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A COMPARISION OF MODES AND DAMPING COLLECTED USING TRADITIONAL MODAL TECHNIQUES VS A BASE INPUT SHAKER FOR A SPACE-BOUND SYSTEM

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ABSTRACT

Environmental testing is frequently the last stop in a Flight Systems' qualification. There is often a need to compress schedule to meet launch dates. Innovative techniques are often needed to reduce schedule and budget. This paper will present modal data collected using traditional modal techniques, i.e., a small modal shaker with localized input versus data collected during a random vibration test on a T-4000 shaker for a small space-bound system. The technique for extracting modal parameters from motion to motion frequency response functions will be discussed. Damping extraction is discussed and shown at each test's highest force input level.

KEYWORDS: Satellite, modal survey, system test, Integration and Test, vibration testing, damping, space-bound system.

INTRODUCTION

Environmental Testing is normally the last stop in the qualification of a satellite system. MIL-STD-1540E recommends space systems undergo a Modal Survey as well as Vibration testing as part of the qualification process. History has shown, for first time designs, the integration portion often extends longer than planned due to anomalies. For this reason, the testing schedule is often compressed. There is a growing need to compress the testing schedule while still collecting all data for verification selloff. This paper will summarize data collected using standard modal techniques versus data collected on a vibration shaker. Modal frequencies are extracted from both tests and compared. Mode shapes are illustrated from the traditional modal test. Damping extraction is discussed and values shown at multiple force levels for each test. A potential end goal would be to collect all required modal data for model validation during the Random Vibration test. This could potentially reduce a test schedule by weeks by eliminating a test with its associated setup and preparation.

REQUIREMENTS DEFINITION

MIL-STD-1540E, Test Requirements for Launch, Upper-Stage, and Space Vehicles establishes the environmental and structural ground testing requirements for launch vehicles, upper-stage vehicles, space vehicles, and their subsystems or components. There are multiple levels of testing. Increasing in amplitude and duration, they begin with Acceptance level, followed by Protoqualification, followed by Qualification levels. Dynamic tests normally increase in increments of 3 dB for each level. This example case underwent Protoqualification level tests, which are defined as, "tests conducted to demonstrate satisfaction of design requirements using reduced amplitude and duration margins[1]." This level is normally required for flight units that have limited production quantities.

A Modal Survey is conducted to obtain data to validate dynamic models for loads analyses. The test is normally conducted at low levels. Modal frequencies, shapes, and damping are extracted and then scaled to flight level using the analytical model. A vibration test is required to demonstrate the unit will survive the launch environment. The test unit normally undergoes a random vibration test in three orthogonal axes with one being parallel to the launch axis.

MODAL RESULTS

The space-bound system was instrumented with 47 triaxial accelerometers for both the random vibration test and the modal survey. Accelerometer locations were optimized to ensure the cross orthogonality matrix would yield the highest diagonal values while minimizing the off diagonals. The structure was bolted to an 11 ton steel mass on airbags for model correlation. Modal data was collected using 47 triaxes or 141 channels of response. Force was input using MB Dynamics 50A modal shakers attached through a modal stinger at rigid locations on the system. Low level responses were collected with 0.5 lbs RMS force input. Single to three shaker inputs were used in sequence to ensure all modes were collected. The frequency response functions (FRFs) from the three shaker test were utilized for the modal analyses. Modal parameter extraction was demonstrated by reconstruction of the complex mode indicator function (CMIF) from the modal parameters and overlaying with the CMIF from the measured data (see Figure 1). The CMIF is calculated by performing a singular value decomposition of the FRF matrix at each frequency line. For three shakers, there will always be three singular values. The strength of the singular values shows the strength of the orthogonal shape vectors in the data. If there were two modes at a single frequency, then one would see the second MIF curve (green) peak at the same frequency of a peak of the first MIF curve (blue). The amplitude of the peak shows how well the mode is excited in the FRF data. In this case, no strong secondary MIF values are seen at peaks in the blue curve indicating we have no significant problems separating closely spaced modal frequencies. The extracted modal frequencies, damping and shapes are used to synthesize the FRFs and create a synthesized FRF matrix. The CMIF calculation is performed on the synthesized FRFs and plotted with the dashed lines. As can be seen, the synthesized primary CMIF (blue dashed) is a very good reproduction of the CMIF from the experimental data, indicating the shape, frequency and damping are accurately extracted.

