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T. Scharton

Cassini Spacecraft And Instrument Force Limited Vibration Testing

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ABSTRACT

During the past three years, force limiting has been used in the vibration tests of most of the instruments and major equipment on the Cassini spacecraft, as well as in the vibration test of the complete flight spacecraft in November of 1996.

Force limits for the Cassini instruments and for the complete spacecraft vibration tests were developed using a simple, semi-empirical method which requires only the acceleration specification and data from a low level pre-test to determine the apparent mass of the test item. This semi-empirical method of predicting force limits was validated with interface force data measured at the instrument/spacecraft interface in acoustic tests of the Cassini spacecraft DTM structure and by comparisons with previously developed two-degree-of-freedom-system analytical models.

The application of force limiting to spacecraft equipment is illustrated with a discussion of the random vibration test of one of the Cassini spacecraft instruments, i.e. the Radio Plasma Wave Subsystem Antenna Assembly.

The environmental test program for the Cassini spacecraft included a force-limited vertical-axis random vibration test. The semi-empirical force limit specified in the Cassini spacecraft vibration test procedure was used without any modifications during the test, Acceleration responses at critical locations were monitored in the spacecraft vibration test, but only the total axial force was used in the control loop to notch the input acceleration. The flight limit loads were achieved at a number of critical locations on the spacecraft, and the instrument responses were similar to those in the component random vibration tests.

SUBMITTED TO:

3rd International Symposium on Environmental Testing for Space Programs, ESTEC, Noordwijk, NL, June 25-27, 1997.

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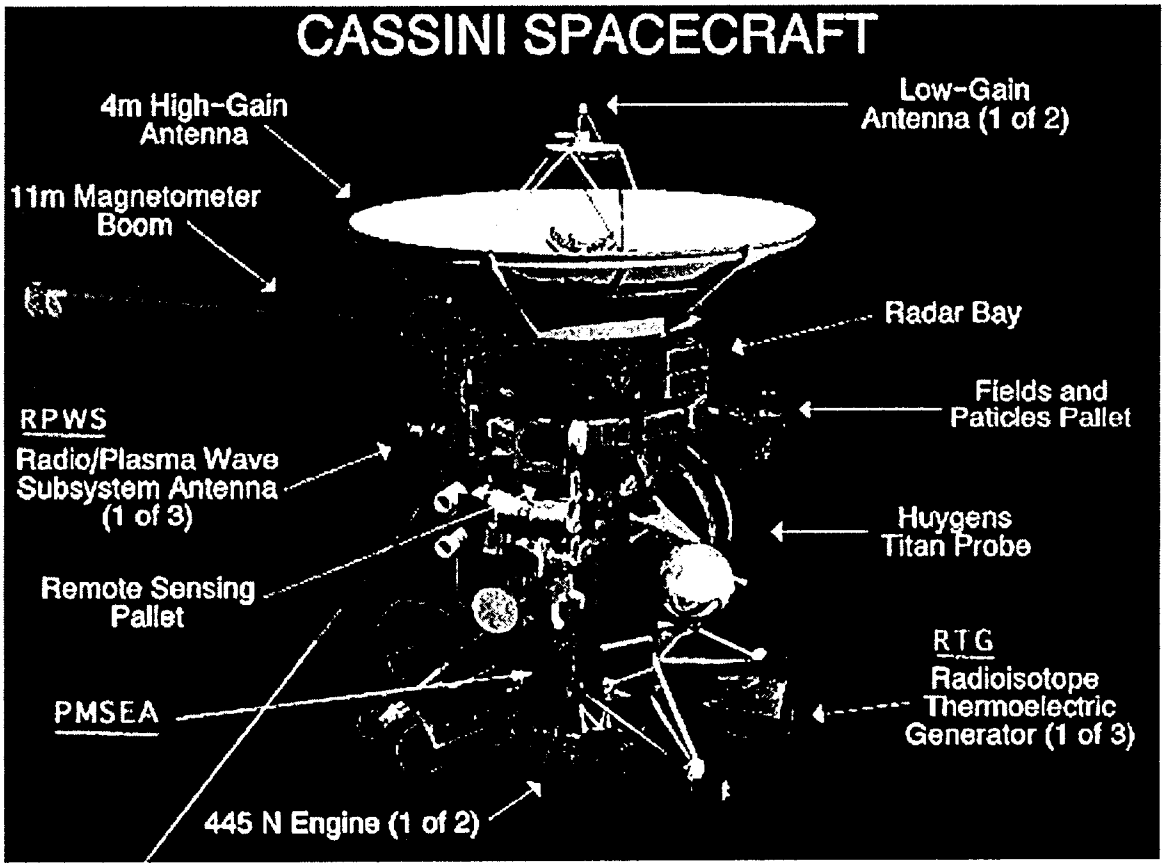
1. DESCRIPTION OF CASSINI MISSION

Saturn and its moon Titan will be the destination for the Cassini mission, a project under joint development by NASA, the European Space Agency (ESA) and the Italian Space Agency. The US portion of the mission is managed for NASA by the Jet Propulsion Laboratory (JPL). Launched in October 1997 on a Titan IV-Centaur rocket from Cape Canaveral in Florida, Cassini will first execute two gravity-assist flybys of Venus, then one each of the Earth and Jupiter to send it on to arrive at Saturn in June 2004. Because of the very dim sunlight at Saturn, solar arrays are not feasible and power will be supplied by a set of three Radioisotope Thermoelectric Generators (RTG's) which use heat from the natural decay of plutonium to generate electricity to run Cassini. After arriving at the ringed planet, the Cassini orbiter (see Fig. I) will release the Huygens probe, provided by ESA, which will descend to the surface of Titan. The launch weight of the Cassini spacecraft and Huygens probe is about 5,800 kilograms (12,800 pounds).



2. SPACECRAFT DTM ACOUSTIC TESTS

Three acoustic tests of the Cassini spacecraft Development Test Model (DTM) have been conducted. Each test involved different configurations of the DTM, flight, and engineering model spacecraft hardware and science instruments [1]. The primary objective of these tests was to provide verification of the predicted acoustically-induced random vibration test levels at equipment locations on the spacecraft. The locations included the attachment interfaces of the science instruments, RTG's, reaction wheels, and other spacecraft assemblies. In many cases, both the interface acceleration and interface force were measured, and in some cases the acceleration response at a position near the equipment static center-of-gravity (CG) was also measured.

3. SEMI-EMPIRICAL METHOD OF PREDICTING FORCE LIMITS

The following semi-empirical relationship between the maximum acceleration and maximum force at the interface between and a source and load in a coupled system was proposed in 1966 [2]:

$$F_l = C\ M_o\ A_s \tag{1}$$

where F_l is the amplitude of the force limit, C is a frequency dependent constant which depends on the configuration, MO is the total mass of the load (test item), and A_s is the amplitude of the acceleration specification. The form of Eq. 1 appropriate for random vibration tests is:

$$S'_{FF} = C^*\ MO^*\ S^s_{AA} \tag{2}$$

where S^l_{FF} is the force limit spectral density and S^s_{AA} the acceleration specification spectral density. In [2], it is claimed that C seldom exceeds 1.4 in coupled systems of practical interest, because of the vibration absorber effect.

