A New Approach in Force-Limited Vibration Testing of Flight Hardware

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ABSTRACT

The force-limited vibration test approaches discussed in NASA-7004C were developed to reduce overtesting associated with base shake vibration tests of aerospace hardware where the interface responses are excited coherently. This handbook outlines several different methods of specifying the force limits. The rationale for force limiting is based on the disparity between the impedances of typical aerospace mounting structures and the large impedances of vibration test shakers when the interfaces in general are coherently excited. Among these approaches, the semi-empirical method is presently the most widely used method to derive the force limits. The inclusion of the incoherent excitation of the aerospace structures at mounting interfaces has not been accounted for in the past and provides the basis for more realistic force limits for qualifying the hardware using shaker testing. In this paper current methods for defining the force limiting specifications discussed in the NASA handbook are reviewed using data from a series of acoustic and vibration tests. A new approach based on considering the incoherent excitation of the structural mounting interfaces using acoustic test data is also discussed. It is believed that the new approach provides much more realistic force limits that may further remove conservatism inherent in shaker vibration testing not accounted for by methods discussed in the NASA handbook. A discussion on using FEM/BEM analysis to obtain realistic force limits for flight hardware is provided.

KEY WORDS: Random vibration, Flight hardware qualification, Force limiting responses, finite element analysis, boundary element analysis

INTRODUCTION

The force limiting technique applied in random vibration qualification tests of flight hardware is discussed in great detail in NASA-HDBK-7004C[1]. This handbook outlines several different methods to obtain the force limit specifications. The rationale for applying force limiting discussed in the handbook is based on the disparity between the impedances of typical aerospace mounting structures and the large impedances of vibration test shakers. The mechanical impedance of the flight mounting structure is defined as the ratio of force to acceleration. In all the methods discussed in the handbook, which contains an extensive cited literature, the information about the source structural apparent mass, in general, is not readily available when a component undergoes random vibration shaker testing.

The depth of a notch in a force limited vibration test, and any other response-limited test, depends on two things: 1) the force, or response limit, and 2) the damping of the resonance being

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limited in the test[2-5]. When the flight mounting structure and test item impedances are comparable, the force or response limit is relatively insensitive to the test item damping, but the damping determines how much the force or response in the test would exceed the limit, and thus the depth of the notch that results from limiting. Therefore, lightly damped resonances are notched much deeper than heavily damped ones.

Recently, analytical and experimental studies were performed[6-8] in an effort to gain more information about the ranges of C^2 that may be used for most aerospace structures. The C^2 is used in semi-empirical formulation (see Equ. (1)) discussed in the handbook[1] and in references 2-5. A few flight measurements related to force limiting have been made in the last several years. Even though for these cases force gages were added between the components and support structure interfaces, the detailed information related to the interface accelerations and forces was not published.[9-10] Since the detailed flight mounting structure data are not available in the above references, in this paper a few electronic boxes attached to two panels subjected to ground test acoustic excitations are examined and considered as being representative of flight data. The measured boxes interface acceleration responses were used to derive acceleration inputs for shaker vibration tests of the boxes. These data are used to re-examine the force limiting specifications. The data gathered from the acoustic tests and shaker testing of the two electronic boxes are used to evaluate existing methods outlined in the handbook and the recent literature[1-5]. In the following section the existing method of estimating the force limiting specification as outlined in the NASA handbook[1] are reviewed with examples given to provide comparison between each method and the force responses measured from acoustic test results.

EXPERIMENTAL RESULTS

Several acoustic tests with various hardware configurations were performed to examine the force spectrums required to realistically limit the electronic boxes during base shake testing. The test articles investigated in this study consisted of a) two support structures, a freely suspended aluminum (Al) panel of dimensions 37.5”x41”x0.25” and of mass 38.4 lb (exclusive of cables), the 155 lb double layer development test model (DTM) Rover top deck of the Mars Science Laboratory (MSL) fixed at its edges to a frame, and b) two components, structurally flight-like electronic boxes A (17.4 lb) and B (45 lb). Three types of tests were conducted: 1) Acoustic tests, 2) tap tests using a calibrated hammer, and 3) vibration tests of the two components, boxes A and B, on a shake table. Force gages were installed at every mounting interface between the boxes and the supporting structures, whether mounted on the panels or on the shake table vibration fixture. In every test, whether the panels were loaded with the boxes or unloaded, accelerometers were mounted near the box mounting positions and response accelerometers were mounted on the boxes themselves. Figure 1 shows the Al panel and boxes A and B, and the MSL DTM Rover top deck, again with boxes A and B. Figure 2 show images of the box A and box B mounted on the shaker head expander.

Eight control microphones were suspended from the JPL reverberant acoustic chamber ceiling to control the sound field to an overall sound pressure level (OASPL) of 145 dB. In addition to using eight control microphones, an array of 36 monitor microphones shown in Figure 1 were positioned very close (~6 inches) to the panels. The array consisted of a rubber net attached to a frame where microphones were carefully tie downed to the net in an approximately 8x8 inch grid pattern. The microphones were held in place using small neoprene ties. This configuration

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decoupled the microphones from any mechanical sources of vibration, eliminating electrical noise that could have resulted from such coupling. The purpose of the array was to map out the sound field close to the surface of the panels discussed in this report. The reverberant chamber sound pressure levels from 8 control microphones and the microphone array are shown in Figure 3a-b. The electronic boxes were instrumented with a series of triaxial accelerometers. Force gages were also used to measure the force responses at the electronic box interface to the panels as shown in Figure 1.

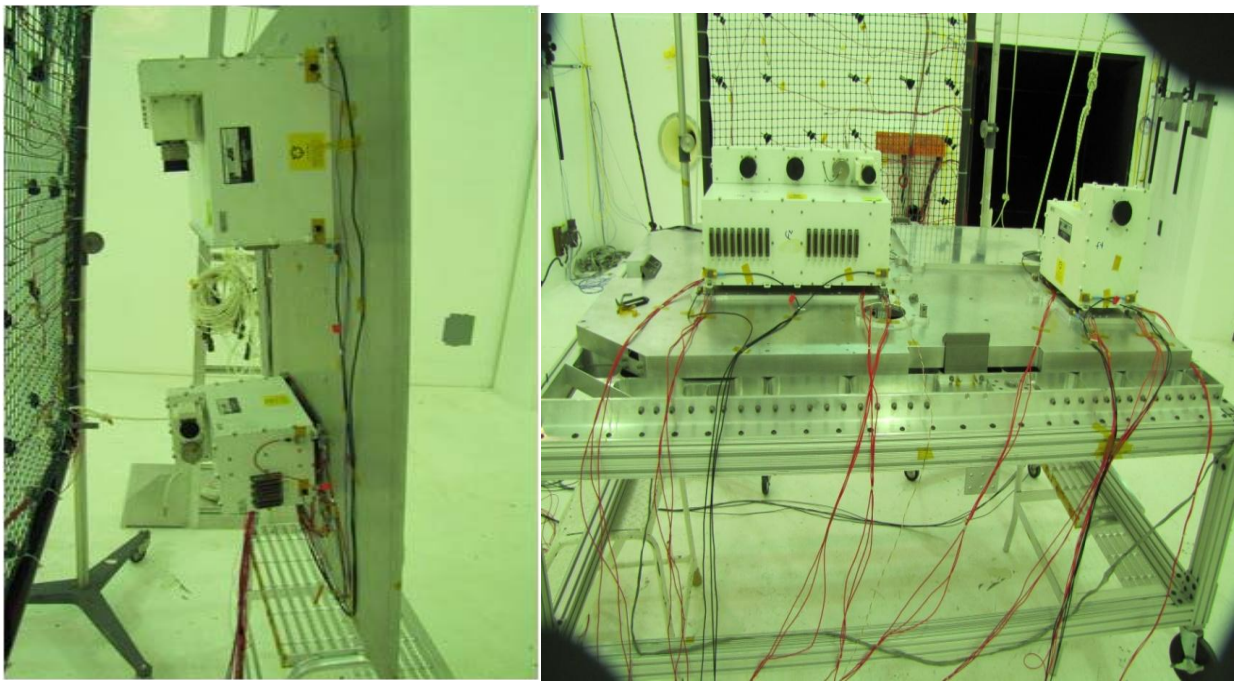

Figure 1: An aluminum panel (left hand photo) and the DTM Rover top deck (right hand photo) with two electronic boxes placed inside the JPL acoustic reverberant chamber. Force gages installed at the box/panel interfaces.

Figure 2: Electronic boxes A (left hand photo) and B (right hand photo) mounted on shaker and tested in the vertical direction.

