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Resonant plate shock test response with isolated damping bars

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Abstract

Resonant plate and other resonant fixture shock techniques were developed in the 1980s at Sandia National Laboratories as a flexible, repeatable, and safe technique to simulate mid-field pyroshock for component qualification. Since that time, many qualification shocks have been specified with these techniques in mind. To control the effective time duration of the simulated shock events, resonant plate techniques require some mechanism to damp the motion of the plate. Damping bars have been used in different configurations to achieve the decay rates required, with varying levels of success. This work will characterize the effect of different isolated damping bar configurations on the response of a resonant plate during a mid-field pyroshock simulation test.

1 Introduction

Satellites, space vehicles, spaceflight, and space launch vehicles are an exciting part of human technical achievement. Advances in spaceflight technology were made possible with great effort on the part of engineers to understand and ruggedize space hardware to the severe environments of launch and space. In the 1960s and early 1970s investigation into system and component failures of spaceflight hardware identified pyroshock events, such as exploding bolts and other separation hardware, as a significant factor in space component electrical system anomalies. In response to these discoveries, ground test methodologies were developed to attempt to simulate launch and flight pyroshock environments.

As the science around pyroshock matured, pyroshock events and the corresponding ground test methodologies began to be categorized as near-field, mid-field, and far-field. According to guidance published by IEST[1], near-field pyroshock is generally characterized by very short acceleration rise times, dominant frequencies above 10,000 Hz, and acceleration levels exceeding ~100,000 m/s2. Simulating a near-field pyroshock often requires the use of pyrotechnic excitation. Mid-field pyroshock events exhibit slightly slower rise times, dominant frequencies from 3000-10,000 Hz, and acceleration amplitudes less than ~100,000 m/s2. Far-field pyroshock contains frequencies below 3000 Hz and acceleration amplitudes less than ~10,000 m/s2. Mid- and far-field pyroshock events can be simulated via mechanical excitation. Accelerometers specially developed for shock testing are used to measure acceleration during these shock events. Analysis of a shock measurement utilizes the Shock Response Spectrum (SRS).

One of the mechanical excitation test methodologies pursued and extensively developed at Sandia National Laboratories is the resonant fixture approach[2]. In this approach, a component to be evaluated is attached to a fixture—usually a plate, bar, or beam—whose dimensions are selected such that the fixture will have a fundamental resonance at a desired frequency. For example, if a 1000 Hz resonant frequency response is desired from a square plate, and we assume the plate will be struck in the center so that the third mode (the "breathing mode") will be excited, a handbook [3] can be consulted to estimate the dimensions required; in this case a 508 x 508 x 51 mm square plate will provide a 1000 Hz mode of the desired shape. The resonant fixture is struck by a projectile at a location and direction that will strongly excite the desired resonance, resulting in a uniaxial, transient acceleration event that is dominated by the desired resonant frequency. It was found that testing conducted in this manner could achieve mid- and far-field shock levels, was repeatable, and could be set up and executed quickly by shock test practitioners. The acceleration amplitude response is easily controlled by a combination of projectile velocity and "programmer" material, such as felt or paper positioned between the projectile and resonant fixture, modifying the input force pulse applied by the projectile. Resonant fixtures are commonly constructed of aluminum, exhibiting long ring-down times due to low damping. To better simulate many environments, shorter ring-down times are achieved by clamping or bolting smaller plates or bars to the resonant fixture. Originally these "damping bars" were bolted directly to the resonant fixture, or perhaps clamped with a paper gasket or lead sheet in between, which disrupted the resonant fixture motion; however, the mechanism that led to increased damping was not understood. Later, rubber sheet was used between the fixture and damping bar to similar effect. Finally, to reduce unwanted high-frequency fixture response induced by the metal-to-metal interfaces in the bolted damping bar system, a fully isolated damping bar approach was developed to eliminate the metal-to-metal interface between the damping bars, bolts, and resonant fixture[4]. An example of a resonant plate shock test assembly is shown in Figure 1.

Figure 1: Preparation of a resonant plate for shock testing. The plate is suspended by rope. A test article can be attached to the front of the plate. The air gun is visible behind the plate.