A refinement of Eq. 2 follows from consideration of Eq. 3, which is Newton's second law for random vibration [3]:

$$S_{FF}\ (6))\ = |\underline{M}_2(\omega)|^2\ S_{AA}\ (\omega) \tag{3}$$

where: $\underline{M}_2(\omega)$ is the apparent mass of the load and $S_{FF}(\omega)$ and $S_{AA}(\omega)$ are the spectral densities of the interface force and acceleration, respectively. (The apparent mass $\underline{M}_2(\omega)$ is the frequency response function defined by Eq. 3.) Consider the frequency envelope of both sides of Eq. 3. As shown in [4] and discussed in [3], both $S_{FF}(\omega)$ and $S_{AA}(\omega)$ have peaks at the same frequencies, i.e. the coupled system resonance frequencies; and the acceleration spectral density $S_{AA}(\omega)$ has notches where the load apparent mass $\underline{M}_2(\omega)$ has peaks. The envelope of the left-hand side of Eq. 3 is S^l_{FF}, the force limit spectral density in Eq. 2. Following the approach taken in [5 & 6], the envelope of the right-hand side of Eq. 3 maybe taken as S^s_{AA}, the envelope of the acceleration spectral density, times the weighted frequency-average of the load apparent mass [7]. Motivated by these considerations, the subject refinement consists of including a factor of one-over-frequency to the nth power on the right hand-side of Eq. 2 at frequencies above the fundamental f_o:

$$S^l_{FF} = C^*\ M_o{}^2\ S^s_{AA}\ , \qquad f < f_o$$

$$S^l_{FF} = C^2\ MO^*\ S^s_{AA}\ /\ (f/\ f,))", \ f \geq f_o \tag{4}$$

Some judgment and reference to test data for similar configurations must be considered to choose the value of C and n in Eq. 4. Comparison of Eq. 4 with the results of the simple two-degree-of-freedom stem (TDFS) method of predicting force limits [3], indicates that a value of C of 1.4 implies a ratio of load to sour-cc masses of 1.5. It has been shown [8] that the frequency-average apparent mass of a rod in dilatation and of a plate in bending both fall off as one over frequency above the fundamental resonance, corresponding to an n of 2 in Eq. 4, and that of a beam in bending falls off as one over the square-root of frequency, corresponding to an n of unity in Eq. 4.

4. CASSINI RAD1O PLASMA WAVE SUBSYSTEM ANTENNA ASSEMBLY

Figure 2 shows the engineering model of the Cassini Radio Plasma Wave Subsystem Antenna Assembly (RPWS) mounted on a shaker for a lateral vibration test. Three small piezo-electric, tri-axial force gages are mounted between the rectangular test fixture and the RPWS triangular base. The use of piezo-electric, tri-axial force transducers for force limited vibration testing is highly recommended over other types of force measurement means such as strain transducers, armature current, weighted accelerometers, etc. The advent of these transducers has made the measurement of force in vibration tests almost as convenient and accurate as the measurement of acceleration [7].

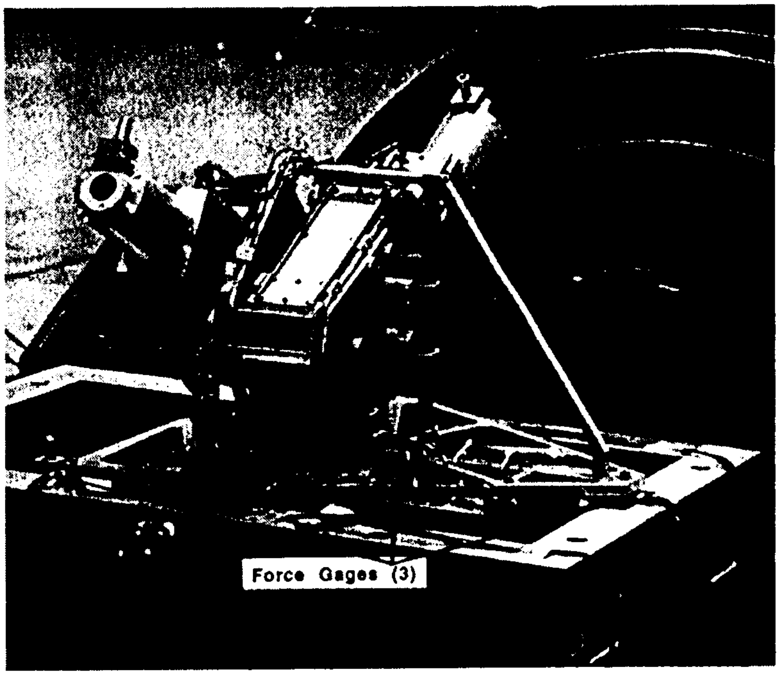


FIGURE 2. Cassini RPWS Antenna Mounted for Lateral Vibration Test

Figure 3 shows the acceleration data measured in different axes at the RPWS base during one of the spacecraft acoustic tests. Also shown in Fig. 3 is the RPWS random vibration test acceleration specification (O. I 5 G^2/Hz). The interface acceleration measured in the spacecraft test exceeds the RPWS test specification by 7 dB at 55 Hz for the z axis (radial) and is about 4 dB below the specification at the higher frequencies, above 200 Hz. The combined RPWS and truss mounting structure apparently has a fundamental radial resonance at 55 Hz and an attenuated response at the higher frequencies.

