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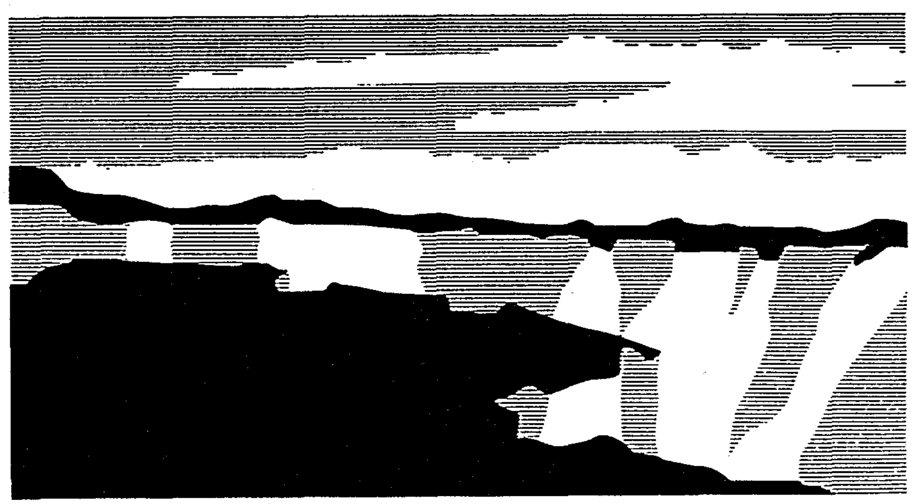

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Viscoelastic struts for vibration mitigation of FORTÉ

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ABSTRACT

FORTÉ is a small satellite being developed by Los Alamos National Laboratory (LANL) and Sandia National Laboratories Albuquerque (SNLA). It will be placed into orbit via a Pegasus launch in 1996. Testing a full-scale engineering model of the structure using the proto-qualification, system-level vibration spectrum indicated that acceleration levels caused by structural resonances exceed component levels to which certain sensitive components had previously been qualified. Viscoelastic struts were designed to reduce response levels associated with these resonances by increasing the level of damping in key structural modes of the spacecraft. Four identical shear-lap struts were fabricated and installed between the two primary equipment decks. The struts were designed using a system finite element model (FEM) of the spacecraft, a component FEM of the strut, and measured viscoelastic properties. Direct complex stiffness testing was performed to characterize the frequency-dependent behavior of the struts, and these measured properties (shear modulus and loss factor) were used to represent the struts in the spacecraft model. System-level tests were repeated with the struts installed and the response power spectral densities at critical component locations were reduced by as much as 10 dB in the frequency range of interest.

Key words: viscoelastic material, graphite epoxy, finite element analysis, passive damping, modal strain energy, direct complex stiffness testing

1 INTRODUCTION

This paper describes the development of viscoelastic struts that were developed to reduce vibrations of the FORTÉ spacecraft when the structure is subjected to the dynamic loading associated with launch and proto-qualification testing. FORTÉ is a small satellite that will be placed in orbit in 1996. The structure weighs approximately 425 lb, and is roughly 80 inches high and 40 inches in diameter. It was developed and built by LANL in conjunction with SNLA for the United States Department of Energy. The FORTÉ primary structure, shown in Figure 1, was fabricated primarily with graphite epoxy, using aluminum honeycomb core material for equipment decks and solar panel substrates. Equipment decks were bonded and bolted through aluminum mounting blocks to adjoining structure. In the photograph, the structure is shown in its modal test configuration. It is mounted to the baseplate by a series of flexures similar to those which will mount the satellite to the launch vehicle separation ring.

The FORTÉ schedule from payload conception to launch was very short, and satellite and payload specifications were written before the design was complete. Random vibration testing of the Engineering Model (EM) of the structure showed that acceleration PSDs for critical components on both decks would exceed proto-qualification levels, and it became evident that some form of vibration suppression was needed. Figure 2 shows the vertical random vibration proto-qualification level for mid deck payload components with the measured PSD

