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Pyroshock Simulation for Satellite Components Using a Tunable Resonant Fixture – Phase 2

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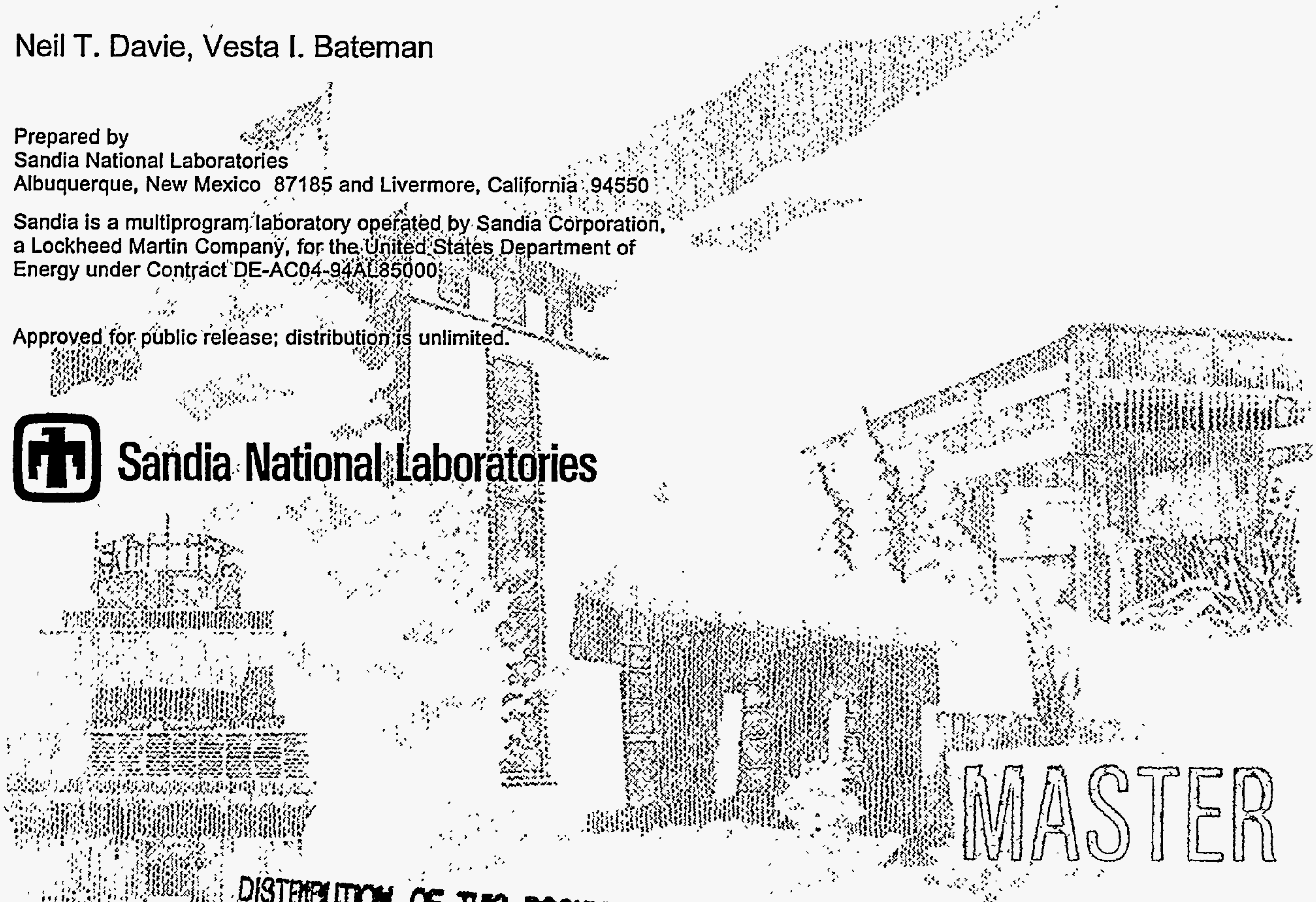

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Abstract

Aerospace components are often subjected to pyroshock events during flight and deployment, and must be qualified to this frequently severe environment. Laboratory simulation of pyroshock using a mechanically excited resonant fixture, has gained favor at Sandia for testing small (<8" cube) weapon components. With this method, each different shock environment required a different resonant fixture that was designed such that it's response matched the environment. In Phase 1 (SAND92-2135) of this research, a new test method was developed which eliminated the need to have a different resonant fixture for each test requirement. This was accomplished by means of a tunable resonant fixture that has a response which is adjustable over a wide frequency range. The adjustment of the fixture's response is done in a simple and deterministic way. This report covers Phase 2 of this research, in which several ideas were explored to extend the Phase 1 results to a larger scale. The test apparatus developed in Phase 1 was capable of testing components with up to a 10"x10" base. The goal of the Phase 2 research was to produce an apparatus capable of testing components with up to a 20"x20" mounting base. This size capability would allow the testing of most satellite and missile components which frequently consist of large electronic boxes. Several methods to attain this goal were examined, including scaling up the Phase 1 apparatus. Only one of these proved capable of meeting the Phase 2 goals. This report covers all details from concept through fabrication and testing of this Phase 2 apparatus.

Acknowledgments

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Table of Contents

Goals of this Project

The goal set forth at the start of this research was to develop a pyroshock simulation technique for satellite components (up to 20" x 20" mounting base) using a tunable resonant fixture apparatus which could be adjusted to any resonate frequency between 500 and 3000 Hz. Previously existing methods were only capable of testing components with less than a 10"x10" mounting base. With these methods a different fixture was required for each different resonant frequency desired, and it was not possible to make small adjustments in fixture response to offset the effects of various size components. The method developed in Phase 1 (Ref. 1) of this research eliminated these problems by using a single resonant fixture that could be tuned in a simple way over the frequency range from 500Hz to 2000 Hz. However, the Phase 1 apparatus was only capable of testing components with up to a 10" x10" mounting base. The goal of Phase 2 was to extend the capabilities of Phase 1 to allow larger components to be tested.

The results described in this report cover the second year of funding (FY93) referred to as Phase 2. This work began with a study to simply expand the size of the Phase 1 apparatus. This study revealed technical difficulties with this approach. A new method was then conceived using a longitudinally resonant bar with an adjustable natural frequency. Laboratory experiments with a small apparatus as well as analytical modeling verified the concept. However, an attempt to enlarge this concept to a reasonable size for testing large components, was unsuccessful. Finally, another method which also uses a longitudinally resonant fixture was successfully developed to yield a tunable fixture technique capable of testing large components.

Background - Pyroshock Simulation

Note: SAND92-2135 (Ref 1) contains detailed background information. Some of the most important details are repeated here.

Satellite components as well as aerospace and weapon components are often subjected to pyroshock events during powered flight or deployment. As a result, system components must be qualified to this frequently severe environment. These shocks are produced by explosive actuation devices such as detonators or linear explosives. Pyroshock-like environments can also be produced by high speed metal-to-metal impacts. The acceleration time history of a pyroshock resembles a decayed sinusoid with one or more dominant frequencies, and is characterized by high frequency, high amplitude, and a duration usually less than 20 msec. The net rigid body velocity change resulting from a pyroshock event is usually negligible. This environment is rarely damaging to structural elements, but can easily damage electronic components and assemblies. The severity of a pyroshock environment is usually characterized using a shock response spectrum (SRS). An SRS is a plot of the maximum response of a single degree of freedom (SDOF) system as a function of the natural frequency of the SDOF. The magnitude of the SRS at a given frequency is the maximum absolute value response that would be produced on an SDOF system with the same natural frequency if it were subjected to the shock time history (base input).