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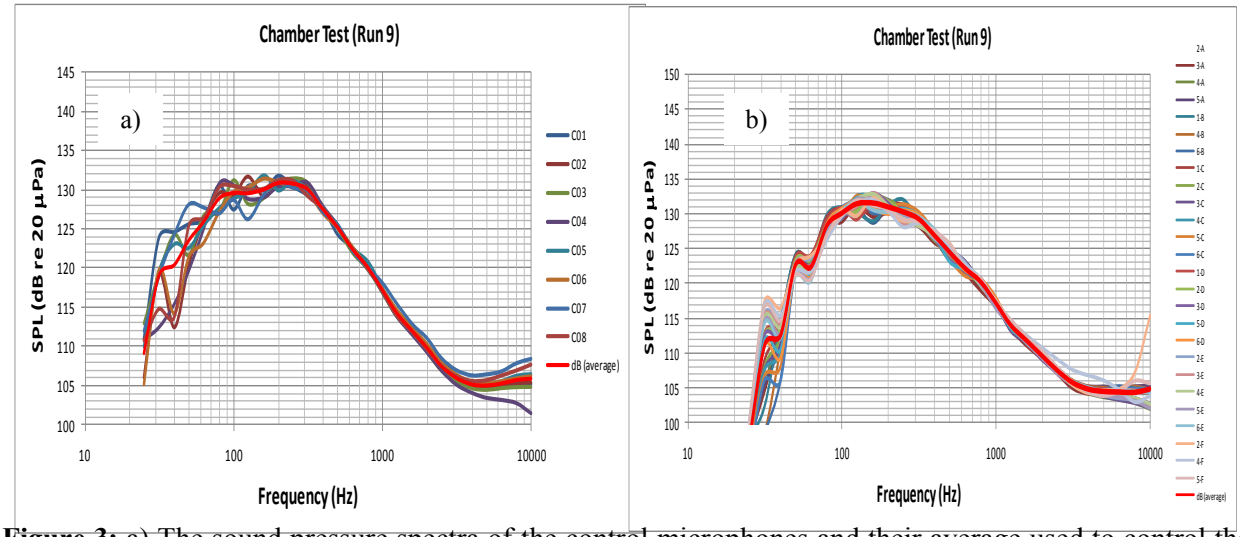

Figure 3: a) The sound pressure spectra of the control microphones and their average used to control the test, and b) the sound pressure spectra of the 36 microphones from the array and their average.

The base shake random vibration tests of the electronic boxes were performed to the average input acceleration spectrum for each box obtained from the acoustic test data. Figure 4 shows the

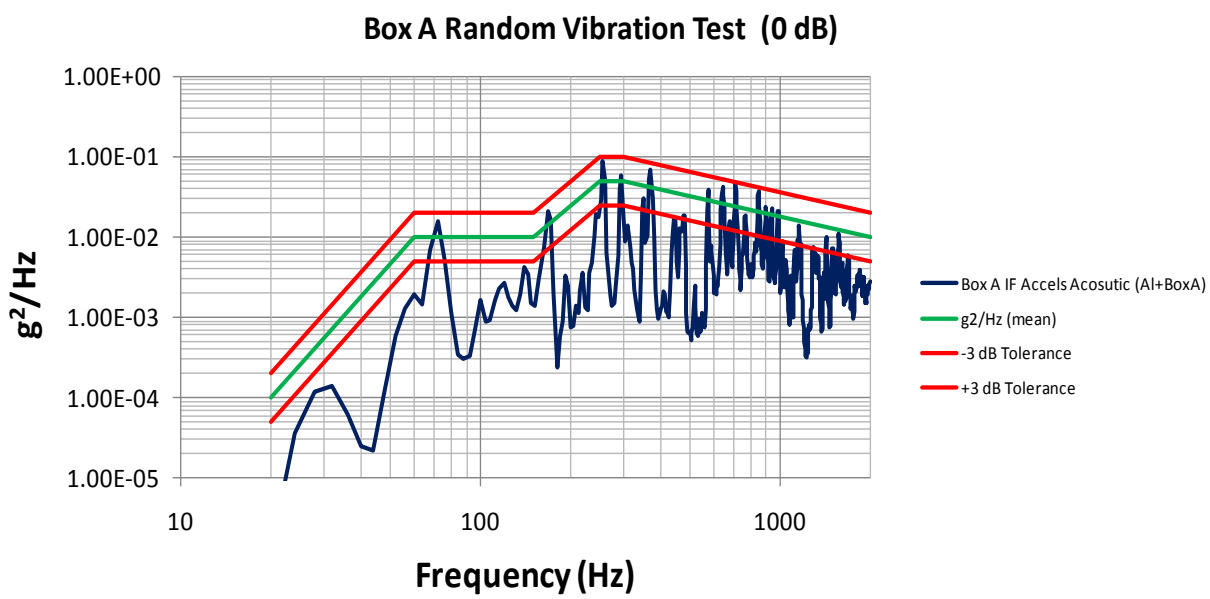

Figure 4: Input acceleration to box A shaker test. The input acceleration spectrum was derived by enveloping the average interfaces acceleration responses obtained from Al panel and Box A acoustic test.

RESULTS AND DISCUSSIONS

In the following sections a few existing methods for deriving the force limiting specifications are discussed using the results obtained from the acoustic and shaker testing of the electronics boxes. In addition to the methods outlined in the NASA handbook new approaches for deriving the force limiting specifications are proposed and discussed in some detail.

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1. Semi-Empirical Method

The semi-empirical method most commonly used in force limited vibration tests is reviewed in this section. The equations for relating the force limit spectrum to the input acceleration spectrum are given by[1]

$$S_{ff}(f) = C^2 M_0^2 S_{aa}(f), \qquad f < f_b \tag{1}$$

$$S_{ff}(f) = C^2 M_0^2 \left(\frac{f_b}{f}\right)^{2n} S_{aa}(f), \qquad f \geq f_b \tag{2}$$

where $S_{ff}(f)$ and $S_{aa}(f)$ are the force and acceleration spectral densities, respectively, C is a dimensionless constant (frequency independent), which depends on the configuration, M_o is the total mass of the test article, f is frequency, f_b is a break frequency, and n is a positive constant.

The purpose of the semi-empirical method is to provide a simple framework for deriving the force limiting specifications with its validity dependent on the accuracy of C, the break frequency f_b, and the exponent n. These parameters depend on the flight mounting structure and test article, and not on the shaker system and mounting configuration.

In these equations the user needs to define the constant C, the break frequency f_b, and the exponent n. The exponent n in Equ. (2) is often equal to unity or can be estimated using a low-level, random vibration shaker run without applying the force limiting to the control system. The break frequency is typically associated with the primary mode of the test article and can be defined using the measured apparent mass without applying the force limiting to the shaker.

The random vibration environment derived from the acoustic test shown in Figure 4 is used to base shake boxes A and B separately. Figures 5 and 6 show the power spectral densities of the average accelerations and summed forces, respectively, measured in the force limited vibration test of box A. The force limit of approximately 100 lb^2/Hz for the vibration test was derived using the semi-empirical Eq. (1) with an input acceleration spectrum S_{AA} of 0.05 g^2/Hz as shown in Figure 5 and box A weight of 17.9 lbs, and a C^2 of 6. The second mode was chosen to be the break frequency, f_b, and above this frequency the force spectrum slope of 6 dB/Oct was selected to derive the force limiting spectrum using the Equ. (1) and (2). These parameters are typical for force limited vibration testing of most flight hardware. This approach resulted in an approximately 18 dB notch at about 410 Hz, the fundamental resonance of the box A in it's test configuration. The maximum force PSD of 10 lb^2/Hz measured in the acoustic test is several orders of magnitude less than the maximum un-notched force response of about 8,000 lb^2/Hz at approximately 410 Hz, the box A frequency as depicted in Figure 6. Figures 7 and 8 are box B average acceleration and summed interface force, respectively, measured in the force limited random vibration test. The force limit of approximately 200 lb^2/Hz for shaker vibration test was derived using the semi-empirical Equ. (1) with an input acceleration spectrum S_{AA} of 0.05 g^2/Hz as shown in Figure 7 and box B weight of 45 lbs, with a C^2 of 2. The break frequency, f_b of 240 Hz and above this the force spectrum slope of 6 dB/Oct were selected to derive the force limiting spectrum using Equ. (1) and (2). The comparisons between the actual force spectra obtained from the acoustic test of box B at the interfaces to the Al panel and the unlimited force spectrum obtained from random vibration test of the box B using the shaker is also shown in Figure 8.

