Proc. of SPIE Vol. 2445, Smart Structures and Materials 1995: Passive Damping, ed. C D Johnson (May 1995) Copyright SPIE

Highly-damped exactly-constrained mounting of an x-ray telescope

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ABSTRACT

Instruments and machines requiring very high stability should be isolated from their normally less stable environment. Exact constraint mounting using six, single-constraint flexures provides a stiff connection between the instrument and its environment while isolating the instrument from low frequency deformations of the environment, such as thermal expansion. Higher frequency disturbances, however, transfer through the flexures and excite vibration modes of the instrument. Traditionally, passive or active vibration isolation is employed to attenuate environmental disturbances reaching the instrument. However, strict alignment requirements for the instrument preclude the use of low-frequency isolation, unless active methods are used. Therefore, the solution is to provide damping in parallel with the flexures to reduce the vibration amplitudes of the instrument. Flexures concentrate strain energy in blades making them excellent candidates for damping treatments. A properly-designed damping treatment across the flexures can provide as much as 8% to 10% viscous damping to the isolation modes and will also help attenuate the instrument vibration modes. Thus, through the use of six damped single-constraint flexures the instrument's requirements for stability, alignment, stress, and vibration may be met.

An application of this approach will be employed on the Reflection Grating Array (RGA) for the X-ray Multi-mirror Mission for the European Space Agency. The RGA is an array of 200 diffraction gratings aligned to sub-micron and sub-arc-second tolerances relative to each other. This produces a coherent wavefront for spectrum analysis. The launch vehicle will be an Ariane 5 scheduled for 1998.

Keywords: passive damping, damped mounts, exact constraints, telescope, optics kinematic mount

1 INTRODUCTION

The Reflection Grating Array on board the European Space Agency's (ESA) X-ray Multimirror Mission (XMM) is a component of a soft x-ray spectrometer for high resolution study of cosmic x-ray sources in the 5-35 angstrom spectral regime. The RGA is a collaboration between Lawrence Livermore National Laboratory (LLNL), the University of California at Berkeley, and the Space Research Organization of the Netherlands. The

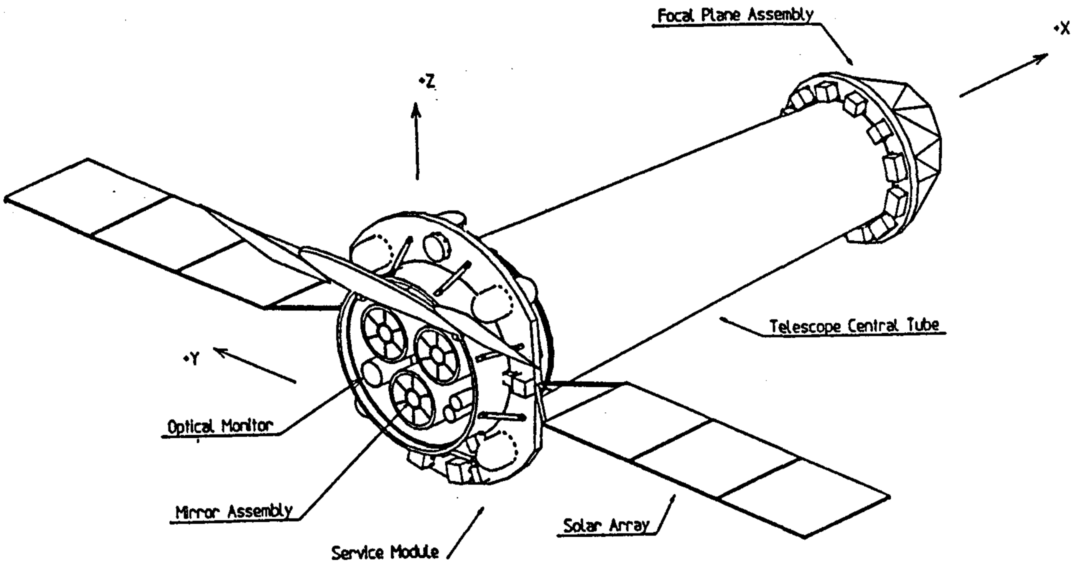

Figure 1: XMM satellite

RGA is a passive optic with no electrical components. Its function is to diffract soft x-ray wavelength light to a focus on CCD detectors at the focal distance of 6.7 meters. The optics are positioned at grazing incidence of 1.57 degrees within the cone of a converging beam produced from a Wolter Type II telescope, see Figure 1. There are three identical telescopes, two of which are used for redundant spectrometers. The grating arrays reside behind the telescopes by approximately one meter. A CFRP tube, three meters in diameter and 8 meters long, separate the telescopes from the focal plane. The thermal environment around the telescopes and RGA's is actively controlled to 20° C ± 10° C.

