N88-18952

Design and Dynamic Testing of an Instrumented Spacecraft Component

Thomas J. Dwyer

This paper presents details relating to the design and subsequent vibration and shock tests of a spacecraft component for a severe vibration and shock environment. The design process and analysis method involves the use of finite element analysis coupled with the Modal Strain Energy method with Risk Graphs to determine and adequacy of the design. The vibration levels experienced by box parts are reduced by the application of passive constrained layer viscoelastic damping treatments that significantly improve component reliability. All significant internal components were fully instrumented in both the random vibration and shock tests, the latter being done on a Mechanical Impact Pyro Simulator now in use at General Electric. In addition information is presented detailing successful testing of another component with the General Electric shock facility and resultant responses are compared with another shock-generating assembly that critically damaged the unit. Correlation between analysis and test data is good, validating the modeling and analysis techniques.

INTRODUCTION

This paper presents details relating to the design and subsequent vibration and shock tests of a spacecraft component for a severe vibration and shock environment. The design process and analysis method involves the use of finite element analysis coupled with the Modal Strain Energy method with Risk Graphs to determine the adequacy of the design. The vibration levels experienced by box parts are reduced by the application of passive constrained layer viscoelastic damping treatments that significantly improve component reliability. All significant internal components were fully instrumented in both the random vibration and shock tests, the latter being done on a Mechanical Impact Pyro Simulator now in use at General Electric. In addition information is presented detailing successful testing of another component with the General Electric shock facility and resultant responses are compared with another shock-generating assembly that critically damaged the unit. Correlation between analysis and test data is good, validating the modeling and analysis techniques.

DISCUSSION

The black box discussed (thought to be a typical black for Aerospace applications) consists of multiple mechanical and electrical components coupled with numerous Printed Wire Boards (PWBs) supported within a rigid container. The PWBs generally contain the most vibration sensitive parts (such as relays) and hence are candidates for vibration attenuation efforts. The PWBs are structurally tied to the box by guide rails, multi-pin electrical connectors and support brackets. PWB weights are generally under 2 pounds, and board surface areas are under 80 square inches.

A significant number of spacecraft and component anomalies have been attributed to the launch vibration environment (Figure 1). In addition vibration is a major cause of failures occurring during ground environmental testing of spacecraft, components, and subsystems. This trend is coupled with increasingly severe vibration and acoustic environments as shown by Figures 2 and 3 and corresponding increases in failures with higher vibro-acoustic levels.

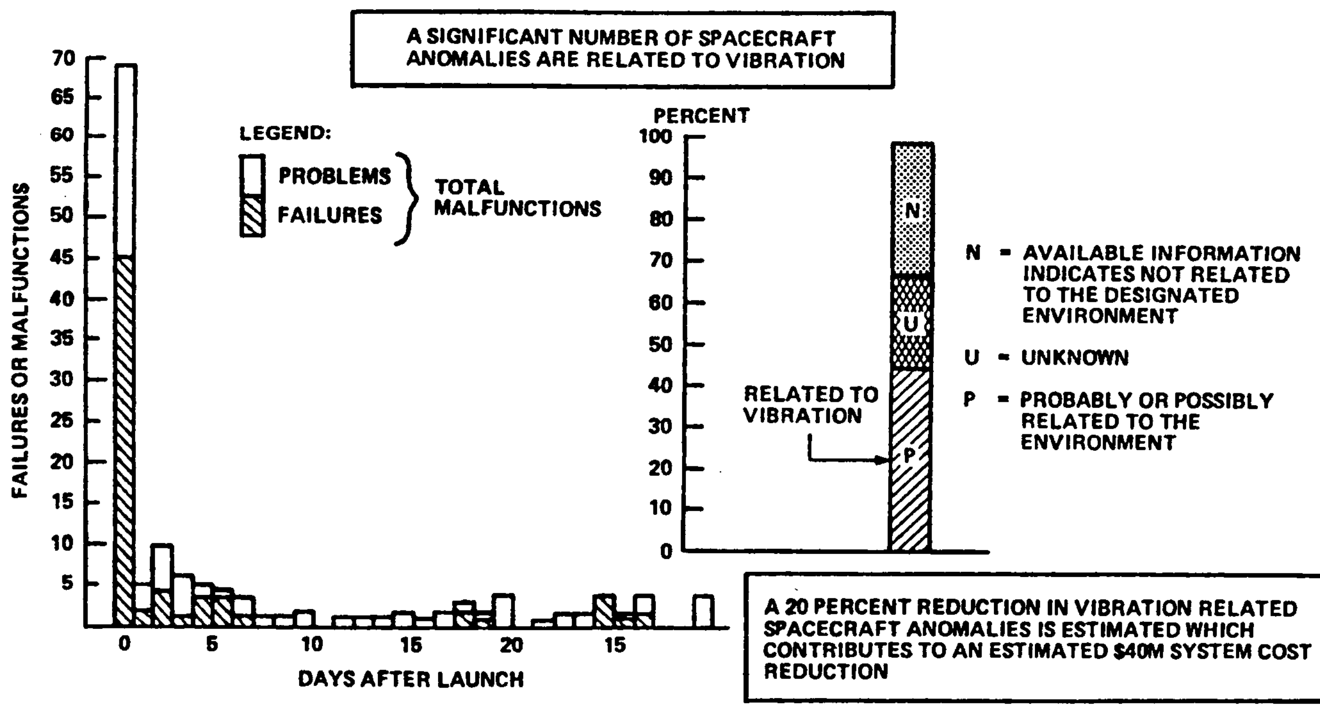

Figure 1 Summary of Early Satellite Anomalies

Integration of passive viscoelastic damping treatments into the design of spacecraft component mounting structures (including PWBs) significantly improves spacecraft reliability and effectively reduces the trend of increased vibration severity. An additional benefit is increased damping in orbit which reduces response to onboard disturbances. Constrained Layer Damping Assemblies (CLDAs) are typically applied in strips running lengthwise across the board with a Viscoelastic Material (VEM) sandwiched between a stiff constraining layer and the

