Determining Damping Trends from a Range of Cable Harness Assemblies on a Launch Vehicle Panel from Test Measurements

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ABSTRACT

The team of authors at Marshall Space Flight Center (MSFC) has been investigating estimating techniques for the vibration response of launch vehicle panels excited by acoustics and/or aero-fluctuating pressures. Validation of the approaches used to estimate these environments based on ground tests of flight like hardware is of major importance to new vehicle programs. The team at MSFC has recently expanded upon the first series of ground test cases completed in December 2010. The follow on tests recently completed are intended to illustrate differences in damping that might be expected when cable harnesses are added to the configurations under test. This validation study examines the effect on vibroacoustic response resulting from the installation of cable bundles on a curved orthogrid panel. Of interest is the level of damping provided by the installation of the cable bundles and whether this damping could be potentially leveraged in launch vehicle design. The results of this test are compared with baseline acoustic response tests without cables.

Damping estimates from the measured response data are made using a new software tool that employs a finite element model (FEM) of the panel in conjunction with advanced optimization techniques. This paper will report on the "damping trend differences" observed from response measurements for several different configurations of cable harnesses. The data should assist vibroacoustics engineers to make more informed damping assumptions when calculating vibration response estimates when using model based analysis approach. Achieving conservative estimates that have more flight like accuracy is desired. The paper may also assist analysts in determining how ground test data may relate to expected flight response levels. Empirical response estimates may also need to be adjusted if the measured response used as an input to the study came from a test article without flight like cable harnesses.

KEY WORDS: Acoustic Test, Reverberant Chamber, Diffuse Acoustic Field, Attenuation, Damping Schedule, Cable Harness, Optimization, Simulation, Patch Density, Vibration Environment, Finite Element Analysis (FEA), Vibroacoustic Response Analysis, Launch Vehicle, Fluid Structure Interaction, Vibration Test Specifications, Secondary Components.

heritage criteria levels or require a delta qualification test program. The test setup, the vehicle panel, and the cable harnesses continue to be a valuable resource for new programs.

INTRODUCTION

The paper presents measured response data to characterize the additional damping provided by cable harnesses which are often supported on light weight vehicle panels. These vehicle panels may be only lightly mass loaded by other equipment. Such a panel might serve as mounting location for a linearly distributed system such as a linear shaped charge or frangible joint used for stage separation, or perhaps for a systems tunnel running lengthwise on a cylindrical launch vehicle.

If a panel were lightly damped and lightly loaded by equipment, the vibration environment resulting from external acoustics or boundary layer FPLs could be very severe, "a screamer". However these panels often support cable harnesses which may provide significant damping. The attenuation effects from cable harnesses may be a benefit for the vibration environment that the separation device or light systems tunnel equipment would need to survive reduced vibration environments usually result in simpler designs and lower costs.

Measured data is presented in order to demonstrate the effect that cable harness bundles have on the vibration of an exterior vehicle panel system. Five representative test article configurations were selected. Each of these configurations was excited using a nearly identical reverberant acoustic field. The cable harness configurations range from a population of cable harnesses referred to as "heavy" to "medium" to "light" and then a case with no cables at all. Also considered is a case similar to our medium configuration but with 1/3 as many cables. The "1/3 medium" set of cables provides a similar mass of cables as the "light" configuration but achieves that mass with fewer but larger cable harnesses.

The results for these 5 cases, will characterize the attenuation effect that cable harness bundles have on the vibration response of an external vehicle panel system. The data will demonstrate the trends that correspond to including greater and greater populations of cable harness bundles have on the vibration response.

Interesting questions might include:

1. What population of cables is necessary before a significant attenuation of vibration response is observed? Is there a threshold below which it makes little difference?

2. Does the attenuation affect continue to increase as more cable harnesses are added?

3. Does the attenuation appear to be a damping effect or perhaps an inertial (i.e. mass loading) effect?

The five cases will be sorted according to total weight of cable bundles in each. They could also be sorted in other ways for instance the number of bundles.

Another phenomenon that will be investigated is the linearity of these attenuation effects. For this part of the study, a comparison of the attenuation provided by the same configuration of cable bundles for excitation at different sound pressure levels is provided. Our data includes measured vibration response cases for the same configuration of cable bundles excited at 4 different spectral levels. The change in attenuation for both the medium and heavy cases are presented. The linearity check is realized by normalizing the vibration response to the same average baseline excitation level. If an overlay of the "normalized vibration response curves" does not collapse to one vibration level, then the response is shown to be nonlinear. A useful comparison of the estimated damping from the most responsive case to the least responsive case is provided. This will serve as an example of the degree to which the damping trends can be nonlinear. This range of possibilities should be of general interest to the launch vehicle vibration environment community.

BACKGROUND

The exterior panels of launch vehicles are subject to significant acoustic and aero-fluctuating pressures during lift-off and ascent. Validation and refinement of the approaches used to estimate the vibration environments associated with these panels is of major importance to new vehicle programs and has been identified by the NASA Engineering and Safety Center (NESC) as an area of uncertainty that is worthy of on-going study [Kaouk 2009 and Kern 2010].

In December 2010, a series of acoustic ground tests were conducted on a flight-like vehicle panel in test facilities at the NASA Marshall Space Flight Center (MSFC) [Frady 2011]. The objective of these tests was to gain a better understanding of the response characteristics of increasingly flight-like launch vehicle structural panel assemblies. A follow-on test series was recently completed to quantify the differences in damping that may be expected when cable harnesses are added to the panel configurations under test. This paper presents a damping identification approach that may be employed to estimate the modal damping of the tested configuration. The damping trends observed from response measurements of similar configurations with and without cable harnesses should allow analysts to make more informed estimates of damping for use in analytical models of vehicle systems.

The flight-like test article used in acoustic ground testing at MSFC is a rib-stiffened aluminum orthogrid panel. It has a curved outer mold line that approximately represents a 45^0 section comprising $1/8^{th}$ of the cylindrical exterior shell of a launch vehicle. The panel is clamped in a baffled condition separating the reverberant chamber from an anechoic room (i.e., a flight-like condition where the exterior side of the panel is excited by the high energy acoustic field). The panel is subjected to acoustic noise excitation energies closely simulating the liftoff event and roughly approximating ascent flight events. Note that fixing the baffled panel so that it is excited by source room energies on one side resembles in-service loading of the launch vehicle. Accelerometer, microphone, and strain gage instrumentation was used to measure the acoustic field and the response of the panel.

The acoustic field is generated in the MSFC reverberant chamber using conditioned air driven by up to four parallel WAS 3000 Modulators that feed sound into the chamber through a single

horn. This acoustic power source is utilized to approximate a diffuse acoustic field in a reverberant chamber. The acoustic field is monitored using an array of microphones in front of the test article. The standard configuration of microphones is to center the microphones in seven sectors one inch in front of the test article. A sketch of the test chamber showing the source and receiver rooms set up with the approximate location of the microphone sensor array is presented as Figure 1.
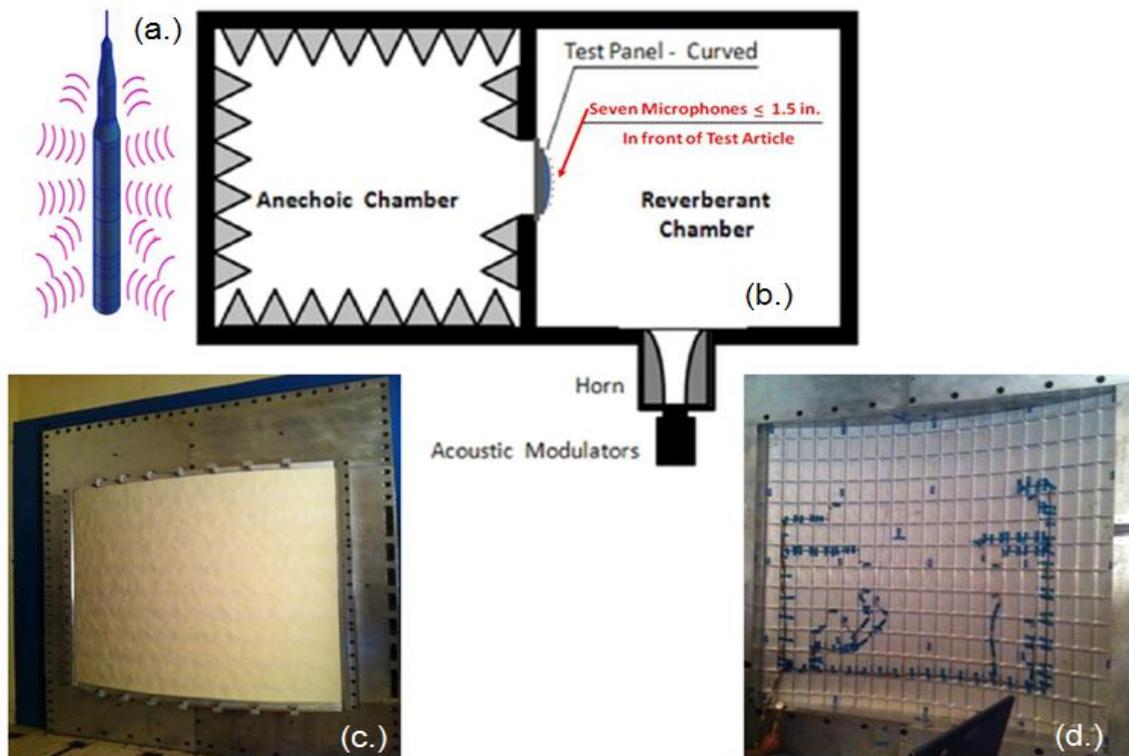

Figure 1. Relating test chamber setup and vehicle panel test article to excitation of a flight vehicle panel. (a.) Acoustic or fluctuating pressures affect the exterior surface of vehicle panels. (b.) Ground test setup. (c.) Flight like excitation of exterior surface in baffled panel test setup. (Reverberant) (d.) View of flight like test article from anechoic receiver room.