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Comparing the force spectral densities of the base shake test and flight-like acoustic test suggests that the uncorrelated interface responses are contributing to the significant differences between the measured force spectra obtained from acoustic tests and the shaker test, where the input for shaker test was obtained from the acoustic test for these cases. This effect will be further examined by considering a few more acoustic examples. The force limited vibration testing performed over the past 20+ years has been based on the differences between the impedances of mounting structures and the large impedances of vibration test shakers and does not account for interface uncorrelated responses of the components.

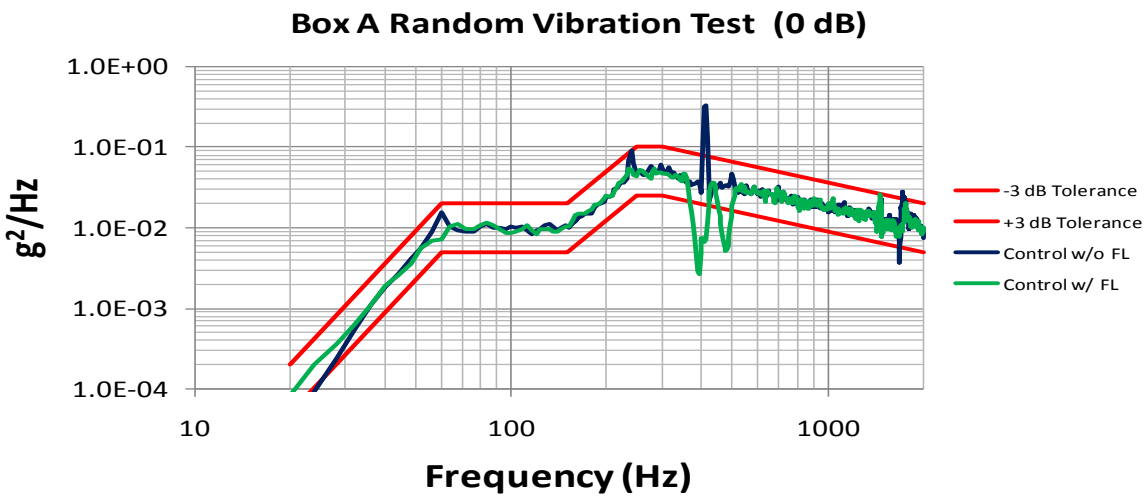

Figure 5: Input acceleration to box A shaker test w/ and w/o force limit at 0 dB. The input spectrum w/o force limiting is scaled to full level from -18 dB.

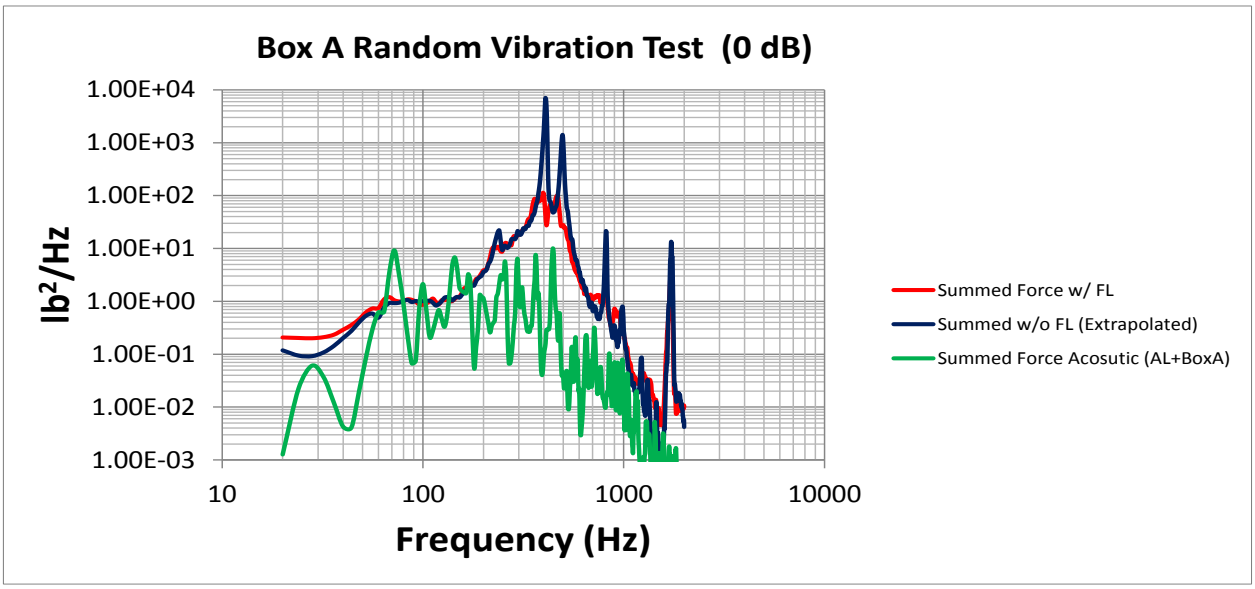

Figure 6: The summed box A interface force with and without force limiting at 0 dB. The force spectrum without force limit was scaled to 0 dB from -18 dB. A C^2 of 6 used for this case. Summed forces obtained from box A interfaces mounted on the AL panel is also shown.

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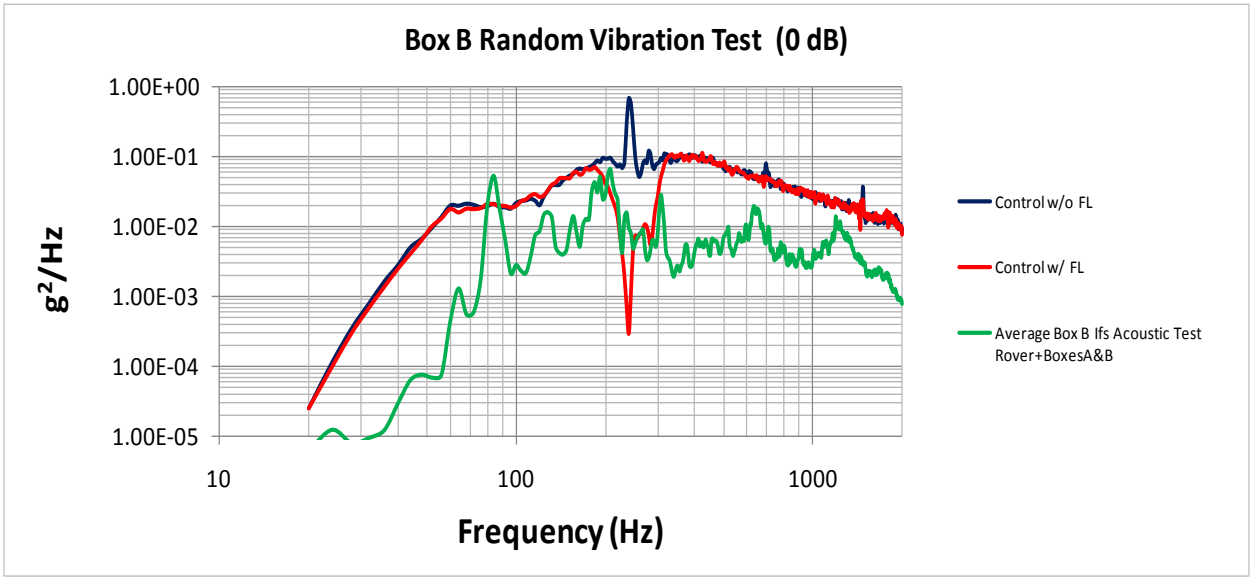

Figure 7: Input acceleration to box B shaker test w/ and w/o force limit at 0 dB. The input spectrum w/o force limiting is scaled to full level from -18 dB. The average box B interface acceleration responses form acoustic is also shown in this figure.