In each RGA, there are 202 variable line density diffraction gratings that are precision-aligned with respect to each other to sub-arc-second in rotation and 40 e -6 inches in translation, see Figure 2. This grating-to-grating alignment is a critical element in the resolving power of the spectrometer. Once aligned on Earth, the array must maintain alignment through launch and a 10 year mission. Stability of the structure that holds the grating array and the internal alignment of the gratings is paramount to the success of this instrument.

This paper will discuss the mounting system which attaches the grating array structure to the telescope, and particularly the passive damping design for this mounting system which is necessary to reduce stresses and dynamic response of the RGA system.

2 MOUNTING SYSTEM FOR THE RGA

For the projected 10 year mission, the mounting system must preserve the precision alignment of the diffraction gratings, which is verified under the Earth's gravity and subject to a severe launch environment. For this reason a simple passive mounting system is highly desirable. In addition, the mounting system must meet several other mission-critical functional requirements:

1. The mounting system must not adversely affect the optical precision of the spectrometer through the attachment to a less precise spacecraft structure. This will be accomplished if the mounting system constrains only the six rigid-body modes of the spectrometer, and thereby isolating it from the unpredictable moment

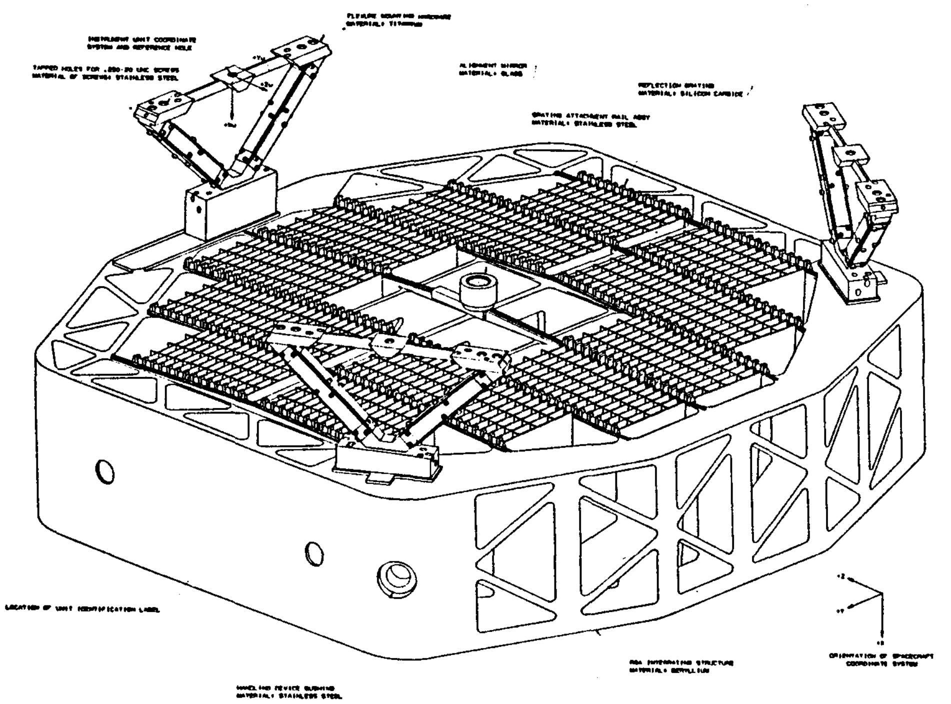

Figure 2: Spectrometer grating array with structure and mounts

loads due to a non ideal interface and differences in thermal expansion rates.

2. The mounting system must act as a dynamic isolator during launch for the higher frequency resonances of the spectrometer, which otherwise would be damaging. This will be accomplished if the six rigid-body modes or so called suspension modes are substantially lower in frequency than those of the spectrometer. In this regard, it is desirable for the mounting system not to affect the natural modes of the spectrometer in such a way as to lower its natural frequencies.
3. The mounting system must limit the amplitudes of the suspension modes and the high-energy flexural modes of the spectrometer. This will be accomplished if sufficient damping is present in the mounting system.
4. The mounting system must occupy a specific annular region around the aperture of the spectrometer.

