FORCE MEASUREMENTS IN VIBRATION AND ACOUSTIC TESTS

Terry D. Scharton
Jet Propulsion laboratory
California institute of Technology
4800 Oak Grove Drive, MS 125-129
Pasadena, CA 91109
U.S.A.

ABSTRACT

The advent of triaxial, piezoelectric force gages and the associated signal processing has opened the door to several dynamics testing innovations. This new technology is being applied in the CASSINI and other spacecraft programs that JPL manages for NASA, with considerable international participation. These applications of force measurements have generated a number of analytical developments and topics for further research.

Force limiting is a proven application. Shaker input forces and moments are now routinely measured and controlled in JPL random vibration tests. Inflight, aerospace equipment mounted on lightweight structure, acts like a vibration absorber and generates a large reaction force to reduce the input motion at antiresonance frequencies. Limiting the input force in the vibration test to that predicted for flight minimizes overtesting and overdesign. New analytical techniques for predicting inflight force limits from structural impedance and modal effective mass have been developed.

Base-drive and acoustic modal testing is a potential application. Modal information recovered from base-drive and acoustic tests of the CASSINI spacecraft, mounted on a dynamic force gage system, will be evaluated against that obtained in the conventional fixed-base modal test of the spacecraft structural model to see if a separate modal test can be eliminated in future spacecraft programs. Relevant experience in the international community is solicited.

NOMENCLATURE

A = interface acceleration
A_o = free acceleration of source
A_s = acceleration specification
F = interface force
F_o = blocked force of source
F_s = force specification
M = dynamic mass
m = modal mass
S_{AA} = acceleration spectral density
S_{FF} = force spectral density
Q = dynamic amplification factor
w = radian frequency
w_o = oscillator natural frequency

FORCE LIMITED VIBRATION TESTING

Background

For lightweight aerospace structures, the mechanical impedance of payloads and of the mounting structure are typically comparable so that the vibration of the combined structure and load involves modest interface forces and responses. Most of the high amplification resonances and mechanical failures in conventional vibration tests are test artifacts associated with the essentially infinite mechanical impedance and unlimited force capability of the shaker. With a recently developed vibration testing technique [1], these artificial failures and the related overdesign penalties are eliminated by limiting the vibratory force in the test to that predicted for flight. Limiting the input force is in theory equivalent to limiting the load response, but force limiting is less dependent on the details of the analytical models and is usually more convenient. ALso, critical response locations are often numerous and not accessible.

Implementation of force limited vibration testing requires: derivation of a force specification (analogous to that for acceleration), vibration test fixturing to accommodate force sensors, and shaker operation with dual control of both acceleration and force.

Equation 1, which may be derived from Norton's and Thevenin's equivalent electrical circuit theorems, provides a theoretical basis for dual control of vibration tests

$$1 = \mathbf{A}/\mathbf{A_o} + \mathbf{F}/\mathbf{F_o} \qquad (1)$$

Equation 1 is exact but difficult to apply because the terms on the right hand side are complex and complicated functions of frequency. (Throughout the paper, boldface type indicates a complex quantity.) The phase of the inputs and the impedances would be difficult to determine, and also phase cannot be specified with currently available vibration test controllers.

An alternative, approximate formulation for the control of vibration tests is provided by the following extremal equations:

$$|\mathbf{A}|/|\mathbf{A_s}| \leq 1 \text{ and } |\mathbf{F}|/|\mathbf{F_s}| \leq 1 \qquad (2)$$

In Eq. 2, the free acceleration and blocked force of Eq. 1 are replaced by specifications which envelope the interface acceleration and force in the coupled system. With extremal control, the shaker current is adjusted in each narrow frequency band so that the larger of the two ratios in Eq. 2 is equal to unity. At frequencies other than the test item resonances, the acceleration specification usually controls the test level; at the resonances, the base reaction force increases and the force specification limits the input.