Achieving a controlled shock acceleration response in more than one axis has become a more recent goal of shock practitioners. One way to achieve this is to extend the resonant fixture technique, utilizing more sophisticated geometries with structural dynamics response in more than one axis to achieve desired shock levels. This approach demands a model based design leveraging computer aided tools such as finite element analysis. The work in this paper was motivated by a desire to provide experimental data to improve resonant plate structural dynamics models, thus enabling the extension of resonant fixture techniques. Experimental activities included experimental modal analysis of a resonant plate with and without damping bars installed, resonant fixture shock tests on the same configurations, estimation of the force due to projectile impact using the SWAT-TEEM technique (Sum of Weighted Accelerations Technique-Time Eliminated Elastic Motion) [5], and operating deflection shape estimation of the plate with damping bars during a shock test. During experimentation, a modification to the isolated damping bar method for resonant plate was envisioned, fabricated, and evaluated with a series of shock tests.

2 Resonant plate experimental modal analysis

2.1 Resonant plate configurations and experimental approach

During shock testing at Sandia National Laboratories, resonant plates are suspended from a frame by rope, with rope eyelets attached at the plate perimeter. The suspension frame holding a resonant plate is clamped to the frame of a gas gun, in front of the muzzle. There are several suspension frames, made to be quickly interchangeable before the gas gun, with each frame holding a resonant plate of different dimensions (to respond at a different natural frequency). An aluminum impact pad, 51 x 51 x 102 mm, is attached to the center of the plate and is struck by the steel projectile from the gas gun. Over time and after several shock tests, the impact pad is deformed and is replaced at the discretion of the shock practitioner.

The configuration and suspension of the resonant plates during the experimental modal testing was the same as in the shock test configuration. For the initial modal analysis and shock experiments, a plate with a responding frequency of approximately 1000 Hz was selected, with dimensions 514 x 514 x 51 mm. Four aluminum damping bars with dimensions 51 x 51 x 514 mm are bolted to the plate, with a layer of neoprene rubber between each bar and the plate. The bolts proceed through the damping bar and plate, then through the damping bar on the back side of the plate. Thin plastic sleeves prevent the bolts from contacting the plate. Two or three steel rope eyelets attach around each edge of the plate. The suspension of the plate by ropes around the four sides of the plate resulted in plate boundary conditions that were nearly free-free. Two 19 mm threaded rods are installed on the damping bars to enable the shock test operator to level the plate with two loops of cord. See the illustration in Figure 2.

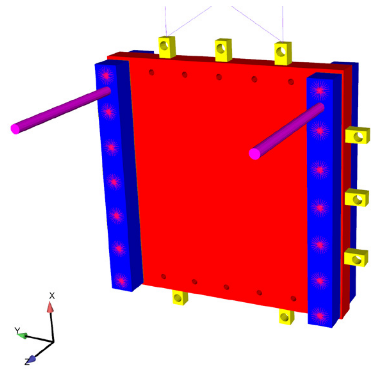


Figure 2: Resonant plate assembly including damping bars.

Experimental modal analysis of the assembly with and without damping bars installed was performed with impact hammers, accelerometers, and a scanning laser vibrometer. Driving points were located at a corner of the plate and additionally at a corner of one damping bar for the damping bar configuration. Natural frequencies, damping ratios, and mode shapes were extracted up to 3.5 kHz. For this paper, the response of interest only extends up to the first three modes with significant plate response. Frequency and damping for the first several modes of each case are listed in Table 1, and the mode shapes of modes with significant plate participation are shown in Figure 3[6].

2.2 Resonant plate without damping bars

Inspecting the mode shapes, it is noted for the configuration without damping bars that a projectile impact in the center of the plate will minimally excite the first two modes (twist and saddle modes) since the impact location point is on a node line. The third mode (breathing) should be strongly excited since the impact location is at a point with high participation for that mode.

2.3 Resonant plate with isolated damping bars

For the configuration with damping bars, the addition of the damping bar assemblies causes the twist mode to drop in frequency. The second plate mode now appears as a bending mode about the X-axis (mode 3 in Table 1) and has participation at the projectile impact location. The third plate mode (mode 9 in Table 1) appears as a bending mode about the Y-axis. It is interesting that the natural frequency of the third plate mode is nearly the same (within 3%) for both configurations; 1020 Hz without and 1002 Hz with damping bars.