PANEL TEST ARTICLE

The AD01panel test article is a cylindrical segment that could be assembled with other similar panels to construct a complete cylinder. For the purposes of configuring the panel in the wall of a reverberant chamber, a smaller representative section of the total circumference was desired. The panel has a smooth outer surface with small orthogrid construction on the interior surface. The material type is an aluminum alloy. It is approximately 81 inches in height. The outer surface is described by a diameter of 216.5 inches. The arc length is approximately equivalent to one eighth of the full cylinder circumference, which is approximately 85 inches.

CABLE HARNESS BUNDLE CONFIGURATIONS

Five test article configurations were selected. Each of these configurations was excited using a nearly identical reverberant acoustic field. The cable harness configurations range from a population of cable harnesses referred to as "heavy," "medium," "light," and then a case with no cables at all, "Brackets only". Also considered is a case similar to the medium configuration but with 1/3 as many cables. The Mass of the 1/3 medium configuration is similar to the light configuration. The mass contribution of the cable bundles is summarized in Table1. Figures 2 and 3 provide mass contribution summary details for the aluminum hardware and describe the dimensions of a footprint over which the cables were installed.

Table 1. Summary of Cable Harness Weights

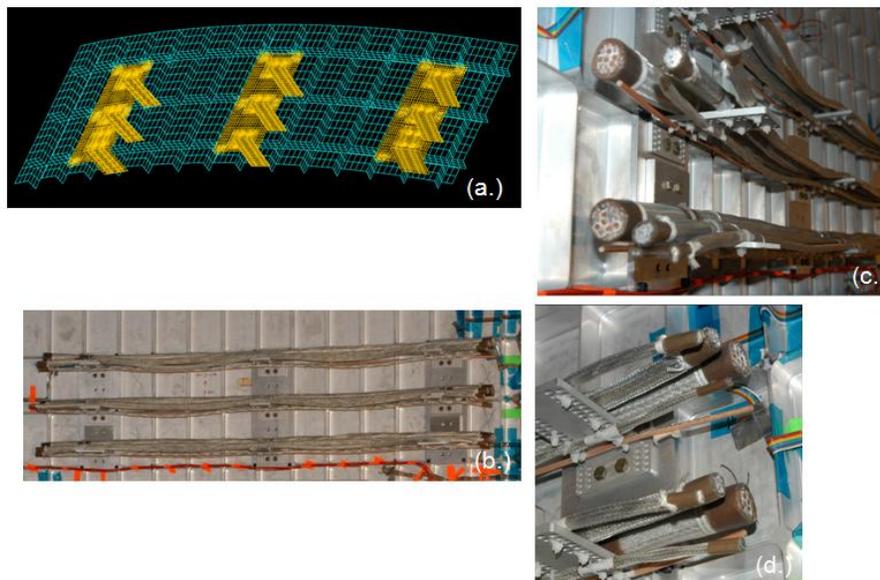| Description	Cable Weight Totals [lb]
Heavy	21.21
Medium	7.11
Light	2.61
1/3 Medium	2.37
Bracket Only	0.00

Figure 2.   Local structure provided for integration of cable harnesses. (a.) FEM view of rib stiffened panel with adapter/brackets configured to support cables. (b.) Photo of same local structures configured with heavy cable set. (c.) close-up of heavy cable set from left end (d.) View of flight like test article from anechoic receiver room.

The different Cable Harness bundles for each of the five configurations appear in Figure 4. The final configuration is simply the same aluminum hardware (panel and brackets) without cables.
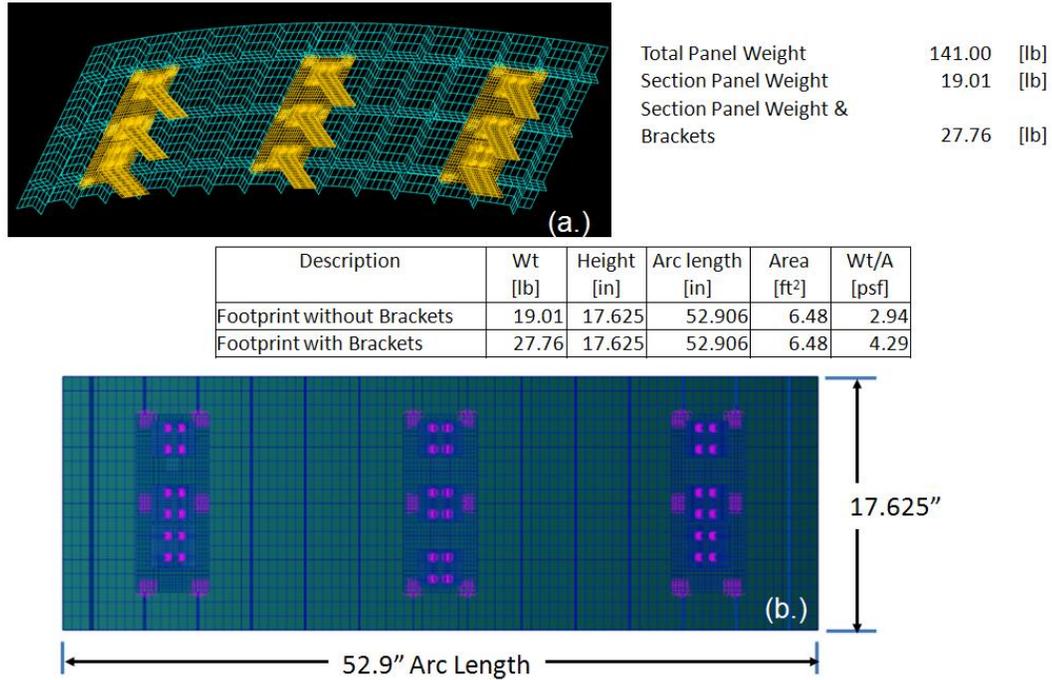

Total Panel Weight	141.00	[lb]
Section Panel Weight	19.01	[lb]
Section Panel Weight & Brackets	27.76	[lb]

(a.)

Description	Wt [lb]	Height [in]	Arc length [in]	Area [ft²]	Wt/A [psf]
Footprint without Brackets	19.01	17.625	52.906	6.48	2.94
Footprint with Brackets	27.76	17.625	52.906	6.48	4.29

17.625"

(b.)

52.9" Arc Length

Figure 3. **Foot print of local panel structure. (a.) FEM view of rib stiffened panel with adapter/brackets configured to support cables. (b.) FEM VIEW OF RIB stiffened panel with dimensions. The weight and weight per unit area are also summarized in the figure.**
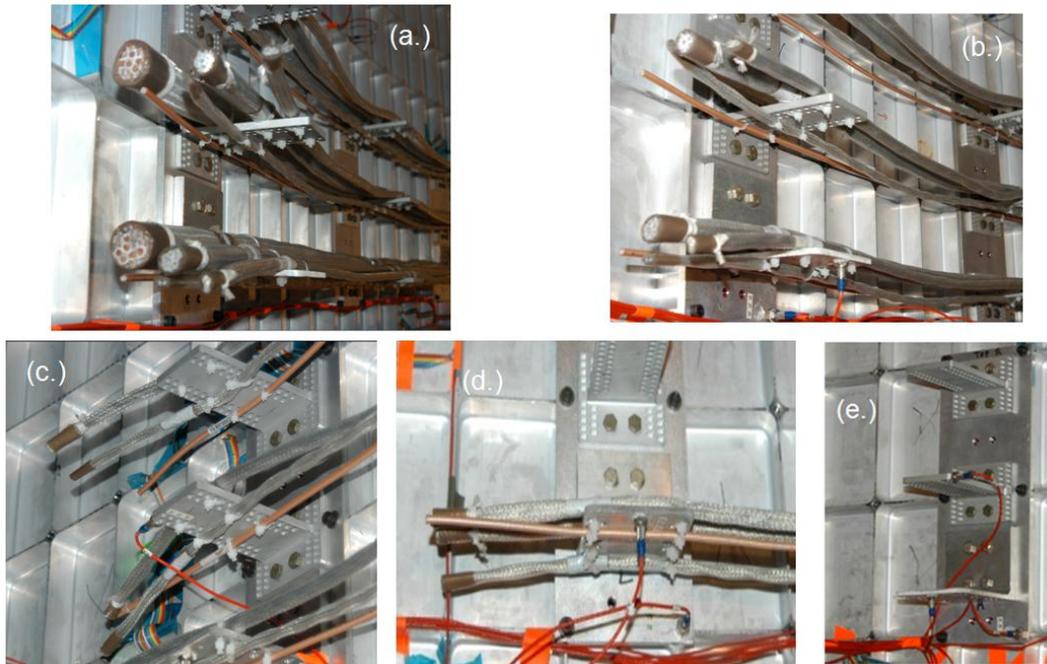