Some of these requirements would seem to be conflicting; however as we shall see, the solution based on exact-constraint or kinematic design, will require only minor compromises. The six rigid-body degrees of freedom of the spectrometer, rotations and translations about three spatial axes, require exactly six independent single-degree constraints or some equivalent combination of constraints to meet the alignment requirement, and these constraints must be sufficiently rigid so that the change from Earth's gravity to zero gravity produces an insignificant or at least a predictable change in alignment. It would turn out that a mandated requirement for the spacecraft, namely, that structures have a minimum natural frequency of 120 hertz, would require even greater rigidity of the mounting system. If one considers adding a seventh constraint to the spectrometer for good measure, the effect would be the constraint of a flexural degree of freedom. Any load carried by this constraint due to any one of several sources would cause the spectrometer to flex and possibly distort its image.

Conceptually, a single-degree constraint is a rigid link with frictionless, backlash-free ball joints at each end. There is an infinite number of ways to arrange six constraints between the spectrometer and the interface of the spacecraft. A few arrangements are prohibited as they would over-constrain one direction while leaving another direction unconstrained. This would occur, for example, if four constraints were parallel, or two were along the same constraint line. Other arrangements fail to lie within the specified annular region available for mounting. The classical three-vee arrangement is well suited for this application, and it leads to a very interesting design.

In this arrangement, six single-degree constraints are paired to form three vee-shaped two-degree constraints. A constraint pair prevents its intersection point from translating in the plane of the vee, while all other motions are free. The three vees attach to the face of the spectrometer near its perimeter so as to define the sides of a triangular prism as shown in Figure 2 . This arrangement provides exact constraint as can be tested by conceptually removing any single constraint and seeing if motion will ensue in the general direction of that missing constraint.

We turn now to the design of a vee-shaped constraint device that will provide some flexibility in the constrained directions so that the suspension modes occur just over 120 hertz. In addition, the device should have several orders of magnitude greater flexibility in the unconstrained directions. The basic structural component that will easily satisfy these requirements is the blade flexure, which is simply a thin plate that acts as an elastic hinge. A series of four blades properly arranged along a line of constraint, will provide a single-degree constraint. This arrangement duplicated along each leg of the vee forms the basic configuration for the constraint device shown in Figure 3.

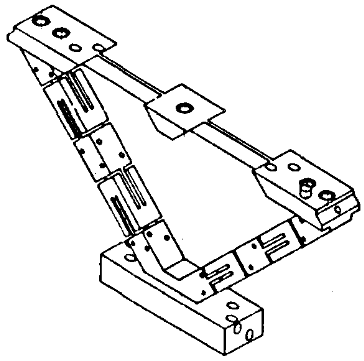

Figure 3: Hale mount for the XMM reflection grating array structure

There are two noteworthy features of this design. First, it is cut from a single plate of titanium using the wire electric discharge machining (EDM) process. This eliminates unnecessary interfaces at the flexures that could slip during launch. Second, the blades overlap somewhat to make the unit more compact, although this is not a significant advantage for this application. A special-purpose design program was developed, based on beam theory, to calculate the stiffness matrix, maximum stress and other useful information from the size of the blades and the material. This allowed a simple iteration process to settle on an acceptable design. Later, the design was incorporated into a dynamic system model using finite element analysis, where close agreement was observed.

A fortunate property of the blade flexure with regard to a damping treatment is that strain energy gets concentrated in the blades. Viscoelastic dampers placed in parallel with the blades will dissipate a portion of the strain energy each cycle, and thereby limit the amplitudes of resonances where the flexures are energetic. Obviously the suspension modes fall into this category, but so do the major flexural modes of the spectrometer that react against the mounts. Because the dampers attach to the surface of the mounts, the dampers act off-axis to the blades, which gives them a moment resistance in addition to their resistance along the constraint axis. This is an advantage for damping but a disadvantage for exact constraint. Fortunately the resistance of the viscoelastic material decreases substantially at low frequencies; therefore, the dampers are unlikely to affect the static precision of the spectrometer.

3 PASSIVE DAMPING OF THE MOUNTING SYSTEM

As mentioned earlier, the mounts provide an exact-constraint, stiff mounting for the RGA which is extremely important to its function. However, these mounts insert dynamics in the form of suspension modes into the system. These dynamics result in large stresses in the flexure blades. Therefore, the obvious candidate location to apply damping is to the mounts. This will greatly reduce the vibration amplitude of the suspension modes