Table 1: Assembly modal test parameters.

assembly without damping bars mode number and description	Freq. (Hz)	ζ (%)	assembly with damping bars mode number and description	Freq. (Hz)	ζ (%)
1. twist	545	0.09	1. rod bending	161	0.02
2. saddle	790	0.27	2. assembly twist	391	0.45
3. breathing	1020	0.17	3. assembly bending about X	582	1.4
4. 2nd twist	1350	0.23	4. damping bar rotation about Z	768	6.1
			5. damping bar rotation about Z	828	4.8
			6. rod bending	923	1.2
			7. rod bending	957	1.2
			8. assy. bending about Y w/bar rotation	989	2.6
			9. assy. bending about Y	1002	2.6
			10. rod bending	1040	2.3
			11. assembly 2nd twist	1288	2.5
			12. damping bar bending about Z	1378	1.7
			13. damping bar bending about Z	1431	3.4

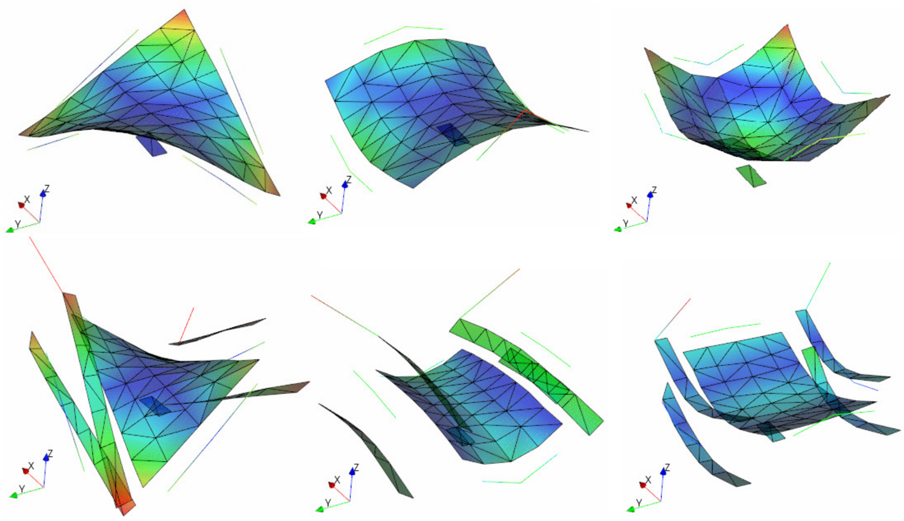


Figure 3: Modes 1, 2, 3 of the plate without damping bars (top), and modes 2, 3, 9 of the plate with damping bars (bottom.)

During modal testing, to characterize the plate response linearity to force input, hammer impacts of increasing force levels were applied. Force pulses applied in the center of the plate were varied from 200 N to 35,000 N. When observing the FRFs generated from these inputs, it was found that for impacts of 2800 N or less, the response of mode 3 (bending about the X-axis) appeared proportional to the input. As impacts greater than 7500 N were applied, it was observed that the response of mode 3 was no longer proportional to the input; increased force input resulted in progressively reduced FRF response. In contrast, the response of mode 9 (bending about the Y-axis) was proportional to input for the entire range of force impacts.

3 Resonant plate shock testing

3.1 Damping bar effect on shock response

Several series of resonant plate shock tests were performed to generate data for finite element model validation. In addition to damping bar configuration, additional test parameters adjusted included the amount of felt programmer material, the projectile velocity, and the projectile mass. These parameters are routinely adjusted by shock practitioners during the shock testing process to generate an acceleration event that will meet some test specification.

To demonstrate the effect of damping bars on the resonant plate shock response, the results of two of these shock tests are presented. Both tests had nearly identical test configurations: the plate was struck with a 10 kg steel projectile having an impact velocity of 8.8 m/sec. A 13 mm thick felt programmer was placed between the projectile and the plate impact pad. The shock acceleration response was measured with Endevco 7270A-20K shock accelerometers perpendicular to the plate (+Z direction) at 29 locations on the plate surface. The response at the location nearest the center of the plate is shown in Figure 4. Without damping bars installed, the resonant plate event has a duration of approximately 150 msec. With damping bars installed, the event duration is reduced to less than 10 msec, or approximately 8 cycles at the plate bending mode about Y (~1000 Hz.)