The amplitude and dominant frequency content of a particular pyroshock environment vary widely depending on the source of the shock, distance from the source, and structural response of the component and surrounding structure. For this reason, a pyroshock simulation technique should be capable of producing a wide variety of environments. Many methods exist for simulating pyroshock environments for component qualification testing. The majority of these techniques are expensive and require a great amount of trial-and-error to attain the desired shock environment. Techniques involving the use of mechanically excited resonant fixtures have gained favor at Sandia due to their low cost, and ability to produce a wide variety of pyroshock environments. Each of the techniques used at Sandia minimizes trial-and-error by using a fixture that is designed so that it's resonant response closely matches the desired pyroshock environment. As explained in Ref. 1, the fixture design is accomplished in a simple and deterministic way. Prior to Phase 1 of this research, each different test specification required the design and fabrication of a different resonant fixture. The test apparatus developed in Phase 1 eliminated the need for different fixtures by using a single fixture with a tunable response.

Due to the high cost and complexity of most aerospace systems, component qualification using the actual pyroshock environment on complete assemblies is not reasonable. In addition, design margin cannot be determined with this approach. For these reasons laboratory simulations of pyroshock environments are conducted on individual components and subassemblies. Traditional haversine pulse tests do not produce an adequate pyroshock simulation with regard to time history or SRS comparison. In general, the use of a haversine pulse test to simulate a pyroshock environment would result in a severe over-test at low frequencies, since the haversine test has considerably more velocity change than a pyroshock with comparable peak G's. Presently, pyroshock environments are simulated in the aerospace industry by one of the following methods (Ref. 1, and 6):

1. Electrodynamic Shaker. This method can accurately produce a desired SRS within closely specified tolerances, but amplitude and frequency limitations of the equipment greatly restrict it's applicability.

2. Live Ordnance with System Structure. Since the actual system structure and live ordnance are used, this method has the potential to produce a shock virtually identical to the expected field environment. All the very high frequencies (>10 KHz) associated with near-field pyroshock events are produced with this method. The cost of the test structure, however, is usually prohibitive, unless large numbers of identical tests are to be conducted. The use of live ordnance may have a wide repeatability tolerance, and does not easily allow the test levels to be increased so that an adequate design margin can be assured.

3. Live Ordnance with Mock Structure. This method has most of the same features as **2.** above, except that some cost savings are attributed to the use of a mass mock-up structure. These savings may be negated by the need for some trial-and-error testing to attain the desired component input, where geometric similarity was used in **2.** to attain the same result.

4. Live Ordnance with Resonant Plate Fixture. This method further reduces test cost, and is a candidate for general purpose testing, due to the use of a generic resonant plate fixture. Since live ordnance is used, all the very high frequencies associated with near-field pyroshock events are produced with this method. However a great amount of trial-and-error testing may be required to obtain the desired component input.

5. Mechanical Impact with Mock Structure. Mechanical impacts do not produce the very high frequencies associated with the stress pulse in the immediate vicinity of a pyrotechnic device. However most components in aerospace systems are isolated by enough intermediary structure such that the shock at the component location is not dominated by these very high frequencies. Instead, the shock at the component is dominated by the structural response to the pyrotechnic device, and has dominant frequencies which are typically less than 10 KHz. For these components, a mechanical impact (e.g. using a projectile or pendulum hammer) can produce a good simulation of the pyroshock environment. Test amplitudes can easily be increased/decreased by simply increasing/decreasing the impact speed. Frequency content can be controlled by the use of various pads at the point of impact.

Simulated pyroshock environments have been produced using mechanical impacts on system structures (or similar mass mock-ups). The idea is to impact the structure at the same point as the actual pyrotechnic device, and experimentally adjust the test conditions so that the response at the component is appropriate. Due to the cost of the test structure, and the large amount of trial-and-error testing required, this method is impractical in most cases.

6. Mechanical Impact with Resonant Fixture. In this method, a resonant fixture (typically a flat plate) is used instead of a mock structure. This significantly reduces cost, and allows for general purpose testing since the fixturing is not associated with a particular structural system. The mechanical impact excites the fixture into resonance which provides the desired input to a test component mounted on the fixture. Historically, test parameters such as plate geometry, component location, impact location, and impact speed, have been determined in a trial-and-error fashion. In general, this method produces a simulated environment which has it's energy concentrated in a relatively narrow frequency bandwidth. This feature may not be desirable for some pyroshock environments.

Expanding on the work of Bai and Thatcher (Ref 2), the Mechanical Shock Lab at Sandia has developed nationally recognized pyroshock simulation technology following **Method 6.** above. Much of the trial-and-error required with **Method 6.** has been eliminated by designing the resonant fixture such that it's dominant lower mode(s) correspond to the dominant frequencies in the component test requirement. This existing technology is documented in References 3 and 4. Using simple design principles, the fixture design is based only on the test requirement, and therefore, automatically has the desired dominant frequency content. Minimal experimental adjustment is required to attain the proper amplitude and mechanical damping. The Phase 1 and Phase 2 research was conducted to provide improvements in this pyroshock simulation method by the introduction of a tunable resonant fixture. The successful implementation of a tunable resonant fixture resulted in easier control of test input, lower test cost, and a reduction in resonant fixture inventory. In addition, it is possible to extend this method to test large test items such as satellite components. The following example illustrates how pyroshock simulation was conducted at Sandia prior to the Phase 1 and 2 research, and includes a discussion of how a tunable resonant fixture would result in the above improvements.

Figure 1 shows a "typical" component test requirement, as specified by an SRS. Note that the SRS exhibits a characteristic "knee" (in this example at 1000 Hz) where the spectrum changes from a steep slope to a nearly constant amplitude. Assume that the component to be tested is an electronic package with a 5"x5" mounting base. At this point, a resonant fixture must be designed such that it's first mode of vibration has a frequency at or near the SRS knee. The fixture must also be large enough to allow the component to fit on an antinodal area of the fixture's first mode. The resonant fixture geometry used by the Shock Lab is either 1) a rectangular aluminum plate which is excited into it's first bending mode, or 2) an aluminum bar which is excited into it's first longitudinal mode. Figure 2 shows resonant fixture dimensions that could be used for this example. Also shown is a schematic illustration of the test set up for each of the two types of resonant fixtures. Figure 3 shows the SRS obtained from the longitudinal bar fixture compared to the test requirement. Similar results can be expected for the bending plate fixture. It should be emphasized that the plate geometry is determined from the test requirement without any trial-and-error testing. Only a minimal amount of experimental adjustment is required to determine impact speed (i.e. SRS amplitude), and fixture damping. The mechanical damping is accomplished by attaching various clamps or metal bars to the resonant fixture.

