

# Virtual Experimental Modal Analysis (VEMA)

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## ABSTRACT

*Experimental modal analysis (EMA) is an accepted methodology for identifying modal parameters of complex system assemblies, but large channel counts and multiple references are usually required to perform a thorough modal survey of such structures. In these cases, modal testing can become very costly and time consuming. Moreover, success in accurately defining all the modes of interest from experimental data depends on a variety of factors, including how and where the excitation is applied, how and where measurement data is acquired, and how boundary conditions or other environmental conditions are simulated.*

*This paper presents a methodology for virtual experimental modal analysis (VEMA), that is, for simulating a complete EMA in the computer, including the test structure, instrumentation, data acquisition and modal parameter identification. This is done by linking two existing commercially-available software packages together, one for simulating the instrumentation, excitation, and response of the test structure, and one for post-processing multi-channel test data. This approach greatly improves the chances for a successful test by allowing one to easily perform a large number of virtual EMA's before entering the lab, and thereby determine optimal locations and ranges for instrumentation and excitation.*

## INTRODUCTION

The chances of performing a successful EMA are significantly improved if answers to the following critical questions are known beforehand;

- How many exciters are required, and where should they be located?
- What type of excitation signals should be used?
- What analysis frequency range should be used?
- How much frequency resolution is needed?
- How many response transducers are required, and where should they be located?
- How can environmental conditions such as component positions or boundary conditions, be accounted for?

- How do operating conditions, such as internally generated forces, excite the modes?

In this paper, we perform a variety of different simulated tests on a satellite model, to show how VEMA is used to answer some of the above questions.

## Satellite Model

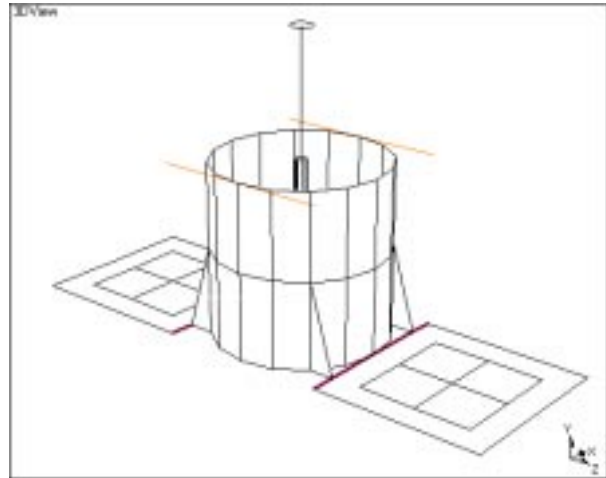


Figure 1. Satellite Model.

The satellite model, shown in Figure 1, represents a typical large communications satellite, with two very flexible solar panels and a high gain antenna. The satellite body is 3 m in diameter and 3 m long. The solar arrays are also (3 m x 3m). The antenna extends out 2 m from the end of its storage cylinder. The total mass of the satellite is about 4500 kg.

There are three momentum wheels inside the satellite body, which are used both for pointing and to stabilize the vehicle's attitude. These wheels were unpowered during the simulations, but could just as easily have been powered.

It should be noted that the solar arrays are so flexible that they have to be reinforced, stowed and locked during flight. It would be extremely difficult, if not impossible, to do actual ground vibration testing on a satellite of this type in a gravitational environment.

## THE ANALYSIS TOOLS

Two different commercially available analysis packages were linked to perform the VEMA; the ADAMS<sup>®</sup> mechanical system simulation package from Mechanical Dynamics, Inc. and the ME'scope<sup>™</sup> experimental modal analysis package from Vibrant Technology, Inc.

### ADAMS<sup>®</sup>

ADAMS was used to simulate the excitation and dynamic response of the complete satellite structure, including the placement and sensitivities of the transducers. ADAMS yields simulated time histories for both the excitation and response signals.