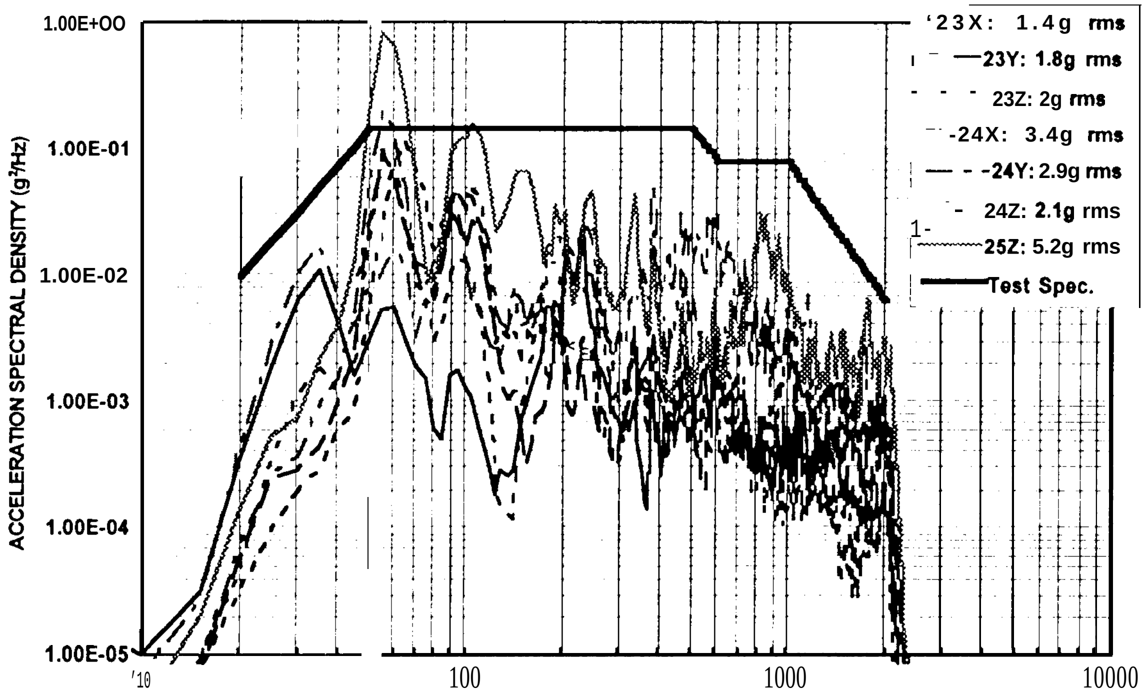


Figure 3. Comparison of Specified and Measured RPWS Interface Accelerations

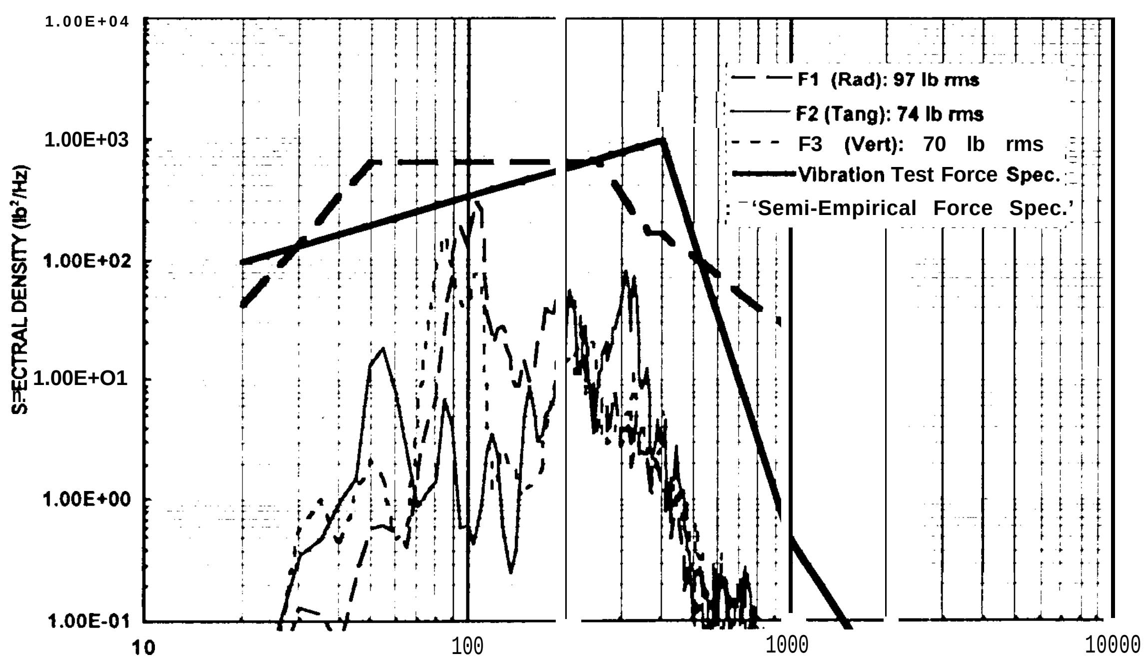


Figure 4. Comparison of Specified and Measured RPWS Interface Forces

Figure 4 shows the force data measured in different axes at the RPWS base during the spacecraft DTM acoustic test together with the force specification used in the RPWS instrument random vibration test. The force specification is 5 dB greater than the data at 55 Hz and much greater at higher frequencies. Also shown in Fig. 4 is a semi-empirical force specification for the RPWS instrument based on Eq. 4 with a C of unity and n of two.

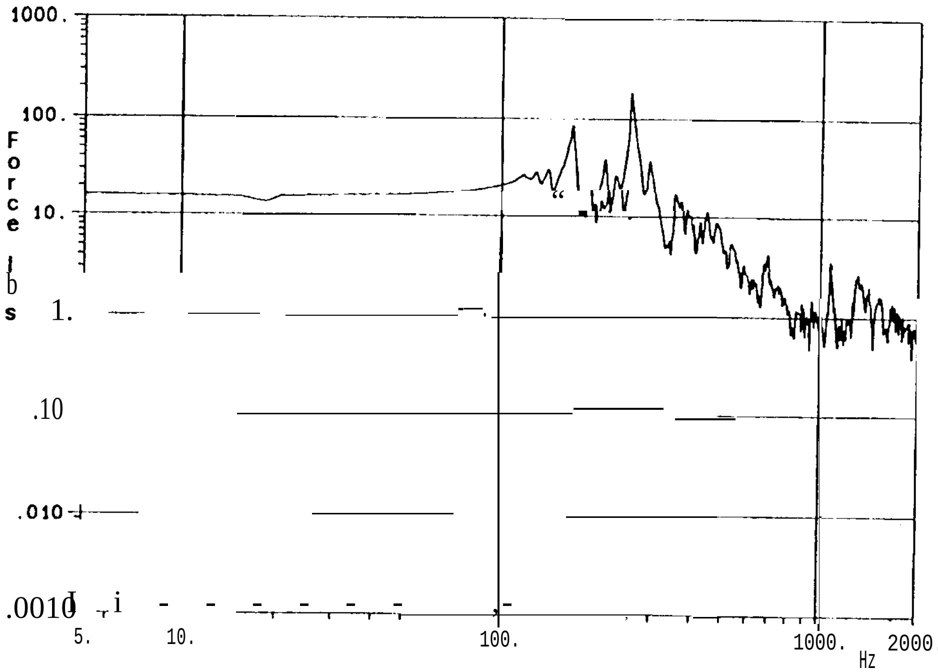


FIGURE 5. Force in 1/4 G Sine-sweep Vertical Vibration Test of RPWS Antenna

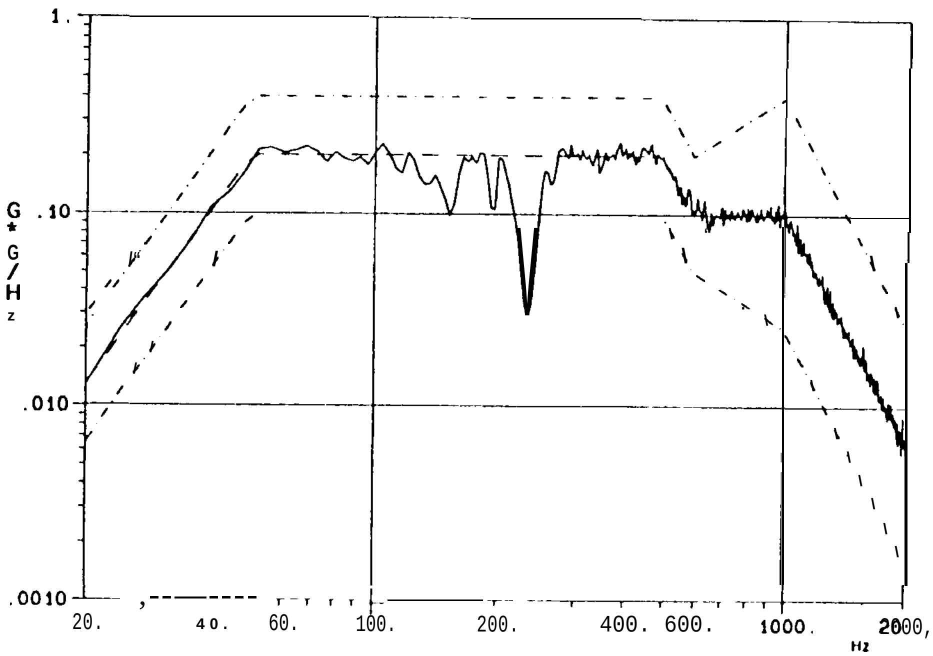


FIGURE 6. Notched Acceleration Input in Vertical Random Vibration Test of RPWS

Figure 5 shows the magnitude (reduced by 4) of the RPWS apparent mass measured in a preliminary low-level (0.25 G) sine-sweep vertical axis vibration test. Notice that the fundamental resonance of the RPWS in the shaker vertical axis test is at approximately 250 Hz, whereas the corresponding radial resonance of the RPWS mounted on the spacecraft DTM (Figs. 3 & 4) is at approximately 55 Hz. This discrepancy explains why the semi-empirical force specification in Fig. 4 must exceed the DTM data in the higher frequency regime. The force at the 250 Hz resonance on the shaker is limited to the maximum force occurring at 55 Hz in the spacecraft configuration. Figure 6 shows the notching that resulted in the full-level vertical random vibration test of the RPWS, when the test force specification in Fig. 4 was utilized.