Since the plates used for the bending configuration are relatively thick, the first bending mode frequency is closely predicted by equations for beam bending frequencies. The following equation is used as a design tool for selecting the plate geometry (Ref 5):

$$f_n = K_n \frac{t}{L^2} \qquad \text{Equation 1}$$

where:

f_n = n^{th} bending frequency, (Hz)

$K_n = \frac{A_n}{2\pi}\sqrt{\frac{E}{12\rho}}$, for a beam of uniform rectangular section

A_n = a constant dependent on the n^{th} mode = 22.4 for 1^{st} mode of free-free or fixed-fixed ends

E = modulus of elasticity, (psi)

ρ = density, (lb-sec^2/in^4)

t = beam thickness, (in)

L = length of beam (or long dimension of rectangular plate), (in)

For aluminum, K_1 = 203,800

Note: This equation applies to beams of various end conditions. The constants A_n are the same for a free-free beam, and a fixed-fixed beam. The free-free condition applies to the present bending plate fixture, and the fixed-fixed condition applies to the tunable resonant fixture.

The corresponding equation for the longitudinal modes of the bar fixture is (Ref 2):

$$f_1 \text{ (free-free)} = \frac{c}{2L} \qquad \text{Equation 2A}$$

$$f_1 \text{ (fixed-free)} = \frac{c}{4L} \qquad \text{Equation 2B}$$

where:

c = wave speed in bar (199,000 in/sec for aluminum)

L = bar length, (in)

For a test requirement with a different knee frequency, the above equations can be used to calculate new resonant fixture dimensions. Sandia's Mechanical Shock Lab is required to simulate pyroshock environments for a wide variety of test requirements. This means that a large inventory of resonant fixtures must be maintained in order to cover the range of SRS knee frequencies encountered. This has not been an extreme burden since most test requirements are for small (<8" cube) weapon components, which means relatively small fixtures. Recent trends have shown more frequent requests for testing of satellite and missile payload components with mounting bases up to 20"x20". Expanding our fixture inventory to allow testing of these large components would be costly and space consuming. This has been the primary motivation to develop a single tunable resonant fixture to replace an entire inventory of fixtures.

Another advantage of a tunable resonant fixture is that it would allow small adjustments in the knee frequency to compensate for the effects that different sized components would have on the response of the resonant fixture. With the previous methods, a resonant fixture designed to give the correct input to a light weight component might not provide quite the same input to a more massive component, since the resonant frequency of the plate would be slightly lowered (Ref. 7, and 8). This difference might be enough to cause the SRS for the massive component to fall outside the test requirement tolerance bounds. In this case, a slightly thicker plate would need to be fabricated to accommodate the massive component.

The previous method also imparts a small rigid body velocity change to the test item. This velocity change is often greater than that of the actual pyroshock being simulated. The Phase 1 tunable resonant fixture concept described below eliminates this rigid body velocity change due to the way the fixture is held. In general, the tunable resonant fixture concept will yield lower cost, more controllable pyroshock simulation, which will in turn serve the interests of the Shock Lab customers both within and outside Sandia.

Tunable Resonant Fixture - Phase 1

Previous research (Ref. 4) led to the development of a tunable resonant bar fixture, for which the first, second or third mode could be selectively excited. With this method, a single fixture could be used to produce pyroshock simulations for three different SRS knee frequencies. However, a continuously adjustable resonant frequency was desired, and the tunable resonant bar does not meet this requirement.

The mechanical system conceived to provide a continuously adjustable resonant frequency was a beam rigidly clamped between two massive blocks. The first bending mode of this system can be roughly predicted from a simple beam with fixed-fixed end conditions. The frequency of the first bending mode can be adjusted by moving the clamping location of the two masses, and thus changing the length of the free span of the beam between the masses. For an ideal beam with fixed-fixed end conditions, the first bending mode is calculated from Equation 1, where L is the length of the beam between the fixed ends. The center of the beam span is the area of maximum response (antinode) for the first bending mode. This would be the logical point of impact to excite the beam into it's first mode. A test component mounted on the beam opposite to the impact would be subjected to a maximum response at the first bending frequency. As with existing resonant fixture test methods, the impact duration must be of the appropriate duration so that the impact energy is delivered to the first mode of the fixture. If the duration is too short, higher bending modes would be excited. This could be desirable for some pyroshock environments that do not follow the characteristic SRS shown in Figure 1. In most cases, however, the impact duration can be adjusted for first mode excitation by using various felt, or cardboard pads at the point of impact.

Figure 4 shows the Phase 1 apparatus consisting of a massive concrete and steel base which houses a 3" ID air gun. An aluminum bar is clamped to the top of this base with adjustable steel blocks near each end of the bar. The length of the beam between the clamps determines the resonant frequency of the beam. This resonance is excited by an impact produced by a captive projectile in the air gun. The clamp positions can be adjusted to produce a resonant frequency between 500 Hz. and 3000 Hz. Reference 1 contains a detailed description of the Phase 1 apparatus, along with measured response data.

Tunable Resonant Fixture - Phase 2

Preliminary Concept

The tunable resonant beam has proven to be easy to use and a reliable tool for simulating pyroshock environments for small to medium sized components up to about a 10" cube. Many satellite and missile components, however, are too large to be tested on the tunable resonant beam. After developing the tunable resonant beam, our goal was to provide a technique with similar capabilities for testing larger components up to about 20" cube. The initial plan was to simply increase the size of the resonant beam apparatus. It should be noted that the amplitude of the resonant beam's response is highest at the center of the beam, and diminishes at locations approaching the clamping plates. If the free span of the beam is adjusted to a dimension approaching that of the component's mounting base, then the portions of the component near the clamping plates would not receive adequate test levels. This condition does not occur if the free span of the resonant beam is at least 1.5 times the length of the component. A 20" component would require a free span of at least 30". In order to obtain a knee frequency of 3000 Hz, the beam must be over 12" thick (Eq. 1). Also, results from the Phase 1 apparatus (Ref. 1) showed that equation 1 tended to overestimate the knee frequency by about 50% at frequencies approaching 3000 Hz. This fact implies that an even thicker beam (probably 24") would be required than predicted from Equation 1. We judged that it would be nearly impossible to force a 24" thick beam to have approximately fixed ends, and hence, we abandoned this approach in favor of tunable resonant bar concepts described below.

Tunable Resonant Bar Concept # 1

The first concept to tune the response of a longitudinally resonant bar is shown in Figure 5, where a relatively large steel block is clamped around a rectangular aluminum bar. If the block is large enough, it will approximate a fixed condition on the longitudinal response of the bar. If this condition could be attained, then the resonant frequency of the protruding end would be dependent only on the length of the bar protruding from the block as determined from Equation 2B with one clamped end. Different SRS "knee" frequencies could be obtained by changing the position of the steel block This concept was evaluated with a very small scale apparatus consisting of a 1/4"x3/4" x 48" long aluminum bar with a 27 lb. steel clamp (2 ea. 5"x5"x2" steel blocks). An accelerometer attached to the left end of the bar was used to measure the bar's response. It turned out that the left end of the bar could be excited into resonance with a longitudinal impact on the right end. The response of the right end was only minimally detected by the accelerometer on the left end. Table I indicates the results of this small scale experiment.

Note that if the actual fixed point of the protruding bar is assumed to be about 3.5" inside the 5"x5"x2" clamping block, then very good agreement is obtained with Equation 2B. In addition, an ALGOR model of the resonant bar accurately predicted the dominant modes shown in Table I.

Two different embodiments of this concept were attempted using a 6" diam. x 72" long steel tube for one and a 2"x10"x96" long aluminum bar for the other. Experiments conducted with these bars did not duplicate the results shown in Table I. The reason that these tests failed has not yet been determined, but it is suspected that better results could be obtained if a larger clamping mass were used. A larger clamping mass might be impractical at the scale required for 20" cube sized components. In any case this concept was abandoned in favor of another described below.