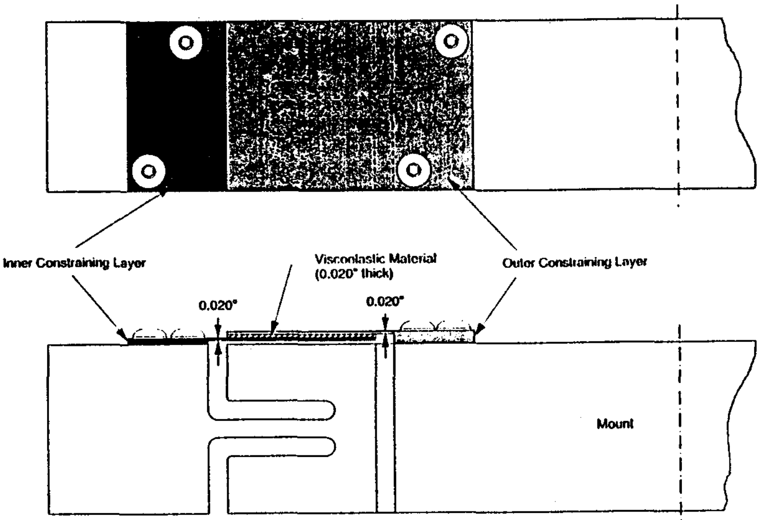

Figure 4: Damping design for the Hale mount

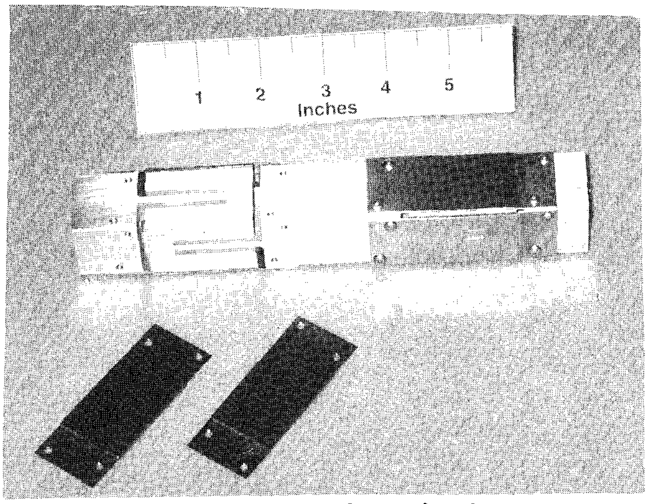

Figure 5: Prototype damped Hale mount

which these mounts introduce, and also will reduce stresses in the overall RGA system.

3.1 Damping design

The basic damping design is shown in Figure 4. This damping treatment spans the flexible portions of the mounts. An inner constraining layer is attached to one side of the flexible region and an outer constraining layer is attached to the other side. These inner and outer constraining layers are attached to each other with a viscoelastic material (VEM). Any relative motion across these flexible regions will move the inner and outer constraining layers such that the VEM is placed in shear strain, which causes the VEM to dissipate energy and thereby damp the motion. Figure 5 shows a photograph of prototype hardware for this damped-mount concept.

Optimal damping design is an iterative procedure. There are many variables to optimize including location of the damping treatment, constraining layer material and geometry, viscoelastic material and thickness, as well as many practical issues such as manufacturability, assembly, etc. The Modal Strain Energy[1,2] (MSE) method was used to analytically determine the damping for the structure. This method is based on the assumption that the modal damping of a structure may be approximated by the sum of the products of the loss factor of each material and the fraction of strain energy in that material for each mode. A well-designed, passively damped structure is one in which high-loss-factor materials, such as viscoelastic materials, are incorporated such that they dissipate energy in the important modes of vibration. Typical VEMs have loss factors on the order of 1.0 while those of metals are comparatively negligible. A good approximation of the modal damping of a treated structure is thus given by

$$\eta^i \approx \frac{\eta_{VEM} SE^i_{VEM}}{SE^i} \tag{1}$$

where η^i = loss factor for mode i , η_{VEM} = loss factor for the viscoelastic material, SE^i_{VEM} = strain energy in the viscoelastic material when the structure deforms in natural vibration mode i, and SE^i = total strain energy in natural vibration mode i.

A finite element (FE) model of the RGA, mounts, and damping was created for use in the damping design. This FE model (Figure 6) was used to calculate frequencies, mode shapes, strain energy distributions, dynamic responses, stresses, and static sag. Many trade studies were run to optimize the damping design. Table 1 shows the modal strain energy distribution for the final damped system model for the first ten modes (six suspension modes + four flexural modes). The viscoelastic damping material which was chosen for this design is Soundcoat DYAD 601 with a thickness of 0.020 inches. This is the best material available which has all the desired properties matching the requirements of the design. These properties include thickness, shear modulus, and loss factor. This material is commercially available. Table 2 lists the frequency and damping of each system mode up to 1000 Hz. The damping values were calculated by multiplying the material loss factor, which is frequency and temperature dependent, by the percentage of strain energy in the VEM. *A value of 0.25% viscous damping is assumed to be inherent in the structure in addition to damping added by the viscoelastic material.*