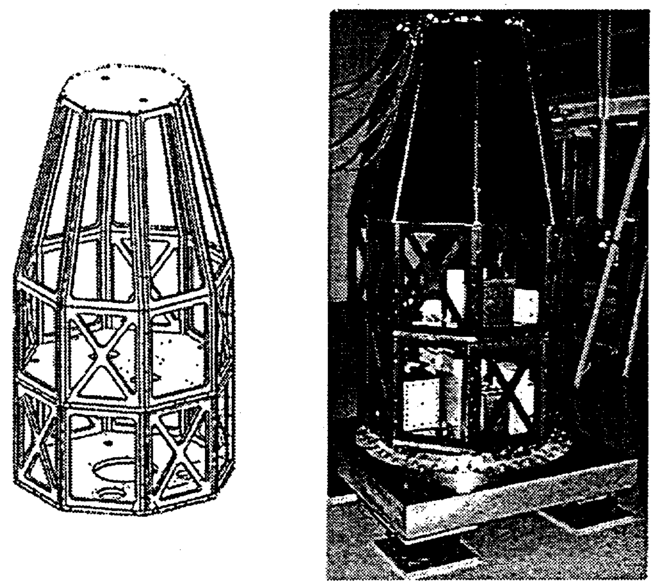

Figure 1: FORTÉ spacecraft structure

for a key component. The structure's design and the FORTÉ equipment layout were virtually complete by this time, so retrofit passive damping treatments were considered in conjunction with force limited random vibration testing. A structural modification was implemented, consisting of high-loss, moderate-stiffness struts installed between the bottom and mid decks of the structure. The struts are shown in Figure 3. Addition of these struts coupled the dynamics of the decks, and shearing of the struts' viscoelastic material (VEM) resulted in dissipation of vibrational energy in an important frequency band and reduction of vibration response at key spacecraft components. The viscoelastic struts were used in conjunction with force limited vibration testing, customized bracketry modified to provide isolation, and manipulation of the system mass distribution, for successful vibration mitigation of FORTÉ.

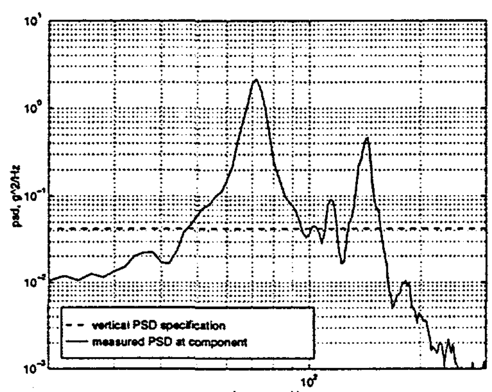

Figure 2: FORTÉ random vibration proto-qualification level and measured PSD

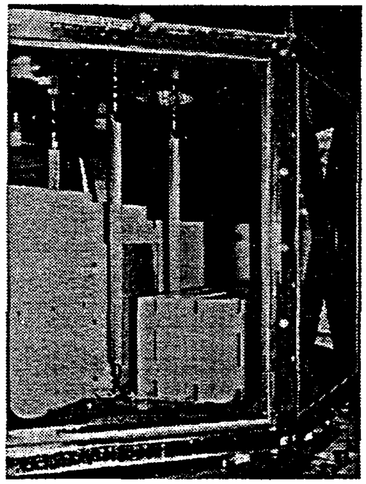

Figure 3: Struts installed between FORTÉ decks

2 DAMPING ANALYSIS WITH STRUCTURAL MODEL

A *Nastran* finite element (FE) model of the FORTÉ structure was used for prediction of system structural response with and without retrofit damping treatments. Boundary conditions consisted of a series of translational springs around the base of the cylindrical structure, representing the flexures on which the structure was mounted on the test stand. These flexures are similar to the mounting of the satellite on the launch vehicle. The model accurately represented global bending and torsion modes, bottom deck and mid deck bending modes, and global plunge modes. Figure 4 shows the predicted shapes for some important modes. Deck modes below 120 Hz and plunge modes around 170 Hz were specifically targeted for vibration suppression.