Figure 4. **Photo depiction of each configuration (a) Heavy (b) Medium (c.) Light (d) 1/3 of Medium (e.) Brackets without cables**
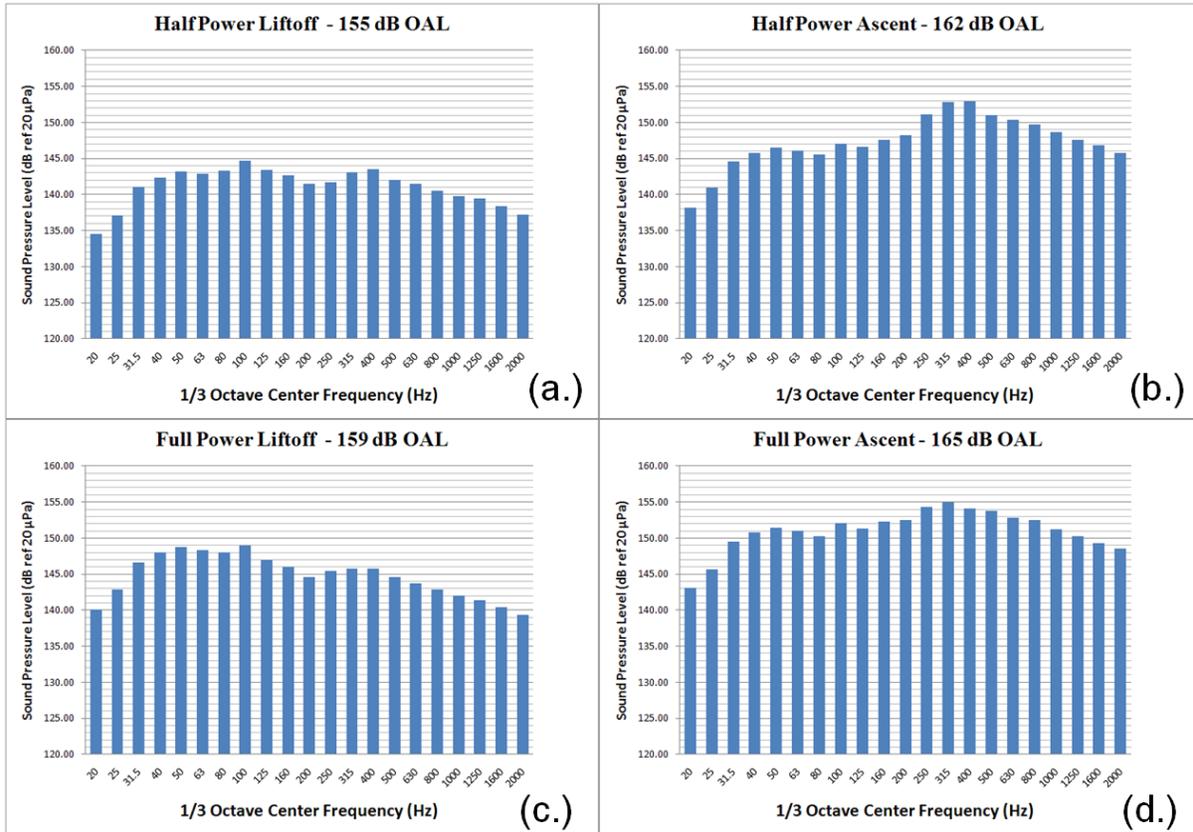
Figure 5. **Average Measured Sound Pressure Level. (a.) Half Power Liftoff. (b.) Half Power Ascent (c.) Full Power Liftoff. (d.) Full Power Ascent .**

ACOUSTIC EXCITATION CASES

The acoustic power source was utilized to create a diffuse acoustic field in a reverberant chamber. The measured Sound pressure levels are presented in 1/3 octave band averages in Figure-5. These spectra represent the average across 7 microphone locations directly in front of the vehicle panel test article.

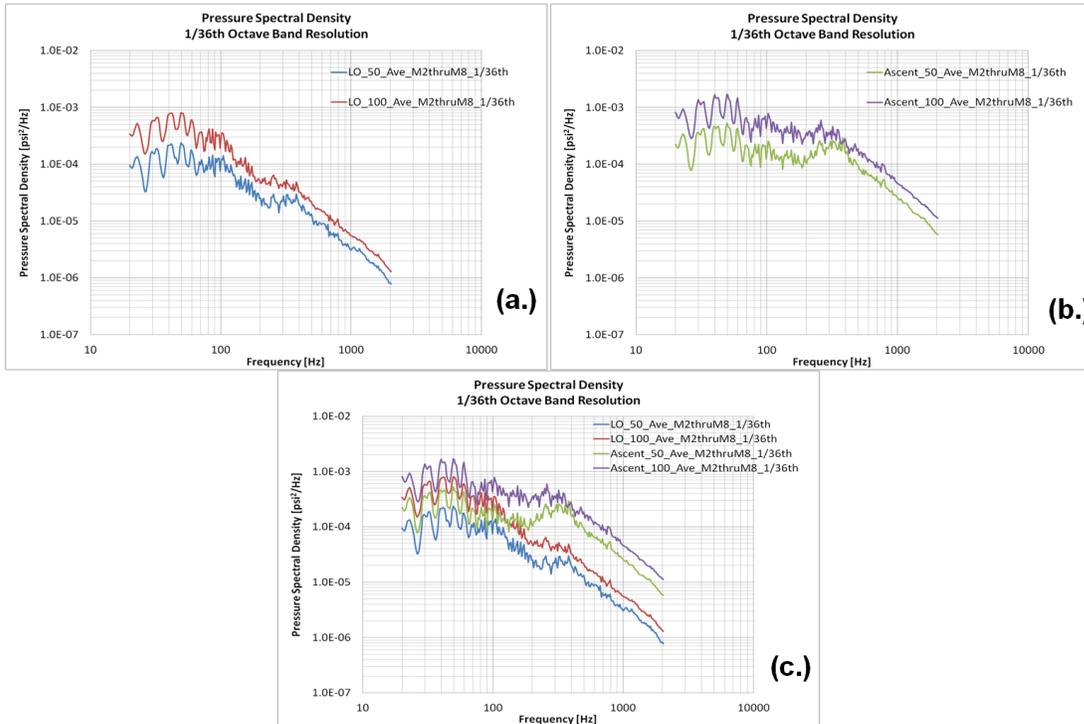The same spectra are presented as narrow band Power Spectral Density (PSD) in Figure-6

Figure 6. Average Measured Pressure Spectral Density presented as 1/36th Octave Band Averaged Spectra. (a.) Overlay of Full Power Liftoff with Half Power Liftoff. (b.) Overlay of Full Power Ascent with Half Power Ascent. (c.) Overlay of all four Test Levels.

DISCUSSION OF MEASURED TEST RESPONSE

The instrumentation set for the acoustic response test included a large number of channels for both accelerometer and strain gauge response measurements. In order to acquaint the reader with a few of the measurement locations, Figure 7 is provided.

The overlay plots provided in Figure 8 each present response at a single common measurement location (Locations1z, 4z and 29z) The overlays compare the difference in response at the same measurement location as the number of cable bundles was increased from "Brackets only" through each of 4 configurations that included cable bundles. The
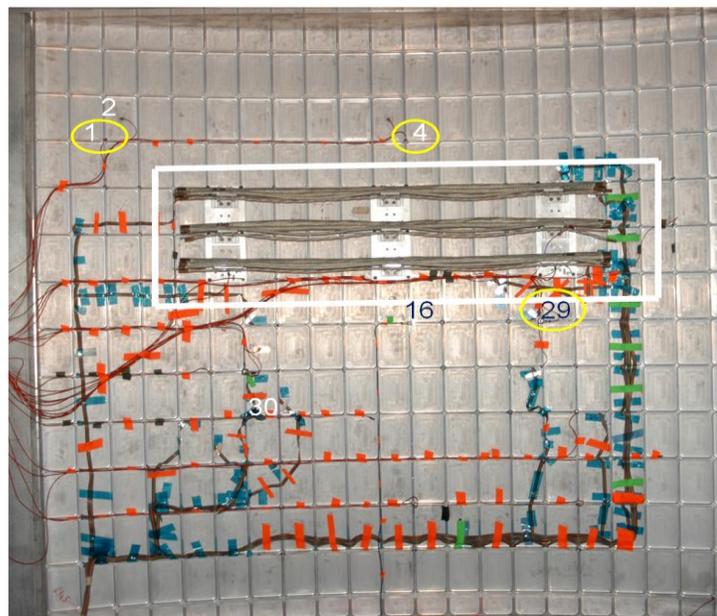

Figure 7. Measurement Locations of Interest just outside the Cable Harness Footprint (outlined in white).