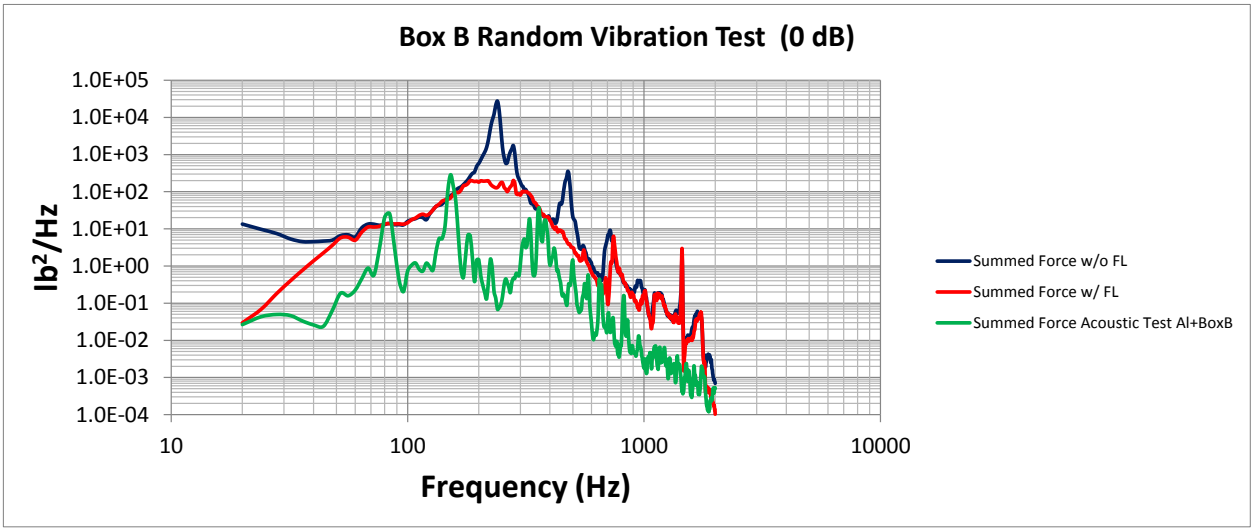

Figure 8: The summed interface box B interface force with and without force limiting at 0 dB. The force spectrum without force limit was scaled to 0 dB from -18 dB. A C^2 of 2 was used for this case. Summed forces obtained from box B interfaces mounted to Al panel is also shown.

Similar results are shown in Figures 9 and 10 for boxes A and B mounted on the MSL Rover top deck panel. An image of the MSL Rover top deck with boxes is shown in Figure 1. The summed forces from the acoustic test with boxes performed using the Rover Deck are also depicted in these figures. As was the case with the Al panel with boxes A and B, the force spectral responses obtained from acoustic tests, which is treated here as flight-like data, are significantly different than the force limiting obtained using the semi-empirical relationship. For semi-empirical relationship in Equ. (1) C^2 of 2 is used.

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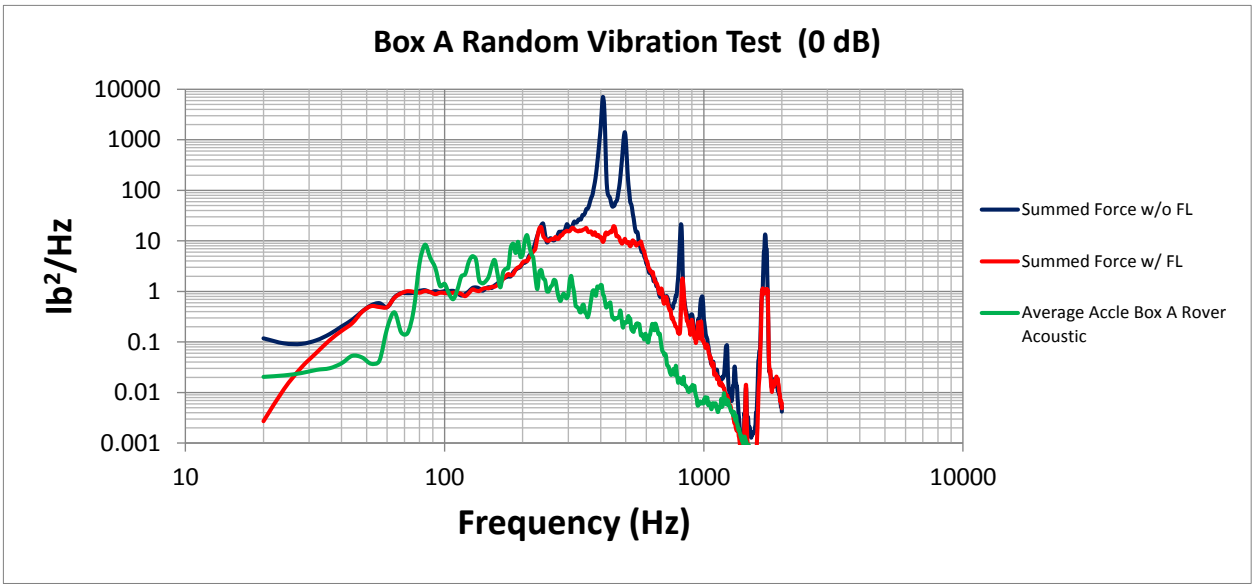

Figure 9: The summed box A interface force with and without force limiting is obtained from shaker test at 0 dB. The force spectrum without force limit was scaled to 0 dB from -18 dB. A C^2 of 6 was used for this case. Summed forces obtained from acoustic test for box A interfaces mounted on the Rover top deck also shown.

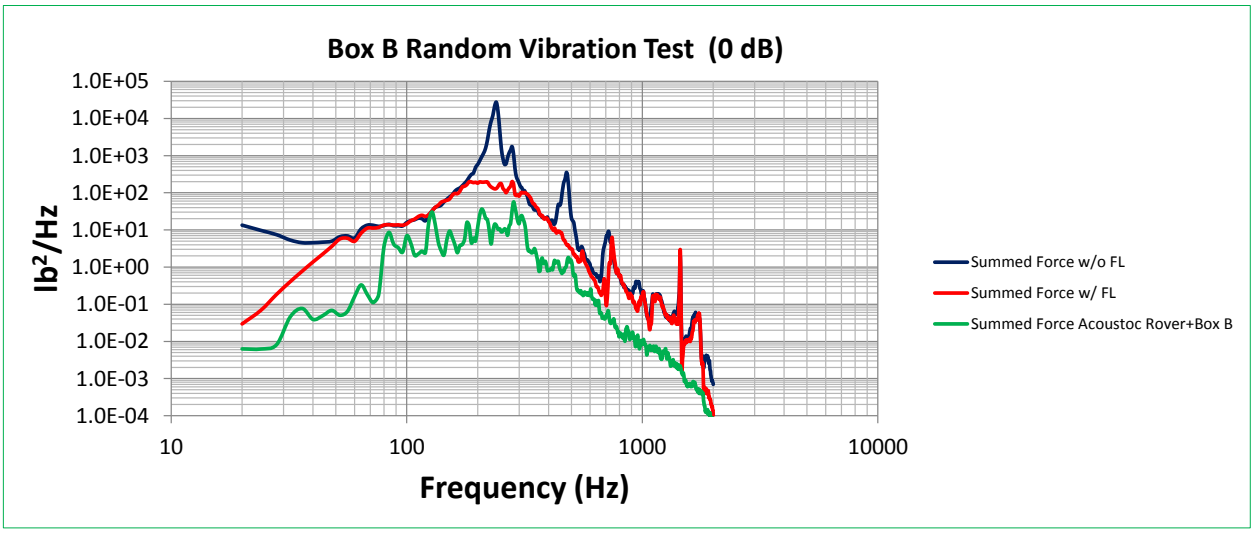

Figure 10: The summed box B interface force with and without force limiting is obtained from shaker test at 0 dB. The force spectrum without force limit was scaled to 0 dB from -18 dB. A C^2 of 2 was used for this case. Summed forces obtained from acoustic test for box B interfaces mounted on the Rover top deck also shown.

2. Methods of Estimating C^2

The factor C can be determined using four different methods outlined in the NASA handbook. The methods discussed in the handbook are: 1) a simple two-degree-of-freedom (TDOF) model, 2) a complex TDOF model, 3) Norton's and Thevenin's equivalent circuit theorems, and 4) finite element method (FEM) analysis of the flight configuration.

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The simple TDOF system method of deriving force limits is described in References 1-5. This method is based on the assumptions that the coupled vibration modes of the source are in the same frequency regime as the vibration modes of the load and the force is the result of the in-axis acceleration and not due to cross-axis or rotational motion. The simple model consists of a source vibration coupled with the component vibration. Figure 11 taken from reference 1 depicts the ratio of the component interface force spectral density and the acceleration spectral density normalized with the mass of the component square as a function of the ratio of the component mass (M_2) to source mass (M_1) for three different quality factors computed using the TDOF model. For the cases with larger mass ratios the effect of damping becomes insignificant in estimating the amount of the notch. To be able to estimate the notch, or the knock down factor, the source and component masses must be determined from FEM analyses or system test or flight measurements as a function of frequency.