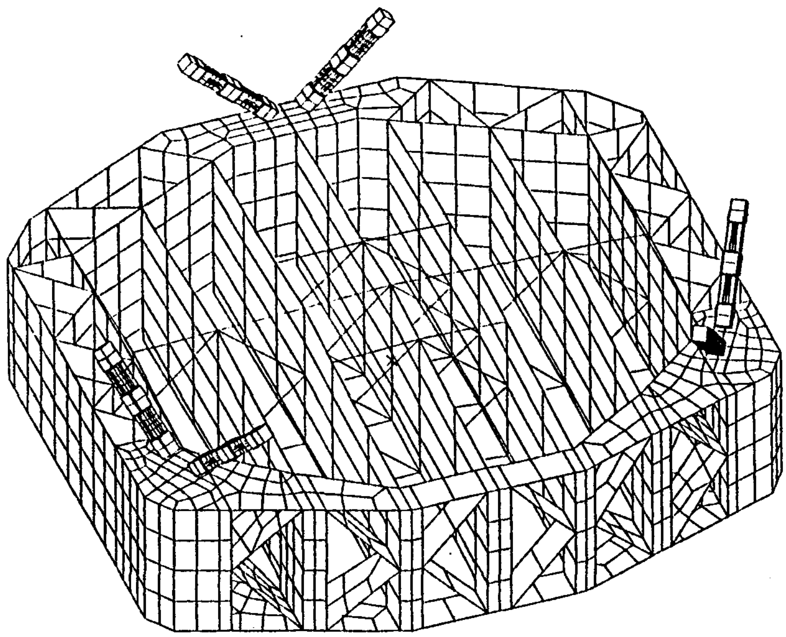

Figure 6: Finite element model of RGA, mounts, and damping

LLNL XMM Integrating Structure and Mounts, Model: is24a										
Damped, Lap-joint dampers										
Modal strain energy values expressed in percent										
Mode number	1	2	3	4	5	6	7	8	9	10
Frequency, Hz	132.7	139.3	200.0	224.7	235.4	243.0	381.0	418.2	426.8	474.6
RING, SOLID	1.0	0.4	0.3	0.8	0.2	0.4	1.9	1.1	0.1	0.1
RING, HORIZ LW	2.0	1.3	1.0	1.3	0.4	1.8	4.7	3.6	0.9	0.4
RING, VERT LW	5.7	0.7	0.7	5.4	0.6	0.3	13.0	8.1	1.3	1.4
RING, ALL	8.7	2.4	2.0	7.5	1.2	2.5	19.6	12.8	2.3	1.8
RIBS, ALL	3.2	0.6	0.5	1.9	5.5	0.4	74.5	86.3	97.7	98.1
LOAD BEAMS	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
INT. STRUCT., ALL	11.9	3.0	2.5	9.3	6.7	2.9	94.1	99.1	100.0	99.9
MOUNTS, ALL	62.0	68.7	70.1	64.4	67.0	69.1	4.2	0.6	0.0	0.1
CL, UPPER	2.6	2.8	2.7	2.6	2.6	2.8	0.2	0.0	0.0	0.0
CL, LOWER	3.4	3.5	3.1	3.2	2.9	3.4	0.2	0.0	0.0	0.0
VEM, ALL	20.2	22.0	21.6	20.5	20.7	21.8	1.3	0.2	0.0	0.0
ALL ELEMENTS	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

Table 1: Modal strain energy distribution for the damped system model

Mode number	Frequency (Hz)	Viscous Damping *	Mode number	Frequency (Hz)	Viscous Damping *
1	132.7	9.7%	16	530.8	0.3%
2	139.3	10.6%	17	535.3	0.4%
3	200.0	10.9%	18	567.0	0.5%
4	224.7	10.5%	19	586.2	0.6%
5	235.4	10.8%	20	608.1	0.3%
6	243.0	11.4%	21	632.3	0.3%
7	381.0	1.0%	22	649.2	0.3%
8	418.2	0.4%	23	678.6	1.3%
9	426.8	0.3%	24	737.4	0.5%
10	474.6	0.3%	25	779.7	0.5%
11	474.8	0.3%	26	790.7	0.3%
12	482.1	0.3%	27	879.2	0.3%
13	503.9	0.3%	28	938.1	0.3%
14	507.5	0.3%	29	999.9	0.3%
15	527.1	0.4%			