response results for the heavy configuration typically provided the most attenuation of response
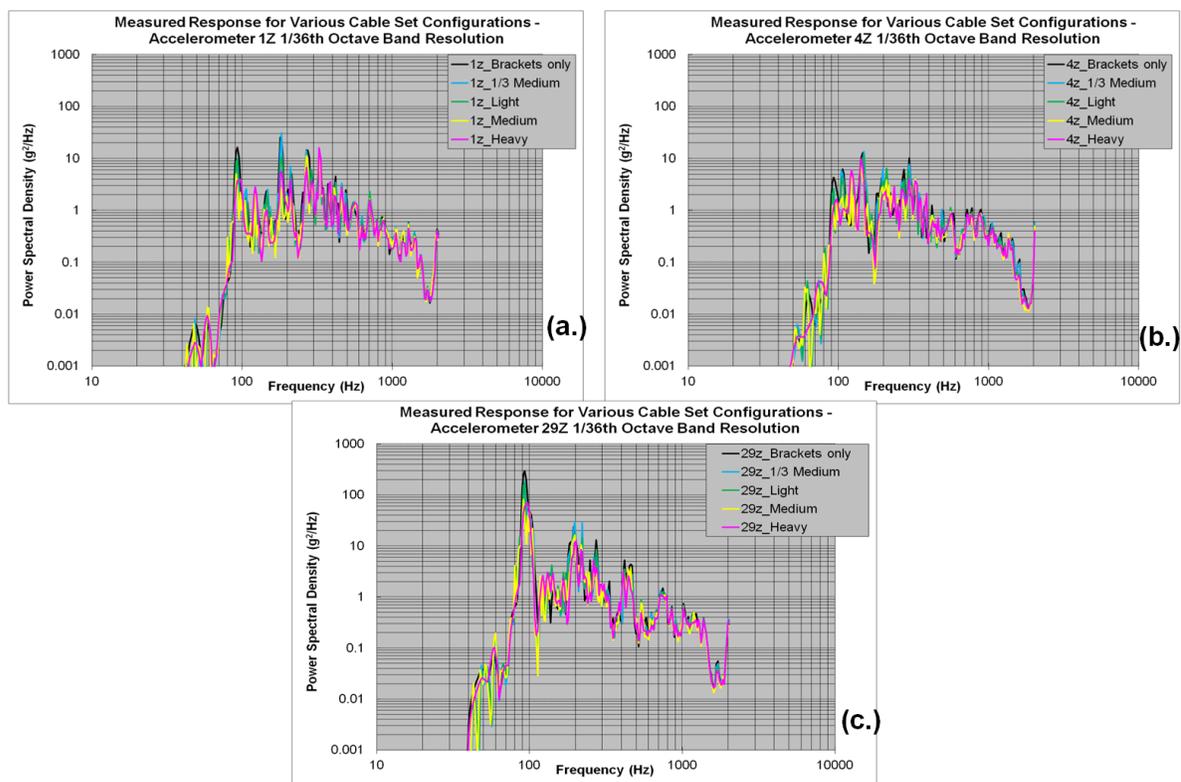
Figure 8. **Measured Response for Various Cable Set Configurations - 1/36th Octave Band Full Power Liftoff. (a.) Accelerometer 1z. (b.) Accelerometer 4z. (c.) Accelerometer 29z.**

as was expected. Response in the low and mid frequency bands exhibited broad band attenuation. But there are a few vibration response peaks which do not appear to have attenuated much at all (the exception and not the rule). Attenuation in the high frequency bands is also observed but appeared to be smaller.

One should note that the response magnitude is highly dependent on location. The most pronounced location to location response differences were observed at low frequency. The
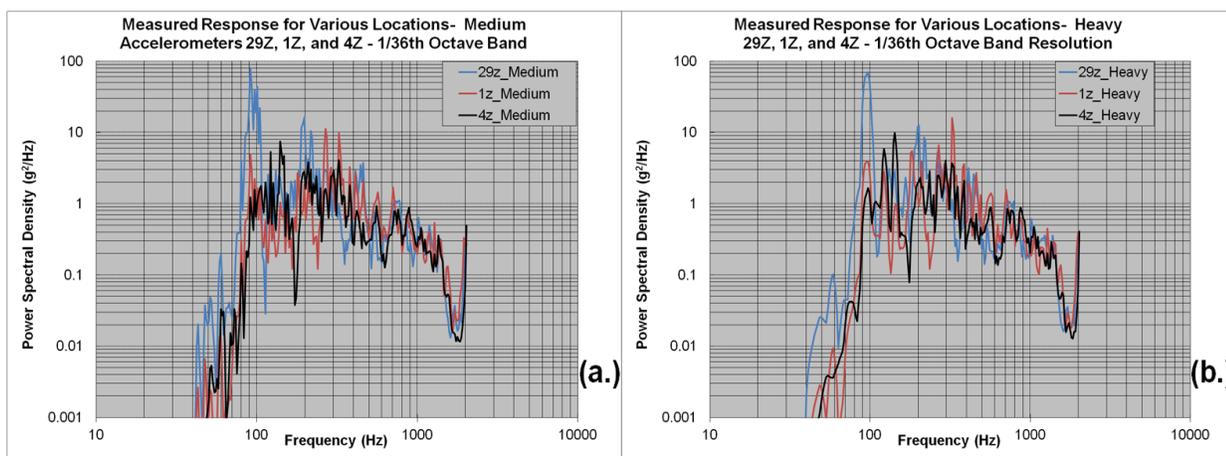

Figure 9. **Measured Response at Various Measurement Locations from Consistent Cable Set Configurations. (a.) Medium Liftoff. (b.) Heavy Liftoff.**

response at different panel locations becomes more similar in the high frequency bands. Figure 9

overlays the response at different measurement locations from each of two cable bundle configurations (Medium and Heavy).

LINEARITY OF MEASURED TEST RESPONSE

The medium and heavy configurations were also examined at four different excitations levels to ascertain any nonlinearity associated with the different excitation levels. The lowest level was selected as a baseline for the nonlinearity study. The measured vibration response from the three higher excitation levels was normalized to the baseline level using a ratio of their corresponding excitation pressure spectral density (recall Figure 6). The overlays of the normalized results are provided for several response channels in Figure 10. If the response were linear with excitation levels then the normalized curves would each lay on top of the other. Figure 10 reveals the not
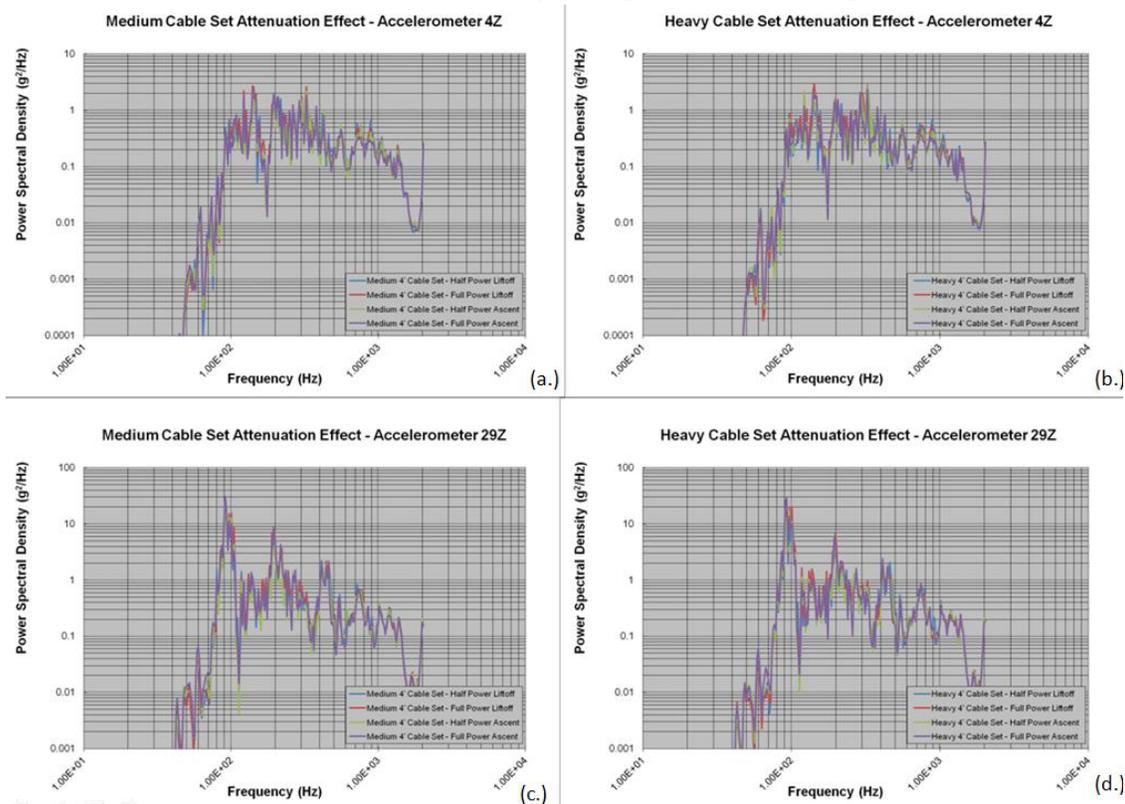

Figure 10. **Overlay of Normalized Response Measurements provided as Linearity Check. Consistent Cable configurations assessed for different excitation levels.. (a.) Accelerometer 4z Medium Configuration. (b.) Accelerometer 4z Heavy Configuration. (c.) Accelerometer 29z Medium Configuration. (d.) Accelerometer 29z Heavy Configuration.**

too unexpected nonlinear behavior in quite a few bands of interest.