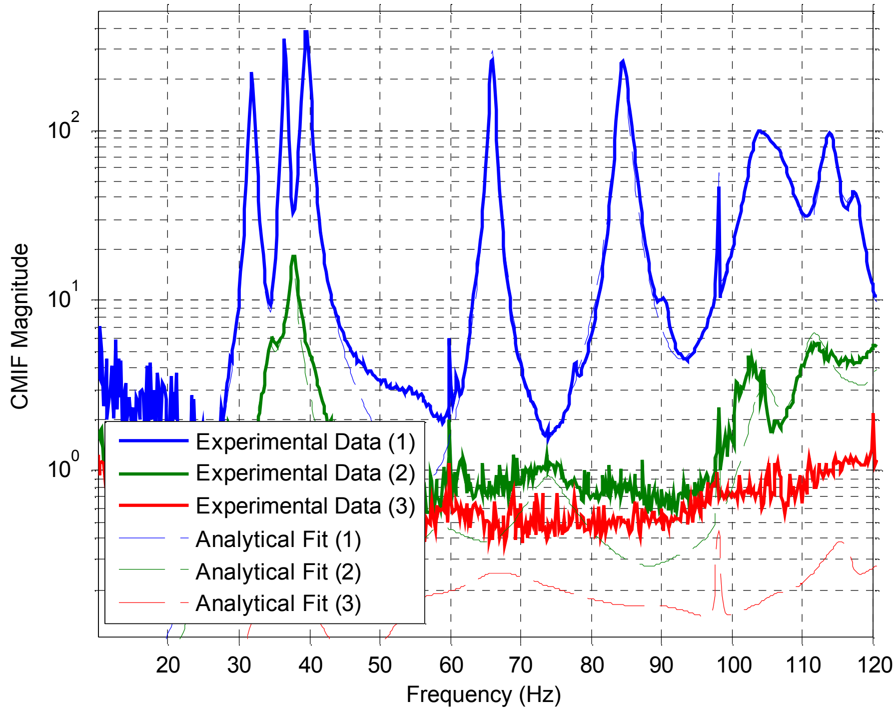

Figure 1: Complex Mode Indicator Function

MODE SHAPES

Five modes are described in Table 1. Figure 2 through Figure 4 show the undeformed system. The system, in a simplified sense, is composed of a head, shown in green, and a body, shown in blue, which are attached by a tripod strut type interface to the base, illustrated in orange. The head normally sits at a slight angle relative to the body. Figure 5 through Figure 9 illustrate graphically the five mode shapes extracted. The undeformed shape is plotted in solid lines and the deformed shape is plotted in dotted lines. The deformed shape is greatly exaggerated to assist in visualization.

Mode	Description	Test Frequency Hz	Test Damping % Critical at 0.5 lbs RMS	Test Damping % Critical at 12 lbs RMS
1	"No" mode	31.8	1.4	Not extracted*
2	"Wag" mode	36.5	0.9	1.4
3	"Yes" mode	39.5	1.4	1.8
4	Lateral	65.9	0.7	2.0
5	Lunge, Head Yaw	84.5	1.0	1.5

Table 1: Modal Test Results

*Only modes 2 through 5 were included in the high level modal test because those four modes were easily excited with a single shaker input.