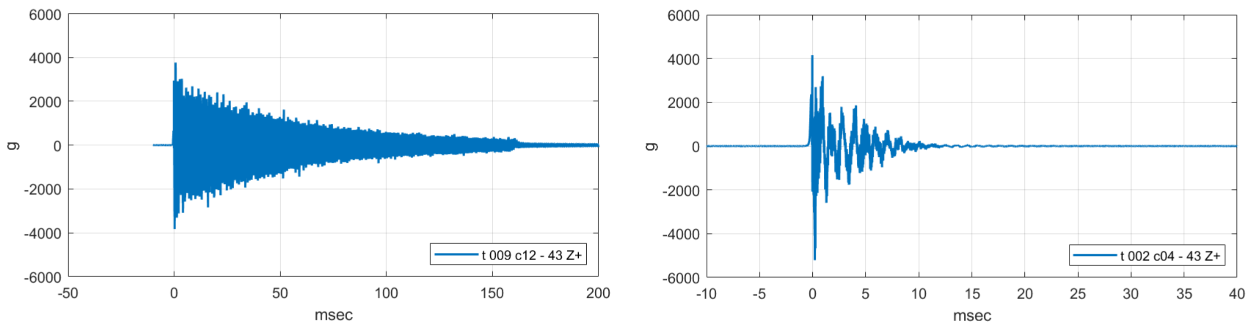


Figure 4: Resonant plate acceleration without damping bars (left) and with damping bars installed (right).

The FFT and shock response spectra calculated from the data are shown in Figure 5. The increase in damping provided by the damping bars (solid line) is evident in the reduced magnitude of peaks in both FFT and SRS. Also evident for the case with damping bars is the presence of the lower frequency bending mode about X (mode 3 in Table 1), at approximately 600 – 700 Hz.

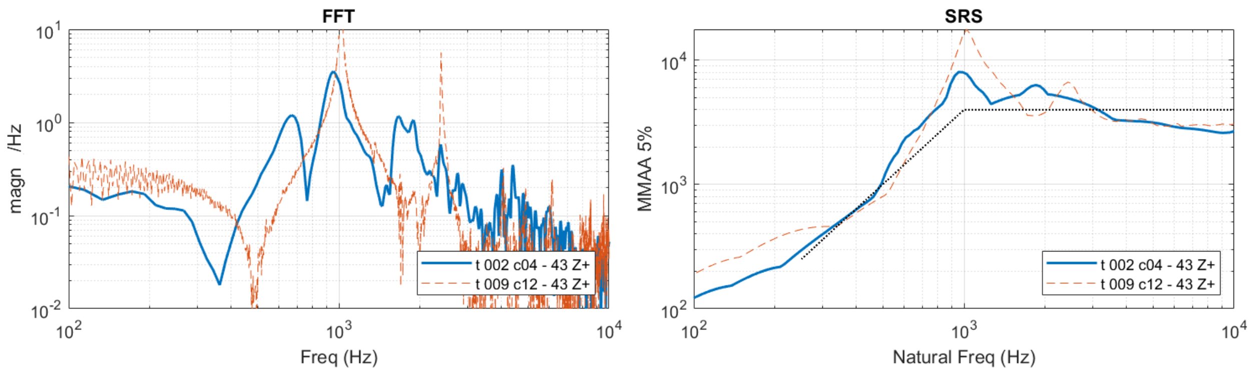


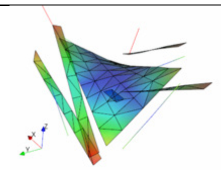
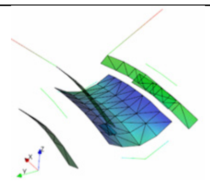
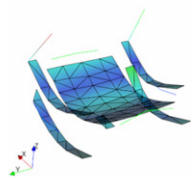
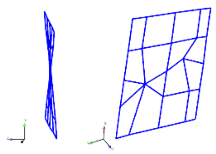
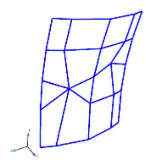
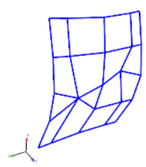
Figure 5: FFT and SRS without damping bars (dash) and with damping bars installed (solid).

3.2 Comparison of shock response to experimental modal analysis

Experimental modal analysis is often performed on mechanical systems at low levels of excitation that are appropriate for understanding vibration environments such as transportation vibration or flight vibration. Structural dynamic response remains nearly linear at such loads. The high force levels required for mechanical shock excitation are known to create significant nonlinearities in many assemblies. Performing experimental modal analysis at shock excitation levels is challenging due to the difficulty in measuring the substantial excitation force applied to the test assembly. In lieu of performing an experimental modal analysis, in this work other methods were used to estimate forces and approximate the relevant modal parameters for the resonant plate with damping bars.

