order of magnitude reduction in vibration induced pointing error. This reduction in pointing error results in decreased power requirements for the laser. Lower laser power leads to longer laser life and higher reliability. This results in a lower system weight and cost.
Introduction

A study was undertaken to evaluate the increased performance benefits of integrating constrained layer damping in the design of a representative structure of an optical system. The representative structure, shown in Figure 1, was designed for a space-based laser communication system. The design required very tight tolerances and a thermally stable structure to meet the on-orbit performance requirements. The result was a graphite/epoxy truss structure with very low intrinsic damping, on the order of 0.2% of critical damping. Constrained layer damping was attractive for this structure to significantly increase the damping and reduce the jitter response. The objective of the damping design was to reduce the nominal line of sight error in the optical system by an order of magnitude.

Another consideration in the damping design was that the modal frequencies above 100 Hz were of interest. Below 100 Hz, an active control system was used for beam steering. For modes above 100 Hz, although magnitudes of displacement response were low, the vibration induced jitter contributed directly to the line of sight error of the system.

The components with high strain were identified from the nominal system structural response. Several constrained layer damping designs were identified which were consistent with the temperature and disturbance frequency bandwidth for the typical structure shown in Figure 1. These designs were analytically evaluated for a 'representative' graphite/epoxy truss component of the system.

The Modal Strain Energy technique\textsuperscript{1,2} was used to calculate the structural system response with constrained layer damping. The constrained layer damping design led to predicted order of magnitude reduction in pointing error for a 1% system weight increase.

Testing was performed on a 'representative' component, a tubular graphite/epoxy truss member to validate the analytical system level predictions. Several constrained layer damping designs were experimentally evaluated to verify the range of damping values predicted in the analytical model.
Approach

Constrained layer damping has been shown to be more efficient than unconstrained layer damping for a given weight. To obtain constrained layer damping, a viscoelastic damping material is sandwiched between the structure and a constraining layer. A cross section of constrained layer damping is shown in Figure 2. The constraining layer forces the viscoelastic layer to deform in shear, which is the optimum manner to dissipate energy in a viscoelastic material. The performance of the constrained layer system depends on stiffness and geometry of the constraining layer and viscoelastic layer, the environmental conditions, and the location of the passive damping in the system.

The stiffness and geometry of the constraining layer and viscoelastic layer is characterized in the shear parameter $g_n$:

$$g_n = \frac{G_2 \lambda_n}{E_C h_C h_0}$$

where:
- $G_2$ Shear modulus of viscoelastic material
- $\lambda_n$ Semi-wavelength for $n^{th}$ mode
- $E_C$ Young's modulus for constraining layer
- $h_C$ Constraining layer thickness
- $h_0$ Damping layer thickness

For this application, a constraining layer stiffness equal to the stiffness of the structure produces the maximum shear strain in the viscoelastic layer.

Environmental conditions which determine the performance of the constrained layer damping system include operating temperature range, magnitude and frequency of dynamic excitation, and the response frequency range. Additionally, the viscoelastic material must be able to "survive" the non-operating temperature range. Outgassing of the viscoelastic material for space applications must be minimal since outgassing can result in material contaminating the optical system or degradation of the performance of the damping design. The environmental conditions, the response of the structure to these conditions, and the damping requirements are important elements in the damping design.

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The performance of constrained layer damping system is influenced by its location in the structural system. Constrained layer damping is applied to areas of the structure which experience the highest strain levels during dynamic jitter response. The Modal Strain Energy approach\textsuperscript{1,2} is a technique to analytically assess the effectiveness of passive damping in a structure.

The Modal Strain Energy method is based on the assumption that the damping in a built up structure may be expressed as the sum of the damping dissipated by each of its structural components. The components with passive damping, having high damping with respect to the remainder of the components in the system, contribute to the equivalent modal damping. This equivalent modal damping is used when predicting the damped system structural response.

The shear parameter, the environmental conditions, and the location of constrained layer damping in the system are important elements that must be merged with system requirements such as performance and weight to develop an acceptable constrained layer damping design.

Analysis

Constrained layer damping was integrated in the design of a representative graphite/epoxy truss structure for a "typical" space-based optical system. The thermally stable graphite/epoxy structure, shown in Figure 1, was attached to the satellite at three points. An on-orbit disturbance was assumed to originate in the satellite due to events such as momentum wheel imbalances, slewing of solar arrays, thruster firing and drive the base of this representative structure. The goal of the constrained layer damping was to reduce the vibration induced jitter which contributes to the line of sight error in the optical system.