Table I

Protruding Beam Length (in)	Measured Dominant Freq. (Hz)	Corresponding Length - Eq. 2B (in)
22	1912	25.3
18	2237	21.6
15	2597	18.6
12	3049	15.9
9	3682	13.2

Tunable Resonant Bar Concept # 2

The second concept for tuning the longitudinal response of a bar fixture is simply a fixed length bar with a variable quantity of mass attached to the impact end. As the mass is increased, the first mode resonant frequency will decrease. This concept and its limits can be understood intuitively by considering the two extreme cases. For a bar with no attached mass, the first mode frequency is given by Equation 2A. For a bar with infinite mass attached, one end condition would be fixed, and the first mode frequency is given by Equation 2B. Note that the frequency for the bar with infinite mass is half the frequency of the no mass condition. Intuitively, for an added mass between these two extremes, the resonant frequency should be adjustable (as a function of mass) between the two frequencies given by Equation 2A and 2B.

Figure 6 shows a resonant bar fixture design which utilizes the above tunability concept. In this design, the resonant bar is comprised of three smaller bars which act in parallel. The resonant bars were designed in this manner to reduce weight, and to accommodate certain damping experiments which are planned for future studies. A magnesium expander attached to one end, provides a larger platform for testing components up to 22"x22". Two different length bars of this design were fabricated; one was 60" long, and the other was 24" long. Equations 2A and 2B predict that the 60" bar should have an adjustable frequency between 800 Hz. and 1600 Hz. Equations 2A and 2B predict that the 24" bar should have an adjustable frequency between 2000 Hz. and 4000 Hz. As shown later, the actual resonant frequencies are lower due to the weight of the expander head.

The resonant bar fixtures were designed to be used with Sandia's 18" Actuator facility. This facility consists of a twin rail track on which sleds are propelled at high speeds by means of an 18" ID pneumatically actuated piston. The resonant bar fixtures were fitted with guides that allowed free motion along the actuator track. An impact from a small sled was used to excite the resonant bar fixture into longitudinal resonance. The resonant frequency of the bar can be reduced by bolting steel plates on the impact end. Figure 7 shows the test configuration, and Figure 8 shows a photo of the 24" bar fixture positioned on the actuator track.

Several tests as depicted by Figure 7 were conducted to evaluate these tunable resonant bar fixtures. For each test, the impact speed was about 40 ft/sec, and the response was measured with an accelerometer attached to the expander head. Figure 9 shows the acceleration time history and SRS for one of the tests on the 60" bar. Figure 10 shows the acceleration time history and SRS for one of the tests on the 24" bar. The SRS shape for each of the other tests was similar to that shown, except that the "knee" frequency was shifted as a function of added mass, as shown in Table II.

The results shown in Table II confirm the tunability of the resonant bar concept. Note that the "knee" frequency of the 60" bar without added weight is close to the frequency predicted by Equation 2A. However, the "knee" frequency for the 24" bar is much lower than predicted by Equation 2A. This is probably due to the fact that the mass of the magnesium expander has a more significant effect at higher frequencies. In either case, Equation 2A serves only as an approximate set up tool for tuning the response of the resonant bar. However, the trend of adding mass to get a lower "knee" frequency allows an operator to set up a desired test with only minimal experimental adjustment of the test fixturing. Together, these two resonant bar fixtures make it possible to conduct pyroshock simulations with "knee" frequencies from 1000 Hz to 2500 Hz. Lower "knee" frequencies could be achieved with a longer bar, but these two bars cover the range most commonly encountered at Sandia. Any attempt to extend the control of "knee" frequencies above 2500 Hz would very difficult for large components. It should also be pointed out that the rigid body velocity change for the resonant bar fixtures is greater than for the resonant beam (which is essentially zero). The velocity change could have been reduced by using a lighter impacting sled.

Table II

Resonant Bar Length (in)	Weight Added on Impact End (LB)	SRS "Knee" Frequency (Hz)
60	0	1500
60	160	1400
60	380	1200
24	0	2500
24	220	2200
24	380	2000

Conclusion

Two new techniques have been developed for simulating pyroshock environments using tunable resonant fixtures. Components with up to a 20"x20" mounting base can be tested. The Phase 1 apparatus uses a tunable resonant beam excited into bending resonance, while the Phase II apparatus uses a tunable resonant bar excited into longitudinal resonance. The dominant resonant frequency of the fixtures can be adjusted in a known manner such that a wide variety of SRS levels can be obtained from each fixture. The fixture's response is approximately determined from simple equations which are used as a starting point for the test set up. Only minimal experimental adjustment is then used to achieve the desired SRS, since the effect of a given adjustment is known in a trend sense. This contrasts sharply with previous methods which used pure trial-and-error to simulate pyroshock environments.

All of the initial goals of this project have been attained with the Phase 1 and Phase II apparatus. The Phase I apparatus was patented (ref. 9) in 1996. The successful completion of these techniques has resulted in the capability to efficiently simulate pyroshock environments for both small and large components. The payoff for this effort is especially important for the testing of satellite components, many of which are large "black boxes". This is an area where we see the potential for increased involvement by Sandia's Shock Lab and other test facilities. The test capability expected from the Phase II tunable resonant fixture, will enhance our ability to serve existing test needs at Sandia, and will help to attract additional work in this area.

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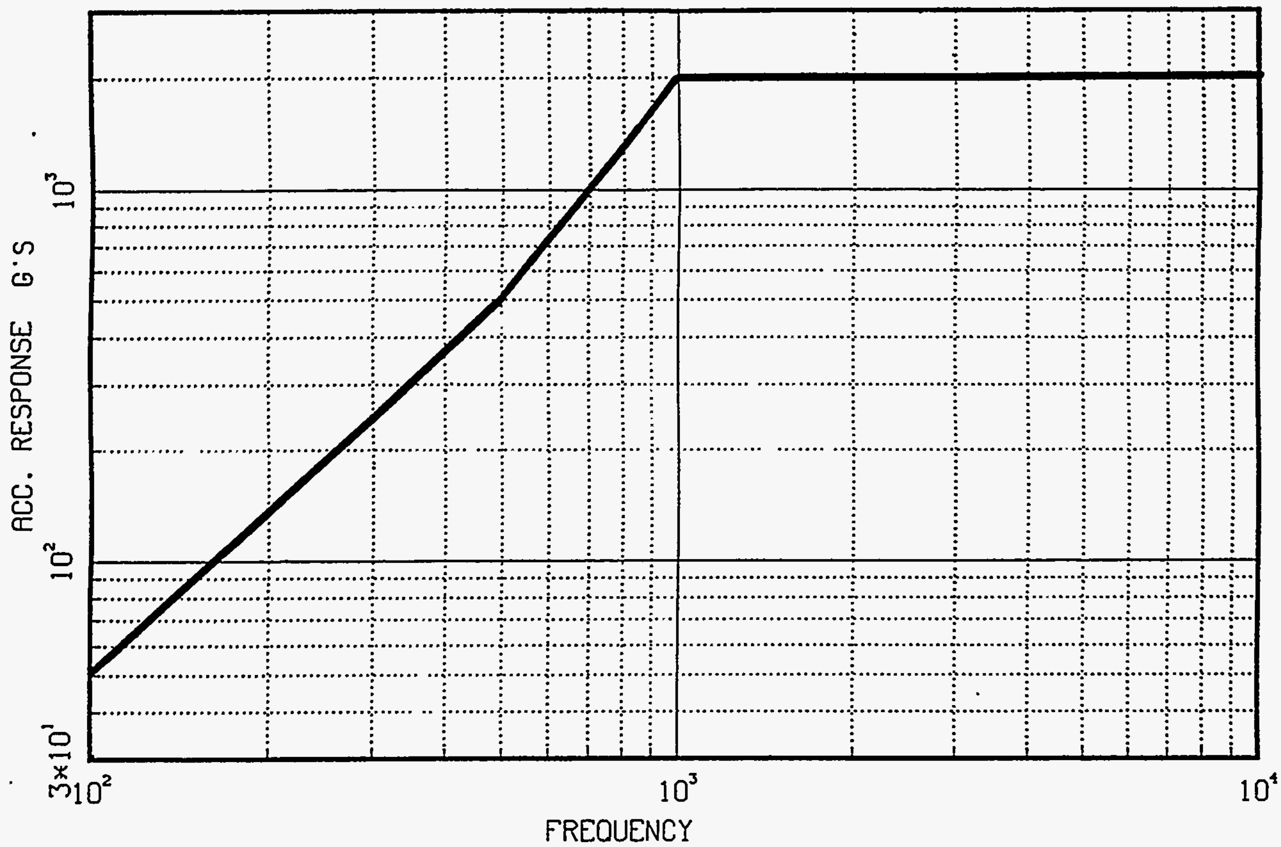