c - 2

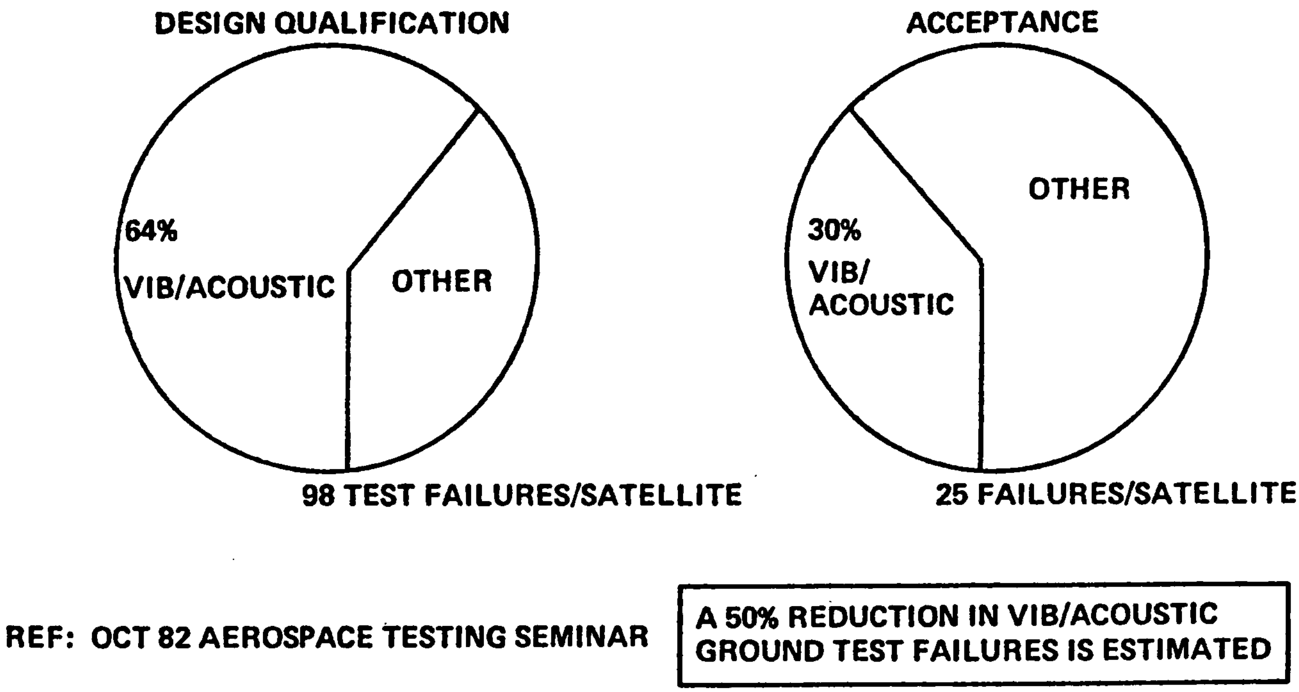

Figure 2 Summary of Vibroacoustic Test Failures

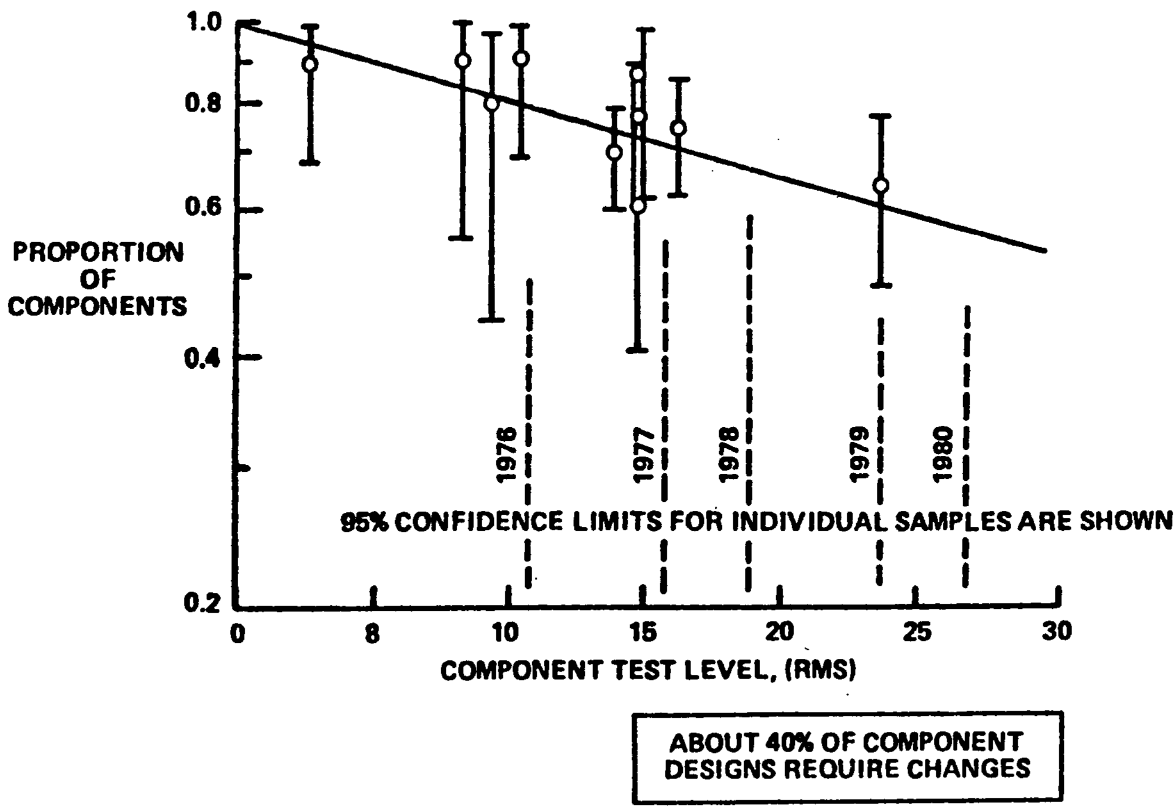

Figure 3 Components Passing Vibration Tests

board surface. The CLDA is placed to maximize the strain energy in the VEM, although this is not always possible due to PWB component mounting. All PWBs within the box have CLDAs, as well as several other critical box surfaces. A typical PWB with CLDA is shown in Figure 4.

To determine the dynamic adequacy of the PWB designs, Finite Element Models (FEMs) of the integrally damped PWBs are constructed (see Figure 5) in which the viscoelastic material is represented by finite element solid elements and the base and upper constraining layer with shell elements with offsets. A standard modal extraction run is executed and the Modal Strain Energy Method used to determine the modal and damping characteristics of the boards. This method is based on the principle that the ratio of composite structural loss factor to viscoelastic material loss factor for a given mode of vibration can be estimated as the ratio of elastic strain energy in the viscoelastic material to the total elastic strain energy in the entire structure when it deforms into the particular undamped mode shape. This ratio multiplied by the viscoelastic material loss factor yields the modal loss factor.

The analysis results are used to determine the dynamic adequacy and effectiveness using a Risk Graph (Figure 6), a system that produces a design classification as a function of fundamental frequency, damping, board characteristics, and the random vibration environment. Basically, for a given random vibration environment, regions of variable amounts of design risks are generated that are based on both experience and analysis considering maximum deflection lines. Approximate response of a single degree of freedom oscillator to an acceleration Power Spectral Density (PSD) which is constant at all frequencies is calculated and used as a basis upon which to classify the design effectiveness. A second criteria used in a Risk Graph is the acceleration, since the Grms response is an indication of the overall severity of the