ADAMS (for Automatic Dynamic Analysis of Mechanical Systems) is a suite of modeling and analysis tools that enables engineers to build "virtual prototypes" and realistically simulate the full-motion behavior of complex mechanical systems. Using ADAMS, hundreds of design variations can easily be analyzed, and system performance refined and optimized.

An add-on package for ADAMS, called ADAMS/VEMA, was used to convert the system geometry and the simulation outputs directly into ME'scope format. ADAMS also links directly with all major CAE packages, including CAD, controls and FEA, allowing for rapid model development, incorporation of flexible, electronic and hydraulic elements, and straightforward design validation.

### ME'scope<sup>™</sup>

ME'scope was used to process the time histories, computer FRFs between the excitation and response signals, and identify the modal parameters of the structure by curve fitting the FRFs.

ME'scope is a family of post-test analysis tools that allow the user to observe, analyze, and document the dynamic behavior of machines and mechanical structures. It contains a number of methods for analyzing vibration data. These include; 1) animation of operating deflection shapes on a 3D-model of the structure directly from a set of time or frequency domain data, 2) modal parameter estimation with display of the mode shapes in animation on a 3D model.

ME'scope also contains MIMO (multi-input multi-output) analysis, which can be used for computing FRFs from simultaneously sampled excitation and response time domain signals. FRFs are computed using spectrum averaging, time domain windowing, and overlap processing, similar to the implementations in most FFT analyzers.

## MULTI-CHANNEL DATA ACQUISITION

A key advantage of the VEMA approach is that it simulates multi-channel data acquisition. Multiple transducer and excitation locations and directions can be easily introduced. Transducer sensitivities and signal processing can also be simulated. All of the excitation and response data is simultaneously sampled, for as many channels as desired. A set of simulated data can be acquired in only a few minutes, whereas acquisition of the actual data in a testing environment might take hours or even days.

Another advantage of VEMA is that tests that would be physically difficult to accomplish, such as underwater or zero-gravity conditions, can be easily done in the computer. Also, systems where it would be impossible to attach the real instrumentation or excitation, for example disk drives, can also be tested.

Figure 2. depicts our simulated test setup. The response accelerometer locations are numbered (1 to 36), and the excitation point indicated. A total of 44 response accelerometers were simulated, 14 on each of the two solar panels mounted normal to the surface, 10 in the X & Y-directions (normal to the axis) on the vertical antenna, and 6 in all three directions on the ends of the body. The 44 sampled response signals, plus the excitation signal(s), were output from the ADAMS simulations.

{ *EMBED PBrush* }Figure 2. Transducer Locations.

### Zero Gravity

Another advantage of VEMA is that gravity can be turned on or off in the ADAMS program. This is particularly useful for simulating ground vibration testing, or for investigating vibration responses in the zero-G space environment.

## FREQUENCY RANGE & RESOLUTION

One of the fundamental questions in any EMA is, “What frequency range and resolution is needed to adequately estimate the parameters of the fundamental modes?”

In many real EMA’s, the easiest way to answer this question is to impact the structure with an instrumented hammer, measure a few FRFs, and examine the FRFs for resonance peaks. Also, a preliminary look at the mode shapes is helpful if a sufficient number of FRFs are measured over the surface of the structure so that one shape can be distinguished from another. This can also be done in VEMA.

## SIMULATED IMPACT TESTS

To determine a suitable frequency range and resolution for identifying the system modes, we first simulated several impact tests on the satellite. As shown in Figure 2, the structure was impacted in the vertical direction on the cylindrical body of the satellite. The measurement location and directions were chosen to give more than sufficient data to identify the modes of the antenna and solar panels. (Virtual instrumentation costs are zero!) The satellite body was made rigid for these tests.

### Test #1. 10 Second Impact Test (0.1 Hz Resolution)

In this first impact simulation, an impulse of 20 Newton’s maximum amplitude and 20-msec. width was used for excitation.