The complex TDFS method of deriving force limits is also described in reference 1. This method requires both the modal and the residual masses of the source and component. In reference 2-3 the normalized ratio of interface force spectral density to input acceleration spectral density for a complex TDFS with Q=20 is tabulated and provided as a nominal value for most applications in the aerospace industry. The Complex TDOF System method is particularly useful in conjunction with FEM analyses, where the effective modal masses can be computed. The normalized force spectra from simple TDOF System and complex TDFS methods may be interpreted as C^2 in the Semi-empirical of Equ. (1).

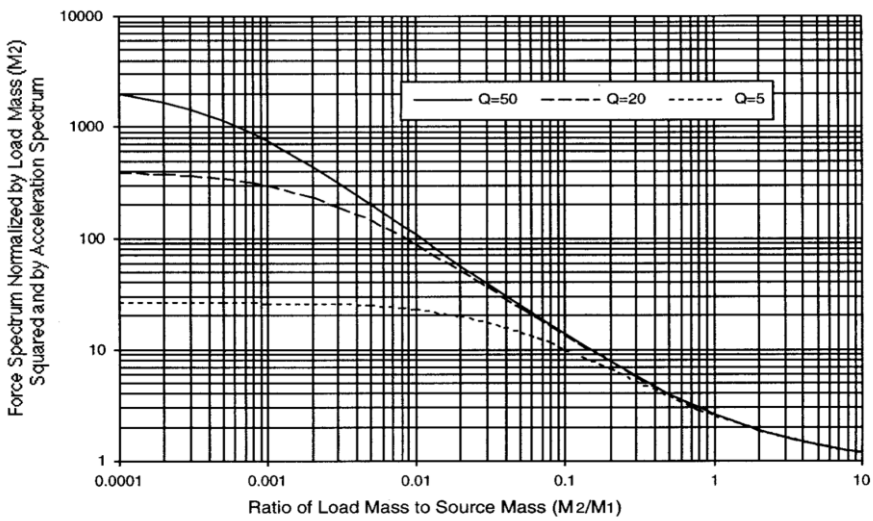

Figure 11: Normalized Force Specification from Simple TDOF System taken from Reference 1.

The apparent mass method is derived by approximating the Norton's and Thevenin's equivalent circuit theorems discussed in references 1-3, where the free acceleration and blocked forces are replaced by the enveloped input acceleration and the force specifications. The ratio of the acceleration spectral density acting at the interface of a coupled source and component to the free acceleration of the source is given by Scharton[2-3],

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$$\left|\frac{ASD_{loaded}}{ASD_{unloaded}}\right| = \left|\frac{M_s(f)}{M_s(f)+M_c(f)}\right|^2, \quad (3)$$

where ASD_{loaded} and $ASD_{unloaded}$ are the acceleration power spectral densities at the component mounting positions on the loaded and unloaded source structure, respectively, and M_s and M_c are the apparent masses of the support structure and the component, respectively.

To be able to estimate C^2 using the simple SDOF from the results shown in Figure 11 requires estimating the source apparent masses. The apparent masses of the bare Al and Rover top deck panels were obtained using tap tests. The tap tests were performed by exciting the panels at the boxes attachment interfaces with and without boxes mounted and measuring the resulting vibration responses at the boxes interfaces. The panels were excited at each of the box interfaces using a calibrated modal force hammer. The acceleration responses were measured with accelerometers positioned very close to the boxes interfaces. The accelerance frequency response function, i.e., the ratio of the response acceleration to the input force was computed at the boxes interfacing the panels. For the unloaded panels, the force data from the instrumented hammer were utilized to compute the accelerances. The transfer function was generated from the average of 5 repeated hits. The analysis was performed with 4-Hz resolution. The accelerances are defined as the acceleration at point i when a force is applied at point j, and the forces at the other interface points are assumed to be zero. The apparent masses, which are defined as the force at point i when an acceleration is applied at point j, and the accelerations at the other interface points are zero, were obtained using MatLab to invert the complex accelerance matrices. The accelerance matrices consisted of the transfer function for each term (i.e. a_i/F_j, where a_i and F_j are respectively, the acceleration at i due to a force at j). Each term in the accelerance matrix included the real and imaginary components. The apparent mass matrix, which was obtained by inverting the accelerance matrix, also included the real and imaginary terms.

Figures 12 and 13 show the apparent mass of the mass bare AL panel obtained using from the method described above for boxes A and B, respectively. In all these figures the diagonal terms and the sum of the diagonal terms of the apparent mass matrices are shown. The bare panel apparent masses shown in Figure 12 and 13 are useful to estimate the C^2 in Equ. (1). Using the apparent mass measured data for boxes A and B, the source apparent mass for the Al panel at the boxes resonance frequencies are estiamted to be 30 lbs and 8 lbs, respectively. This is the source mass (M_2) of the SDOF plots shown in Figure 11. The ratios of the boxes masses to the source masses provide a C^2 of 4 and 1.5, respectively, using the Q=20 curve from Figure 11. These values are close to the C^2 values used for force limiting during the random vibration testing of these boxes.

The impedance of the loaded and unloaded panels may also be determined using FEM models. The approach is to analytically excite the panels at the boxes attachment interfaces and compute the resulting vibration responses at the boxes interfaces. In the case of Al panel loaded with boxes A and B, all four interface points for each box were constrained using RB2 elements in the FE model. The base shake analysis of the bare Al panel, panel with box A, panel with box B, and panel with boxes A and B were performed by applying constant acceleration spectral density

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at each interface. The input acceleration PSD with a constant energy (white noise) was applied at each of the boxes interfaces and the resulting acceleration and force responses were computed at the boxes interfaces. The frequency response function relating the predicted force response to the input acceleration was measured at all the interfaces. Figure 14 shows the apparent mass of the bare Al panel estimated at box A interfaces. The total apparent mass is obtained by summing all force interface impedances (the diagonal terms in the apparent mass are included). The apparent mass changes as ~*1/f*, which is the case of the infinite extended panel. Figure 15 shows the apparent mass of the Al panel loaded with box A, estimated using FE analysis. Unlike the unloaded panel, who's apparent mass behaves like an infinite extended panel (i.e. it's spectral characteristics behaves as *1/f*), the loaded panels' apparent mass spectral characteristics behave as $1/f^2$ above the coupled frequency for the panels and boxes.

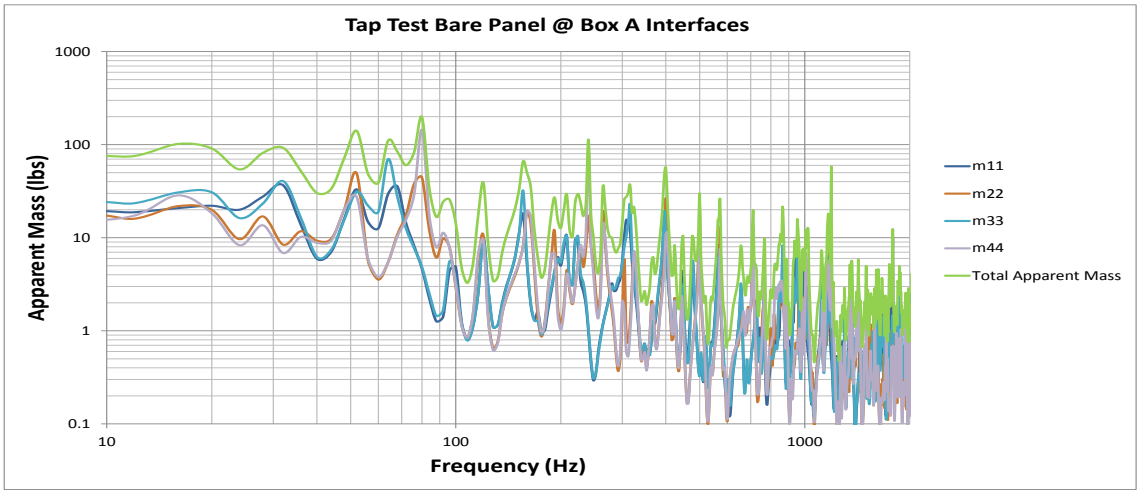

Figure 12: Apparent mass of the bare Al panel at box A interfaces obtained from tap test. The apparent masses are computed by inverting the complex accelerance matrix. Only diagonal terms and sum of the diagonal apparent masses are shown in this figure.