5. MEASUREMENT OF QUASI-STATIC ACCELERATION IN VIBRATION TESTS

The design loads for aerospace equipment are often given in terms of the "quasi-static" acceleration of the center-of-gravity (CG), and it is often desired to limit the CG acceleration in low frequency vibration tests to something less than the design load. With the test item mounted on force gages in a vibration test, it is easy to monitor the CG acceleration; which, by Newton's second law, is equal to the measured external force divided by the total mass of the test item. However, it is relatively difficult to measure the CG acceleration of vibrating equipment with accelerometers [9]. Sometimes the CG is inaccessible, or there is no physical structure at the CG location on which to mount an accelerometer. However, there is a more serious problem: in the case of a deformable body, the CG is not generally a fixed point on the structure.

The non-fixity of the CG of a deformable body is demonstrated with an example in Fig. 7, which illustrates the third vibration mode of a three-mass, two-spring vibratory system.

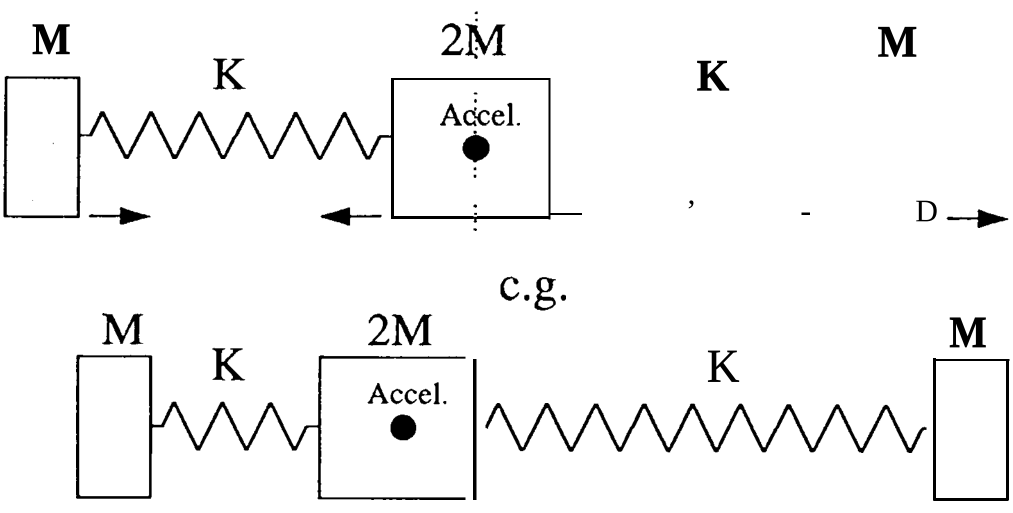


FIGURE 7. Third Mode of Three-Mass, Two-Spring System Showing Non-fixity of CG

The mass value of the middle mass is twice that of the end masses and the two springs are identical. The upper sketch in Fig. 7 shows the system at rest with the CG clearly located at the center of the middle mass. The lower sketch in Fig. 7 shows the system displaced in it's third mode, with the middle mass moving one unit to the left and the two end masses both moving one unit to the right. (The first mode involves rigid body translation, and the second mode involves zero motion of the middle mass and the two end masses moving an equal amount in opposing directions.) It is a characteristic of modal motion that there are no external forces acting, so by Newton's second law, the modal displacement illustrated in the lower sketch of Fig. 7 can't involve motion of the CG. However, since the middle mass, as well as the end masses, is clearing moving in the third vibration mode, the CG is not at a physical point. Clearly, one could not attach an accelerometer at the CG position.

The difficulty of measuring the CG acceleration with an accelerometer is further illustrated with the data in Figs. 8 and 9 obtained on the RPWS instrument in the Cassini spacecraft DTM acoustic test . In addition to the accelerometers and force transducers at the instrument and spacecraft interface (see the data in Figs. 3 and 4), there was also a tri-axial accelerometer located approximately at the static CG of the RPWS instrument. Fig. 8 shows the ratio of the total external radial force to the static CG radial acceleration for the RPWS in the DTM spacecraft acoustic test. If the accelerometer at the approximate static CG location actually measured the CG acceleration in the vibration test, the curve in Figs. 8 would be a horizontal line equal to the total weight of the RPWS, approximately 65 lb.

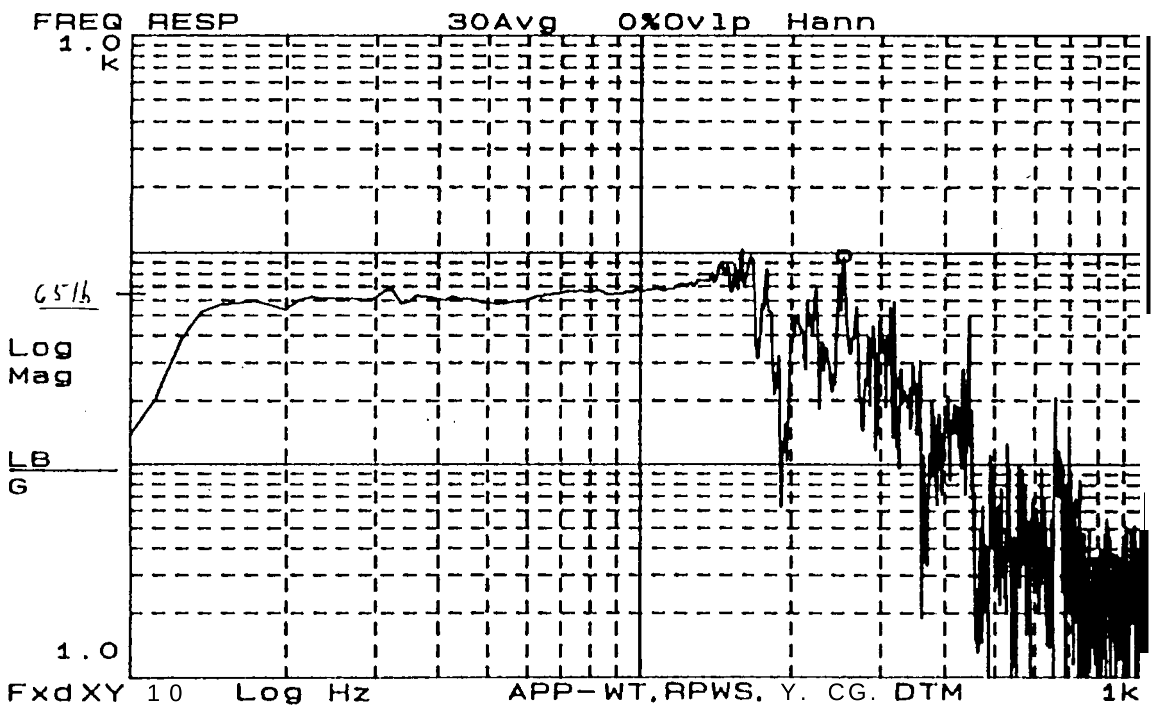


FIGURE 8. Ratio of Radial External Force To Radial Acceleration at Static CG for RPWS Instrument in Cassini DTM Spacecraft Acoustic Test

It is interesting to compare Fig. 8 with Fig. 5, after multiplying the ordinate in Fig. 5 by a factor of four to compensate for the one-quarter G input. Figure 8 shows that the measured ratio of external force to static CG acceleration falls off above 160 Hz. Figure 5 shows that there is an RPWS fixed-base resonance at approximately 160 Hz in the RPWS vertical test, which corresponds to the spacecraft radial direction. Figure 9 shows that in the RPWS lateral test, the corresponding ratios of external force to static CG acceleration measured in the spacecraft DTM acoustic test roll off at even lower frequencies. That the measured ratios of external force to static CG acceleration are less than the total weight, indicates that the measured static CG accelerations are greater than the true CG acceleration at the higher frequencies. Therefore response limiting based on acceleration measurements at the static CG position will generally result in undertesting at frequencies above the fixed-base resonances of the test item. The true center-of-gravity tends to remain at rest.