environment that the piece-parts (diodes, crystals, resistors, relays) must endure. A 30 Grms response is recommended as a design goal for all boards although 50 Grms designs have been employed in cases where the boards do not have vibration-sensitive items (such as relays) in the most severe vibration axis and where spacing and other parameters limit damper applications. The final boundary is based on the amount of structural damping predicted by an analysis method, currently the MSE approach. This method has been employed effectively and successfully in numerous General Electric spacecraft and development programs.

The box successfully passed a fully instrumented vibration test with no vibration-related anomalies. The random vibration testing employed standard electrodynamic shakers and the shock testing was done on a Mechanical Impulse Pyro Simulator (MIPS) in limited use throughout the industry. Figure 7 illustrates the MIPS test setup used for component shock testing at General Electric.

The box was tested to random vibration levels of 18 Grms in one axis and approximately 13 Grms in the remaining two axes with no vibration-related anomalies related to the vibration testing validating both the passive damping treatment and the analysis method. All boards within the box as well as various other critical locations both in and on the container were instrumented with microminiature accelerometers during the test, and data from these accelerometers are summarized in Table 1. The accelerometers were located in the area of the maximum response. Measured Composite Loss Factors were obtained using circle fits to the high level random response data and are on the order of 0.25 for all boards except one. The fundamental board resonances are in the 130 to 250 Hz range and did not couple with the container resonance which is above 400 Hz. It is clear that the passive damping treatment, which is introduced with a minimum weight impact to the component, provides an effective method of reducing the board response to the vibro-acoustic energy and hence greatly reduces the chances of vibro-acoustic related failures. In addition because of the excellent correlation