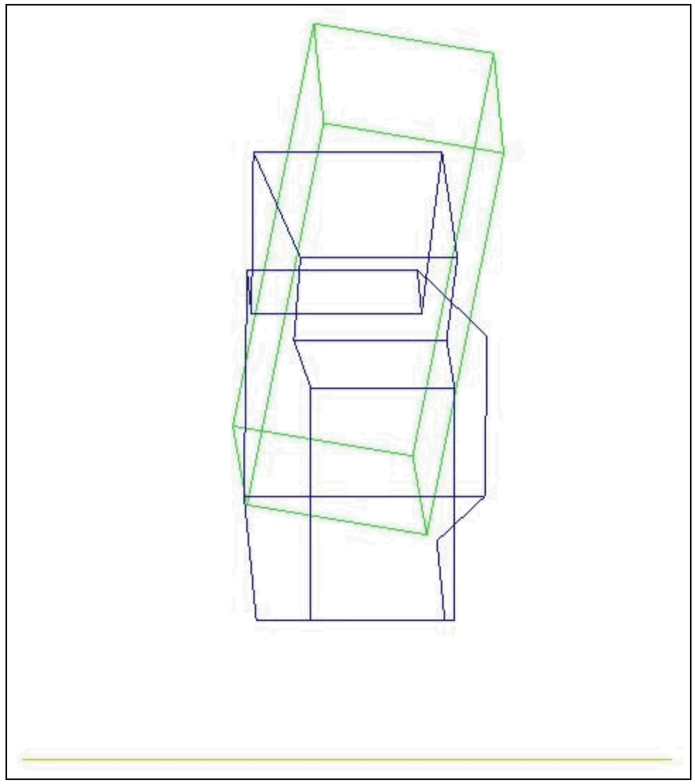

Figure 2: Front View of System

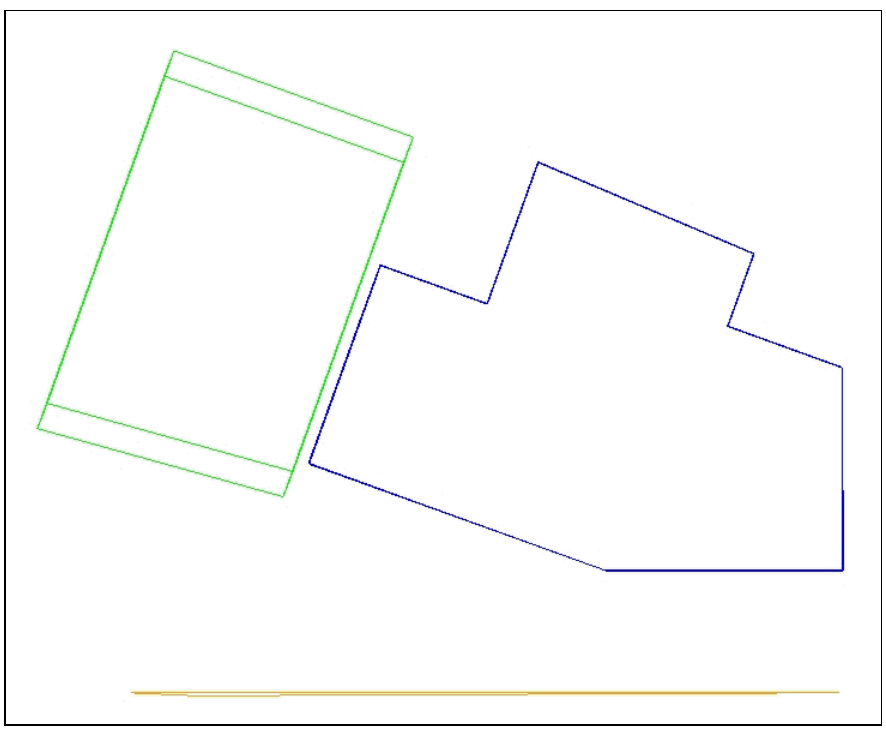

Figure 3: Side View of System

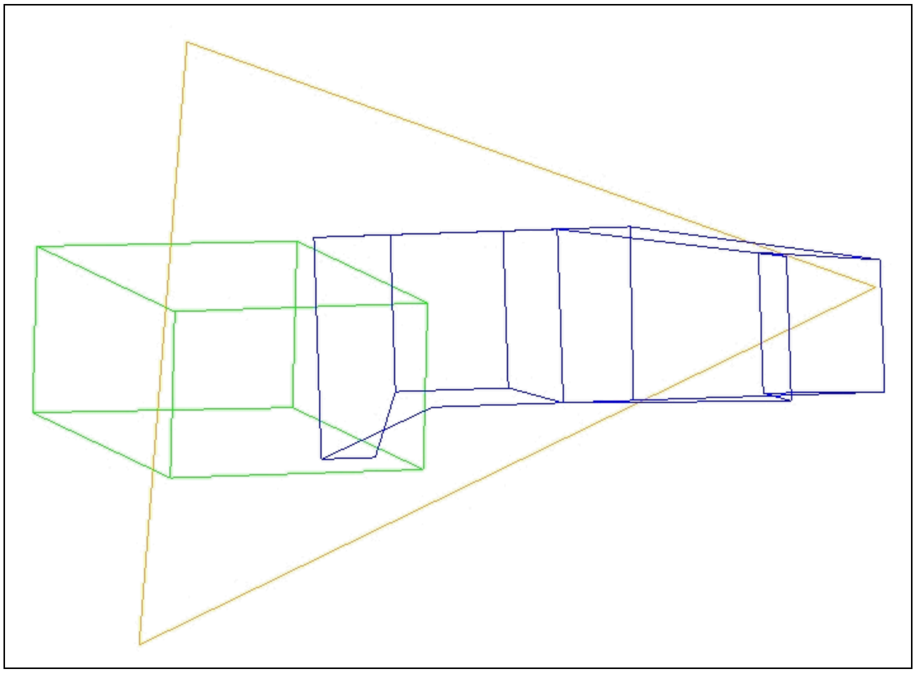

Figure 4: Top View of System

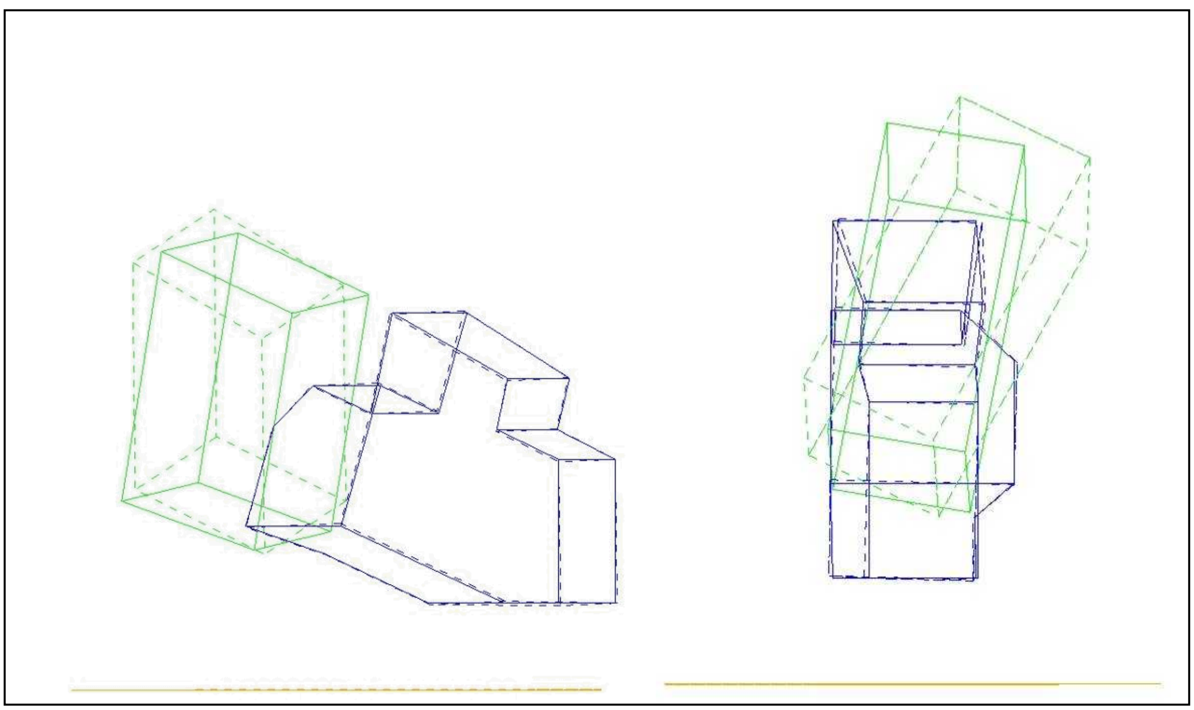

Figure 5: No Mode 31.8 Hz

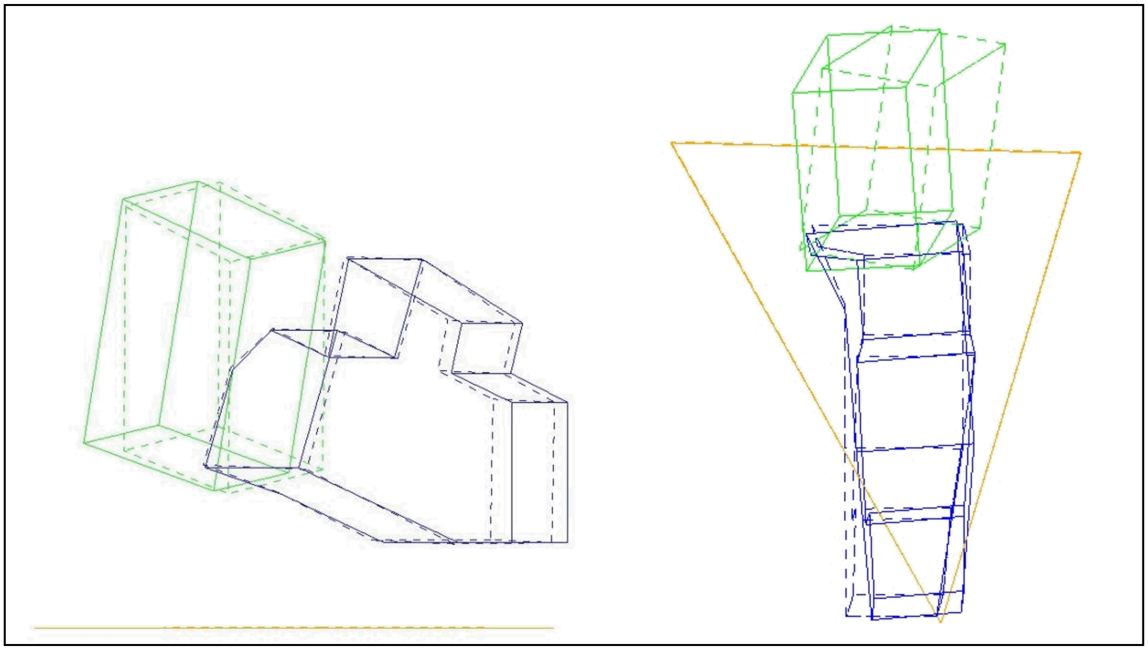

Figure 6: Wag Mode 36.5 Hz

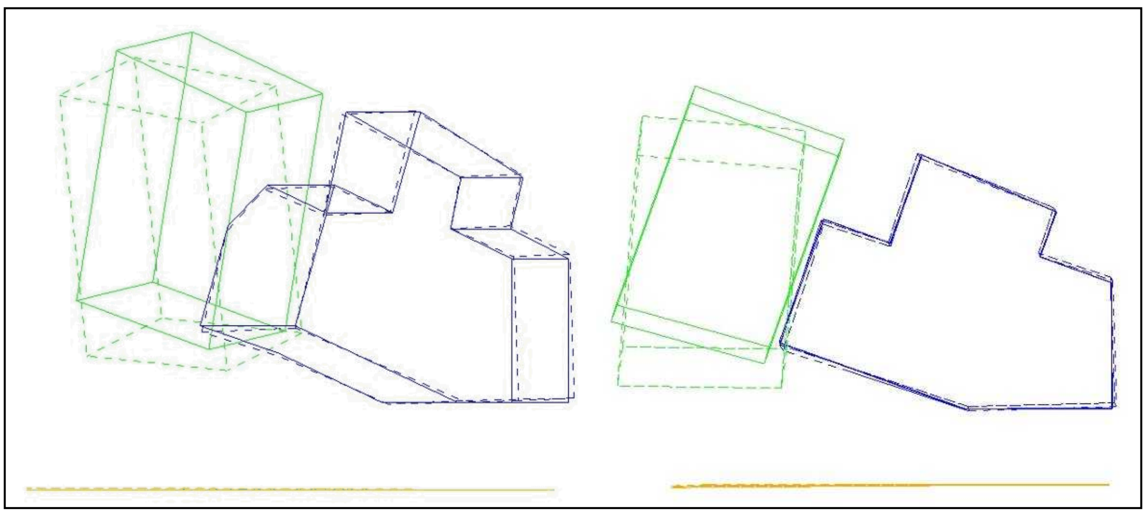

Figure 7: Yes Mode 39.5 Hz

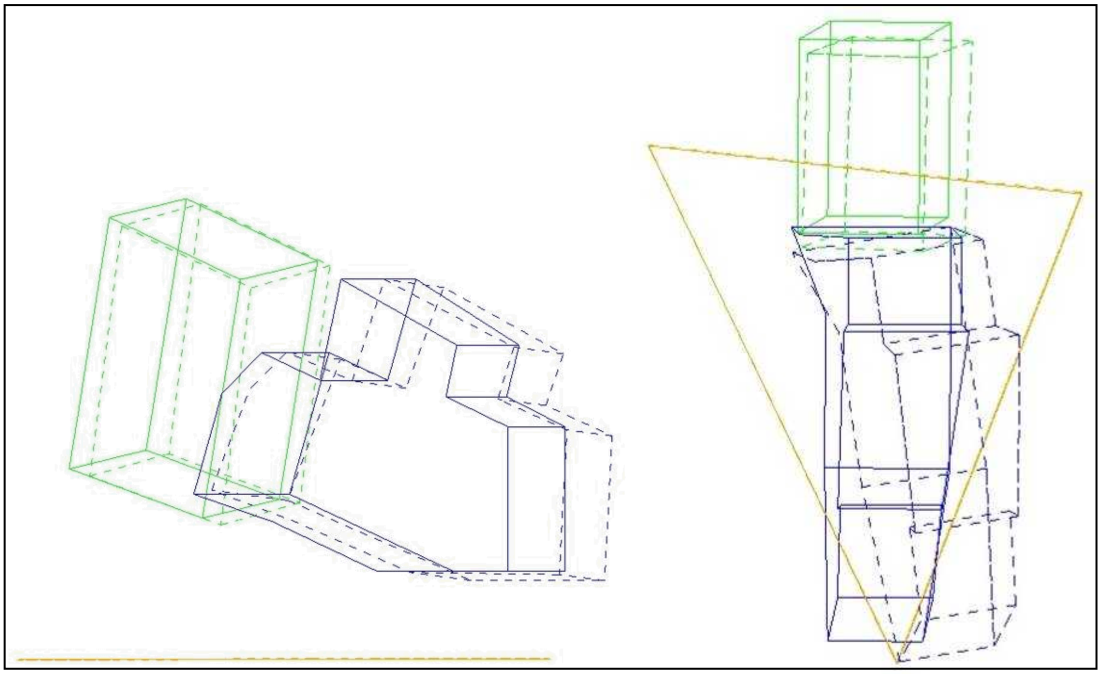

Figure 8: Lateral Mode 65.9 Hz

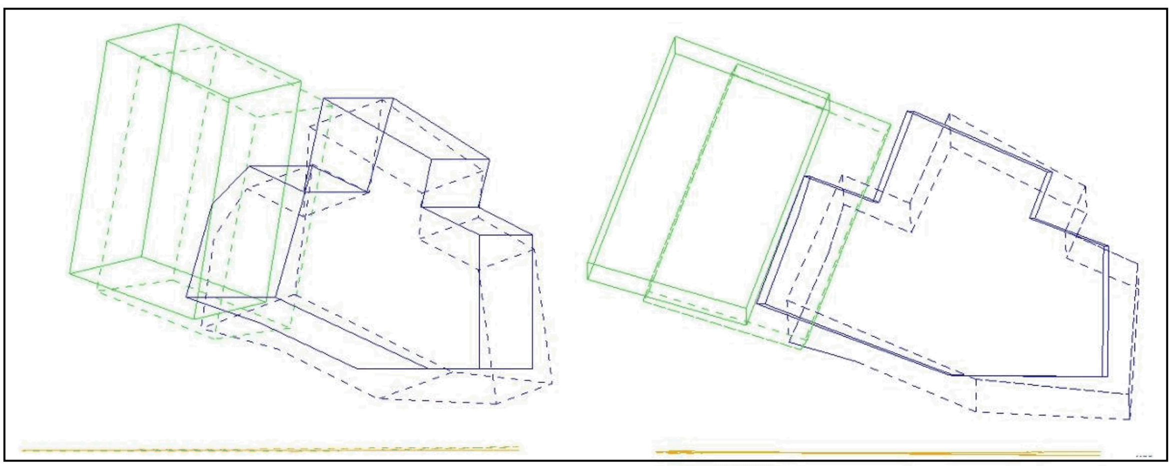

Figure 9: Lunge Mode 84.5 Hz

MODAL DAMPING

The damping values in the various modes of vibration are important quantities, that are required to be measured because they cannot be predicted using the analytical model. Further, the model uses these damping values so that it can predict the magnitude of the loads during the launch environment. Historically, it has been observed that the damping in a mode is not constant but increases with the amplitude of the response. Consequently, it is desired to measure the modal damping at the highest levels that the system