The excitation and response signals were sampled using a sampling rate of 409.6 Hz., corresponding to a time between samples of  $\Delta t = 0.00244$  sec. The signals were sampled for 10 sec., giving a total of 4096 samples per trace.

Since  $\Delta f = 1/T$ , this yields a frequency resolution  $\Delta f = 0.1$  Hz. Due to the Nyquist criterion, the sampling rate yields FRFs with a frequency range of (0 to 204.8 Hz).

The 45 time histories (the impulse and 44 responses) were then transferred to ME’scope, where MIMO analysis was used to compute 44 FRFs, between the single excitation DOF and the 44 response DOFs. A plot of the log magnitudes of all 44 FRFs overlaid on one another is shown in Figure 3.

Figure 3 indicates that there is a cluster of fundamental modes in the 0 to 10 Hz region, plus modes at approximately 100, 140, and 200 Hz.

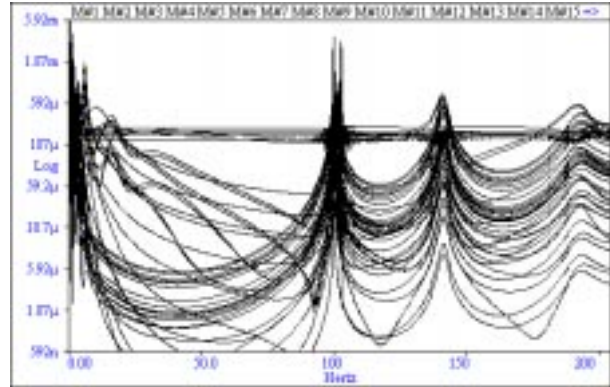


Figure 3. Test #1, 44 Overlaid FRF Magnitudes (0 to 204.8 Hz).

## Usable Frequency Range

Although the modes above 100 Hz appear to be excited in Figure 3, we should check the Auto Power spectrum on the excitation to verify this. Figure 4 shows the power spectrum of the impulse signal, indicating that it is providing adequate energy to excite the modes of the structure approximately over the range (0 to 50 Hz).

Since the fundamental modes are all well within this range, we can assume that they were adequately excited, provided the excitation wasn’t close to a nodal point (zero amplitude) of one of these modes. We will therefore focus our attention on the modes below 50 Hz.

{ EMBED Word.Picture.8 } Figure 4. Auto Power Spectrum of the Impulse.

Figure 5 is a zoomed display (0 to 8 Hz) of the *imaginary parts* of the 44 FRFs, overlaid on one another. The resonance peaks in the imaginary part of (acceleration/force) or (displacement/force) FRFs typically have the narrowest peaks, making it easier to identify modes. Figure 5 clearly shows the resonance peaks of the fundamental modes.

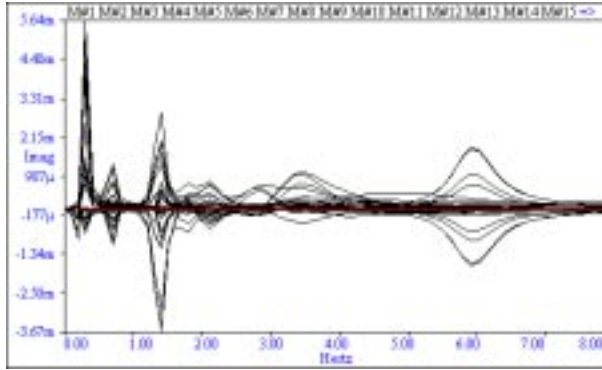


Figure 5. Test #1, 44 Overlaid FRF Imaginary Parts (0 to 8 Hz).

### Modal Parameter Estimation

These 44 measurements were curve fit using ME'scope, to estimate the modal parameters of the modes in the ( 0 to 7 Hz ) range. The frequencies and damping of these modes are given in Figure 6.

The first two modes are very lightly damped compared to the other modes. The (first two) narrow resonance peaks in Figure 5 also indicate this. The higher frequency modes are more heavily damped, also verified by the wider resonance peaks in the FRFs.