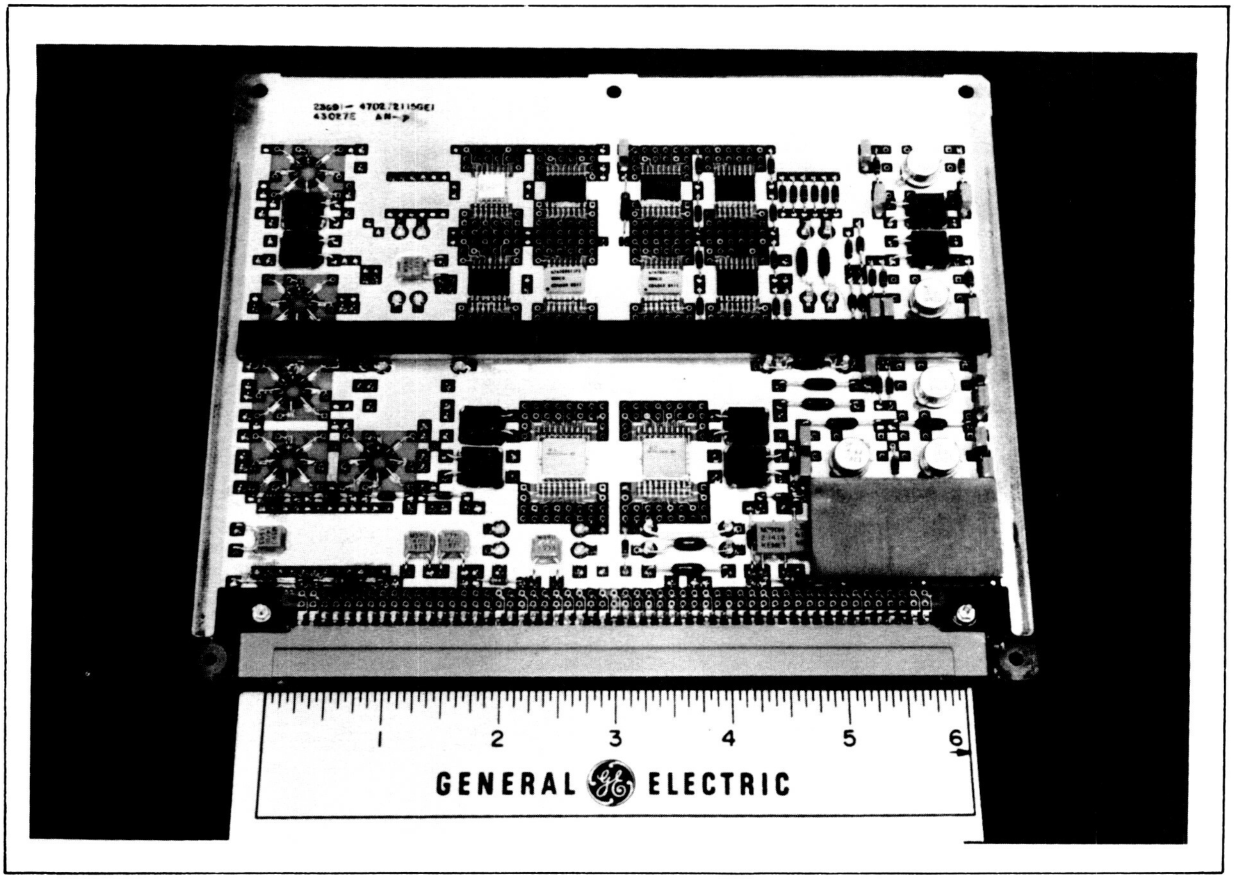

Figure 4 Typical Damped PWB with Damper shown on Board Center

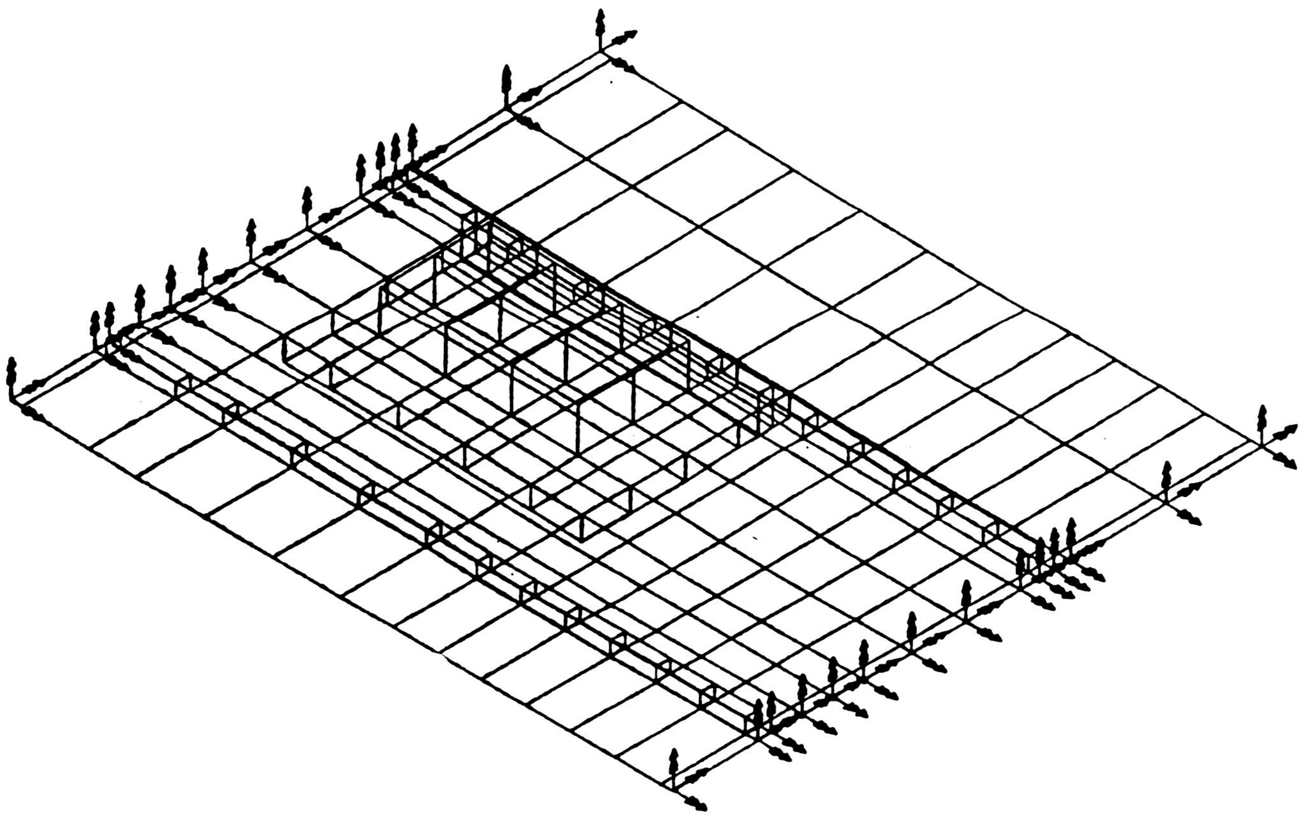

Figure 5 Typical PWB FEM with Tantelum Shield Representation

ORIGINAL PAGE IS
OF POOR QUALITY

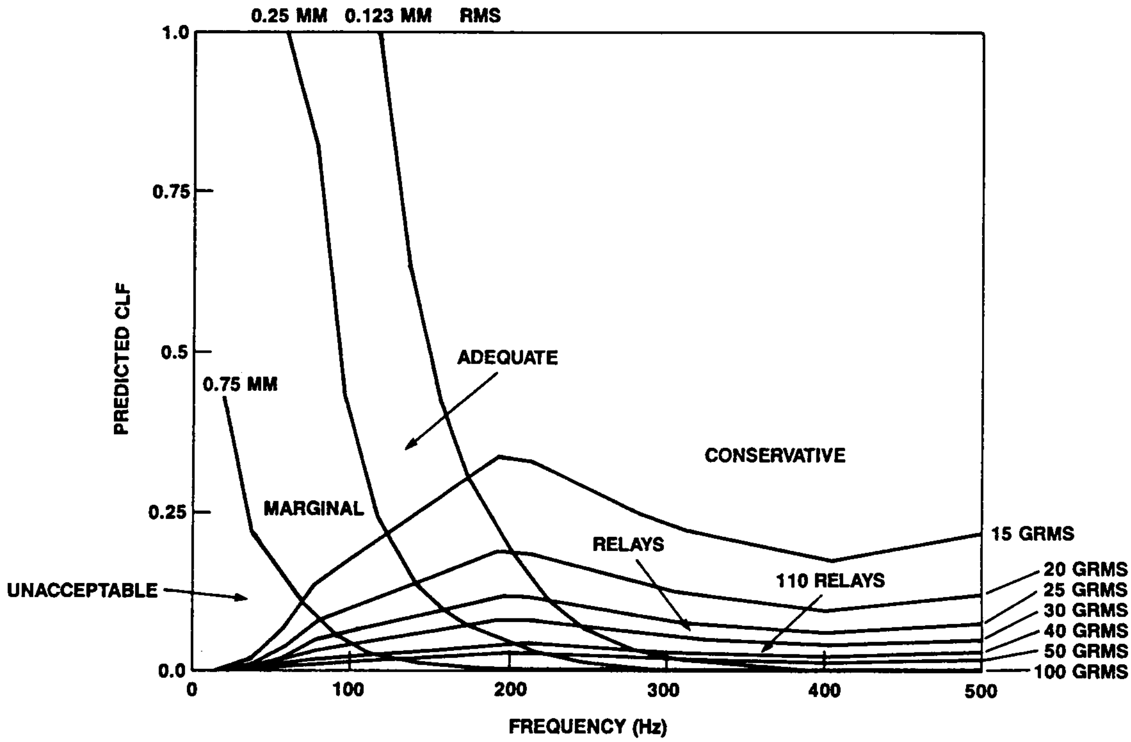

Figure 6 Risk Graph used to Classify Design Adequacy

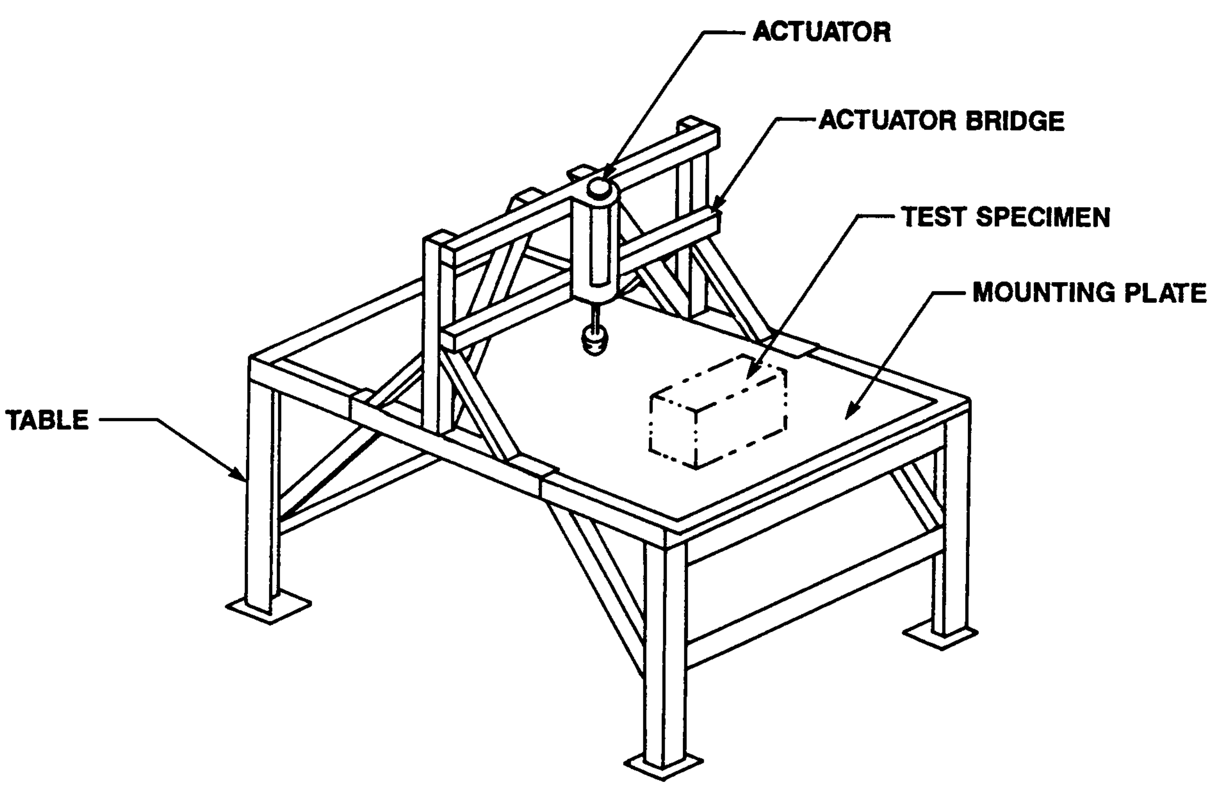

Figure 7 GE MIPS Facility

Table 1 Comparison between Vibration Test and Analysis Results

PWB	TEST F_1	TEST η_1	TEST GRMS	ANALYSIS F_1	ANALYSIS η_1	F_1 $\frac{\text{ANALYSIS}}{\text{TEST}}$	η_1 $\frac{\text{ANALYSIS}}{\text{TEST}}$	RISK GRAPH CLASSIFICATION TEST	ANALYSIS
5	189	.37	31.4	156	.31	.83	.84	CONSERVATIVE	ADEQUATE
6	178	.31	33.5	187	.32	1.05	1.03	CONSERVATIVE	CONSERVATIVE
7	202	.22	33.6	147	.22	.73	1.00	CONSERVATIVE	ADEQUATE
8	144	.25	28.6	153	.34	1.06	1.36	ADEQUATE	ADEQUATE
9	226	.23	35.6	229	.20	1.01	.87	CONSERVATIVE	CONSERVATIVE
10	243	.24	37.1	290	.19	1.19	.79	CONSERVATIVE	CONSERVATIVE
12	173	.13	27.4	147	.20	.85	1.54	ADEQUATE	ADEQUATE
13	127	.26	20.2	123	.21	.97	.81	ADEQUATE	ADEQUATE