* This damping is due to the VEM plus an assumed 0.25% viscous inherant damping

Table 2: Frequencies and damping of the system model

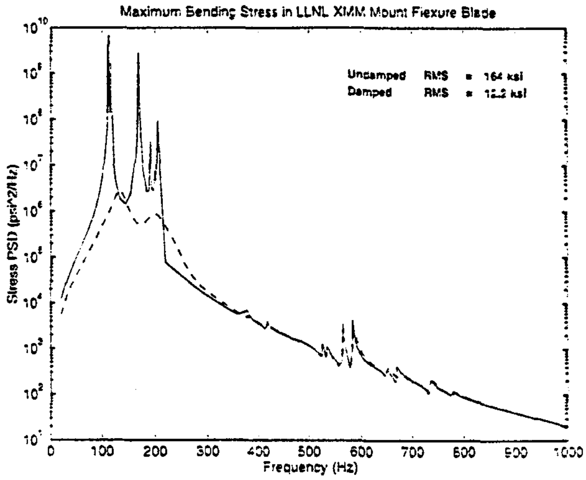

Figure 7: Stress PSD response at worst flexure blade location

3.2 Dynamic Response due to Launch Loads

The system dynamic model was run to calculate random PSD stress responses. Stresses are highest in the flexure blades of the Hale mounts. Stresses are very low in the RGA. Table 3 summarizes the stress requirements and the undamped and damped system model results for key locations. Figures 7 and 8 show the stress PSD response plots for the worst location on a flexure blade and on the RGA, respectively. *Note that the undamped and damped stresses reported in Table 3 represent PSD rms stresses output by the models which have then been multiplied by a crest factor of 3.0. The stress PSD plots of Figures 7 and 8 have not been multiplied by the crest factor.*

Location	Design Limit	Undamped Stress*	Damped Stress*
Flexure blade (worst location)	65 ksi	492 ksi	36.6 ksi
RGA (worst location)	10 ksi	37.5 ksi	5.7 ksi

* This stress is PSD rms stress multiplied by a crest factor of 3.0

Table 3: Stresses in undamped and damped system model

3.3 Testing of the damping design

Several tests have been performed to assure that the damping design will function as predicted. These include viscoelastic material testing, modal testing of a single damped flexure, and temperature testing of the damping treatment. A full-scale mock-up vibration test is planned.

3.3.1 Viscoelastic material characterization testing

Analytical predictions of damping using viscoelastic materials are very dependent on accurate knowledge of the complex shear modulus of the VEM as a function of temperature and frequency. This complex shear modulus

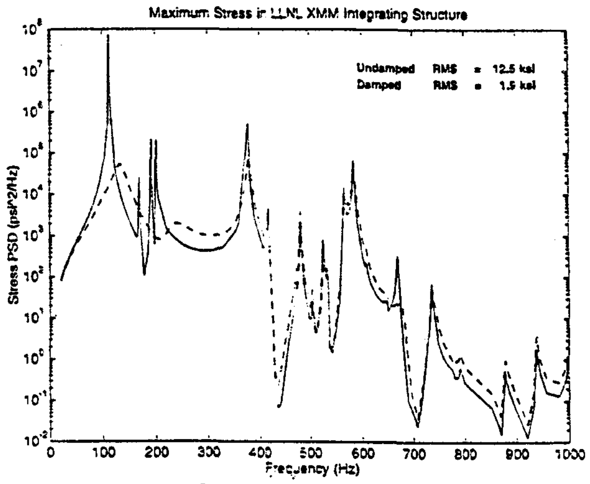

Figure 8: Stress PSD response at worst RGA location

is often referred to in terms of its components: real shear modulus and loss factor. A direct complex stiffness (DCS) test was performed on the DYAD 601 VEM to characterize these properties. In a DCS test, a measured dynamic force is applied in shear to a small VEM specimen which is held at a controlled temperature. The force and resulting shearing deformation are transduced and the frequency response between input force and output displacement is measured by means of a discrete Fourier transform analyzer. This complex function of frequency is then processed to obtain the complex shear modulus of the material. The defining characteristic is that the measurement is direct; force through and displacement across the VEM specimen are transduced and processed without need for any theoretical model relating modal properties to material properties. Uncertainties are reduced to problems of measurement: transducer amplitude and phase accuracy, fixturing dynamics, temperature accuracy, etc.

The temperature range for this test was 0° C to 40° C. The frequency range for this test was 50 Hz to 400 Hz. Figure 9 shows constant temperature curves for the DYAD 601 shear modulus versus frequency for temperatures of 0° C to 40° C. Figure 10 shows constant temperature curves for the DYAD 601 loss factor versus frequency.