Shape	Frequency (Hz)	Damping (%)	Damping (Hz)
1	245.122E-3	1.043	2.557E-3
2	291.023E-3	1.196	3.482E-3
3	652.733E-3	2.575	16.811E-3
4	1.363	5.269	71.907E-3
5	1.814	2.292	41.576E-3
6	2.078	8.181	170.607E-3
7	2.819	11.362	322.441E-3
8	3.289	14.401	478.577E-3
9	5.910	6.398	378.888E-3

Figure 6. Frequency & Damping Estimates of Fundamental Modes ( $\Delta f = 0.1$  Hz).

### More Resolution Needed

Figure 7 is the same as Figure 5, except it is zoomed still further to ( 0 to 5 Hz ). It is clear from Figure 7 that the frequency resolution of these measurements is marginal for estimating the modal parameters of these low frequency modes. There are simply not enough frequency samples to define each resonance peak.

As a “rule of thumb”, 5 or more samples above the 6-dB points, or the noise floor, of a resonance peak is considered acceptable for good curve fitting.

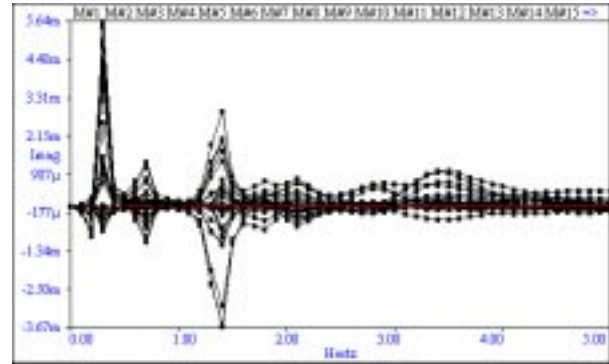


Figure 7. Test #1, 44 Overlaid FRF Imaginary Parts (0 to 5 Hz).

### Test #2. 20 Second Impact Test (0.05 Hz Resolution)

In a second simulated test, we increased the frequency resolution of the FRFs by sampling the simulated signals for 20 seconds instead of 10 seconds. Hence, the frequency resolution of the second simulation was  $\Delta f = 0.05$  Hz, instead of  $\Delta f = 0.10$  Hz.

Figure 8 is a zoomed display ( 0 to 8 Hz ) of the imaginary parts of the FRFs overlaid on one another. It is apparent that the resonance peaks are better defined by the increased resolution.

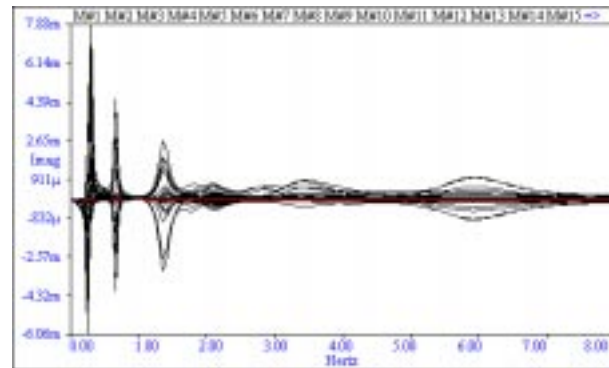


Figure 8. Test#2, 44 Overlaid FRF Imaginary Parts (0 to 8 Hz).

These measurements were also curve fit to estimate the modal parameters of the modes in the (0 to 8 Hz) range. The frequencies and damping are given in Figure 9.

Notice that the damping of the first mode is considerably less than the estimate from the previous impact test. This nearly 50% change in modal damping lead

us to simulate a third impact test, with still more resolution.