Most vibration controllers have the capability for extremal control, but older controllers allow only one reference specification. To implement dual control in this case, a filter must be used to scale the shaker force feedback signal to an equivalent acceleration. New controllers allow separate specifications for limit channels, so Eq. 2 may be directly implemented. Force limiting has been used primarily for random vibration tests, but the application to swept sine tests is also practical and beneficial.

Frequency Shift Method

There are virtually no flight data and little system test data on the vibratory forces at mounting structure and test item interfaces. Currently force limits for vibration tests are therefore calculated using analytical or measured structural impedances of the mounting structure and the test items, together with the conventional interface acceleration specification. Herein an improved "frequency shift" method of calculating the force limits is described and applied to a simple two-degree-of-freedom system (TDFS). The two oscillators shown in the upper right-hand corner of Fig. 1 represent coupled resonant modes of the source and load, so the oscillator masses are equivalent to modal masses of the distributed systems.

For both the flight configuration with a coupled source and load and the vibration test configuration with an isolated load, the interface force spectral density is related to the interface acceleration spectral density as

$$S_{FF}(w) = |M_2(w)|^2 S_{AA}(w) \qquad (3)$$

The load dynamic mass is a frequency response function (FRF) which includes mass, damping, and stiffness effects. The frequency dependence is shown explicitly in Eq. 3 to emphasize that the relation between force and acceleration applies at each frequency.

For white noise base motion or external force excitation of the coupled system in Fig, 1, the interface acceleration and force spectral densities both peak at the same frequencies, i.e. the coupled system natural frequencies. The load dynamic mass, evaluated at one of these natural frequencies, may be interpreted as the ratio of the force spectral peak to the acceleration spectral peak at that natural frequency.

The frequency shift method of deriving force specifications consists of multiplying the conventional acceleration specification, which is assumed to properly envelope the acceleration spectral peaks, by the load dynamic mass, evaluated at a coupled system resonance frequency. A central point of the method is that the load dynamic mass must be evaluated at the coupled system, or shifted, resonance frequencies, The values of the load dynamic mass at the coupled system resonance frequencies are considerably less than the peak value at the load uncoupled resonance frequency.

Two-Degree-of-Freedom System Example

As an example of the frequency shift method, the force limit is calculated for the TDFS in Fig. 1 with different masses of the source and the load oscillators. The maximum response of the load and therefore the maximum interlace force occur when the uncoupled resonance frequency of the load equals that of the source [2]. For this case, the characteristic equation is that of a classical dynamic absorber [3]:

$$(w/w_0)^2 = 1+(m_2/m_1)/2 \pm [(m_2/m_1)+(m_2/m_1)^2/4]^{0.5} \qquad (4)$$

The ratio of the interface force to acceleration spectral densities, calculated as in Eq. 3 from the magnitude squared of the load dynamic mass, is

$$S_{FF}/(S_{AA}\, m_2^2) = [1+(w/w_0)^2/Q_2^2] / \{[1-(w/w_0)^2]^2 + (w/w_0)^2/Q_2^2\} \qquad (5)$$

The force spectral density, normalized by the load mass squared and by the acceleration spectral density, at the two coupled system resonances is obtained by combining Eqs. 4 and 5. For this TDFS the normalized force is just slightly larger at the lower resonance frequency of Eq. 4. The maximum normalized force spectral density, obtained by evaluating Eq. 5 at the lower resonance frequency from Eq. 4, is plotted against the ratio of load to source mass for three values of Q_2 in Fig. 1.

In Fig. 1, for very small (0,0001) values of the ratio of load to source mass, the load has little effect on the source, and the maximum normalized force approaches Q squared, For larger ratios of the masses, the maximum force is smaller because of the vibration absorber effect at the load resonance frequency.