OPTIMIZATION OF FEM RESPONSE RESULTS SIMULATING MEASUREMENTS

Consolidation of FEM and test responses was accomplished using a constrained optimization technique. A finite element optimization code called DampID [Davis 2012 and Smith 2012] was used to estimate system modal damping schedule for each of the test cases described herein. The DampID code was used to cast the minimization problem in terms of an objective function composed of multiple optional weighted objectives. For example, one objective might be the

minimization of the difference between maximum peaks in the test and FEM PSD within each user defined frequency band. Another objective might be to minimize the difference between Root Mean Square (RMS) values in the same frequency bands. The weights may be chosen rather arbitrarily and according to their relative importance to the overall objective. DampID is built around the **fmincon** nonlinear constrained optimization function found in the MATLAB Optimization Toolkit.

The test correlated FEM [Maasha 2012] and excitation scheme used to complete the consolidation are depicted in Figure 11.

DampID requires the user to divide the response into a set of fixed frequency bands, typically 20-50 in number. The modal damping at an "anchor" frequency at the center of each band is considered one independent variable. For example a response divided into 40 bands would iteratively vary 40 independent damping values as variables in the optimization. At each iteration a spline fit between the frequency-anchored damping values is applied to populate the damping schedule at frequencies between the anchors. The new damping schedule is then used to update the associated acceleration/pressure transfer function and output acceleration PSD. The difference between the updated FEM response and test response is then evaluated, and the process is repeated until convergence is achieved.

In this project, both the peak PSD response and the RMS response in each band were included in the minimization objective. As stated above, the user may assign relative weights to a number of metrics, depending on how important these criteria are to the user. Several other metrics available to the user are listed in Figure 12.

A special feature in DampID is the ability to include response measurements from several channels simultaneously in the optimized damping solution. Our team has had success on this project by just addressing the peak and RMS response in each of the selected bands. The tool produces a best estimate of system damping that optimizes the response across several measurement channels/locations as illustrated in Figure 13.

The utility of the tool is illustrated by the fine convergence of both measurement location 29 z and 4z simultaneously. Inspection of Figure 9 shows that the lower frequency resonant affects around 90 to 100 Hz differed for these two locations by more than an order of magnitude. Even so the same resulting system damping schedule serves to simulate the response at all three of the measurement locations.
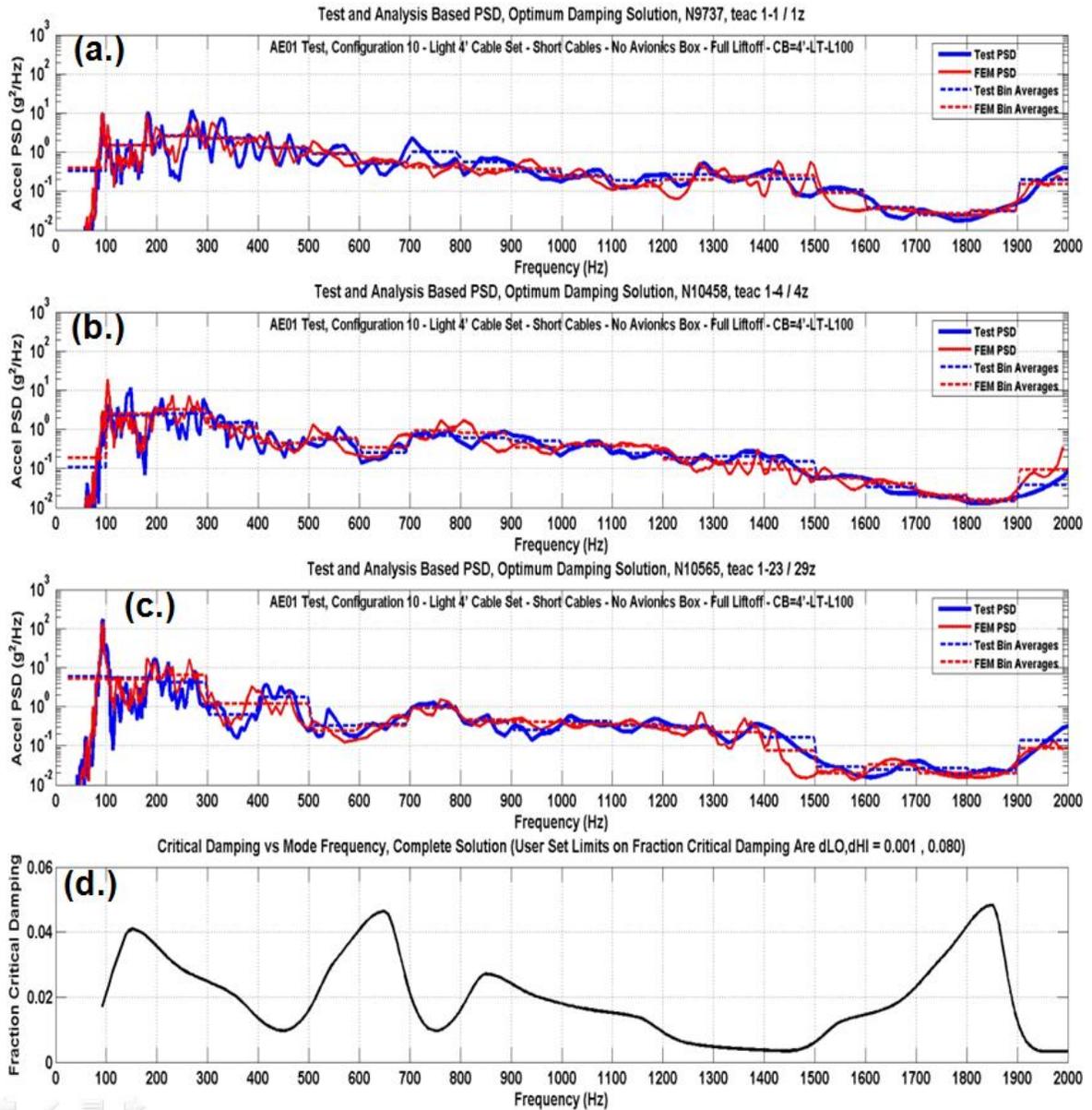

Figure 13.　　　**Example Comparison Response at Locations in the Optimization Set vs. Measured Response for Light Cable Set (Case10) Full Power Liftoff. (a.) Accelerometer 1z - 9737. (b.) Accelerometer 4z - 10458. (c.) Accelerometer 29z - 10565. (d.) System Damping Estimate.**

Furthermore we can spot check measured response from channels that were not used as targets in the optimization and see that the analytical response also did an admirable job of estimating those measured responses, Figure 14.
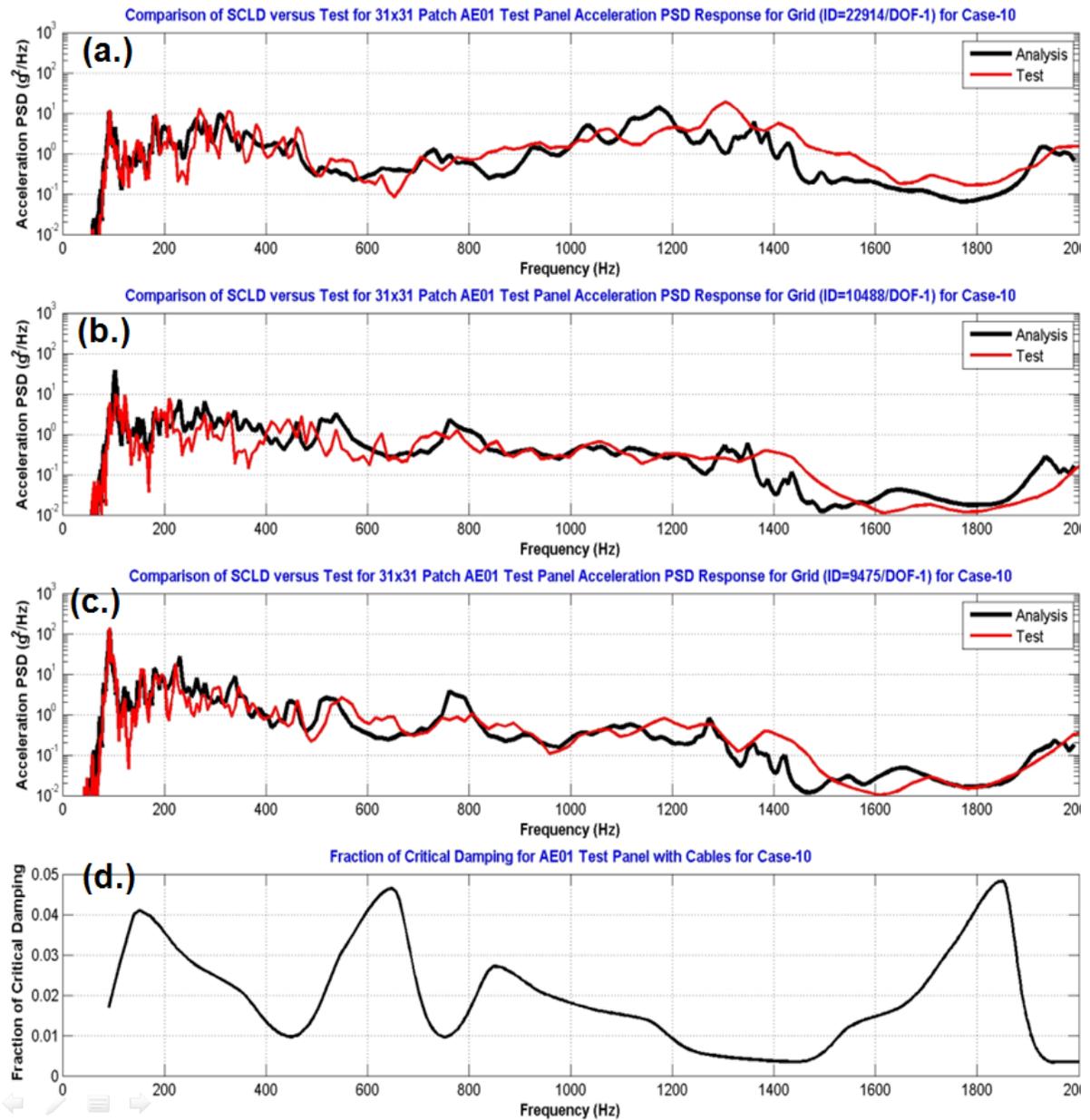