would be exposed. Extraction of the modal damping from FRFs is an estimation process, and there are uncertainties in the estimated values. Damping estimation is always more uncertain than modal frequency estimation percentage wise. Because damping tends to increase with the response level, the highest damping values are the most critical. For most of the damping reported here, the modal extraction technique used was the SMAC algorithm[2.] Damping estimates were extracted at force levels from 0.5 lbs to 12 lbs RMS and were shown in Table 1.

THEORY OF EXRACTING MODES FROM A SHAKER TABLE VIBRATION TEST

To perform the modal analysis, one analyzes data either in the form of impulse response functions or frequency response functions (FRFs). Both require a measure of the input force, which is generally not available during a vibration qualification test. Carne[3] follows Beliveau[4] in deriving accelerance FRFs from transmissibility functions with a reference accelerometer mounted on the shaker table in the direction of the input motion. He assumes that the shaker table is rigid and has motion only in the single direction of motion, i.e. it is fixed in the other five degrees of freedom of the rigid table. With Carne's approach, all that is required is to subtract one from each transmissibility with response in the direction of the table motion, and the transmissibilities with response orthogonal to the table input motion are left unchanged. The resulting FRFs are equivalent to fixing the base and applying a body force to the rest of the structure that is parallel with the direction of input motion. In reality for the vibration test, the shaker table is not rigid, neither is it perfectly constrained in the five other degrees of freedom besides the translation in the direction of input. By subtracting one from appropriate transmissibilities, the only response that is truly mathematically constrained to zero is the reference accelerometer degree of freedom which is intentionally mounted in the direction of input motion of the shaker table. The boundary conditions everywhere else are whatever is naturally provided by the shaker system. For this reason, the modal analysis results provided here are