Use of Fig. 1 to define force specifications requires that the oscillator masses in Fig. 1 be interpreted as effective masses of the distributed source and load system, which masses vary with frequency. It has proven convenient to define the masses in one-third octave frequency bands. In most previous force limited vibration tests the effective masses have been taken as the smoothed FRF's of the ratio of drive point force to acceleration as measured with a shaker or an impact hammer. This smoothing is defined by geometric averaging in the frequency domain, and the result is synonymous with what is sometimes called the critically damped, asymptotic, or skeleton FRF's. Alternately, using the results of finite-element-model (FEM) analyses, the masses in Fig. 1 have been taken as the suitably normalized [4] residual masses, i.e. the sum of the masses of all modes with resonance frequencies in anti above the excitation frequency band. Application of the frequency shift method to a more complex TDFS model, with both modal and residual masses, is described in [1].

Application to CASSINI Spacecraft CDA Instrument

The German Aerospace Research Establishment in Berlin GR conducted the vibration qualification tests of the Cosmic Dust Analyzer (CDA) instrument being developed jointly with the Max Planck Institute in Heidelberg GR for the CASSINI spacecraft being integrated by the Jet F'repulsion Laboratory for NASA. The shaker force limiting technique, described herein was used in the vibration tests of the prototype CDA in Fig. 2 and for most other instruments on the CASSINI spacecraft [5]. The CASSINI spacecraft is shown mounted for a system vibration test in Fig. 3.

In force limited vibration tests, the acceleration input to the instrument under test is automatically notched at the equipment resonances by limiting the shaker forces to values predicted for flight. Ideally, the acceleration and force specifications used in the tests would envelope the peaks in the spacecraft/in strument interface environment during the launch, with a desired test margin. However, since interface acceleration data are not often available at the time of the instrument vibration tests, the acceleration specification is usually scaled from previously obtained flight or system test data. Since no flight data and little system test data on the interface forces are available, force limits are usually derived from measurements of the mounting structure mechanical impedance (apparent weight) and two-degree-of -freedom models described herein and in [1]. Interface force data were measured during acoustic tests of the CASSINI spacecraft Development 7 est Model (DT M), and that data provided verification of the force limit prediction methods used for the CDA [6].

In the CDA tests, which were conducted in three perpendicular axes, force limiting was used in both tile sine and the random vibration tests. The 16 Kg CDA prototype instrument, which is also the flight spare, was mounted on four medium sized triaxial force gages using an aluminum adapter ring which weighed 0.6 Kg, see Fig.?. In every case the total force, the sum fronr the four gages, in the direction of shake was limited to the predicted fright values. Fixture and procedure checkout was accomplished with a mass simulator of the CDA. [n the random vibration tests, force limiting was used to notch the acceleration in real-time using the extremal (peak) control mode. Since the controller used did not support independent reference specifications for limit channels, a flat force limit was used, and scaled into a pseudo-accele ration which the controller compared with the acceleration reference spectrum. This technique has been used in many force limited random vibration tests and worked well in the CDA tests. In the sine vibration tests of the CDA, the measured force was compared off-line to the calculated force limits and to the equivalent rigid body acceleration design limits, and manual notching was utilized. Manual notching was used because of the absence of the automatic notching feature in the controller and insufficient experience with the scaled pseudo-acceleration control in conjunction with the fast, six octave per minute, sweep rate.

Both the sine and random vibration tests of the CDA prototype instrument in three axes were successfully completed in three working days. Notches between six and twelve dB resulted at the CDA resonance frequencies on the shaker. There was a general consensus among those present at the test that these notches, necessary to avoid overtesting, were essential to the completion of the test without damage to the CDA unit.

Figure 4a and 4b show the input force and control acceleration in a lateral random vibration test of the CDA instrument mass simulator. The data in Figs.4 were measured in low level (15 dB down from full level) tests conducted to verify the force limits and acceleration input notching; Fig. 4a is without force limiting and Fig, 4b is with force limiting. As previously mentioned, the force signal and limit are scaled into a psuedoacceleration acceleration to accommodate the controller, which did not support specification of response limit spectra. Comparison of Figs. 4 a and 4b show that force limiting reduced the input force, control acceleration, and response about 10 dB at the fundamental resonance of 93 Hz.