Figure 1. Typical Pyroshock Test specification

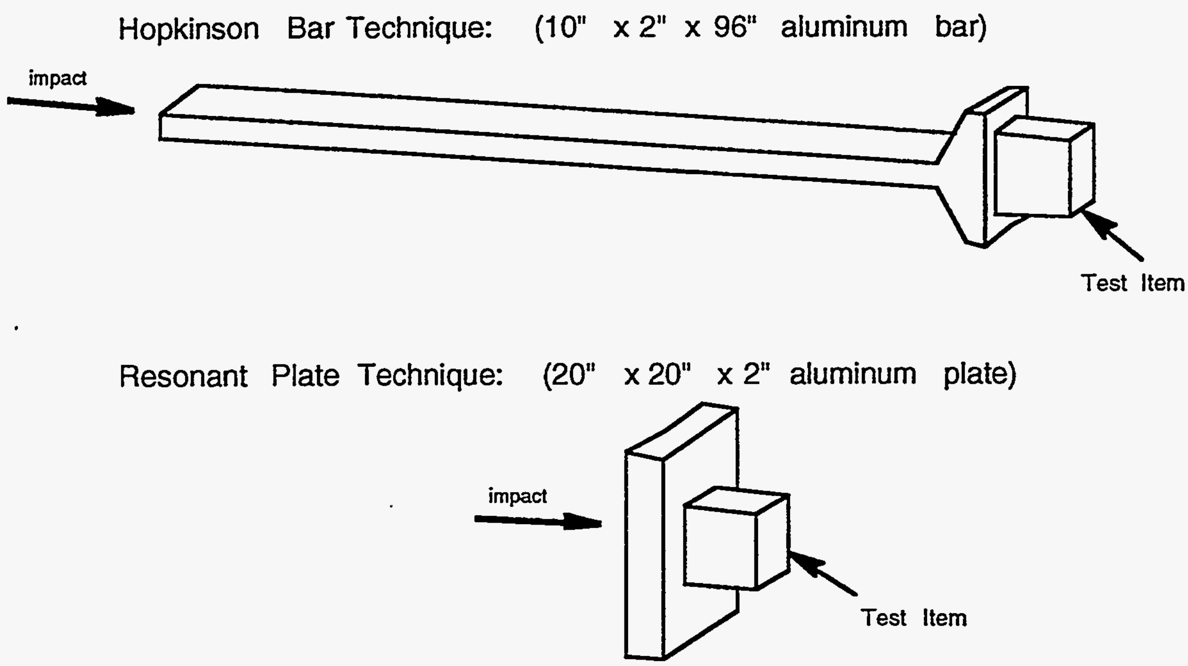
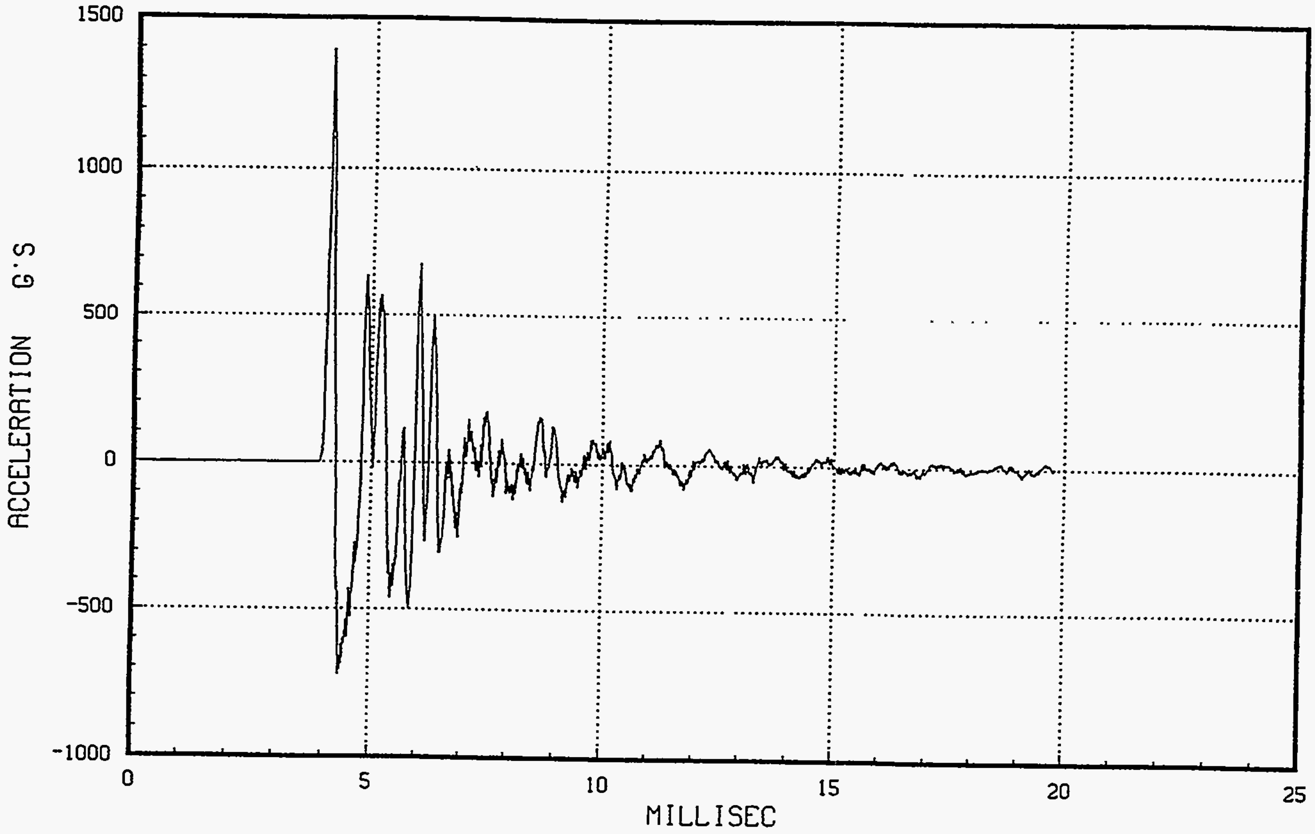