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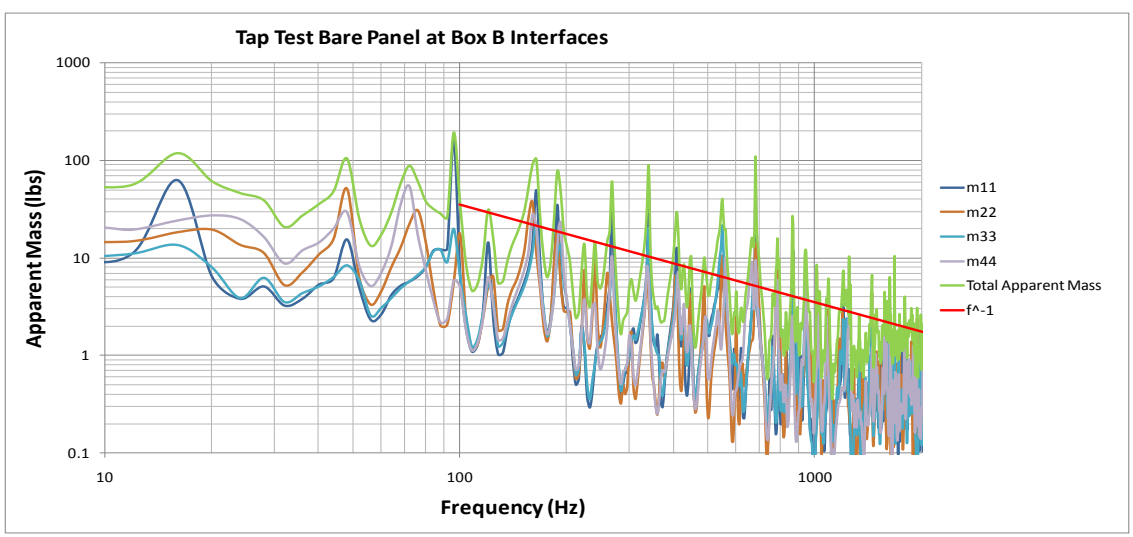

Figure 13: Apparent mass of the bare Al panel at box B interfaces obtained from tap test. The apparent masses are computed by inverting the complex accelerance matrix. Only diagonal terms and sum of the diagonal apparent masses are shown in this figure.

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Similar C^2 estimate can be made using the FEM results shown in Figures 14 and 15. Based on these results the estimated apparent mass of the bare panel at box A interfaces at the box's resonance is approximately equal to 10. This provides a C^2 value of close to 2 using Figure 11.

The finite element analysis of the loaded and unloaded panel at boxes interfaces can be used to predict the force limiting specification. The load apparent mass, M_l, is equal to the ratio of the interface force S_{ff} to the interface acceleration S_{aa}, To obtain the force limiting specification, Equ. (3) may be rewritten as[1]:

$$\left|\frac{S_{ff}(f)}{S_{aa}(f)}\right| = \left|\frac{M_s(f) x M_l(f)}{(M_s(f)+M_l(f))}\right|^2, \tag{4}$$

Equation 4 may be used to derive a force limit if the free acceleration of the source is known. This method of prediction, where the component interface cross spectral densities are included, is currently being considered.

The differences in the C^2 estimates discussed above range between ~ 2 to 6, which provides up to 6 dB differences in the notched frequency depending which method is selected in deriving the force limiting specification. However, the force responses measured at boxes interfaces from the acoustic tests conducted using Al and Rover top deck panels with boxes A and B mounted on them are much different than the estimated force responses using the semi-empirical relationship of Equ. (1) as shown in Figures 6 to 10. The major physical reason for these differences is that most components' interface responses mounted on flight-like structures are excited in a non-correlated fashion (i.e. the components in most aerospace applications do go through "rock-n-roll" excitations). The impact of this phenomena on excessive conservatism in current force limit derivation approaches is the major thrust of this paper and efforts are underway to come up with recommended practices accounting for the uncorrelated component structural responses in force limiting specifications. Current force limiting methods reviewed in this paper and discussed in great detail in the NASA handbook are based on the assumption that the interface responses are correlated and there are no component cross interface responses.

Estimating C^2 values that account only for the mismatch in impedances between the shaker and flight mounting configuration of the component does not adequately approximate the physics of the flight configuration and still provides excessive conservatism in random vibration testing. In the following section new approaches to account for the uncorrelated component responses that lead to new force limiting specifications are discussed.

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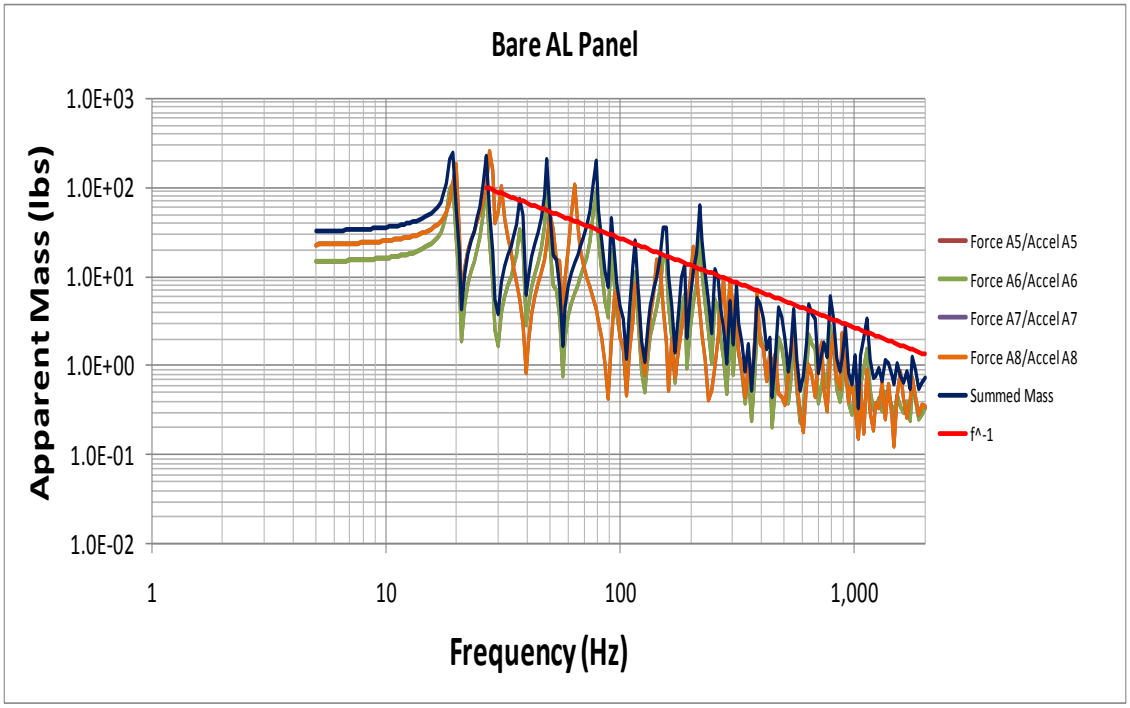

Figure 14: Apparent mass of the bare Al panel at box A interfaces obtained from FEM analysis by applying constant spectral input (white noise) on the panel at box interfaces. The apparent masses are obtained by computing the impedance at each interface, with the total mass being the sum of the four impedances**.**

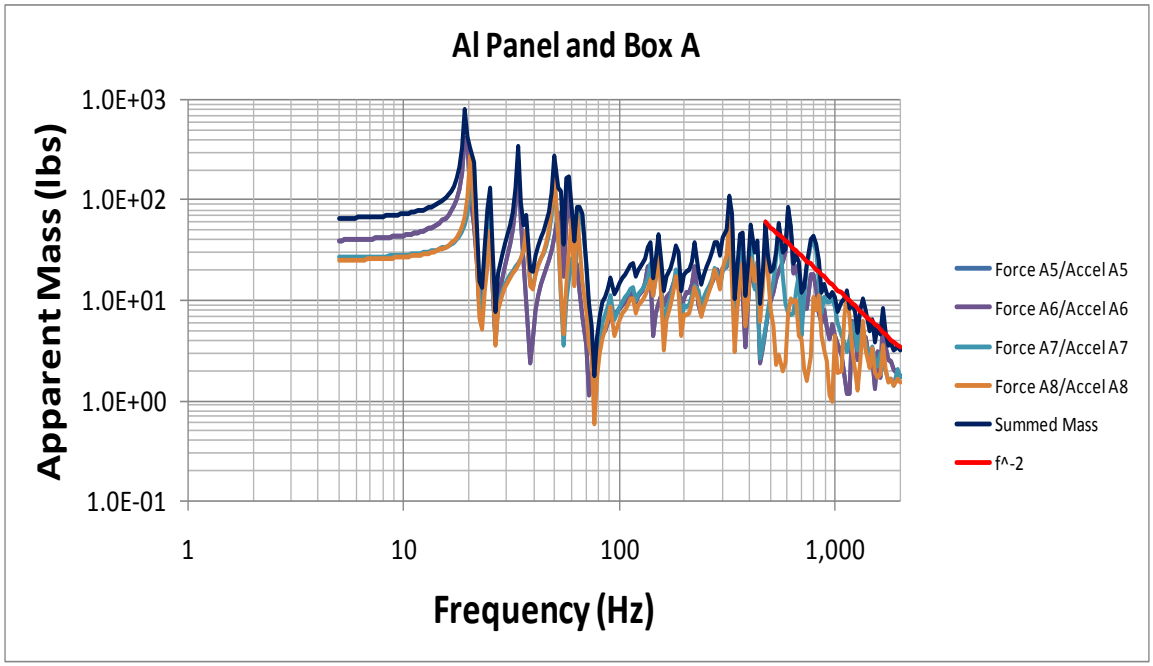

Figure 15: Apparent mass of the Al panel with box A at box A interfaces obtained from FEM analysis by base shaking the panel and the box. The apparent masses are obtained by computing the impedance at each interface, with the total mass being the sum of the four impedances**.**