The SWAT-TEEM method was used to estimate the force pulse that the projectile imposed on the resonant plate. SWAT-TEEM utilizes the free decayed time response from the shock test acceleration measurements to estimate the force applied to the test assembly. This approach enabled the modeling team to develop a realistic range of force input pulses—amplitudes and durations—for use when modeling the shock. The projectile force pulse peaks estimated during shock testing were on the order of a hundred times those measured with the impact hammer during modal testing (~50,000 N vs. ~500 N.) Experimental modal analysis was performed utilizing estimated forces and accelerations measured on the instrumented resonant plate to estimate mode shapes, natural frequencies, and modal damping[7]. These results are compared to the experimental modal analysis results for the most important plate modes in Table 2. The twist mode shows the least difference between the two tests; the frequency drops 3%, and the damping is unchanged at 0.4%. The bending mode about X is significantly different, with frequency increasing 20%, and damping increasing from 1.4% to 9.6%. Since the frequency of this mode is well below the desired plate frequency, the significant damping increase is beneficial and ensures the response of this undesirable mode is very low during the shock test. This result agrees with the linearity evaluation performed during modal testing. The bending mode about Y is reduced in frequency 5% while the damping is doubled, from 2.6% to 5%. This damping increase was not indicated during the modal test linearity evaluation. Mode shapes derived from the experimental modal analysis at shock levels were like those extracted during the traditional experimental modal analysis; however, no attempt was made to quantify the similarity.

Table 2: Comparison of experimental modal and shock test response

	Peak Input Force	Twist mode		Bending about X		Bending about Y	
Experimental modal	~500 N						
		391 Hz	ζ=0.4%	582 Hz	ζ=1.4%	1002 Hz	ζ=2.5%
Shock test	~50,000 N						
		379 Hz	ζ=0.4%	697 Hz	ζ=9.6%	953 Hz	ζ=5.0%

3.3 Damping bar configuration study

Modal analysis of the resonant plate with damping bars demonstrated that a second, potentially unwanted responding mode at a frequency lower than the plate "design frequency" is present when damping bars are installed on two sides of the plate. On the 1000 Hz plate, the high damping of the additional mode reduced response enough to allow the bending mode at the design frequency to dominate. Based on shock testing experience on other plates, it was known that a 500 Hz plate of dimensions 514 x 514 x 25 mm exhibited undesirable response at a frequency lower than designed when 51 x 51 x 514 mm damping bars were installed on two sides. As a potential mitigation for this issue, evaluation of circumferential damping bar configurations was proposed, with damping bars to be attached on four edges of the plate. Photos of the test setup are in Figure 6. To investigate the potential effect of damping bar thickness, thinner 25 x 51 mm section damping bars were fabricated.

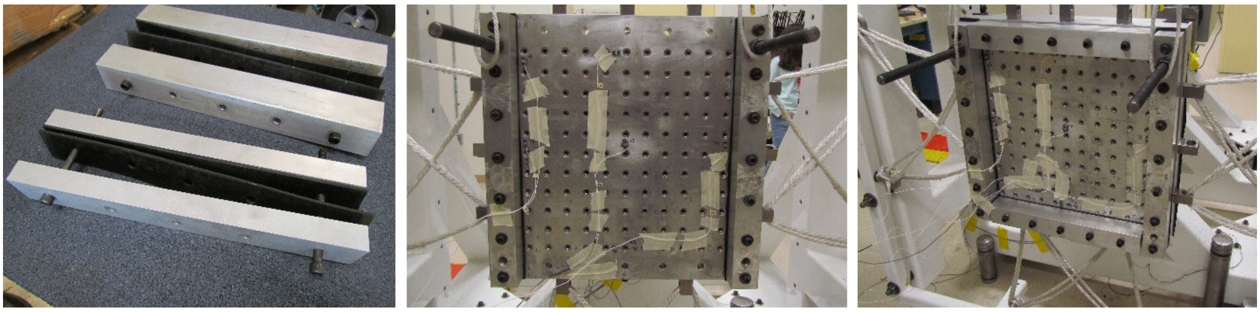


Figure 6: 25 mm and 51 mm thick damping bars (left,) damping bars on two sides (center,) circumferential damping bars (right.)