between test results and analysis predictions, the MSE approach is a reliable method of predicting PWB performance in what is thought to be a typical vibration environment. Responses in this box are thought to be typical for similar applications throughout the industry and can be scaled for other random environments by analysts wanting to determine PWB response characteristics for their application.

The peak Shock Response Spectrum (SRS) that the box was exposed to was approximately 12000 g and represents the input at the interface with the table mounting surface. The data summarized in Table 2 provides a definition of the environment within the box resulting from the MIPS simulation of the pyrotechnic environment. It indicates relatively high SRS levels for parts mounted directly to the PWBs - on the order of 1000 to 2000 g. The PWBs do provide a significant attenuation of the MIPS plate environment (2000 g or less for the 12000 g input) and the component structural environment as measured by wall response (3000 to 7000 g). This attenuation is expected in view of the relatively low resonant frequencies of the PWBs. The PWB having the lowest resonant frequency is PWB 13 while PWB 10 has the highest resonant frequency. As one would expect, the PWB 13 SRS is significantly less than PWB 10 SRS. Again responses in this box are thought to be typical for similar applications throughout the industry and can be scaled for other shock environments by analysts wanting to determine PWB response characteristics for their application.

Another example of a MIPS component illustrated in Figure 8 which sustained catastrophic internal damage and mounting feet deformation during prior pyro shock testing of the unit at another facility. That facility (Figure 9) is quite different that the MIPS facility thus affecting the path taken by the shock wave from the impact point to the unit. In addition in the previous test the flight unit itself was used during system calibration with some 67 hits made during the calibration resulting in probable unit over-testing. To resolve the question of whether the component failures resulted from being over-tested/over-exposed or whether the design was susceptible to normal-axis shock, another unit was tested at the General Electric MIPS facility.