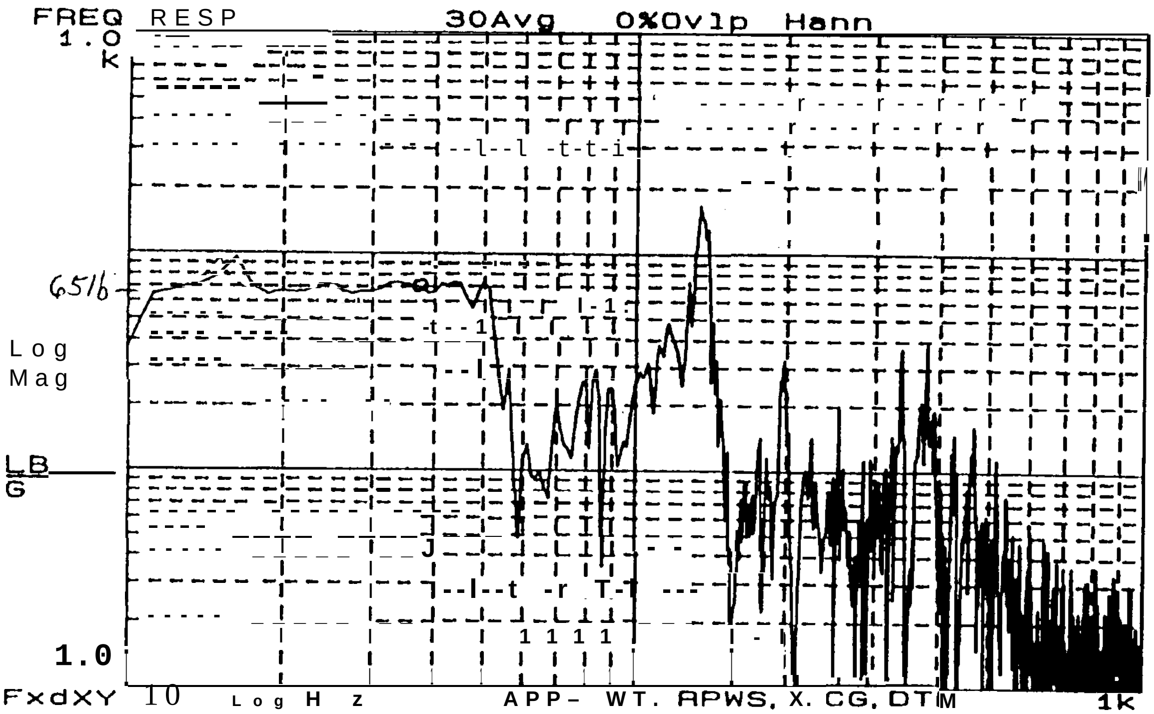


FIGURE 9. Ratio of Lateral External Force To Lateral Acceleration at Static CG for RPWS Instrument in Cassini DTM Spacecraft Acoustic Test

6. CASSINI SPACECRAFT SYSTEM RANDOM VIBRATION TEST

Figure 10 is a photograph of the Cassini flight spacecraft mounted on the shaker for the vertical random vibration test which was conducted at JPL in November of 1996 [7, 10]. The weight of the Cassini spacecraft for the vibration test was 3809 kg (8380 lb.), which is less than the weight at launch because for the test the tanks were loaded to only 60% of their capacity with referee fluids. The schematic in Fig. 1 indicates the location of major equipment on the Cassini spacecraft. Most of the orbiter twelve instruments are mounted on the Remote Sensing Platform shown at the upper left and on the Fields and Particles Platform shown at the upper right in Fig. 10, and seven other instruments are located in the Huygens probe at the right in Fig. 10.

Figure 11 shows the plan view of the spacecraft mounting ring before the spacecraft is attached. The black offset weight positioned in the upper right quadrant of the ring is for the purpose of proof testing the shaker and mounting configuration. The spacecraft bolts to the ring at eight positions corresponding to the mounting feet locations on the spacecraft/launch-vehicle adapter. A large tri-axial force transducer is located under the mounting ring at each of these eight positions. The shaker fixture is restrained from moving laterally during the spacecraft vertical vibration test by three hydraulic bearings. The force capability of the shaker is 35,000 lb. Virtually all of this capability was used to vibrate the spacecraft and the fixtures, which weighed 6,000 lb. in addition to the 8380 lb. spacecraft.