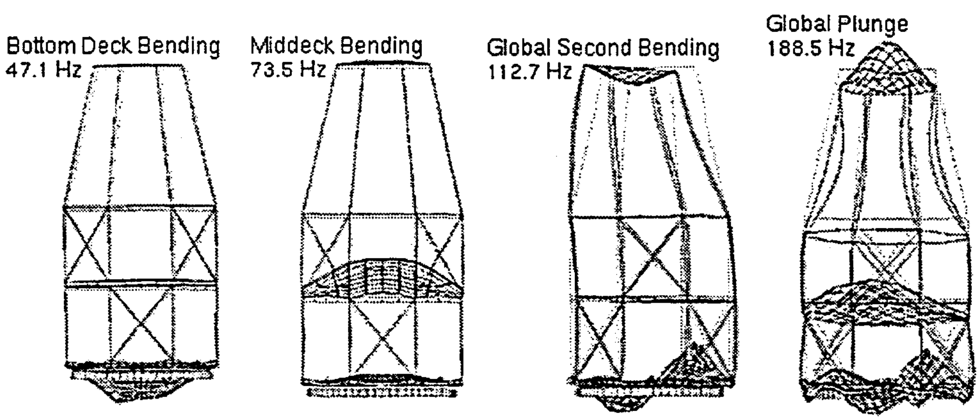

Figure 4: *Nastran* model predicted mode shapes

Several approaches were considered for implementation of passive damping in the FORTÉ structure, including constrained-layer damping, viscoelastic struts, and viscous struts. Viscoelastic strut configurations and variations of a proposed viscous strut were investigated. Performance predictions were made using the system FE model.

The system FE model was used to evaluate modal strain energy distribution in the modes that contributed significantly to the response of the payload components. The rationale for using high-loss struts mounted between the bottom deck and mid deck of the satellite is that the strut viscoelastic material is strained in shear due to local deck bending modes as well as global bending, torsion and plunge modes. Equipment on the bottom deck is very closely spaced, restricting placement of the struts, but mounting the struts symmetrically around the antenna can was found to be effective. Strut mounting locations on the bottom deck are shown in Figure 2. System analysis was performed using the *Nastran* model to predict performance improvements that might be expected with various viscoelastic strut configurations, and variations on a viscous strut that was proposed.

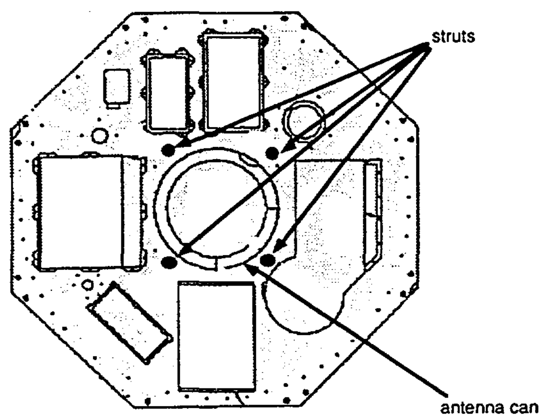

Figure 5: Bottom deck equipment configuration and strut locations

A schematic of the viscoelastic strut configuration is shown in Figure 6. Early models of this strut concept used estimates of stiffness and loss from hand calculations based on experience with VEMs. A refined component-level FE model of the strut, shown in Figure 7, included springs to account for compliance in the strut end fittings, and was tuned to results from component-level direct complex stiffness testing described in Section 4 of this paper. The tuned component FE model of the strut was then modified to investigate different strut configurations, with variations on the VEM thickness and VEM shear area.

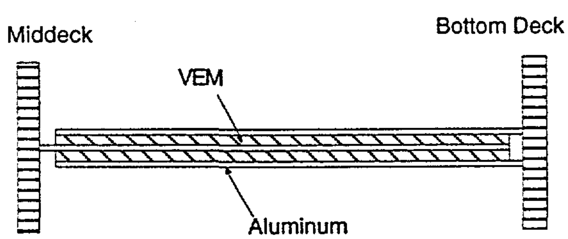

Figure 6: Schematic of viscoelastic strut configuration

For each strut configuration of interest, the component FE model was exercised to compute strut stiffness and strut loss versus frequency. These functions were then used to model the struts in the system FE model, as beam elements having the appropriate frequency-dependent stiffness and loss characteristics. *Nastran* provides for the frequency dependent material behavior in random response analysis, so the FORTÉ system model was subjected to the proto-qualification random vibration inputs, with the struts installed in the model, and response predictions were computed for comparison with baseline response and other strut designs.

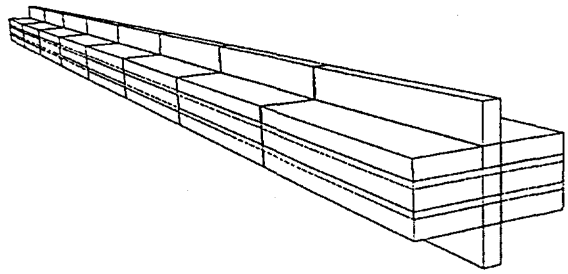

Figure 7: Finite element model of viscoelastic strut

Viscous struts were modeled as beam elements in parallel with viscous dashpots. Stiffness of 54 lb/in and damping of 4.4 lb-sec/in were included as specified by the strut vendor. Struts with other dashpot constants ranging from 0.8 to 110 lb-sec/in were also analyzed.