The following environmental conditions were assumed to develop the constrained layer damping design:

- Operating temperature range: 0°F to 86°F
- On-orbit dynamic disturbance: 100 to 150 Hz; 1 microradian at the base.
Using a finite element model of the truss structure, the on-orbit dynamic disturbance was input at the attach points. The line of sight error for the dynamic disturbance, as a function of frequency, is shown in Figure 3. The two modes, which contribute to the peak response shown in Figure 3, were selected for vibration control. The high strain areas for these modes were identified. About 25% of the strain area for the two modes resided in two pairs of truss members shown in Figure 4. These four truss member were selected as candidate for application of constrained layer damping.

A finite element model of a representative graphite/epoxy truss component was developed to evaluate different constrained layer damping designs. A 50 inch graphite/epoxy truss tube was modeled as shown in Figure 5. To simulate the dynamic behavior (modal frequencies) of the component in the structural system, the translation was restrained at one end and a mass moment of inertia was applied to the other end. The constrained layer damping design chosen consisted of four 0.25" width damping strips, covering the center two-thirds of the truss tube. The stiffness of the constraining layer and viscoelastic material were varied during the analysis (Figure 6). Strain energy increased with increasing stiffness of the constraining layer and viscoelastic material. DYAD 606, ISD 110 and SMRD 100F90 were candidate viscoelastic materials since they have been qualified and used in other space applications\(^4\). The SMRD 100F90 and a 0.20" graphite/epoxy constraining layer was one damping design that yielded a 2% equivalent modal damping for the two modes of interest. The resultant line of sight error, shown in Figure 7, met the objective of an order of magnitude reduction in dynamic jitter. This damping design resulted in a 1% increase in system weight.

Test Program

A test program was developed for a representative component of the system to validate range of damping values predicted analytically. The test configuration is shown in Figure 8. Six graphite/epoxy truss tubes, 54 inches in length, were fabricated as test articles. The lay up and fabrication procedure for the graphite/epoxy tubes were identical to the components in the structure of the optical system. Soft springs were attached to each end of the tubes to approximate free-free end conditions. Testing was conducted at Soundcoat, Inc.

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Due to time constraints some compromises were made in selection of material for the constraining layer and the viscoelastic material. Steel was chosen for the constraining layer because of lower cost and ready availability. Three different steel constraining layer thicknesses were chosen, 1/16", 3/32" and 1/8" thick. SMRD 100F90 could not be obtained in the time available. The DYAD 606 and ISD 110 were readily available from the manufacturers. DYAD 601 was chosen for testing since its performance was better in the 0 to 86°F operating temperature range than the DYAD 606, which was analytically evaluated. The material compositions of DYAD 601 and 606 are very similar, therefore outgassing properties should be the same.

The graphite/epoxy tubes were excited by a force impulse from an instrumented hammer with a piezoelectric force transducer. Impact measurements were made at the center of the tube and 26 additional points spaced two inches apart. The accelerometer was mounted at an antinode for mode two of the beam, 12 inches from the end. The tubes were mass loaded at their ends and third points, in order to obtain the second mode frequency of 110 Hz. The intent was to approximate the second modal frequency and mode shape of the component in the complete structural system.

The test matrix is shown in Figure 9. The objectives of the test were threefold. First, to determine damping and modal characteristics for the first three mode, with mode two of particular interest. Second, determine damping measurement dependence on shear parameter, temperature and frequency. Third, to determine if the damping is the same for low and high excitation force levels.

**Test Results** - The six test configurations were tested with and without constrained layer damping treatments at room temperature. For two test specimens the temperature was varied from 30 to 80°F.

Second mode test results for the six test configurations, before applying the constrained layer damping treatment and with the constrained layer damping treatment, are presented in Figure 10. The results are compared to a value of 2% damping which is required to obtain an order of magnitude reduction in the vibration induced jitter. The intrinsic damping in the component without

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constrained layer damping is partly due to the weights bolted to the end fittings and the weights attached at the third points.

A comparison of damping versus temperature for sample 2 and sample 4 is shown in Figure 11. The DYAD 601 loss factor for 100 Hz peaks between 50 and 60°F. The ISD 110 material loss factor peaks above 80°F. Material data for the DYAD 601 and ISD 110 confirm the loss factor peaks (at 100 Hz) are at 50°F and 110°F respectively. The DYAD 601 with a 3/32" steel constraining layer meets the 2% damping objective over the test temperature range (30 to 80°F). The material data for the DYAD indicates that below 30°F the damping value would fall below 2%. Another viscoelastic material is needed in the damping design to achieve the 2% damping over the entire operating temperature range.

The stiffness of the constraining layer versus damping relationship, shown in Figure 12, exhibits the same trend as the analytical data. Measured damping increased with stiffer constraining layer. The objective of 2% damping was met by all damping designs (at room temperature) except the DYAD 601 with 1/16" thick steel constraining layer.

Linearity of response with respect to input level was verified for the dynamic range of the test system. The measured damping response for low excitation force (response acceleration of .06 g) was the same as measured response at the high excitation force (response acceleration of 3 g).