The unit was exposed to an approximate SRS peak input of 5000 g in both tests. However, upon examination of the test data presented in Table 3, it is clear that the component responses were grossly different between the two test methods. For an approximate normal axis input of 3300 g, responses at the top of

Table 2 Summary of Shock Response Spectra Maximum Values and Frequency

PWB	DESCRIPTION	RUN 16 (Y-X)		RUN 17 (Y-X)		RUN 20 (Y-Z)		RUN 22 (Y-Z)	
		PEAK SRS (G)	FREQ (Hz)	PEAK SRS (G)	FREQ (Hz)	PEAK SRS (G)	FREQ (Hz)	PEAK SRS (G)	FREQ (Hz)
	INPUT-X	12930	5079	12623	5079	1706	8063	1808	9050
	INPUT-Y	6083*	8063	6131*	8063	6314*	2850	6280*	3200
	INPUT-Z	1694	4525	1902	8063	10629	5701	10560	5701
	PLATE CORNER (X)	3406	5701	3363	5701	4728	5701	4623	5701
	PLATE CORNER (Y)	5921	5079	5761	4525	5522	6400	5524	5701
	PLATE CORNER (Z)	4341	3200	4179	6400	3147	5701	3289	5701
5	PWB	873	1131	928	1131	393	5079	458	5079
6	PWB	3771*	4031					557	5701
7	PWB					763	897	650	4031
8	PWB	1086	1131	1280	1131	1207*	3591	1371	3591
9	PWB	1514	1600	1654	1269	938	3200	1043	6400
10	PWB	980	1795	1248	800	1771	10159	1869	10159
11	PWB								
12	PWB	1174	5701	856	2539	1918	4031	2007	5079
13	PWB	427	448	650	503	532	2015	1096	1600

*REFERENCE SHIFT IN SIGNAL

the component for the MIPS test were on the order of 2500 g, whereas responses with the Impact Facility illustrated by Figure 9 were some 12000 g. It is obvious that the unit as tested on the MIPS assembly experienced grossly lower response levels than experienced on the other test apparatus. The primary differences are due to the path the energy takes before reaching the component and a 2000 Hz component resonance that was magnified by a system resonance with the Impact Facility in the same frequency band. It is concluded that the MIPS set-up is a better representation of the actual environment that the unit will experience in flight application and imposes far less structural risk due to the test assembly than the Impact Facility.

CONCLUSIONS

Details have been presented relating to the design and subsequent vibration and shock tests of a spacecraft black box for a severe vibration and shock environment. The design process and analysis method involves the use of FEM coupled with the MSE method with Risk Graphs to determine the adequacy of the design. The vibration levels experienced by box components are reduced by the application of CLDAs which significantly improve component reliability. All significant internal components were fully instrumented in both the random and shock vibration tests, the latter being done on a MIPS now in use at General Electric. In addition information is presented detailing successful testing of another component with the General Electric shock facility and resultant responses are compared with another shock-generating assembly that critically damage the