The damping bar configuration investigation proceeded with factors of 1) plate design: "1000 Hz" and "500 Hz," 2) damping bar configuration: two sides and four sides, and 3) damping bar thickness: 25 x 51 mm and 51 x 51 mm. Acceleration response for each test configuration was measured for three air gun pressures (10.3, 17.2, and 27.6 kPa). Only the results of testing at 17.2 kPa are included in this paper. FFT and SRS were calculated. Responses measured or evaluated include the significant responding frequencies picked from peaks in the FFT magnitude and SRS, damping estimated by the half power point method, and a subjective evaluation of the conformance of the SRS to an expected profile. If the plate natural frequency matched the design frequency within ±15% the configuration was considered successful or a "pass" for frequency. Damping less than or equal to 10% was considered a "pass," since damping above 10% results in too few cycles in the shock ringdown event. The SRS profile used for comparison was based on legacy resonant plate tests and consists of an SRS response with a positive slope of +12dB per octave up to the plate design frequency, and flat (0dB/octave slope) at frequencies higher than the plate design frequency. The profile is shown as a dashed line in the SRS plots. If a configuration produced an SRS response that conformed to this shape within approximately ±6dB, it was considered a "pass" or successful configuration. The time domain acceleration response, FFT, and SRS are shown in Appendix B, Figure 7 to Figure 14. Table 3 contains a summary of factors and responses for the investigation.

Reviewing the results, the key findings of this investigation are: Damping bars installed on four sides appeared to eliminate the bending mode about X on both 500 Hz and 1000 Hz plates. The 1000 Hz plate response was acceptable with both thin and thick damping bars on two sides of the plate. Although a lower frequency mode (bending about the X axis) was present, the response was low enough to enable the higher frequency bending mode about the Y axis to dominate the SRS response. The 1000 Hz plate with damping bars on four sides provided acceptable response for both thin and thick damping bars. The 500 Hz plate with damping bars installed on two sides always exhibited an undesirable modal response. Damping bars on four sides of the 500 Hz plate eliminated the unwanted bending mode; response was acceptable with thin bars, but damping was too high with the thick bars in this configuration. Due to these findings, the use of thin (25 x 51 mm) bars, installed on four sides, has been adopted as a standard procedure for tests utilizing the 500 Hz resonant plate.

Table 3: Damping bar configuration investigation.

Nominal Plate	Bar Thickness	2 sides Frequency Hz	Deviation %	Freq. Pass/Fail	ζ ** %	Profile Pass/Fail	4 sides Frequency Hz	Deviation %	Freq. Pass/Fail	ζ ** %	Profile Pass/Fail
"1000 Hz" 508 x 51mm	51mm	668			6		935	-6.5	Pass	10	Pass
		954*	-4.6	Pass	6	Pass					
	25mm	668			7		847	-10.3	Pass	10	Pass
		916*	-8.4	Pass	4	Pass					
"500 Hz" 508 x 25mm	51mm	277					553	+10.6	Pass	15	Fail
		677*	+35	Fail	N/A	Fail					
	25mm	339*	N/A	Fail	1	Fail	448	-10.4	Pass	2	Pass
		503									
		*Dominates in the SRS			**Half power point damping estimate						

4 Conclusion

Resonant plate shock testing has proven to be an efficient, repeatable method for performing mid-field pyroshock simulation at Sandia National Laboratories. Most of the test methodology was developed without finite element modeling and did not make use of experimental modal analysis to inform engineers and test operators of the structural dynamics of the test equipment. Armed with this basic structural dynamics knowledge, improved test equipment and methods can be developed for simulation of these severe shock environments.

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This paper describes objective technical results and analysis. Any subjective views or opinions that might be expressed in the paper do not necessarily represent the views of the U.S. Department of Energy or the United States Government.

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Appendix

A Nomenclature

ζ Damping ratio

B Damping bar configuration response plots

Response, FFT, and SRS plots from the damping bar configuration study are shown. All tests were performed with a 10 kg projectile, 17.2 kPa of air gun pressure yielding approximately 8.8 m/s projectile velocity, and 13 mm of felt programmer between the projectile and plate. Acceleration was measured in the center of the plate.