Conclusions

It has been shown analytically that integrating a constrained layer damping treatment in the design of a space-based optical system can yield an order of magnitude reduction in the vibration induced line of sight error for the system. Testing of a graphite/epoxy truss component confirmed the range of damping used in the analytical model. Reducing line of sight error for space-based optical systems can have significant system benefits in terms of reduced laser power requirements, increased communication reliability, and longer laser life. Reduced pointing error can also result in performance improvements such as increased communication data rates, lighter weight designs, and better target discrimination.

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References


TYPICAL PRECISION STRUCTURE FOR SPACE-BASED OPTICAL SYSTEMS

Figure 1

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CONSTRAINING LAYER

VISCOELASTIC MATERIAL (VEM) DAMPING LAYER

EXTENSIONAL STIFFNESS RESISTS ELONGATION AND "CONSTRAINS" THE VEM

BENDING CAUSES SURFACE ELONGATION

- DIFFERENCE IN ELONGATION CAUSES VEM TO DEFORM IN SHEAR
- VIBRATION CYCLE DISSIPATES ENERGY UNDER STRESS-STRAIN HYSTERESIS LOOP OF VEM

CONstrained LAYER DAMPING

Figure 2

JITTER RESPONSE WITH INTRINSIC DAMPING (0.2 % Cc)

Figure 3
AREAS OF HIGH STRAIN ENERGY FOR 107 AND 115 HZ MODE

Figure 4

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FINITE ELEMENT MODEL OF GR/EP TUBE WITH CONSTRAINED LAYER DAMPING

Figure 5

PARAMETRIC STUDY OF CONSTRAINED LAYER DAMPING DESIGNS

Figure 6
JITTER ATTENUATION WITH CONSTRAINED LAYER DAMPING

Figure 7

TEST CONFIGURATION

Figure 8
<table>
<thead>
<tr>
<th>SAMPLE NUMBER</th>
<th>VISCOELASTIC DAMPING MATERIAL DESCRIPTION</th>
<th>CONSTRAINING LAYER DESCRIPTION</th>
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<tr>
<td>1</td>
<td>DYAD 601A - 80 MILS</td>
<td>STEEL - 1/16&quot; THICKNESS</td>
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<tr>
<td>2</td>
<td>DYAD 601A - 80 MILS</td>
<td>STEEL - 3/32&quot; THICKNESS</td>
</tr>
<tr>
<td>3</td>
<td>DYAD 601A - 80 MILS</td>
<td>STEEL - 1/8&quot; THICKNESS</td>
</tr>
<tr>
<td>4</td>
<td>ISD 110 - 90 MILS</td>
<td>STEEL - 1/16&quot; THICKNESS</td>
</tr>
<tr>
<td>5</td>
<td>ISD 110 - 90 MILS</td>
<td>STEEL - 3/32&quot; THICKNESS</td>
</tr>
<tr>
<td>6</td>
<td>ISD 110 - 90 MILS</td>
<td>STEEL - 1/8&quot; THICKNESS</td>
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PASSIVE DAMPING DESIGNS FOR COMPONENT TESTING

Table for Measured Damping Values with and Without Constrained Layer Damping

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<tr>
<th>SAMPLE NUMBER</th>
<th>SAMPLE DESCRIPTION</th>
<th>INTRINSIC DAMPING $%C_c$</th>
<th>DAMPING W/ VEM $%C_c$</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>DYAD 601, 1/16&quot; STEEL CONSTRAINING LAYER</td>
<td>0.7</td>
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<tr>
<td>2</td>
<td>DYAD 601, 3/32&quot; STEEL CONSTRAINING LAYER</td>
<td>0.5</td>
<td>2.5</td>
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<tr>
<td>3</td>
<td>DYAD 601, 1/8&quot; STEEL CONSTRAINING LAYER</td>
<td>0.5</td>
<td>2.5</td>
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<tr>
<td>4</td>
<td>ISD 110, 1/16&quot; STEEL CONSTRAINING LAYER</td>
<td>0.5</td>
<td>3.2</td>
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<tr>
<td>5</td>
<td>ISD 110, 3/32&quot; STEEL CONSTRAINING LAYER</td>
<td>0.5</td>
<td>3.9</td>
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<tr>
<td>6</td>
<td>ISD 110, 1/8&quot; STEEL CONSTRAINING LAYER</td>
<td>0.6</td>
<td>4.3</td>
</tr>
</tbody>
</table>

MEASURED DAMPING VALUES WITH AND WITHOUT CONSTRAINED LAYER DAMPING

Figure 10

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COMPARISON OF MEASURE DAMPING VALUES VERSUS TEMPERATURE

Figure 11

MEASURED DAMPING VALUES VERSUS STIFFNESS OF CONSTRAINING LAYER

Figure 12