Figure 14. **Example Comparison Response at Locations not used in the Optimization Set vs. Measured Response for Light Cable Set (Case 10) Full Power Liftoff. (a.) Accelerometer 2z -22914. (b.) Accelerometer 16z - 10488. (c.) Accelerometer 30z - 9475. (d.) System Damping Estimate.**

All five of the configurations presented in Figure 4 and Table 1 were evaluated using DampID in order to estimate the system damping schedule of each. Selected results similar to those provided in Figure 11 are presented at the end of the paper covering the other configurations evaluated using the liftoff excitation level (Figures 17- 20).
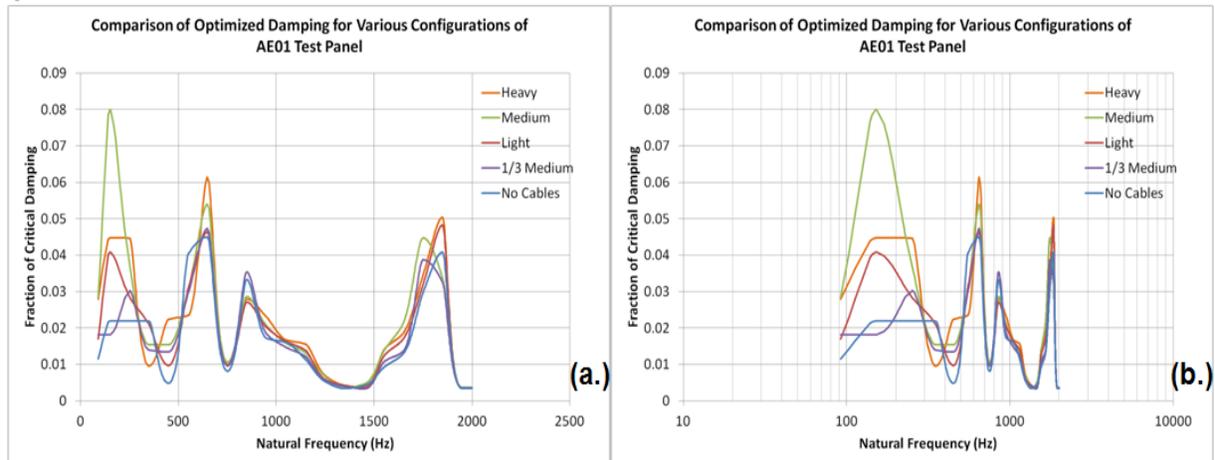

Figure 15. **Estimated System Damping Schedule for Various Cable Set Configurations. (a.) Linear Frequency Scale Liftoff. (b.) Logarithmic Frequency Scale Liftoff.**

The system damping estimates from each of the five cases have been overlaid for comparison in Figure 15. The estimate from the medium case may be suspect in the frequency range between 100 and 200 Hz. Otherwise the estimates seem to correspond to expectations from 80-1400 Hz.

The results from 1500 to 1800 Hz are very curious but consistent with a previous study [Smith April 2012]. In this set of optimizations the forcing function was applied using a 31 x 31 patch density which would correspond to a 2.6 inch center to center distance between adjacent patches[Smith June 2012]. This patch density was finer than that used in a previous assessment [Smith April 2012] and should have been adequate from 1500-1800 Hz. Spatial correlation indicates that the forcing function is diffuse above 200 Hz [Jones 2012]. Also spectral calculations produced similar pressure spectra at several different measurement locations which indicates a uniformity at measurement positioned near the vibrating panel test article [Jones 2012]. The FEM mesh density is adequate for the frequency range of interest, but the modeling approach using shell elements leaves out details like fillets and rib intersection bosses. This non intuitive result could be an indication that a solid element modeling approach may be required to improve simulation accuracy in the high frequency bands. The authors have not ruled out that the result represents a real physical phenomenon that would merit future study.

Figure 10 was used to reveal nonlinearities in damping at different excitation levels using the normalized response for two different cable configurations was used to illustrate the nonlinear trends. Figure 16 presents the range of differences the damping schedules can assume relative to the extremes of the excitation levels used in the test series for one of these configurations. The medium cable configuration was assessed at the highest and the lowest excitation levels to produce this illustration. The full power ascent and half power liftoff excitation levels were used in the comparison.
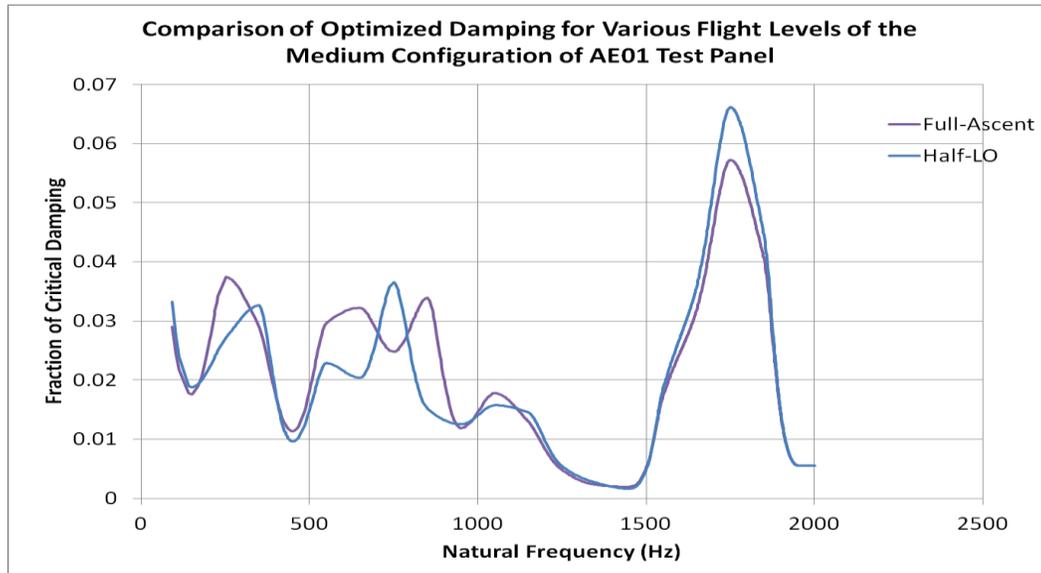
Figure 16. **Range of Damping Schedule Differences produced by Exciting the same Cable Configuration at Different Levels Medium Cable Configuration**

SUMMARY AND CONCLUSIONS

The results of this development test examine the effect of cable bundles on the vibroacoustic response of a flight like vehicle panel subject to acoustic excitation. The test data obtained will be used as a point of reference for different methods of reducing vibroacoustic response to levels more representative of flight like assemblies. In particular, the results discussed in this paper apply to vehicle panel system response which is lightly loaded by equipment, but includes supported cable bundles. The measured data has demonstrated that cables installed on a flight-like panel will lower measured response through an increase in damping. The critical damping fractions were estimated by conducting response optimization over several measured channels using the DampID tool for five different cable assemblies, but lightly loaded with other equipment. The damping values were presented in the format of system damping schedules. The damping schedules were further validated comparing calculated response at measured locations not included in the optimization set with favorable results.

The particular critical damping values were significant in certain frequency ranges. Most significant was the increase in damping with the addition of cable bundles. These results should serve to guide analysts in the choice of damping parameters when assessing similar vehicle structures.

The team also evaluated the same data to reveal damping nonlinearities at different excitation levels. The normalized response for two different cable configurations was used to illustrate the nonlinear trends. One of these configurations was used as an example to illustrate the range of differences the damping schedules could assume relative to the extremes of the excitation levels used in the test series.