Table 1 gives RMS response predictions, in g's, for various strut designs, subjected to the vertical proto-qualification-level random vibration input. Both the viscous and viscoelastic struts were found to produce favorable results, so selection of the appropriate treatment was based on fabrication and in-service concerns. Ultimately, the viscoelastic struts were preferred because of the simplicity of fabrication and LANL's reluctance to include viscous fluids in a spacecraft application.

		viscoelastic struts loss = .3			viscous struts stiffness = 54 lb/in			
		strut stiffness, lb/in			strut damping coefficient, lb-s/in			
	baseline	50000	30000	10000	0.88	4.4	22.0	110.0
input	1.55	1.55	1.55	1.55	1.55	1.55	1.55	1.55
center of mid deck	7.69	4.01	3.74	4.60	6.29	4.30	3.15	3.79
scan wheel, inner	6.76	3.30	3.08	3.87	5.71	3.81	2.74	3.40
scan wheel, outer	5.33	2.53	2.32	2.92	4.53	2.97	2.33	3.38
corner of mid deck	4.23	3.70	3.59	4.04	3.81	3.69	3.42	3.26
corner of antenna can	2.28	3.00	2.55	2.87	2.16	1.94	2.26	3.51
corner lower deck	2.17	1.97	1.92	2.10	2.34	2.26	2.11	2.06

Table 1: Predicted responses, RMS g's to 300 Hz, to vertical random vibration

3 STRUT DEVELOPMENT AND MATERIAL TESTING

As described above, trade studies were performed with the FE model to determine the appropriate stiffness and loss characteristics of the struts. Analytical predictions of strut stiffness and loss were correlated with direct complex stiffness test results from an initial design of the strut that was fabricated and tested. The figures of merit for optimization of the design were (1) RMS response at the sensitive equipment locations and (2) reduction of peak response levels to the specification level to which they were tested, i.e., 0.042 g^2/Hz.

Four damped struts plus one spare were built. The viscoelastic material, 3M 9473 pressure-sensitive adhesive, was configured with a thickness of 0.040 inches, and a shear area of about 18 square inches. Attachment to the bottom and mid decks of the structure was achieved with 10-32 threaded inserts in the decks. The end fittings of each strut were configured to provide a turn-buckle mechanism for installation of the struts. It should be noted that any compliance in the struts at the deck-attachment locations works against the function of the struts.

An important challenge in this design was to minimize the compliance of the strut end fittings. Figure 8 shows a closeup of the strut attachment at the FORTÉ mid deck. Material testing was performed to quantify the mechanical properties and outgassing characteristics.

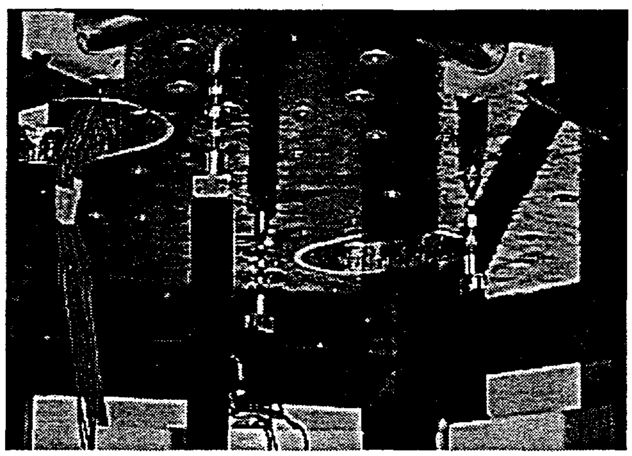

Figure 8: Strut attachment at FORTÉ mid deck

Material testing – mechanical properties. The FORTÉ temperature environment is benign, since both launch and testing environments are thermally controlled, so temperature variation is not an issue. But knowledge of the viscoelastic properties as functions of frequency was critical to development of the struts. The material was tested, and the temperature-frequency nomogram that describes the relevant mechanical properties, shear modulus and loss factor, is shown in Figure 9. This nomogram conveys information about the material's shear modulus and loss factor as functions of temperature (-25°F to 185°F) and frequency (to 600 Hz). Isotherms of shear modulus and loss factor for the FORTÉ strut material at 70°F, are plotted versus frequency in Figure 10.