Outgassing tests were performed on the Soundcoat DYAD 601 viscoelastic material. The test results showed that the VEM outgassing properties are within acceptable limits.

3.3.2 Prototype Mount Analysis and Test

The ability to accurately model the Hale mount is fundamental to the design of a dynamic system which uses these mounts. A prototype Hale mount was built from aluminum for the purpose of verifying analytical model predictions of stiffness and damping. The simplest test to measure these properties on a single mount is a modal test. Figure 11 shows the FE model of the test configuration. Rigid masses weighing 29 lb were attached to each end of the mount. This assembly was then suspended from cables attached to these masses. The masses are necessary to produce test frequencies in the range expected in the system model. The method of pendulum suspension assures that the mode shapes will be effectively free-free boundary conditions. Mode shape measurement was limited to modes in the horizontal plane. This set of shapes contain all useful deformation patterns of the flexures. Therefore, correlating an analytical FE model to match these test results provides assurance that the mount is well-understood and accurately modeled for use in system analyses. Both an undamped and a damped modal test were conducted and the results are summarized in Tables 4 and 5, respectively. These tables show that both the stiffness and damping can be predicted within engineering accuracy.

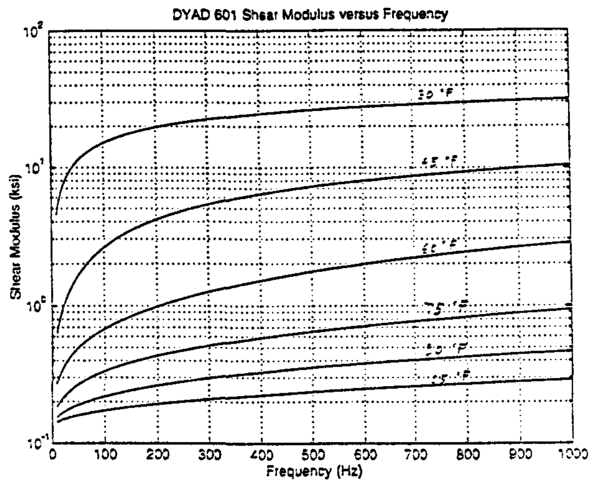

Figure 9: Constant temperature curves for DYAD 601 shear modulus

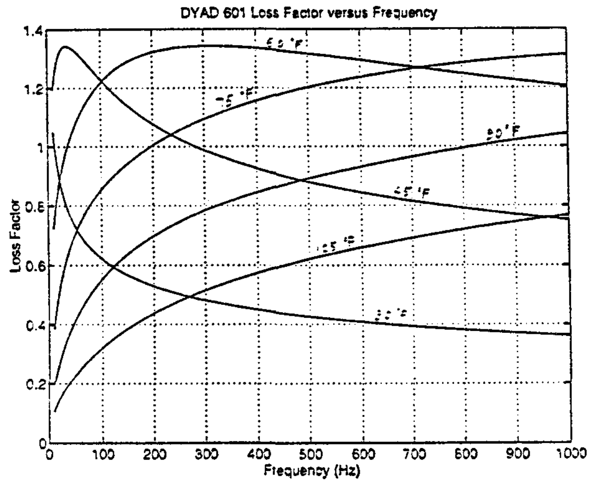

Figure 10: Constant temperature curves for DYAD 601 loss factor

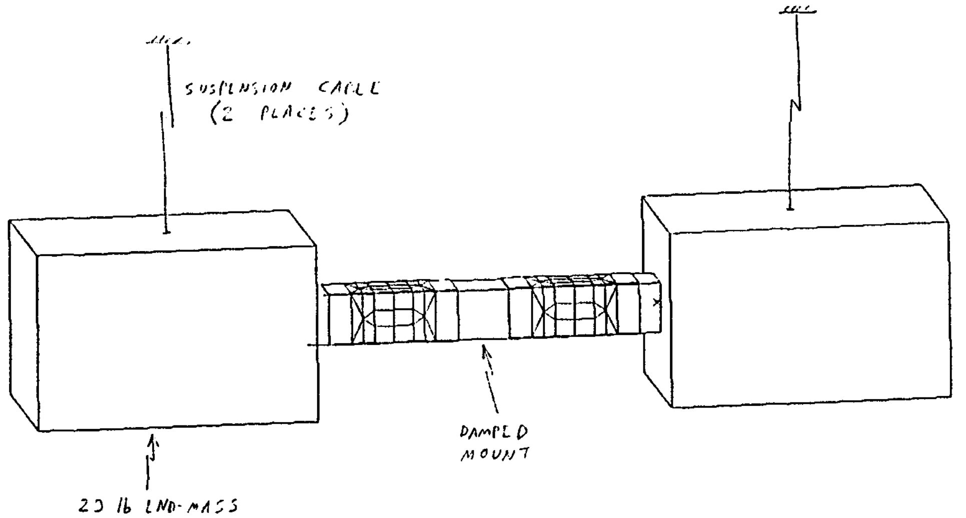

Figure 11: Finite element model of modal test configuration

Modal Test of Undamped XMM Hale Mount

Mode Shape	Frequency (Hz)	
	Test	Analysis
First Bending	7.4	7.05
Torsion	10.9	10.7
Second Bending	26.7	25.2
Axial	291.6	290.8

Notes:
1. Mount was tested with a 29 lb end-mass on each end
2. Test article was suspended at CG of each end-mass with kevlar string
3. All modes were measured in the horizontal plane

Table 4: Undamped analytical and modal test results

Modal Test of Damped XMM Hale Mount

Mode Shape	Frequency (Hz)		Viscous Damping	
	Test	Analysis	Test	Analysis
First Bending	51.8	57.6	28%	24%
Torsion	55.0	63.0	13%	16%
Second Bending	205.7	221.4	22%	25%
Axial	417.5	424.2	16%	19%

Notes:
1. Mount was tested with a 29 lb end-mass on each end
2. Test article was suspended at CG of each end-mass with kevlar string
3. All modes were measured in the horizontal plane

Table 5: Damped analytical and modal test results

3.3.3 Temperature testing