This set of results is an extension of the work published earlier [Smith 2012]. The expanded results provide the opportunity to reach more definitive conclusions for the design problem of lightly mass loaded vehicle panel systems.
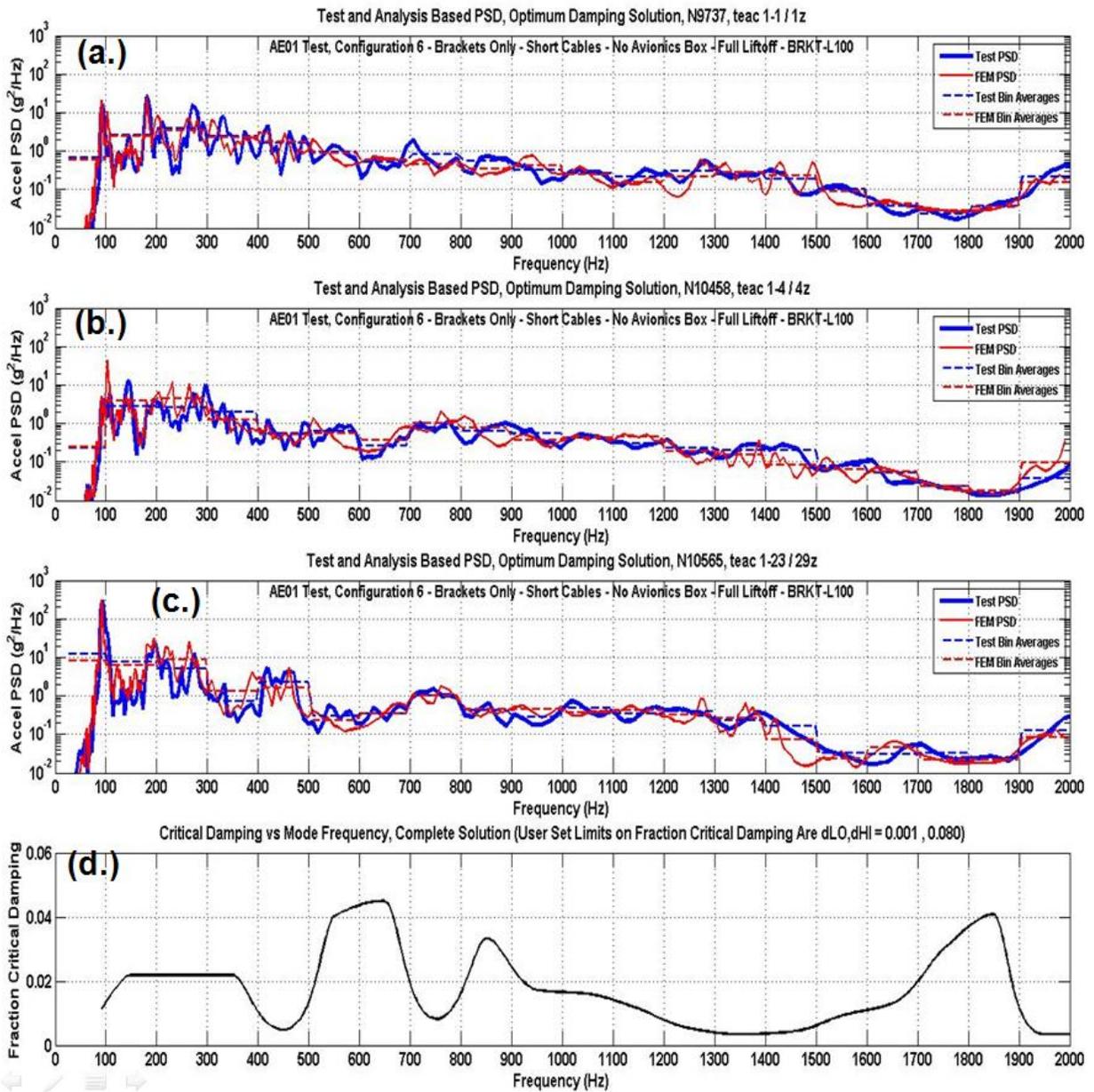

Figure 17. Response at Locations in the Optimization Set vs. Measured Response for "Brackets Only" (Case 6) Full Power Liftoff. (a.) Accelerometer 1z - 9737. (b.) Accelerometer 4z - 10458. (c.) Accelerometer 29z - 10565. (d.) System Damping Estimate.

Figure 18. **Response at Locations in the Optimization Set vs. Measured Response for "1/3 Medium" Cable Set (Case 18) Full Power Liftoff. (a.) Accelerometer 1z - 9737. (b.) Accelerometer 4z - 10458. (c.) Accelerometer 29z - 10565. (d.) System Damping Estimate.**