not "fixed base", but simply fixed at one point on the shaker table in the direction of input motion. As will be observed in the results, some modes of this test are clearly not fixed base, and the modal frequencies extracted should not be considered to be fixed base frequencies. However, we assume that the mode shapes are similar enough to those extracted in the original modal test on the seismic mass that the damping ratios estimated for each mode are useful for FE predictions. Modal Assurance Criterion (MAC) values are provided in Table 3 to assess the similarity of each mode shape on the vibration table to the seismic mass modal test shapes.

Since Carne's method requires subtracting one from the FRFs only in the direction of the input motion, this creates an additional complication in the data analysis. Most of the triax accelerometers are in coordinate systems that are not lined up with the direction of input motion. A significant amount of work would be required to develop the transformations of all the FRFs into motions parallel or orthogonal to the direction of input motion. The data were to be analyzed with the SMAC modal analysis package[2]. We observed that if the real modes option in SMAC is utilized, only the imaginary parts of the FRFs are fit to determine the mode shapes. Thus, subtracting one (a real number) from the transmissibility FRFs as in Carne's approach has no effect on the mode shape estimate. In fact, with the real mode assumption, most of the FRF data utilized to fit the frequency and damping are imaginary, although there are small real components. The remaining question was whether the frequency and damping estimate would be impacted in any significant degree by simply not performing the subtraction of one in the direction of the input motion. Previous experimental transmissibility data were analyzed with SMAC to see if the subtracting of one for certain directions had a significant effect. The results are in Table 2 below. It appears that the effect of subtracting one is negligible when utilizing the real modes assumption in SMAC. Therefore the data from these tests were analyzed without transforming and subtracting one from parallel transmissibilities.