3. New Force Limiting Specification Approaches

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The semi-empirical formulations in Equ. (1) and (2) are most commonly used to derive the force limit specifications, where an estimated C^2 value is required. This estimate is based on the point impedances of components coupled with the source structure. The data obtained from acoustics tests performed on Al and Rover top deck panels coupled with two boxes mounted on them and discussed above demonstrate that the current methods of estimating force limit specifications are limited to taking into account the mismatch in the impedances between the shaker and flight mounting configurations of the components. The inclusion of the uncorrelated component interface forces provides force limit specifications much different than the conventional estimate of the force limit spectrum. This can be illustrated in Figures 16 and 17. Figure 16 is the same as Figure 6 for the case of Al panel and Box A, except the force spectrum obtained from acoustic test is enveloped to derive the force limiting specification. Clearly the differences between the force limited spectral densities derived for Box A shaker test and Box A acoustic test are significant. Figure 17 is the same as Figure 9 except the envelop of the interface summed force of Box A mounted to the Rover top deck panel obtained from acoustic tests is included. These are just a couple of examples of the enveloped force spectrum obtained from acoustic test (assumed to be flight-like conditions) illustrating the differences between the conventional force limiting specification and the force response of the components in flight configuration. The differences in the force limiting specification by more than 10 dB across broader frequency for these cases and in case of the box A mounted on Rover top deck with the break frequency much lower than the one chosen for shaker testing suggest that the existing force limiting specification derivation requires major improvements.

These observations compelled us to re-examine the force limiting specifications. An extensive research and development efforts is underway to improve the force limited random vibration testing. A few different approaches are currently being considered. First is to use FE models of the source and component structures in flight configurations and to predict the component interface uncorrelated responses. The FE models are also being used to predict multi-point impedances of the components. The multi-point impedance estimates include the cross-spectral predictions of the force and acceleration responses. Second, applying boundary element method (BEM) to predict the component interfaces forces excited by the acoustic field. Figure 19 shows an example of the box A and Al panel BEM predicted force responses compared with the measured data. Envelop of the predicted forces provide a more accurate force limiting specifications that can be used in vibration testing of flight hardware. Finally, the generalized mass attenuation (GMA) method, which is a new method outlined in reference 11 predicts the force responses at the component interfaces. Preliminary results from this method are promising.

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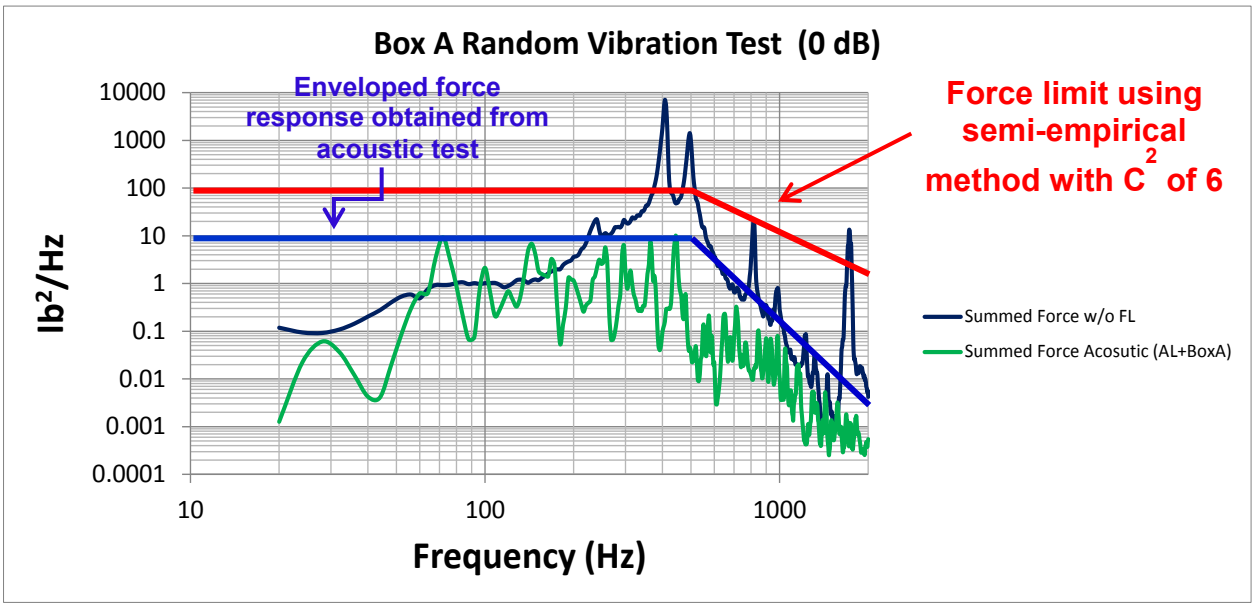

Figure 16: The same as Figure 6, except the enveloped force responses of box A mounted on the AL panel from acoustic test is used to illustrate the differences in the semi-empirical method and flight-like component interface force measurements.

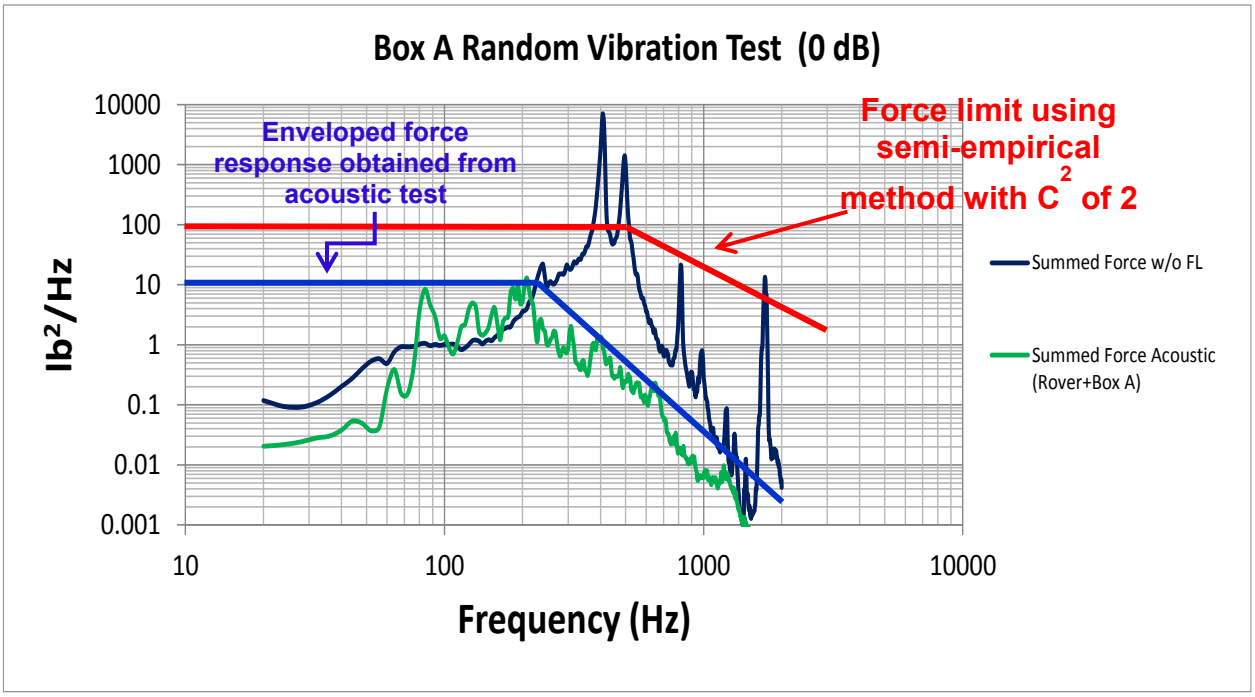

Figure 17: The same as Figure 9, except the enveloped force responses of box A mounted on the Rover top deck from acoustic test is used to illustrate the differences in the semi-empirical method and flight-like component interface force measurements.