Figure 19. **Response at Locations in the Optimization Set vs. Measured Response for Medium Cable Set (Case 14) Full Power Liftoff. (a.) Accelerometer 1z - 9737. (b.) Accelerometer 4z - 10458. (c.) Accelerometer 29z - 10565. (d.) System Damping Estimate.**
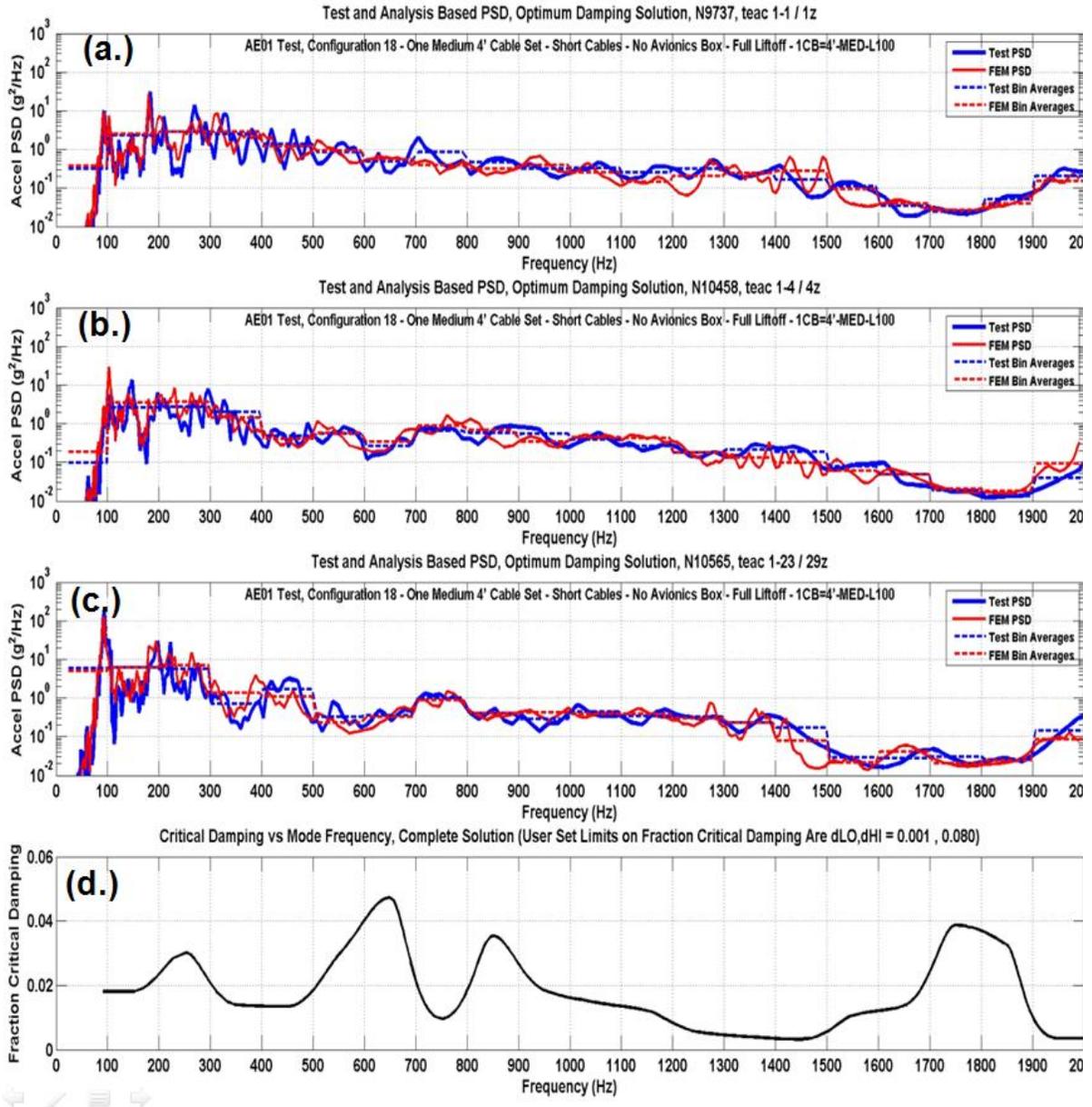
Figure 20. **Response at Locations in the Optimization Set vs. Measured Response for Heavy Cable Set (Case 22) Full Power Liftoff. (a.) Accelerometer 1z - 9737. (b.) Accelerometer 4z - 10458. (c.) Accelerometer 29z - 10565. (d.) System Damping Estimate.**
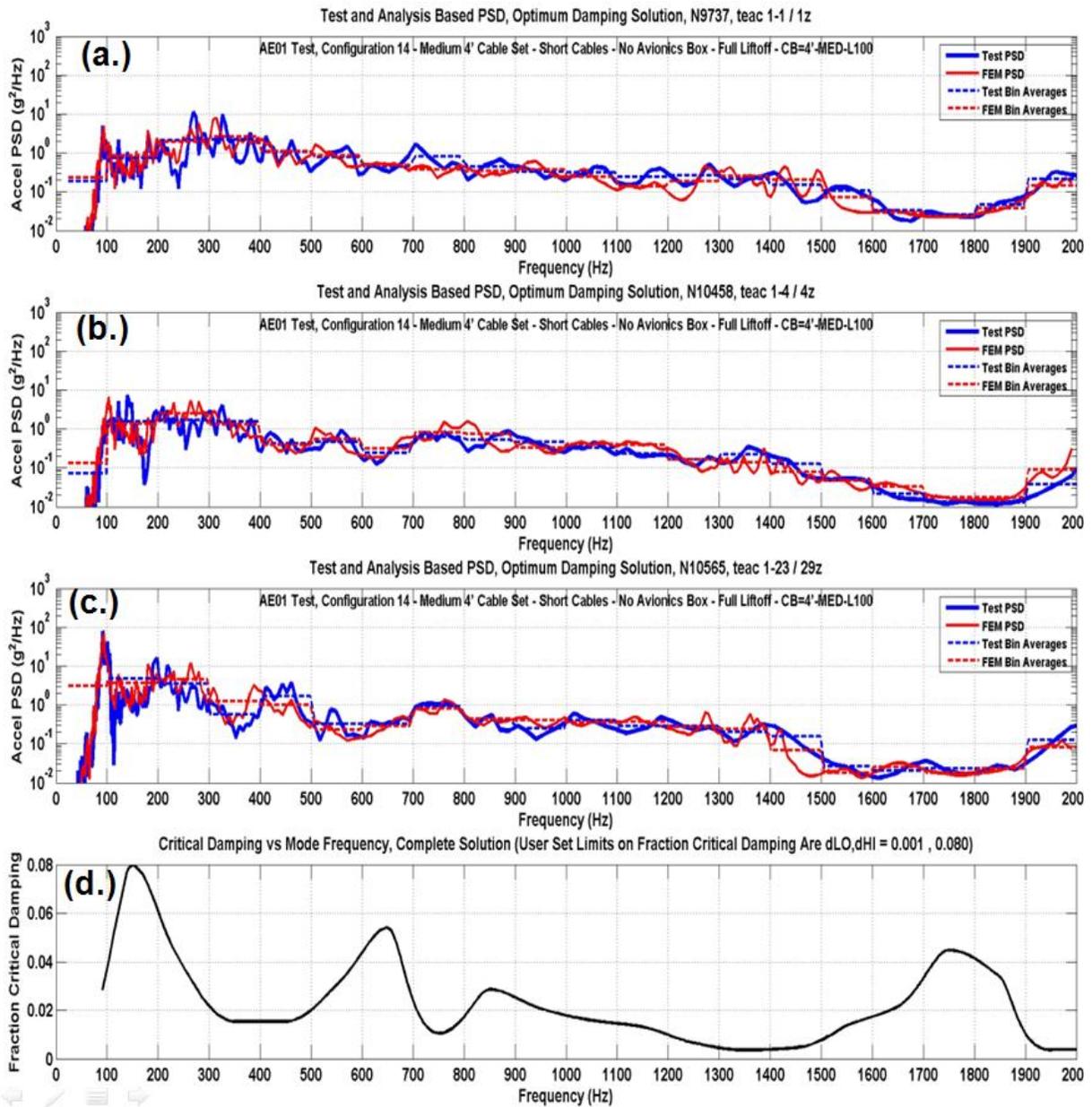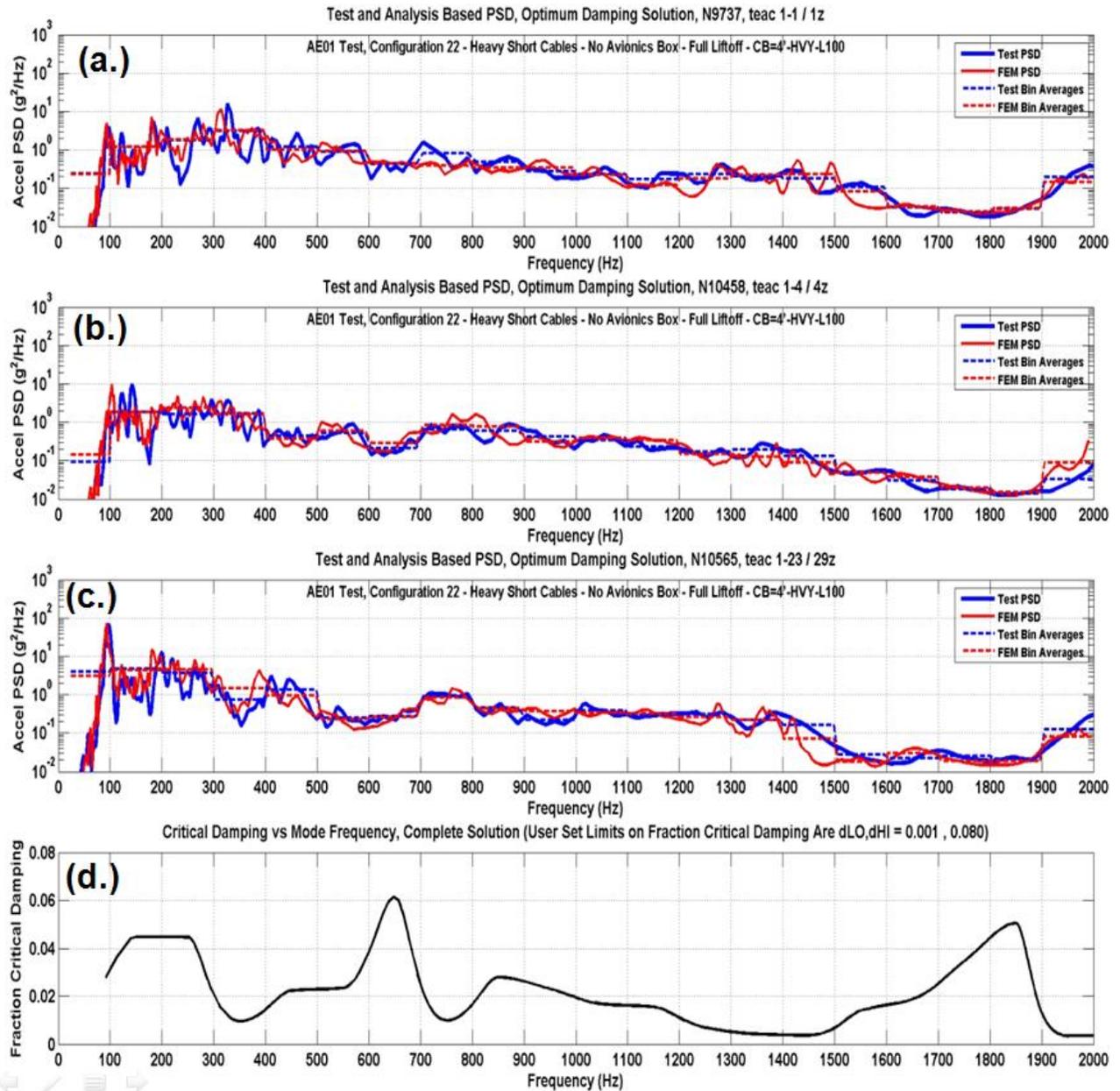
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BIOGRAPHIES

Andrew Smith – NASA – MS Mechanical Engineering, Tennessee Technological University – Currently providing NASA-Marshall Space Flight Center with Vehicle Vibroacoustics support and leadership. He is the chief investigator for a series of Acoustic Panel Response Tests at MSFC. These included observation of attenuation effects stemming from the integration of flight like avionics boxes, wire rope isolators, particle dampers, and damping due to cable harnesses. Future work will demonstrate environments for a shelf mounted avionics panel concept. Andrew's duties include processing and analysis of the measured data as well as developing vibration, internal acoustic, and shock environments for the SLS launch vehicle. Email – Andrew.m.smith-2@nasa.gov

Dr. Robert Ben Davis – NASA – is a propulsion structural dynamics and acoustics analyst at the NASA Marshall Space Flight Center. His professional and research experience spans the areas of acoustics, structural dynamics, nonlinear dynamics, elastic stability, and acoustic-structure interaction. As a former NASA Graduate Student Researchers Program (GSRP) fellow, Dr. Davis developed analytical techniques for assessing acoustic-structure interaction in rocket engines. These techniques have recently been applied in the development of the J-2X engine turbopumps and in the analysis of the Ares I First Stage thrust oscillation issue. He holds a B.S.E. in Mechanical Engineering from Duke University, an M.S. in Mechanical Engineering from Georgia Tech, a Diplome Gradué en Ingenerie from the École Nationale Supérieure d'Arts et Métiers (ENSAM) in France, and a Ph.D. in Mechanical Engineering from Duke University. Email – Robert.b.davis@nasa.gov

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Bruce LaVerde - ERC Inc. - BS Physics, Clarion University of PA - is an engineer supporting Marshall Space Flight Center through the ESTS contract. He has 23 years experience providing response estimates for random vibration and has conducted acoustic, pyroshock, static fire propulsion, and flight tests to establish environments. He is an experienced finite element analyst who has also used statistical energy analysis to produce analytical response estimates of the same. He has published results using both SEA and Hybrid FEA/SEA. He is interested in the differences in response resulting from different phase correlated pressure fields. Email - bruce.t.laverde@nasa.gov

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