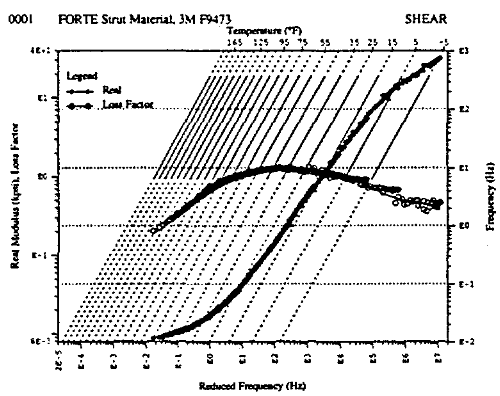

Figure 9: Temperature-frequency nomogram for FORTÉ viscoelastic material

Material testing – outgassing. Outgassing was not considered a major issue for the FORTÉ spacecraft, but for completeness, and to insure that this did not become an issue at some point in the future, the selected material was characterized for outgassing, in accordance with ASTM E-595 and NASA SP-R-0022A specifications. The total mass loss (TML) was measured at 0.68%, and the collected volatile condensible material (CVCM) was 0.03%. The NASA standards for acceptable spacecraft materials are TML of 1.00%, and CVCM is 0.10%.

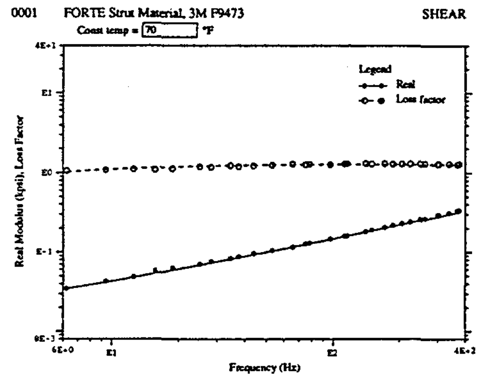

Figure 10: Isotherms of FORTÉ viscoelastic material properties at 70°F

4 DIRECT COMPLEX STIFFNESS TESTING OF STRUTS

Stiffness and loss of the FORTÉ struts were measured by means of a test procedure referred to as direct complex stiffness testing. The test rig consisted of virtually rigid "book ends" (mounting brackets to support the strut under test) bolted to a large work plate, and a hydraulic shaker positioned to excite the strut axially. A strain-gage-type load cell was mounted in series with the strut. A schematic of the test configuration is shown in Figure 11. The strut was excited axially with a controlled random force, and axial displacement was measured

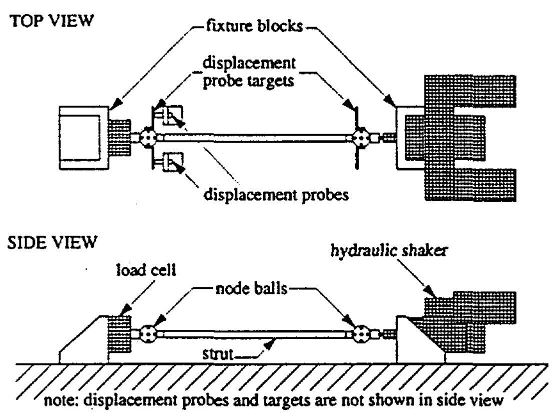

Figure 11: Schematic of strut test configuration

with eddy current probes referencing the angle-bracket "flags" attached to each node ball. Complex stiffness was calculated by dividing the input force by the measured displacement. Measured stiffness functions are shown in Figures 12 for input force levels of 100 lbf, 200 lbf, and 300 lbf. Measured strut loss versus frequency is shown in Figure 13.