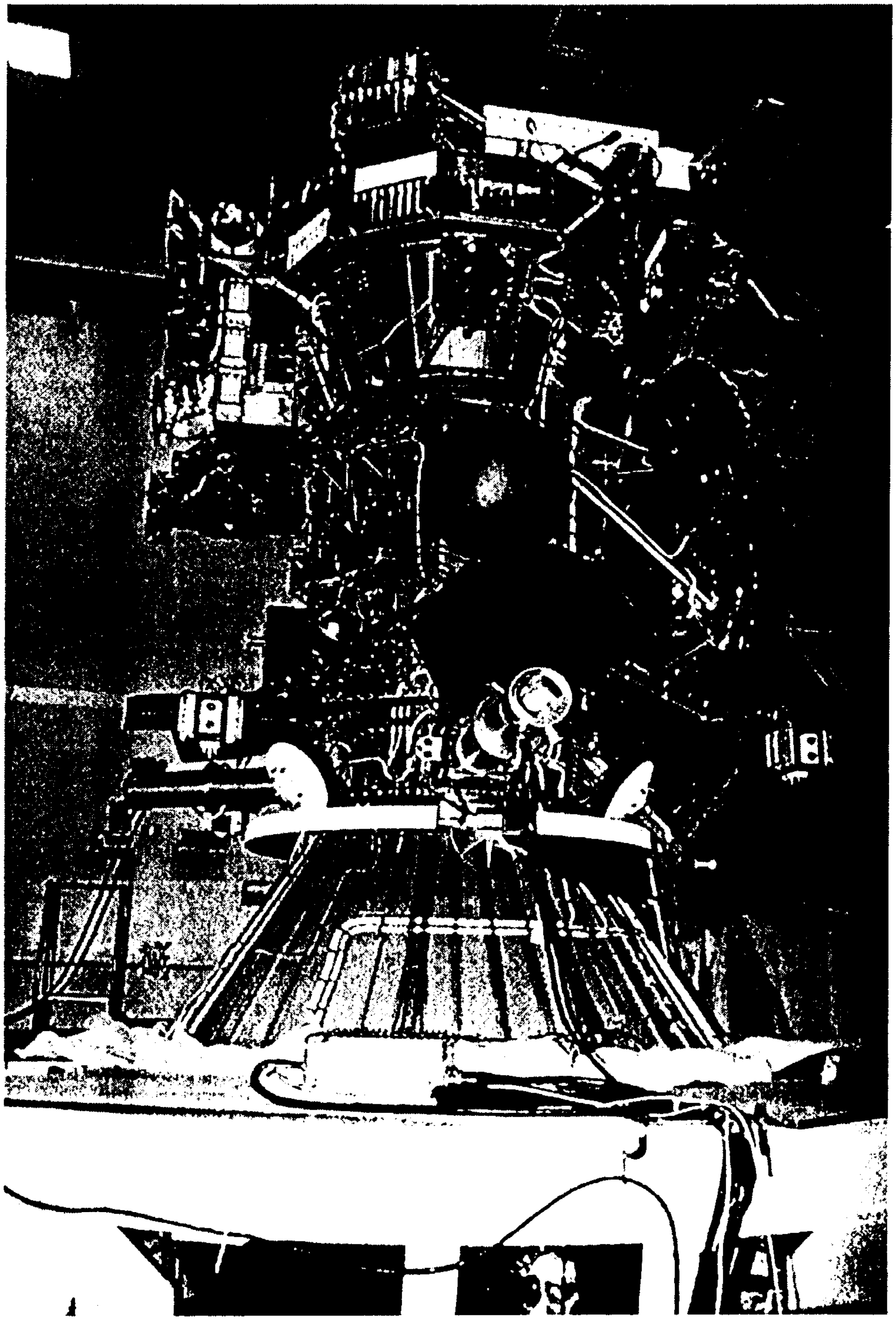
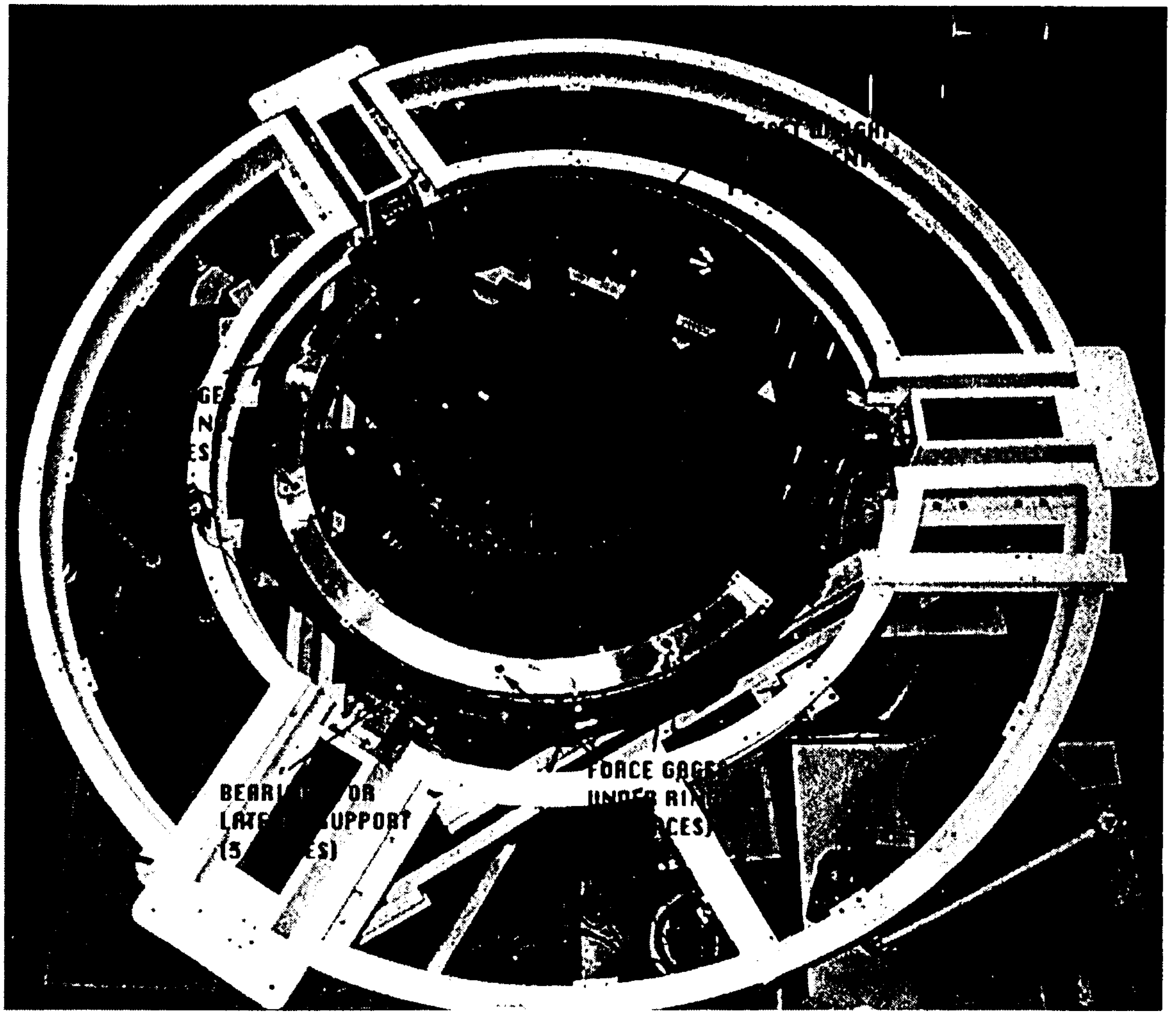


FIGURE 10. Cassini Spacecraft Mounted on Shaker for Vertical Random Vibration Test



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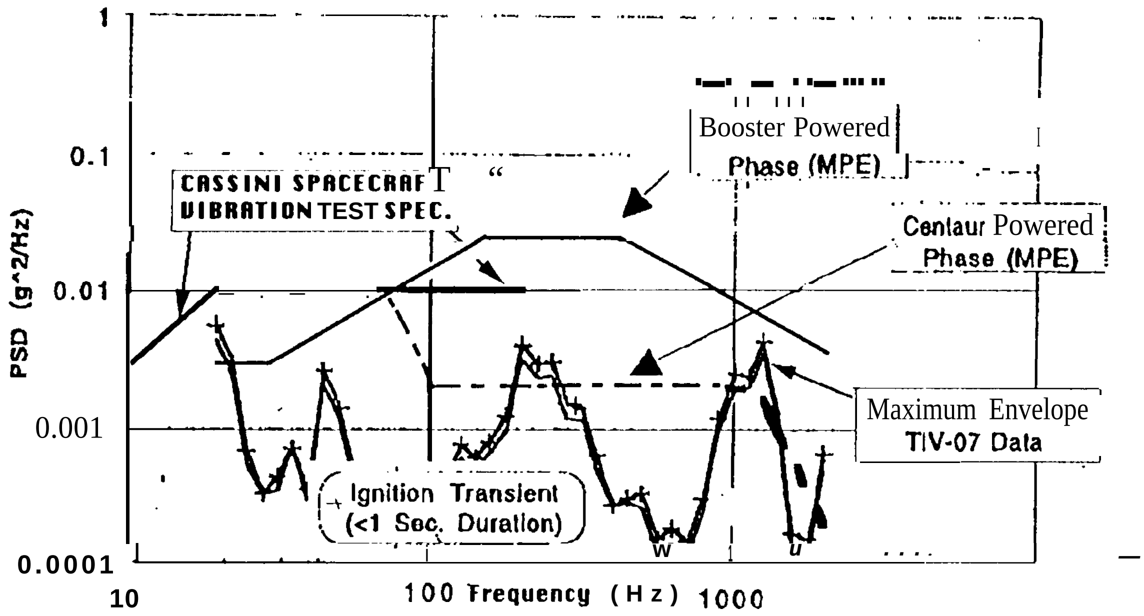

FIGURE 12. Comparison of Cassini Spacecraft Random Vibration Acceleration Input Specification with Launch Vehicle Specifications and Flight Data