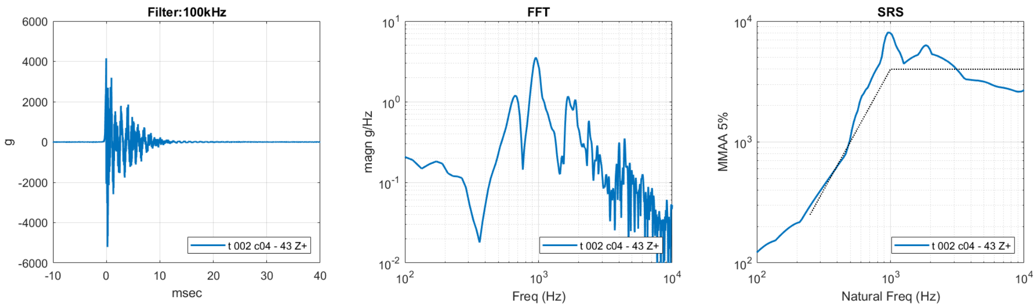

Figure 7: "1000 Hz" plate, 51 mm bars, 2 sides.

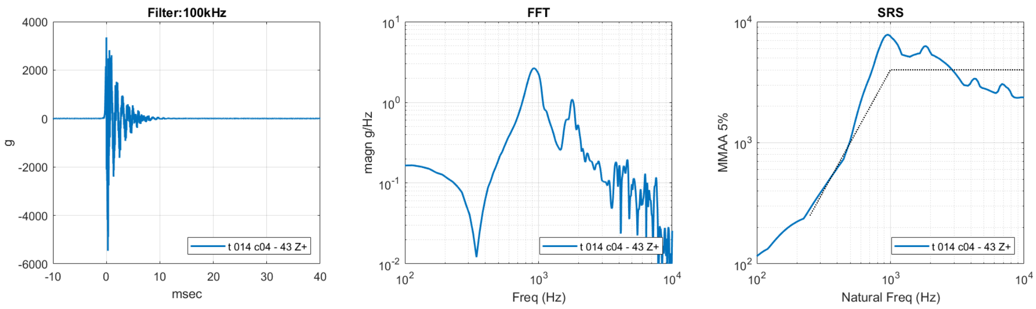

Figure 8: "1000 Hz" plate, 51 mm bars, 4 sides.

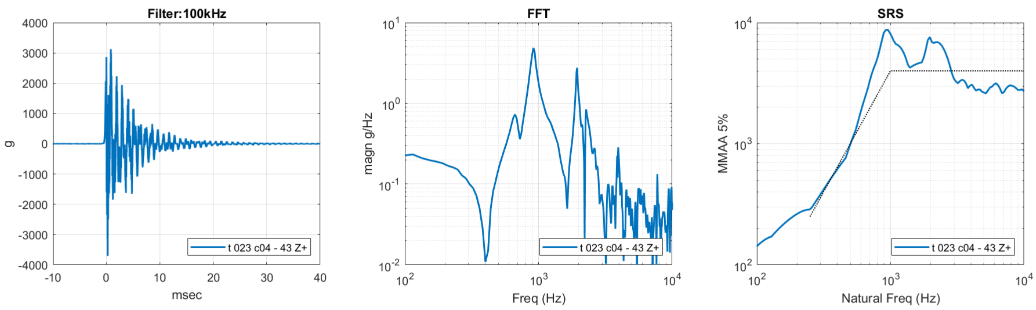

Figure 9:: "1000 Hz" plate, 25 mm bars, 2 sides.

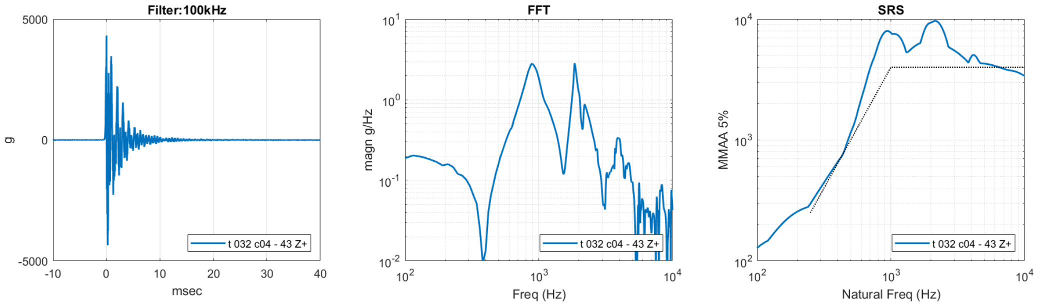

Figure 10: "1000 Hz" plate, 25 mm bars, 4 sides.