Figure 2. Previous Pyroshock Simulation with Resonant Fixtures



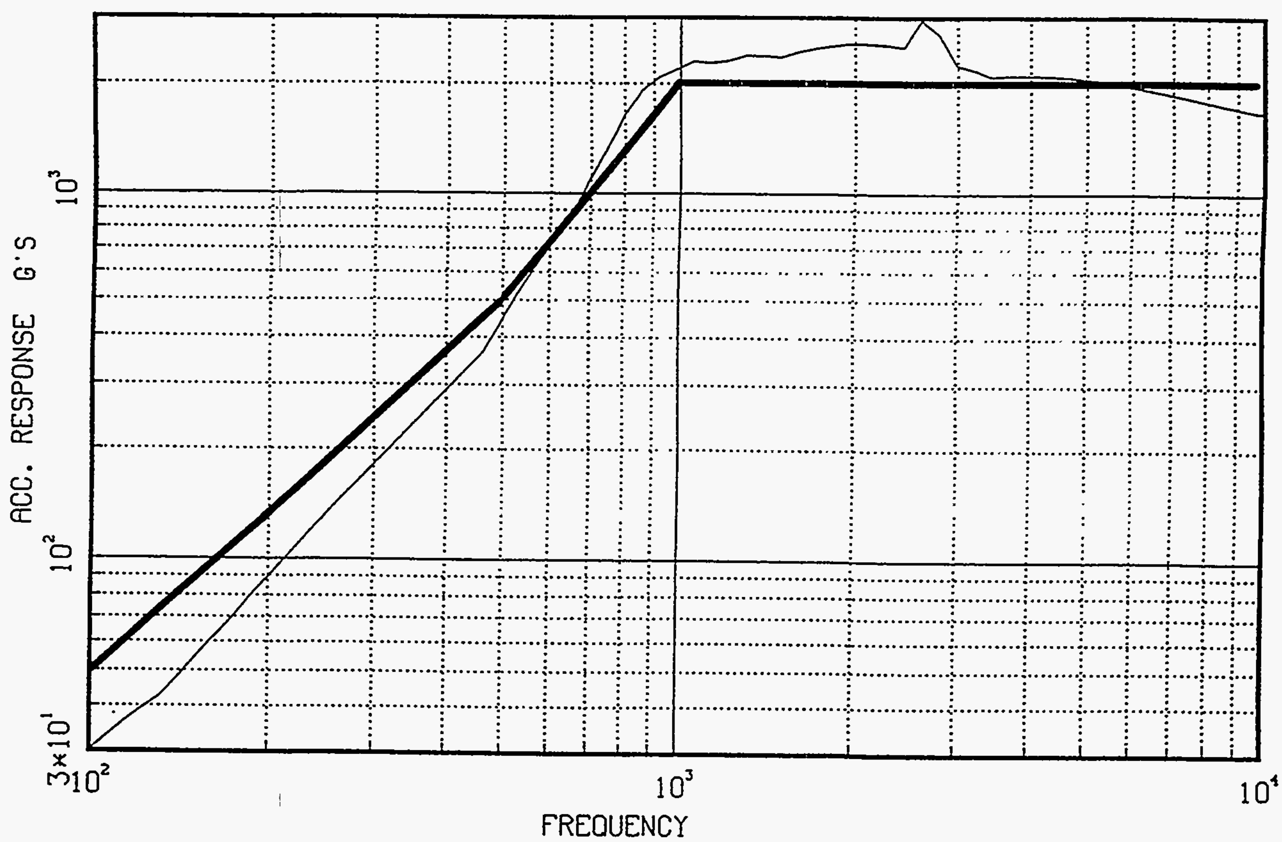


Figure 3. SRS for 1000 Hz bar with superimposed test specification

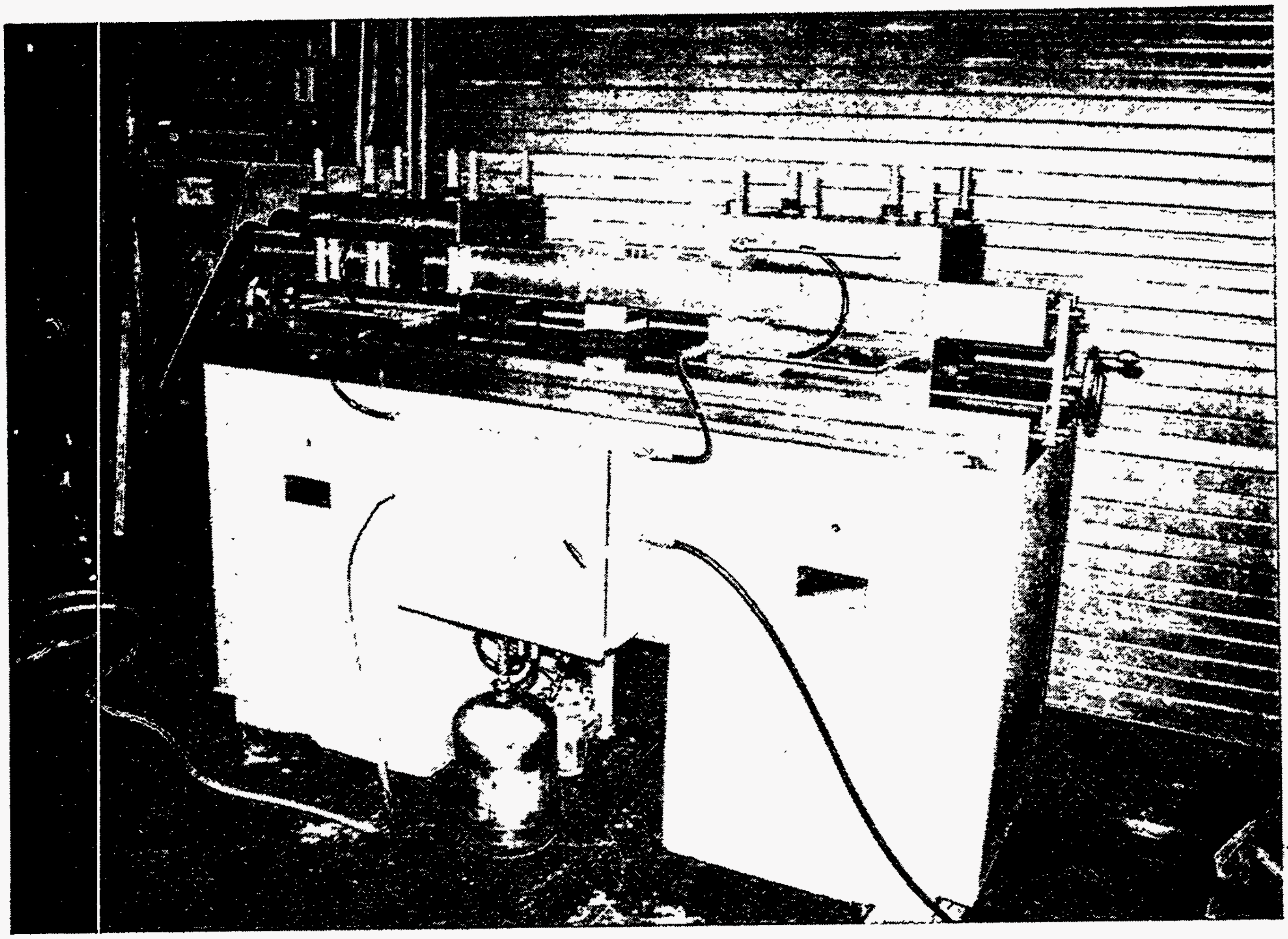