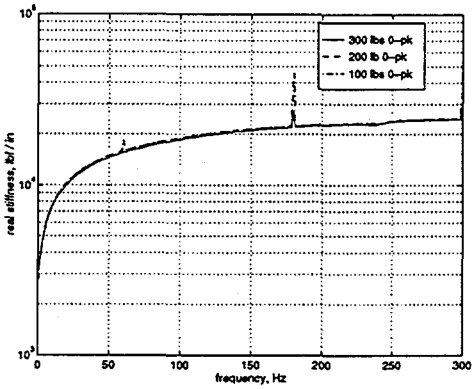

Figure 12: Measured strut stiffness versus frequency

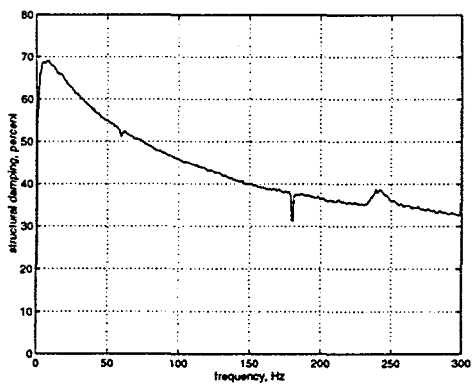

Figure 13: Measured strut loss versus frequency

5 RANDOM VIBRATION TESTS

Random vibration testing was performed as part of the proto-qualification testing for FORTÉ. The figures of merit for vibration mitigation design were based on measured responses when the structure is subjected to random vibration testing: (1) minimization of RMS responses at sensitive equipment, and (2) reduction of peak responses to 0.042 g^2/Hz, the level to which equipment was tested. It was especially important to reduce peak levels below 0.042 g^2/Hz in frequency bands where component resonances were present.

Figures 15 through 17 present comparison plots of PSD response at key locations on the structure with and without the viscoelastic struts. Figure 14 shows responses at the scan wheel, a very important payload component. The reduction of 3 orders of magnitude around 70 Hz shown in this plot was especially significant, because this component is known to have an internal resonance around 65 Hz. Response levels above 100 Hz for the scan wheel were reduced with force limited testing. Figure 15 shows responses at the corner of the antenna can on the bottom deck. Figure 16 shows responses at the corner of the mid deck and Figure 17 gives responses at the center of the mid deck.

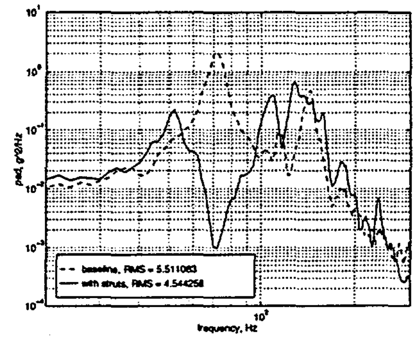

Figure 14: Random vibration response at scan wheel

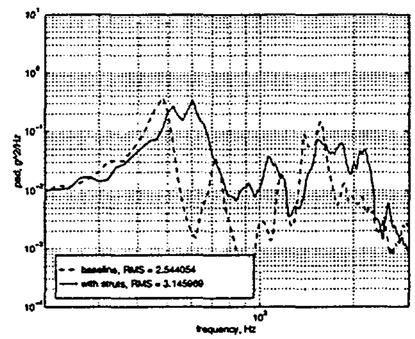

Figure 15: Random vibration response at corner of antenna can

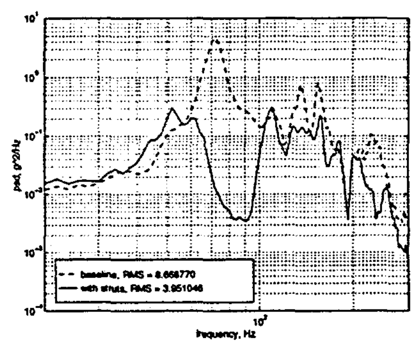

Figure 16: Random vibration response at corner of mid deck

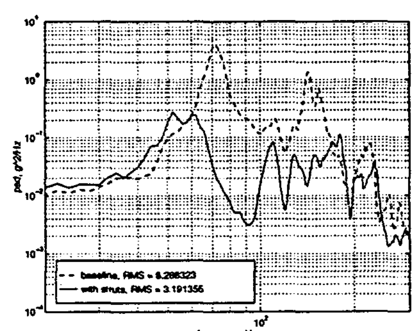

Figure 17: Random vibration response at center of mid deck

6 SUMMARY

Viscoelastic struts were designed, built, and tested as part of a program to reduce vibrations of the FORTÉ spacecraft. The technical objective of the work was reduction of response at the location of payload components when the structure is subjected to the dynamic loading associated with launch and proto-qualification testing. A *Nastran* finite element model of the FORTÉ structure was used for system analysis and damping design. Analytical trade studies were performed and strut FE model was built to determine the best design configuration for a viscoelastic strut. Material testing and direct complex stiffness testing of the struts were performed. Addition of the viscoelastic struts coupled the dynamics of the decks, and shearing of the viscoelastic material resulted in dissipation of vibrational energy in an important frequency band and reduction of vibration response at key spacecraft components. The viscoelastic struts were used in conjunction with force limited vibration testing, customized bracketry modified to provide isolation, and manipulation of the system mass distribution, for successful vibration mitigation of FORTÉ.

Acknowledgment

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