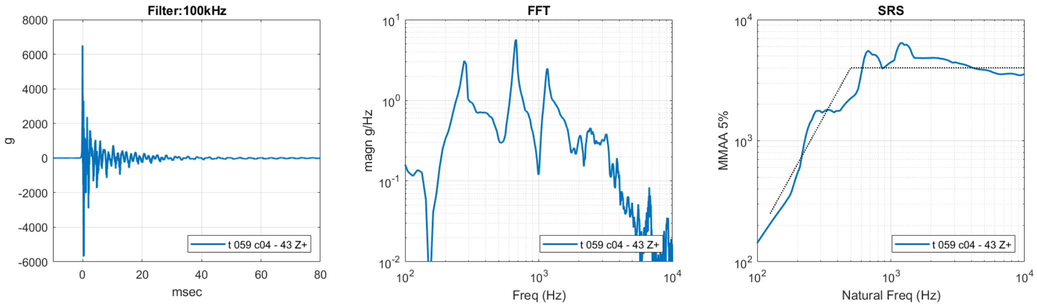

Figure 11: "500 Hz" plate, 51 mm bars, 2 sides.

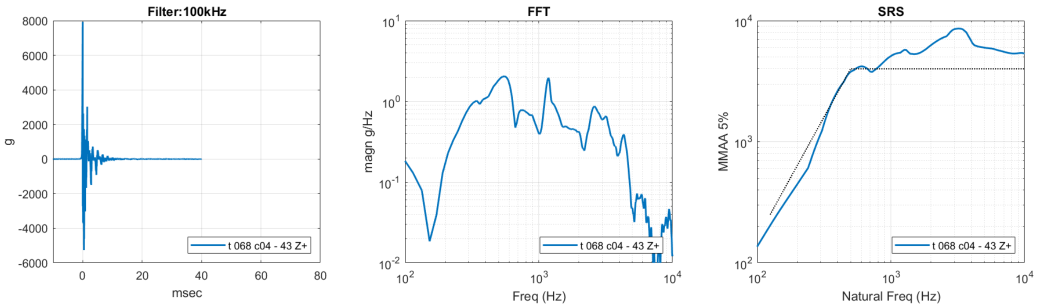

Figure 12: "500 Hz" plate, 51 mm bars, 4 sides.

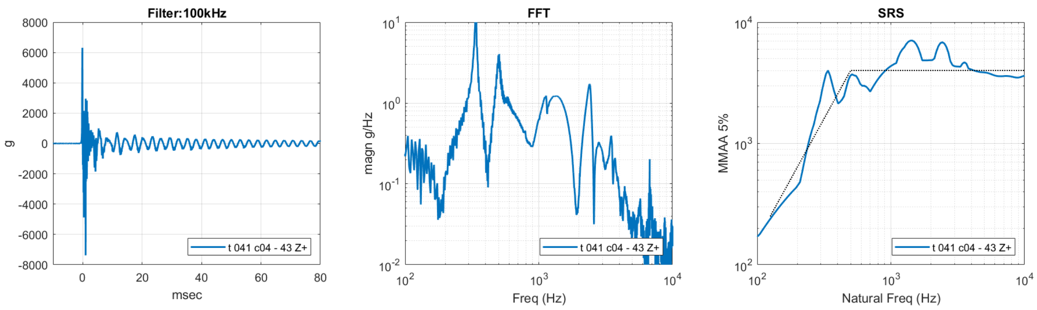

Figure 13: "500 Hz" plate, 25 mm bars, 2 sides.

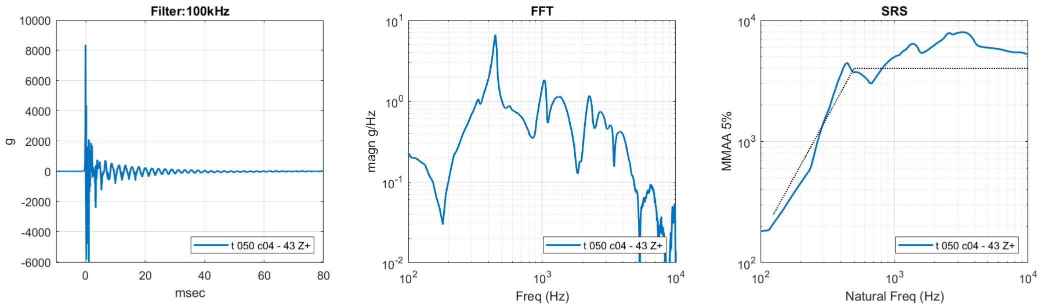

Figure 14: "500 Hz" plate, 25 mm bars, 4 sides.