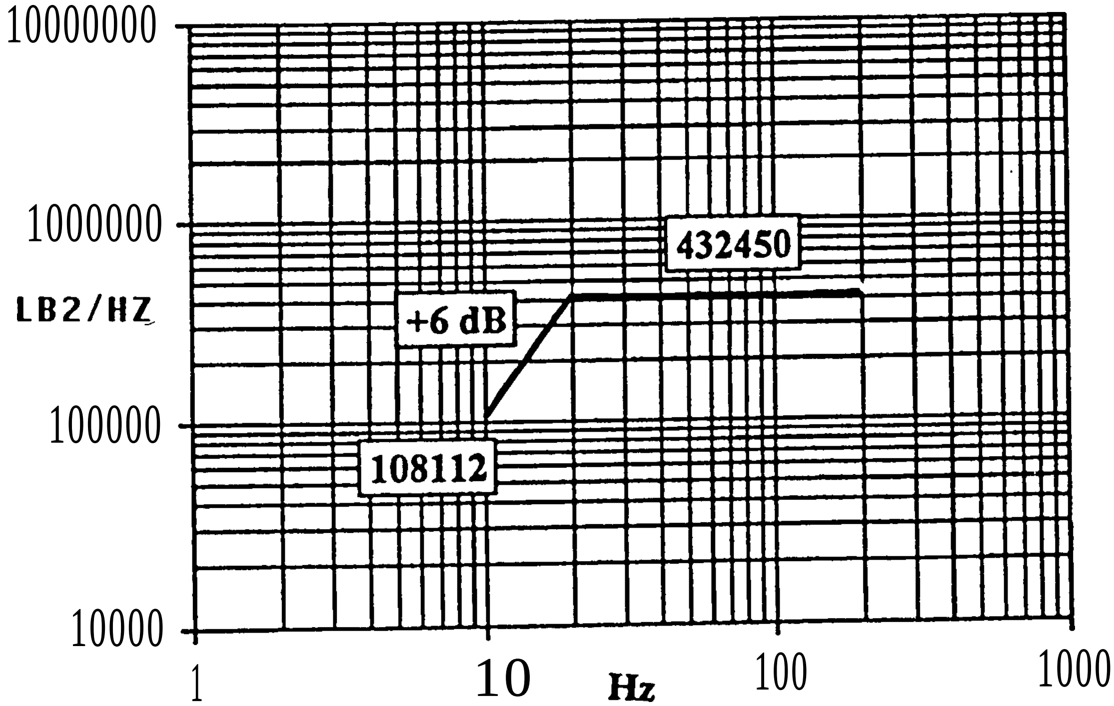

FIGURE 13. Cassini Flight Spacecraft Random Vibration Test Force Specification

The choice of C* equal to one-half was selected on the basis of the pre-test analysis and in order to keep the proof test, which had a margin of 1.25 over the test limit loads, within the shaker force capability. The force specification was not rolled off at the spacecraft fundamental resonance, because neither the FEM pre-test analysis nor the actual vibration test data showed a distinctive, fundamental resonance of the spacecraft in the vertical axis. During the test, it was not necessary to modify or update the force limit specified in the test procedure, which is quite remarkable considering the complexity of this test .

Figures 14 and 15 respectively, show the input acceleration and force spectra measured in the actual full-level random vibration test of the spacecraft. Comparison of the measured acceleration input with the specification of Fig. 12 shows notching of -8 dB at the Huygens probe resonance of approximately 17 Hz, and of -14 dB at the tank resonance of approximately 38 Hz.

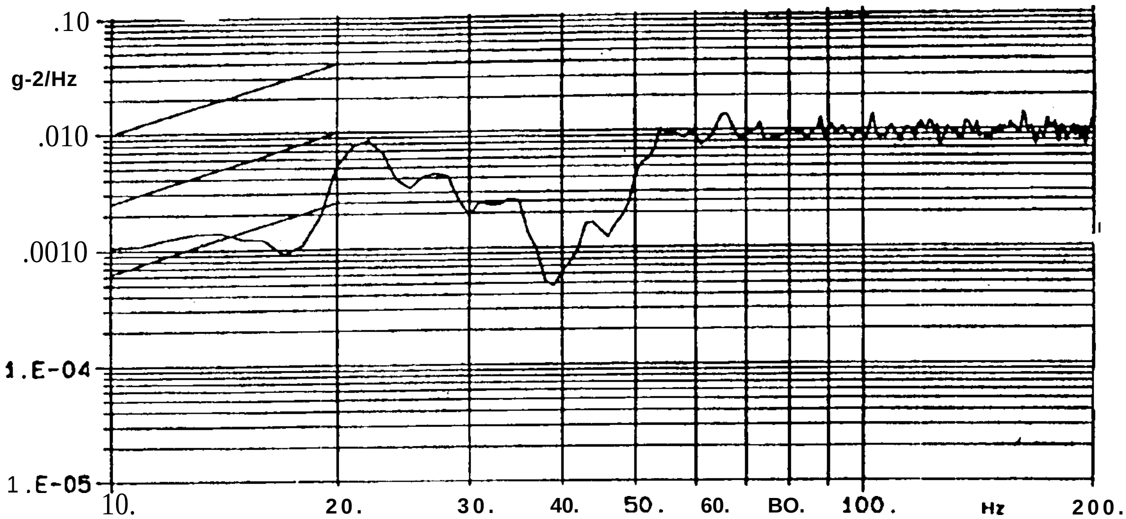

FIGURE. 14 Cassini Flight Spacecraft Random Vibration Test Full-Level Input Acceleration Spectral Density (Notches from Specification Due to Force Limiting)

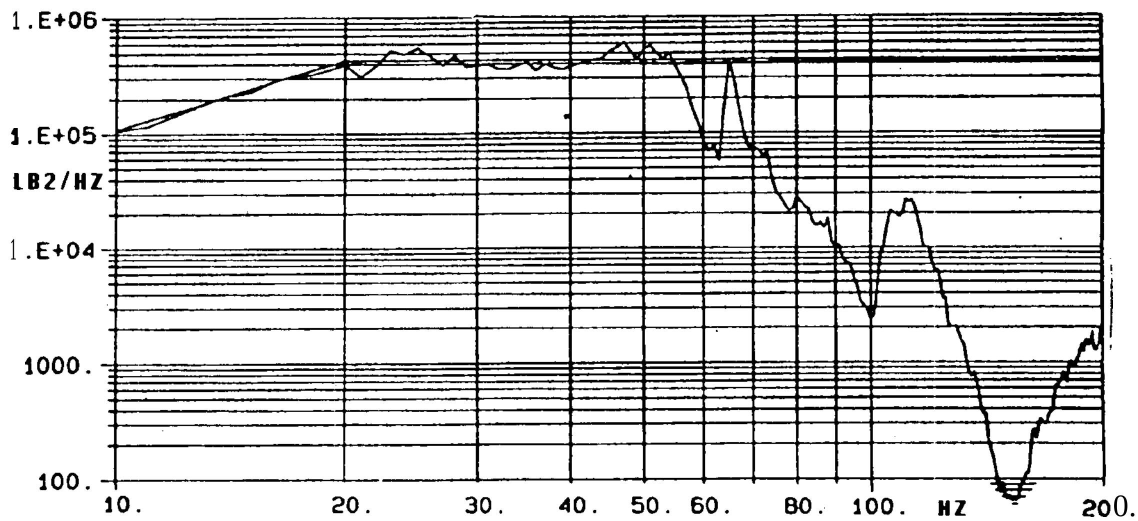

FIGURE 15. Total Vertical Force Measured in Cassini Flight Spacecraft Full-Level Random Vibration Test (Comparison with Vertical Force Limit Specification)

The other five components of the total input force vector, as well as the responses at over a hundred critical positions on the spacecraft, were monitored during the test, but only the total vertical force signal was used in the controller feedback to notch the acceleration input. Comparison of the measured force with the specified force in Fig. 15 verifies that the force was at it's limit over the entire frequency range where notching occurred in the input acceleration. The choice of C* of one-half resulted in the input acceleration being 3 **dB** less than the acceleration specification at frequencies below the first spacecraft resonance, e.g. at 10 Hz. This situation is generally to be avoided,

Figures 16 and 17 show the acceleration inputs measured near the feet of a number of instruments mounted on the Fields and Particles and on the Remote Sensing Pallets, respectively. Comparison of these measured data with the random vibration test specifications for the instruments, which are also indicated in Figs. 16 and 17, verifies that many of the instruments in the spacecraft vibration test, reached their component random vibration test specifications. In addition, several major components of the spacecraft including the Huygens probe upper strut, the three RTG's, the magnetic canister struts, and the Fields and Particles Pallet struts reached their flight limit loads during the spacecraft vibration test. The only anomaly after the test, other than possibly those associated with spacecraft functional tests for which data are not available, was that the electrical resistance between the engineering mode] RTG and the spacecraft structure was measured after the test and found to be less than specified. The insulation between the RTG adapter bracket and the spacecraft was redesigned to correct this problem.