Figure 4 Phase I Apparatus

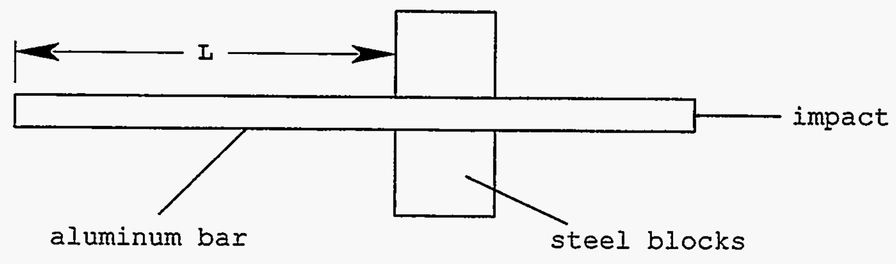


Figure 5 Tunable Resonant Bar - Concept 1

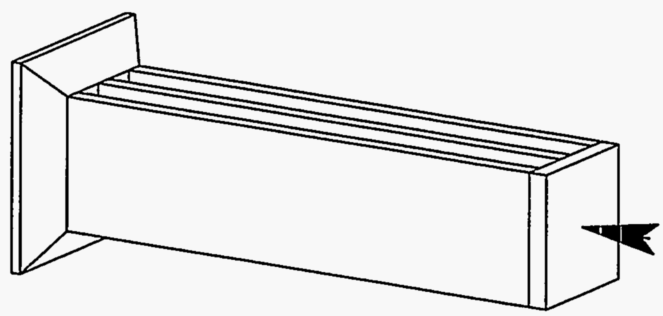


Figure 6 60" Tunable Resonant Bar - Concept 2

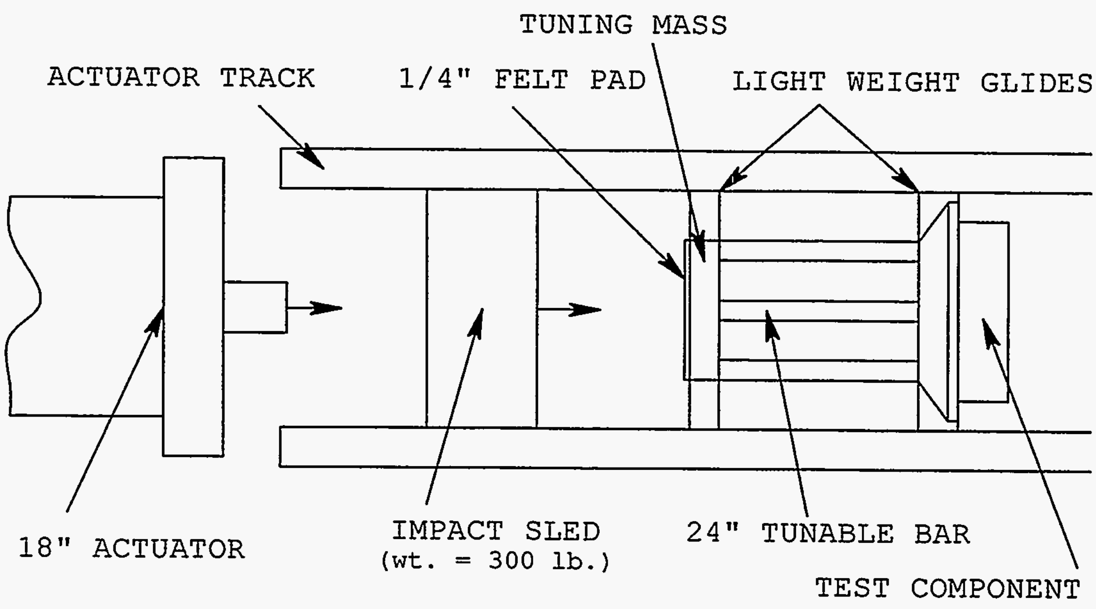