Frequency with Subtraction (Hz)	Frequency without Subtraction (Hz)	%Damping with Subtraction	%Damping without Subtraction
119.20	119.19	1.06	1.05
361.94	361.90	0.66	0.64
472.27	472.20	0.26	0.26

Table 2: Comparison of SMAC Real Modes Modal Parameters on Transmissibility Data with and without Carne subtraction

VIBRATION RESULTS

The vibration environments were achieved by attaching the space-bound system to an electro-dynamic shaker table and applying proto-flight level loads and acceleration environments to the structure. All tests were conducted on the T-4000 electro-dynamic shaker, first on the slip table for the lateral directions (with a 90 degree rotation between tests), then on an expander head for the vertical direction. The mode shapes could be correlated both visually and by MAC calculations (Table 3) to the similar mode shape previously extracted from the traditional modal test on the seismic mass. If the MAC is one they are the same and if it is zero they are very dissimilar. In every case, the damping is significantly higher in the high level vibration tests with full level damping shown in Table 4. The boundary conditions of the vibration table tests are different from fixed base, so the modal frequencies from these extractions are not of great value from a model validation perspective. Damping increases with response level for any particular test setup. In the case of the vibration table test, the mode shape is utilized to find correspondence with the previous seismic mass modes. For mode shapes that correspond, we assume that the associated damping values can be compared. Also, we assumed that corresponding mode shapes from the seismic mass and vibration table tests will exercise the structure in about the same manner, though the overall amplitude may be much higher in the high level vibration tests.

MAC -15dB	MAC -12dB	MAC -9dB	Original Seismic Mass Frequency	-15dB Vibe Frequency	Mode Shape Description
0.97	0.98	0.97	31.8	29.6	No
0.94	0.94	0.93	36.9	33	Wag
0.98	0.98	0.98	39.5	36.8	Yes
0.94	0.94	0.91	65.9	54	Lateral
0.88	0.89	0.89	84.5	65.7	Lunge

Table 3: MACs between Lateral Vibration Test and Original Modal Test

Description	Lateral full level		90° Lateral full level	
	frequency	damping	frequency	damping
No	29	2.6	28.8	2.7
Wag	32.5	2.8	33.1	2.1
Yes	36.1	2.4	36	3.1
Lateral	51.3	4.1	51.7	5.1
Lunge	*	*	63	6

Table 4: Full Level Modal Damping Estimate

*The lunge mode almost disappeared in high level lateral direction testing, therefore we estimated ten percent damping as a value that essentially eliminates the effects of this mode.

SUMMARY & CONCLUSIONS

During the Vibration tests, all the target modes could easily be observed in the transmissibility data. Transmissibility is a frequency domain ratio of acceleration response divided by an input acceleration. For these transmissibility data, the input acceleration was from an accelerometer attached to the slip table or the expander head of the shaker. The transmissibility functions, measured from a driven-base vibration test, can be used to easily extract modal parameters of a system fixed in the reference accelerometer degree of freedom. There was clearly some flexibility in the slip table and fixtures used during the vibration test. All of the modal frequencies were shifted somewhat lower from those of the seismic mass modal test; some were significantly shifted. However, the mode shapes actually remained similar as shown from MAC values given in Table 3; therefore we assume the modal damping values measured during the vibration test are representative of the damping in the mode for flight environments. For the vibration test in vertical direction, the compliance in the head expander was sufficiently great that the driven base could not be approximated by a rigid base. Consequently, modes shapes changed radically from the seismic mass test. The modal damping extracted from the vertical-input vibration test was not discussed here.

As a note for future satellite qualification testing, Mayes[5] has demonstrated, on a small scale, the theory for acquiring modal test data from a random vibration test, and mathematically manipulating the data to estimate fixed base modal parameters which could be used for model validation directly, including modal frequencies. This capability requires some additional measurements on the vibration table, and still needs to be proven on a large scale system, but it is step in the direction towards replacing the traditional modal survey with test on a vibration shaker table.

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BIOGRAPHIES

Thomas G. Carne received his PhD in Engineering Mechanics from Caltech in 1972. Following graduation he worked at the General Motors Research Laboratories. In 1977 he moved to Sandia National Laboratories where he worked in finite element modeling and modal testing of structural dynamic systems. He is currently consulting for MannaTech Engineering.

Mr. Martinez works in Experimental Environment Simulation and Test Integration Department at Sandia National Laboratories in Albuquerque. He has performed modal tests on various articles including weapons systems and satellites. He holds a BS and MS in Mechanical Engineering from the University of New Mexico.

Mr. Mayes is employed at Sandia National Laboratories as an engineer working in the experimental structural dynamics area. His interests include modal testing, model validation and force reconstruction. He began his career at Sandia in 1979 as a structural analyst. In 1989 he moved to the experimental modal analysis group. He holds a BS and MS in Mechanical Engineering from Texas Tech University.

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