BASE-DRIVE AND ACOUSTIC MODAL TESTING

In future spacecraft programs, there will probably be less system testing in order to reduce cost and schedule. It is anticipated that the traditional separate vibration qualification, modal, and acoustic tests may be replaced by a single test, In this case, it is important to obtain as much information as possible from the one test. JPL has been experimenting to determine the added value of measuring the base reaction forces in all three types of these tests. Triaxial force gages are mounted under the test item, as in the Ct3A instrument vibration test application discussed previously, and a summing network is used to determine in real time the six resultant force and moment components. In the case of a vibration qualification test, or a base-drive modal test, the force gages provide the effective mass of the test item modes [7]. In the case of a fixed-base modal test, the force gages provide the reaction forces required for recently developed substructuring models [8]. In the case of an acoustic test of a ground supported test item, the measured reaction forces provide the natural frequencies of the lower order fixed-base modes.

Figure 3 is a schematic of the CASSINI spacecraft, vertical, random vibration test which will be conducted in the fall of 1996. The spacecraft weighs approximately 5700 kg and stands about 9 m high mounted on the shaker. The CASSINI spacecraft is being built by JPL for NASA with considerable international participation, e.g. the Huygens probe is being provided by ESA and the high gain antenna by Italy. In the vibration test, a force ring incorporating eight triaxial force gages will be located between the shaker head expander and the spacecraft adapter. The individual force gage signals will be combined in real time to provide the six resultant input forces and moments, which will be controlled to flight limit values, A limited number of accelerometers will be used to monitor the response of the critical spacecraft elements, This approach is expected to be considerably simpler and less time consuming than the traditional JPL approach which involved measuring and limiting the response with over a hundred accelerometers mounted on the spacecraft.

The reaction force measurements in the CASSINI spacecraft vibration test will also be used to obtain modal data for comparison with that previously obtained in fixed-base modal tests of the spacecraft. Following the vibration test, an acoustic test will be conducted with the CASSINI spacecraft still mounted on the force transducers, Acoustic tests of partial stack-ups of the Development Test Model (DTM) of the CASSINI spacecraft have already been conducted, primarily to determine the random vibration input at instrument locations [6]. Figure 5 shows the vertical reaction force measured in one such acoutic test of a partial DTM stack-up which weighed approximately 4100 km. (The fuel mass simulators accounted for most of this weight.) Major resonances at 60 Hz and 100 Hz are evident.

CONCLUSIONS

The advent of triaxial piezoelectric force transducers has lead to a number of innovations in vibration and acoustic testing of aerospace hardware. The principal development is the limiting of shaker force to effectively control the source mechanical impedance in vibration tests. Secondary developments are the determination of effective mass in base-drive modal tests, of boundary reactions in fixed-base modal tests, and of global mode resonance frequencies in acoustic tests. There is great need for flight measurements of the vibratory forces at equipment and mounting structure interfaces to complement the existing acceleration data base and to validate this arid other force limit prediction methods. Two programs are underway to obtain flight force data, one for the space shuttle and one for an expendable launch vehicle.

ACKNOWLEDGMENTS

The author wishes to thank Robert Bamford of JPL and David Smallwood at Sandia National Laboratory for their help in the development of the frequency shift method. My [5,6] co-authors, Kurng Chang, Heiko Jahn, and Swen Ritzmann, are greatfully acknowledged, Thanks are also due to NASA Headquarters Office of Chief Engineer for funding the development of NASA guidelines and standard practices for force limited vibration testing. 1 he work described in this paper was carried out by the Jet Propulsion Laboratory, California Institute of 1 ethnology under a contract with the National Aeronautics and Space Administration.

REFERENCES

1. Scharton, T. "Vibration-lest Force Limits Derived from Frequency-Shift Method", *Journal of Spacecraft and Rockets,* Vol. 32, No. 2, March-April 1995.

2. Crandall, S. and Mark, W. , *Random Vibrations,* 1st cd., Academic Press, NY, NY, 1963, p. 96

3. Den Hartog, J., *Mechanical Vibrations, 3rd* cd., Mcgraw-Hill, New York, 1947, p. 115

4. Wada, B., Bamford, R., and Garba, J., "Equivalent Spring-Mass System: A Physical Interpretation", Shock and Vibration Bulletin, No, 42, 1972, pp. 215-22'5.