Shape	Frequency (Hz)	Damping [%]	Damping (Hz)
1	246.167E-3	560.938E-3	1.381E-3
2	291.096E-3	1.204	3.504E-3
3	653.075E-3	3.056	19.965E-3
4	1.362	6.295	85.917E-3
5	1.760	7.857	138.706E-3
6	2.071	9.877	205.582E-3
7	2.820	13.521	384.807E-3
8	3.238	16.428	539.240E-3
9	5.857	10.844	638.898E-3

Figure 9. Frequency & Damping Estimates of Fundamental Modes ( $\Delta f = 0.05$  Hz).

### Test #3. 30 Second Impact Test (0.033 Hz Resolution)

In a third simulated test, we increased the frequency resolution of the FRFs yet again, by sampling the simulated signals for 30 seconds. Since  $\Delta f = 1/T$ , the resolution of these FRFs is 0.033 Hz.

Figure 10. is a zoomed display ( 0 to 1 Hz ) of the imaginary parts of the 44 FRFs from the third impact test, overlaid on one another. Evidence of a third mode,(between the 0.246 and 0.291 Hz modes), is shown by the imaginary peaks in Figure 10.

{ EMBED PBrush }Figure 10. Test #3, 44 Overlaid FRF Imaginary Parts (0 to 1 Hz).

These measurements were also curve fit to estimate the modal parameters of the modes in the ( 0 to 7 Hz ) range. The modal frequencies and damping are given in Figure 11.

Shape	Frequency (Hz)	Damping [%]	Damping (Hz)
1	240.876E-3	1.248	3.007E-3
2	254.535E-3	665.566E-3	1.694E-3
3	289.917E-3	1.524	4.420E-3
4	651.651E-3	3.516	22.928E-3
5	1.358	7.287	99.192E-3
6	1.755	10.717	189.213E-3
7	2.052	11.444	236.367E-3
8	2.681	15.423	418.432E-3
9	3.275	17.503	582.107E-3
10	5.759	15.142	882.156E-3

Figure 11. Frequency & Damping Estimates of Fundamental Modes ( $\Delta f = 0.033$  Hz).

### Comparison of Mode Shapes

To better understand the three fundamental modes found in the third set of test data, their mode shapes were displayed in animation. From the three mode shape displays, Figures 12 to 14, it is clear that the first bending mode of the antenna falls between the first symmetric and anti-symmetric bending modes of the solar panels.

In Figure 12, only the solar panels are moving, in symmetric bending. In Figure 13, only the antenna is moving, in first bending. In Figure 14, the panels are moving in anti-symmetric bending, along with the antenna in second bending.

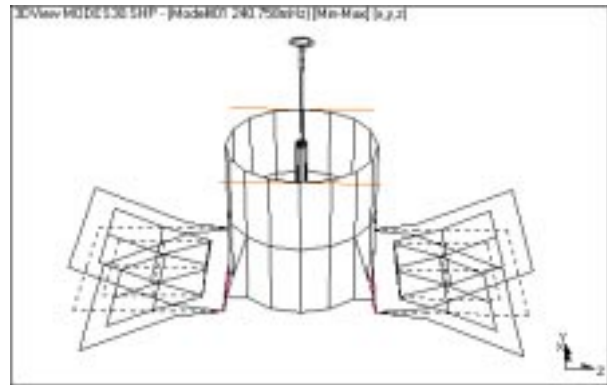


Figure 12. Symmetric Bending (0.241 Hz)

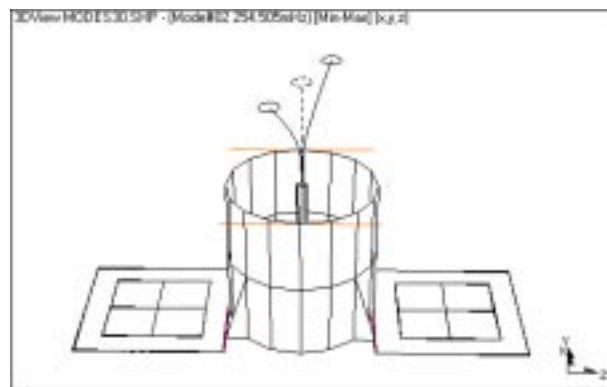


Figure 13. Antenna First Bending (0.245 Hz)