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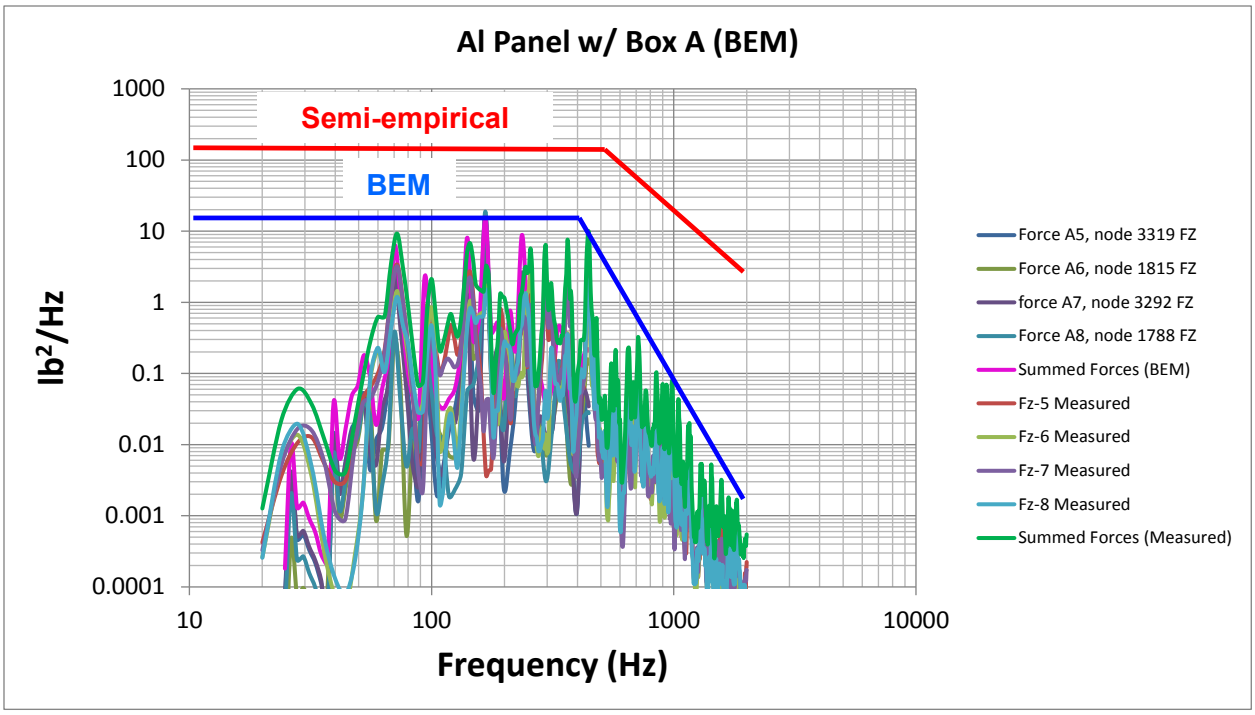

Figure 18: The BEM predicted forces at the box A interfaces mounted on Al panel. The envelop of the predicted forces are used to compare with the semi-empirical derived force limiting specification.

SUMMARY

The force-limited vibration test approaches discussed in NASA-7004C were developed to reduce overtesting associated with base shake vibration tests of aerospace hardware where the interface responses are excited coherently. Some of the methods discussed in the handbook were used to specify the force limits and are compared with the forces measured at two electronic boxes mounted on the Al and Rover top deck panels that underwent acoustic testing. The rationale for force limiting is based on the disparity between the impedances of typical aerospace mounting structures and the large impedances of vibration test shakers when the interfaces in general are coherently excited. The observations made from the acoustic tests indicate the inclusion of the incoherent excitation of the components at mounting interfaces result in force specifications much different (in terms of force limiting magnitude, break frequency, and roll-off slope) than predicted forces using the semi-empirical method. These observed differences provide the basis for studying more realistic force limits for qualifying flight hardware using shaker testing. Elaborate efforts at JPL are currently being undertaken to come up with new approaches based on considering the incoherent excitation of the structural mounting interfaces. These approaches are to use FE models of the sources and loads (components) and predict the interface forces either directly or predict apparent masses that account for the cross spectral responses of the components' interfaces, or use BEM to predict the interface forces, and finally use GMA method, which is a classical random vibration approach that only considers the acceleration responses of the sources. All these methods are being extensively investigated. The final product from these investigations will lead to the modification of the force limiting handbook, where recommendations will be made to improve the force limiting specifications and to remove still existing excessive conservatism from random vibration testing by employing improved force limiting methods of deriving the specifications.

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REFERENCES

1. Force Limited Vibration Testing, NASA-HDBK-7004C, currently is going through release signature cycle.
2. Scharton, T. D., "Vibration-Test Force Limits Derived From Frequency-Shift Method", AIAA Journal of Spacecraft and Rockets 32(2), 1995, pp. 312–316.
3. Scharton, T. D., "Force-Limited Vibration Tests at JPL--A Perfect Ten", ITEA Jour. of Test and Evaluation, Vol. 14, No. 3, September 1993.
4. Chang, K. Y. and Scharton, T.D., "Verification of Force and Acceleration Specifications for Random Vibration Tests of Cassini Spacecraft Equipment", Proc. of ESA/CNES Conference on Spacecraft Structures, Materials & Mechanical Testing, Noordwijk, NL, March 1996.
5. Scharton, T. D. and Chang K., Y., "Force Limited Vibration Testing of the CASSINI Spacecraft and Instruments", Proceedings of the Institute of Environmental Sciences, 43rd Annual Technical Meeting, 1997.
6. Soucy, Y. and Montminy, S., "Investigation of Force Limited Vibration Based on Measurements During Spacecraft Vibration Testing", Proceedings of the 2007 Spacecraft and Launch Vehicle Dynamic Environments Workshop, Los Angeles, CA, June 2007.
7. Soucy, Y., Dharanipathi, V., and Sedaghati, R., "Comparison of Methods for Force Limited Vibration Testing", Proceedings of the IMAC XXIII Conference, Orlando, FL, Jan. 31 – Feb. 3, 2005
8. Soucy, Y., Vijayamohan, V., and Sedaghati, R., "Investigation of Limit Criteria for Force Limited Vibration", Proceedings of the 2005 Spacecraft and Launch Vehicle Dynamic Environments Workshop, Los Angeles, CA, June 2005.
9. Scharton, T. D., "Force Limits Measured on a Space Shuttle Flight", Proceedings of the 47th Annual Technical Meeting of the Institute of Environmental Sciences and Technology, ESTECH 2001, April 22, 2001.
10. Scharton, T. D., "In-flight Measurements of Dynamic Force and Comparison with the Specifications Used for Limiting the Forces in Ground Vibration Tests," European Conference on Spacecraft, Structures, Materials, and Mechanical Testing, Braunschweig, GR, November 1998.
11. Arya Majed, Ali R. Kolaini, and Ed Henkel, "Generalized Mass Attenuation and Force Limiting Methods," in preparation.

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Dr. Ali R. Kolaini has been a Member of the Technical Staff at JPL since 2005. He currently has a position as a Principal Engineer in the Dynamics Environments group of the Mechanical Systems Division. Prior to joining JPL, Dr. Kolaini was an Engineering Specialist at The Aerospace Corporation, an associate professor at the University of Mississippi. He has a B.S. degree in Mechanical Engineering from the Lawrence Tech University, and a M.S. and a Ph.D. in Mechanical Engineering from the University of California, Santa Barbara. He has more than 20 years of experience in the fields of vibration, shock, and acoustics.

Mr. Dennis Kern has worked in vibration, acoustic and shock prediction and testing for aerospace structures and components for 41 years. He has supervised the dynamics environments activities at the Jet Propulsion Laboratory since 1978, supporting all JPL flight projects and managing numerous technology development programs. He has played a major role in the development of several NASA and industry standards and handbooks and has organized the annual NASA/USAF/Industry Spacecraft and Launch Vehicle Dynamics Environments Workshops since 1988. Mr. Kern holds B.S. and M.S. degrees in Mechanical Engineering from the California State University, Long Beach.

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