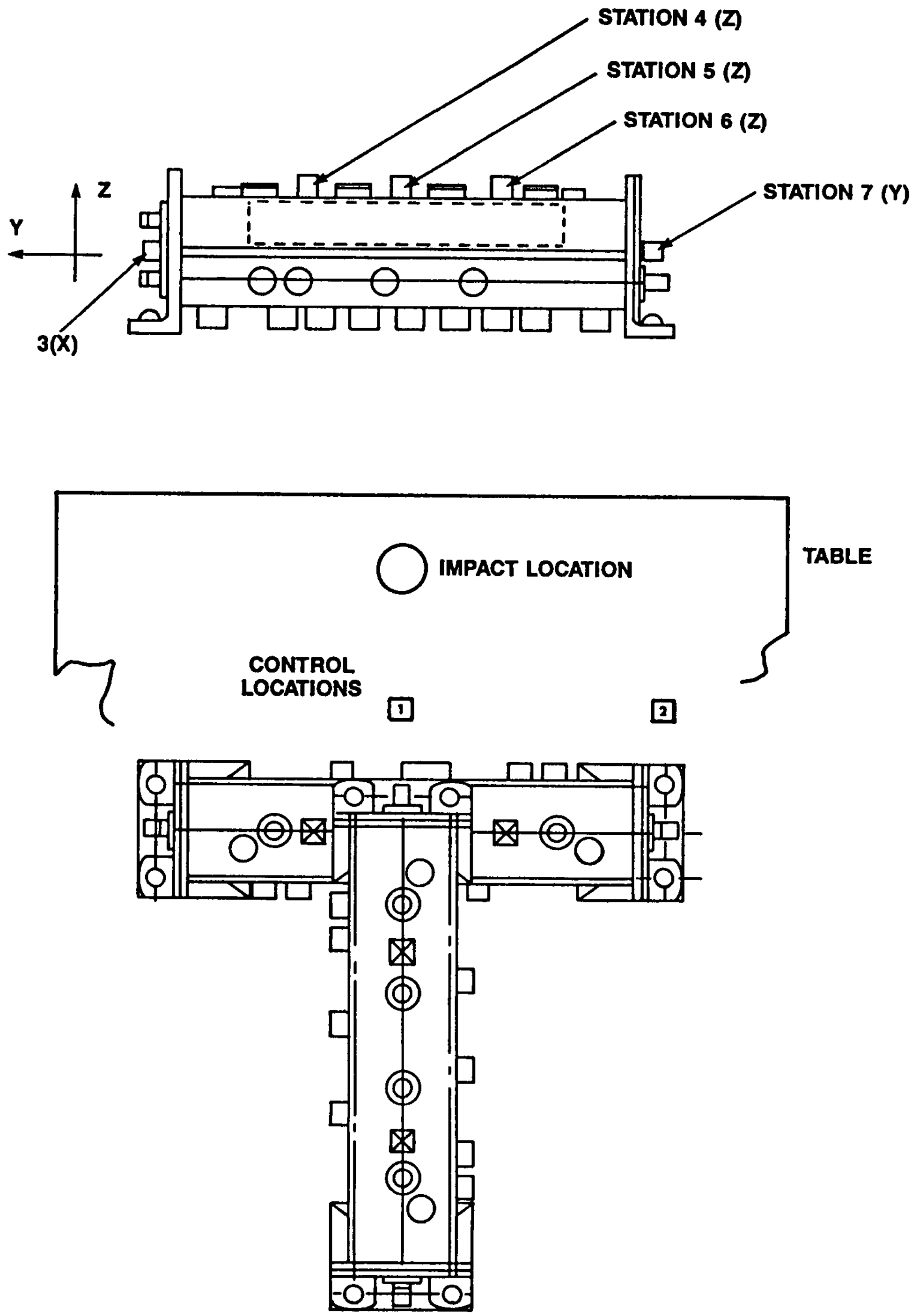

Figure 8 Component Tested on General Electric MIPS Facility

unit. It is sufficient to say that for approximately the same input to the unit, any component response and damage potential is significantly reduced with a MIPS shock test assembly. Responses documented for this vibration environment are thought to be typical for similar applications throughout the industry and can be scaled for other vibration environments by analysts wanting to determine PWB characteristics for their application.

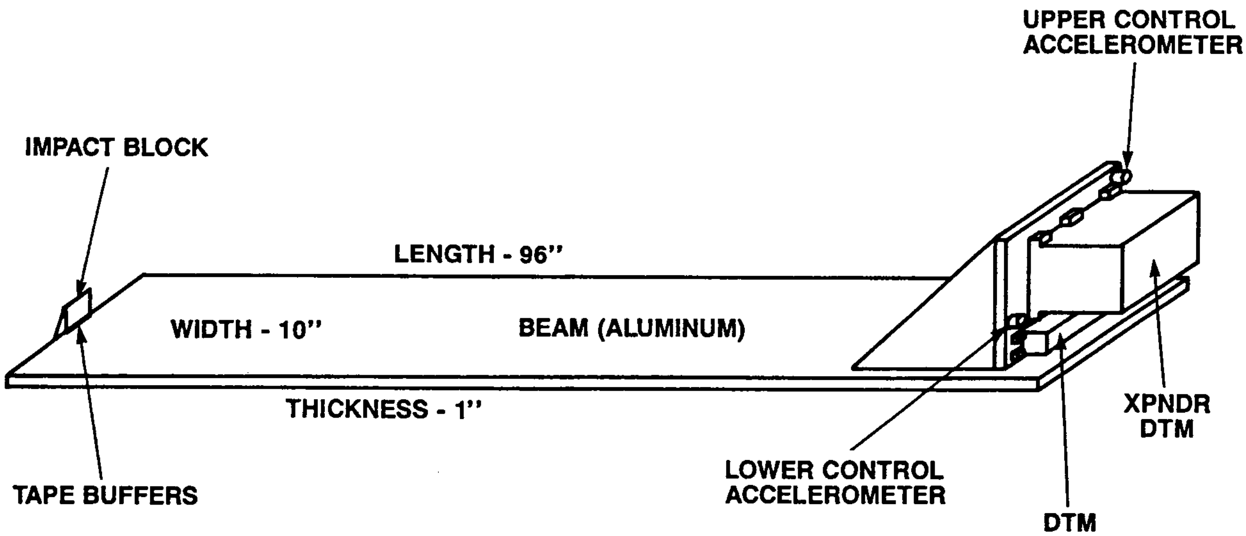

Figure 9 Test Assembly that Damaged Component

Table 3 Time History Summary

RUN #6 YZ			
DESCRIPTION	STATION	MIPS TIME HISTORY PEAK G'S O-P	IMPACT TIME HISTORY PEAK G'S O-P
CONTROL	1X	800	--
CONTROL	1Y	4900	--
CONTROL	1Z	3400	3363
CONTROL	2Z	1900	--
SIDE NEAR J1/J2	3X	500	--
TOP END	4Z	3400	(5443)*
TOP CENTER	5Z	3300	11954
TOP END	6Z	2900	(5443)*
SIDE NEAR J3	7Y	1800	--

RUN #7 XZ			
DESCRIPTION	STATION	MIPS TIME HISTORY PEAK G'S O-P	IMPACT TIME HISTORY PEAK G'S O-P
CONTROL	1X	3400	--
CONTROL	1Y	800	--
CONTROL	1Z	3200	3363
CONTROL	2Z	1700	--
SIDE NEAR J1/J2	3X	1100	--
TOP END	4Z	2900	(5443)*
TOP CENTER	5Z	2500	11954
TOP END	6Z	2300	(5443)*
SIDE NEAR J3	7Y	850	--

*MEASURED IN Z DIRECTION ON THE SIDE OF UNIT