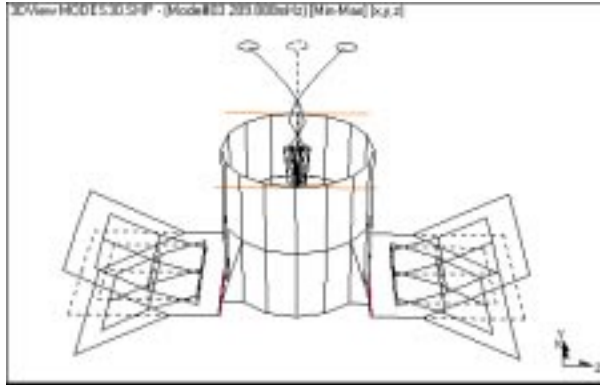


Figure 14. First Anti-Symmetric Bending (0.289 Hz)

Since more resolution showed the first antenna mode to have a different frequency from the panels, one might ask, “Does the second bending mode of the antenna have a separate frequency from the anti-symmetric panel mode?” More frequency resolution is needed to answer that question.

### SIMULATED ONE-SHAKER TEST

Next, shaker testing was simulated and the results compared with impact testing results. The simulated shaker was attached to the same point as the impact point on the satellite body, with its axis oriented in the vertical direction.

A swept sine wave excitation signal was synthesized in the ADAMS software, with a continually varying frequency from 0.1 Hz to 7 Hz. A rapidly sweeping sine wave of this type is called a *chirp signal*.

#### Test #4. 30 Second Chirp Test (0.033 Hz Resolution)

The satellite was excited for 60 seconds using chirp excitation. The excitation and response signals were sampled using a sampling rate of 68.27 Hz., corresponding to a time between samples of  $\Delta t = 0.01465$  sec. Each time domain trace contained 4096 samples.

Since  $\Delta f = 1/T$ , this yields a frequency resolution of  $\Delta f = 0.0167$  Hz. Due to the Nyquist criterion, the sampling rate yields FRFs with a frequency range of ( 0 to 34.14 Hz ).

A partial plot of the chirp signal is shown in Figure 15.a, and its Auto Power spectrum is shown in Figure 15.b.

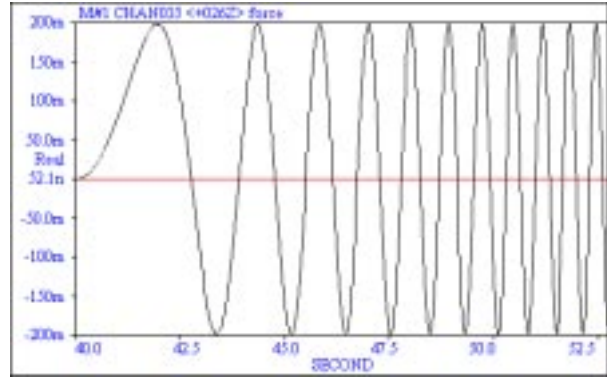


Figure 15.a. Partial Plot of Chirp Signal

Figure 16 is a zoomed display ( 0 to 5 Hz ) of the imaginary part of the 44 FRFs overlaid on one another. The first five modes are clearly shown in the resonance peaks.