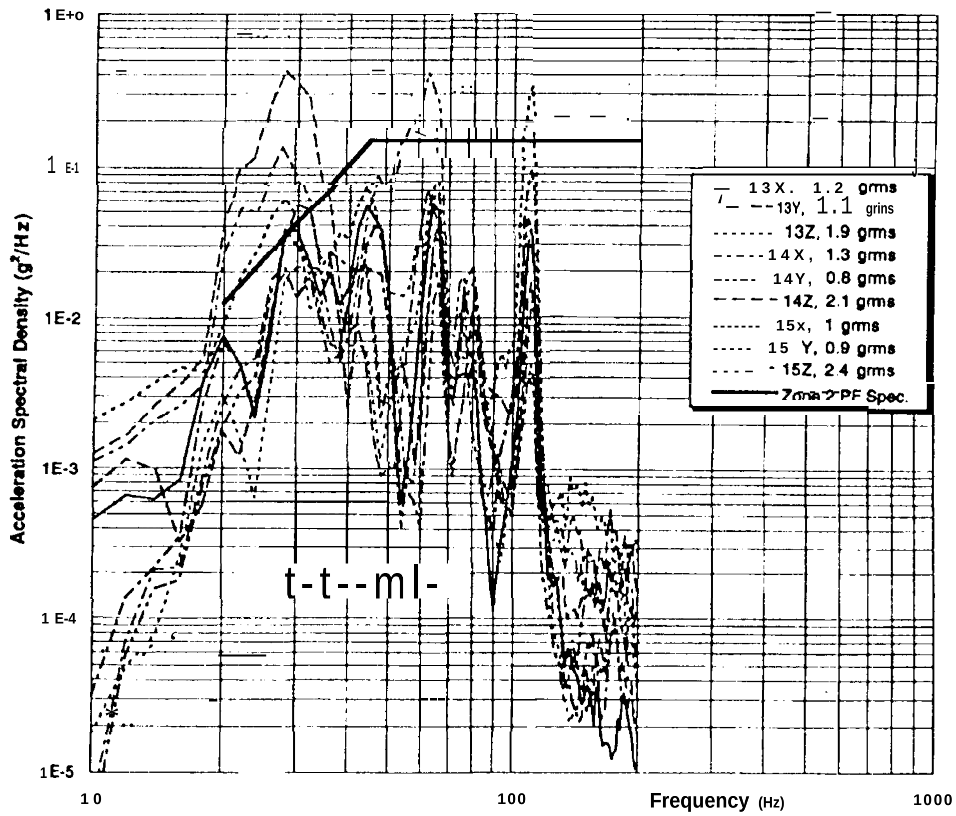

FIGURE 16. Acceleration Inputs to Fields and Particles Pallet Instruments in Cassini Full-Level Random Vibration Test (Comparison with Instrument Random Vibration Test Specification)

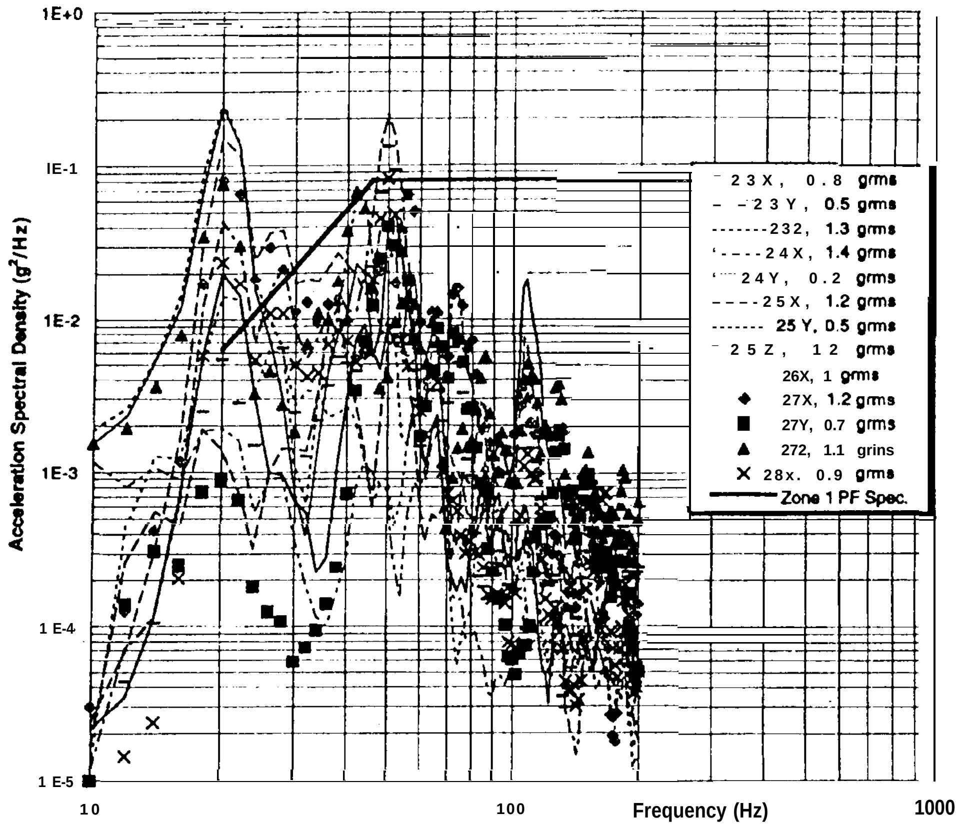

FIGURE 17. Acceleration Inputs to Remote Sensing Pallet Instruments in Cassini Full-Level Random Vibration Test (Comparison with Instrument Random Vibration Test Specification)

7. CONCLUSIONS

1. Force limiting has been used extensively in the vibration tests of many of the instruments and sub-assemblies on the Cassini spacecraft, as well as in the vertical-axis random vibration test of the complete spacecraft. In all cases, the use of force limiting reduced the degree of overtesting, without compromising the test objectives. In many cases, and particularly in the case of the spacecraft system test, the use of force limiting greatly simplified and expedited the conduct of the test.

2. Semi-empirical force limits were used in a number of the Cassini vibration tests. Semi-empirical force limits require only the acceleration specification and data from a low-level vibration pre-test and are therefore much simpler to develop than previously described force limits based on analytical models and measurements of the mounting structure mechanical impedance. However, some caution and reference to validation data for similar configurations are essential in order to properly employ semi-empirical force limits.

3. With piezo-electric, tri-axial force transducers the acceleration of the center-of-gravity (CG) maybe accurately measured in vibration tests, via Newton's second law. By contrast, it is very difficult to measure the true CC acceleration of a deformable body with accelerometers. This capability to measure the true CG acceleration with force transducers provides a convenient means of limiting the quasi-static acceleration, commonly used for design purposes, to some fraction of the design limit in a vibration test.

4. The vertical-axis, random vibration test of the Cassini spacecraft was conducted using only the total vertical force for notching. The force limits were derived using the semi-empirical method together with an extensive pre-test analysis and consideration of the shaker and amplifier capabilities. The force specification in the test procedure was used without any modifications during the test. The flight limit loads were achieved at a number of critical locations on the spacecraft, and the instrument responses were similar to those in the component random vibration tests.

8. ACKNOWLEDGMENTS

The work described in this paper was carried out by the Jet Propulsion Laboratory (JPL), California Institute of Technology under a contract with the National Aeronautics and Space Administration (NASA). The many contributions of the personnel in the Cassini Spacecraft Project, the JPL Environmental Test Laboratory, the JPL Dynamics Environments Group and its supervisor, Dennis Kern, are appreciated.

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