It was desired to know whether or not the VEM would self-heat during operation and lose its effective damping. That is, viscoelastic materials provide damping to structures by dissipating vibrational energy into heat energy, and they incur a temperature rise in the process. This temperature rise causes a reduction in both the stiffness and loss factor in the VEM, which for this damping treatment, reduces the amount of passive damping to the structure. A test was designed which actually measured the temperature rise and also the stiffness change in the VEM. The details of this test will not be discussed here, but the results are briefly summarized. The test article consisted of a dual shear lap-joint specimen of the same constraining layer and damping material design. This test article was subjected to random vibration loads similar to those seen in the system dynamic model, and temperatures of the VEM and constraining layers were measured. Figure 12 shows the temperature versus time for the viscoelastic material and constraining layer. This shows that for representative input disturbances, the VEM temperature increased by only a modest 3.8° F.

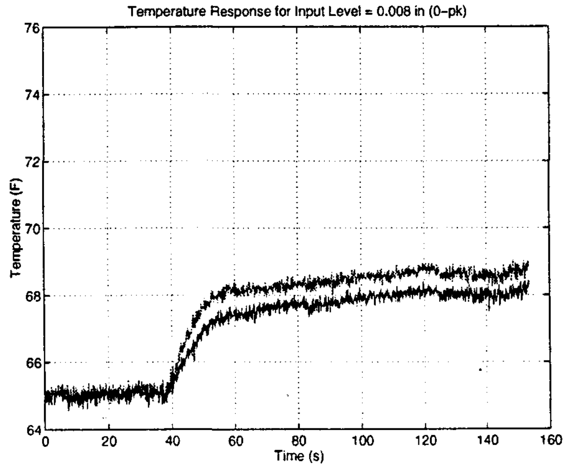

Figure 12: Temperature rise of VEM and constraining layer under representative dynamic loads

3.3.4 Full-scale mockup vibration testing

A full-scale mockup vibration test is planned for the XMM RGA on damped mounts. This test will use a mass simulator for the RGA, but will use actual CDR-level designs for the mounts and damping treatment. This test will verify the damping performance, the damping self-heating, the suspension mode frequencies, and the hysteresis of the system.

4 CONCLUSIONS

The nature of the XMM mission requires very precise and stable alignment of the reflection grating array. This places a premium on the mounting system design. The analyses and test which have been completed show that the mounting system should perform well as a kinematic or exact-constraint support for the RGA. System frequencies are above the minimum requirement of 120 Hz using these mounts. The passive damping treatment provides sufficient damping to reduce stresses in the RGA and mounts to levels below the requirements. In the mount flexure blades, the damping reduced the stress by more than an order of magnitude. The prototype tests show that the stiffness and damping of the mounts is predictable within engineering accuracy.

The combination of passive damping and exact-constraint mount design results in a mounting system which provides the XMM reflection grating array with alignment stability and high damping, as well as isolation from thermal expansion, spacecraft interface deformations, and high-frequency dynamic excitations.

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