5. Jahn, H., Ritzmann, S., Chang, K., and Scharton, T. "Force Limited Vibration Testing of CASSINI Spacecraft Cosmic Dust Analyzer", Presented at ESA Conference on Spacecraft Structures, Materials and Mechanical Testing, Noordwijk, NL, 27-29 March 1996.

6. Chang, K. and Scharton, T. "Verification of Force and Acceleration Specifications for Random Vibration Tests of Spacecraft Equipment", Presented at ESA Conference on Spacecraft Structures, Materials and Mechanical Testing, Noordwijk, NL, 27-29 March 1996.

7. Fullekrug, U., "Determination of Effective Masses and Modal Masses from Base-driven 1 ests", German Aerospace Research Establishment (DLR), Proceedings of 14th International Modal Analysis Conference, Dearborn, MI, February 1996.

& Craig Jr., R, Blades, E., and Cutshall, W., "A Reduced Order Substructure System Identification Method", Paper AIAA-96-1200-CP, Presented at AIAA Dynamics Specialty Conference, Salt Lake City, Utah, April 18-19, 1996.

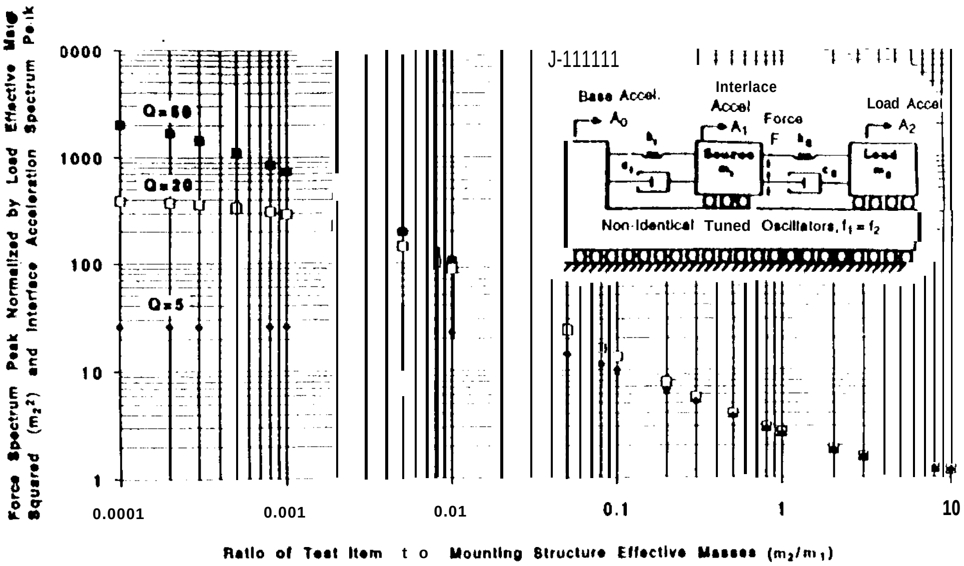

Fig. 1. Maximum Normalized Force for 1 wo-Degree-of-Freedom System

Fig, 2. Cosmic Dust Analyzer (CDA) Instrument Vibration lest

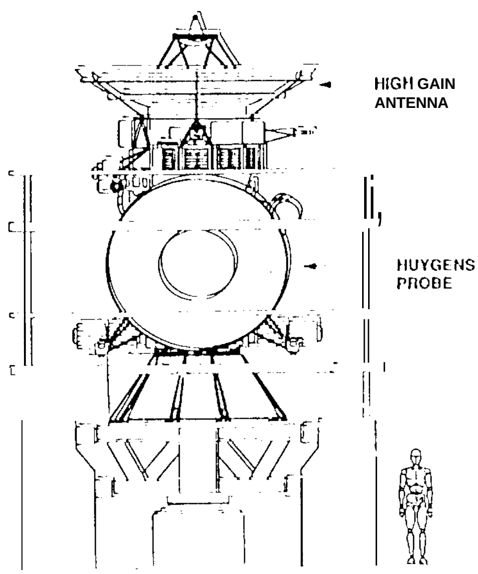

Fig. 3. CASSINI Spacecraft Mounted for System Vibration Test

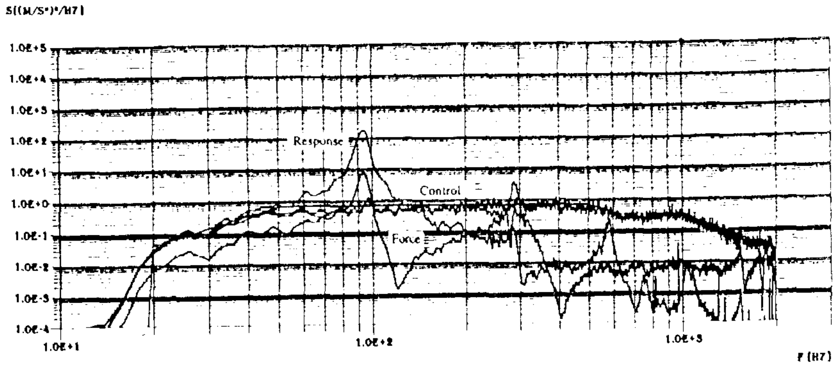

Fig. 4 a. Random Vibration Test (-15dB) without Force Limiting

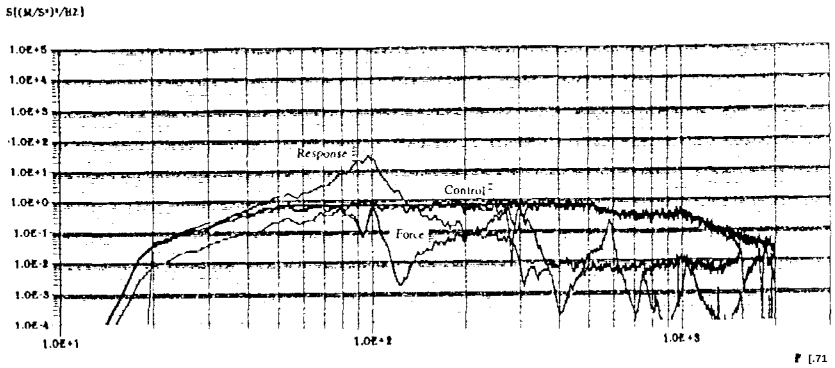

Fig. 4 b. Random Vibration Test (-15dB) with Force Limiting

Fig. 4. Random Vibration Test Data for CDA Instrument Mass Simulator

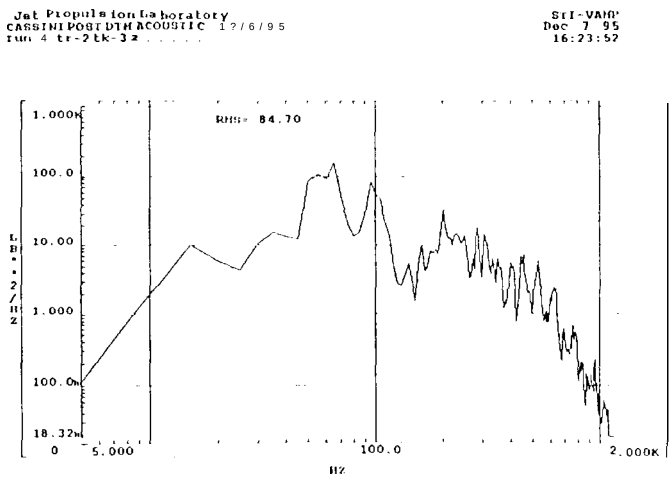

Fig. 5. Base Vertical Reaction Force Measured in Acoustic 1 est of CASSINI Spacecraft Partial Development Test Model (DT M)