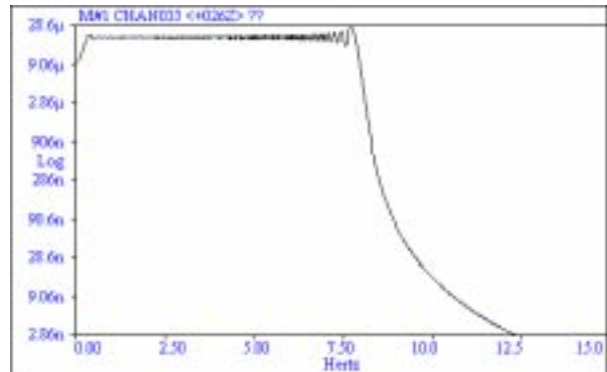


Figure 15.b. Auto Power Spectrum of Chirp Signal

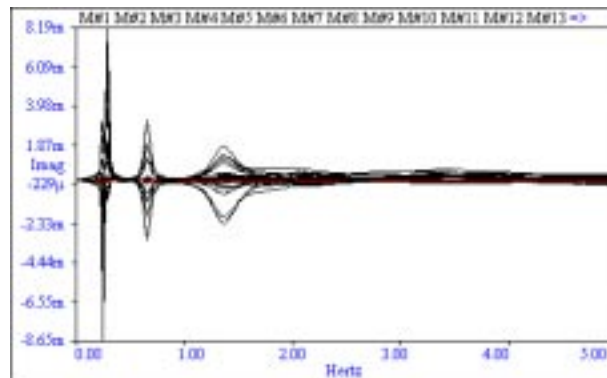


Figure 16. Test #4, 44 Overlaid FRF Imaginary Parts (0 to 5 Hz).

### Modal Parameters

Finally, the FRFs from the shaker test were curve fit to obtain the modal parameters of the first five modes. In table 1, these results are compared with the modal parameters from impact test #3. The Modal Assurance Criterion (MAC) values of the mode shapes are also included. MAC = 1.0 means that the two shapes are

identical. A value less than 1.0 means the two shapes are dissimilarity.

Impact Freq. (Hz)	Shaker Freq. (Hz)	Impact Damp. (%)	Shaker Damp. (%)	MAC
0.2408	0.2409	1.248	1.872	0.999
0.2545	0.2547	0.665	1.276	0.998
0.2899	0.2902	1.524	2.224	0.999
0.6516	0.6517	3.516	4.945	0.999
1.358	1.346	7.287	9.559	0.999

Table 1. Comparison of Impact and Shaker Modes.

## CONCLUSIONS

We have introduced a new method for simulating an Experimental Modal Analysis (EMA) using a combination of two commercially-available software tools, ADAMS and ME'scope. We have named this new simulation method Virtual Experimental Modal Analysis, or VEMA. The primary advantage of this completely analytical approach is that a variety of questions can be addressed regarding the testing of complex structures before the actual testing is begun.

The ADAMS software package was used to simulate the testing of a satellite structure. The simulation included modeling the mass, elastic, and damping properties of the satellite, placement of excitation and response transducers, excitation of the satellite, and multi-channel data acquisition of the time domain excitation and response signals. The sampled time domain signals were then transferred to the ME'scope software package, where they were post-processed.

In ME'scope, a set of FRF measurements was computed from the excitation and response signals. Then, modal parameter estimation (curve fitting) was performed on the FRFs to obtain the modal parameters of the satellite.

We illustrated the use of VEMA by simulating 4 different tests, 3 impact tests to determine adequate frequency resolution, and a swept sine (chirp) shaker test to compare modal parameters with impact test results.

Since impact test #3 allowed us to identify the first three modes, which were lightly damped yet heavily coupled, attention was focused in the simulated shaker test on the first few fundamental modes. The chirp

signal was synthesized between .1 and 8 Hz, which adequately excited these modes. The results in Table 1. show that mode shapes of the first five modes obtained by impact and shaker tests are identical. The frequency and damping estimates are also in very good agreement, especially for the first three heavily coupled modes.

Random testing was also simulated, with equally encouraging results. However, random signals require the use of a Hanning window, to reduce leakage (due to the non-periodicity of the signals), which smears the resonance peaks together. Therefore, better frequency resolution is needed to clearly separate the first three modes, so the random results were not presented.

We have demonstrated that VEMA can be used to simulate the most common types of modal testing; impact and single shaker testing. Other cases need to be simulated, including multi-shaker testing, but based on these results, we are confident that VEMA can be used to effectively address all of the questions about modal testing listed in the Introduction.

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