Figure 7 Tunable Resonant Bar Set Up

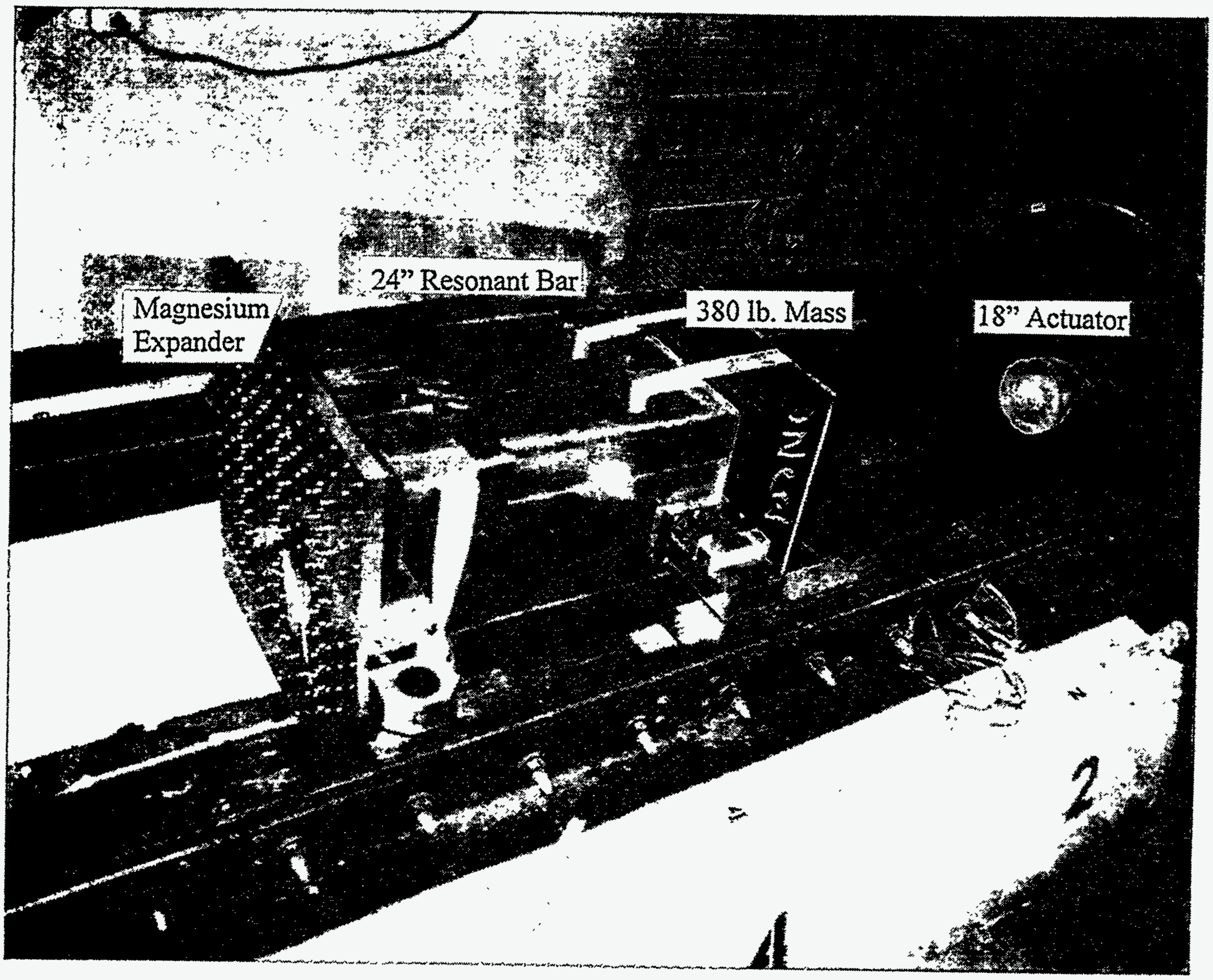
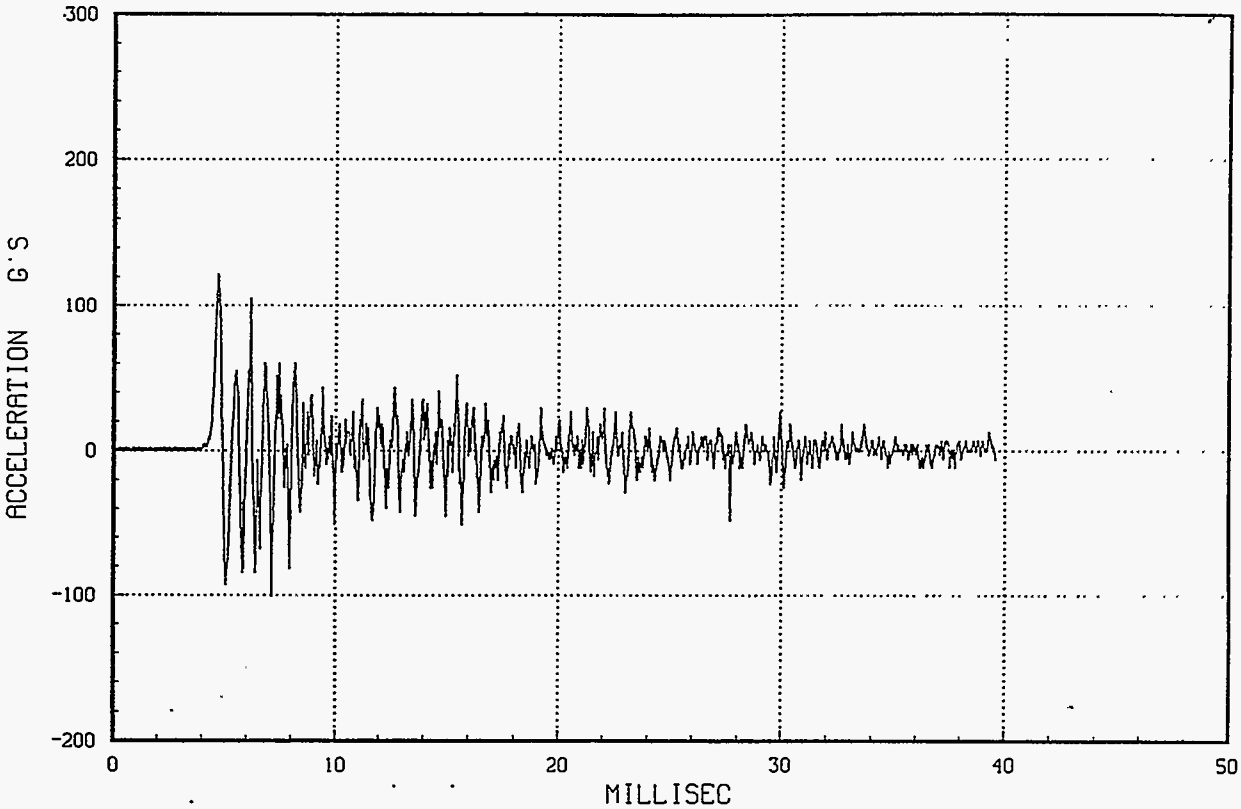


Figure 8 Tunable Resonant Bar on 18" Actuator Track



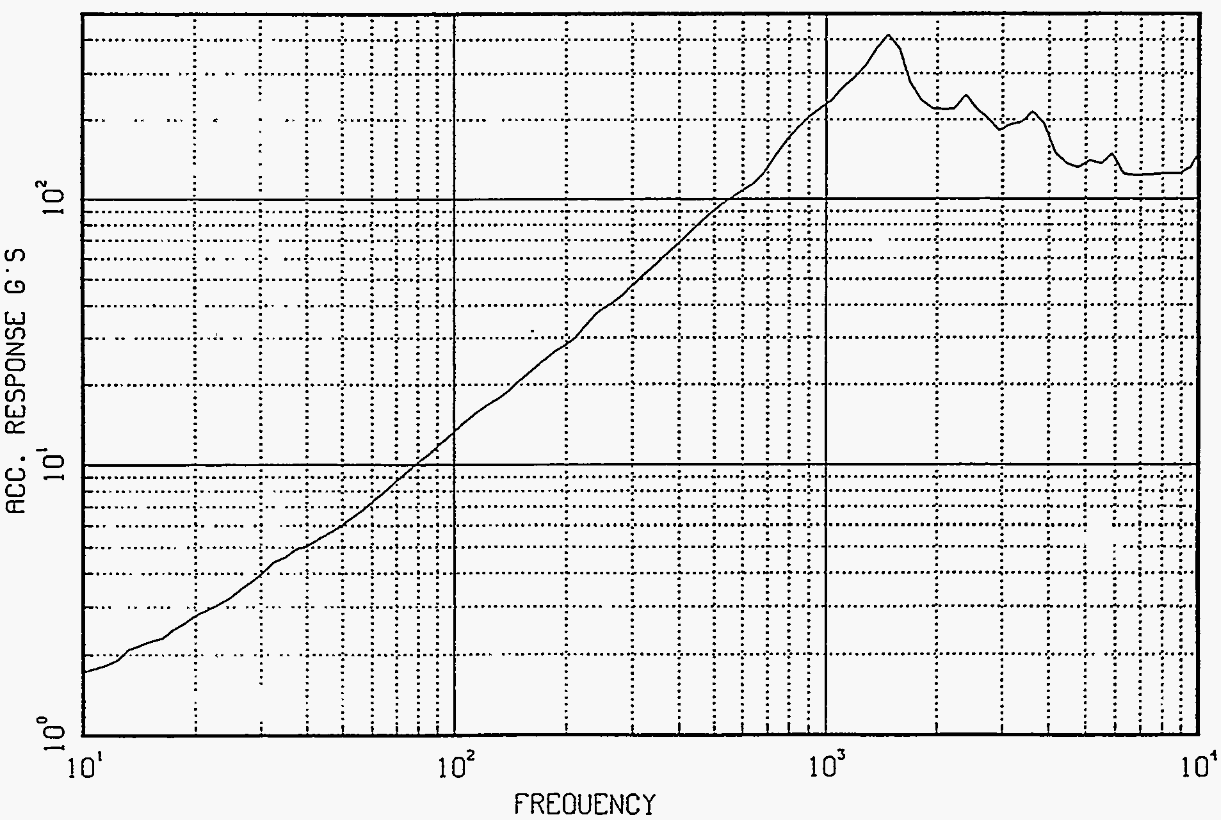
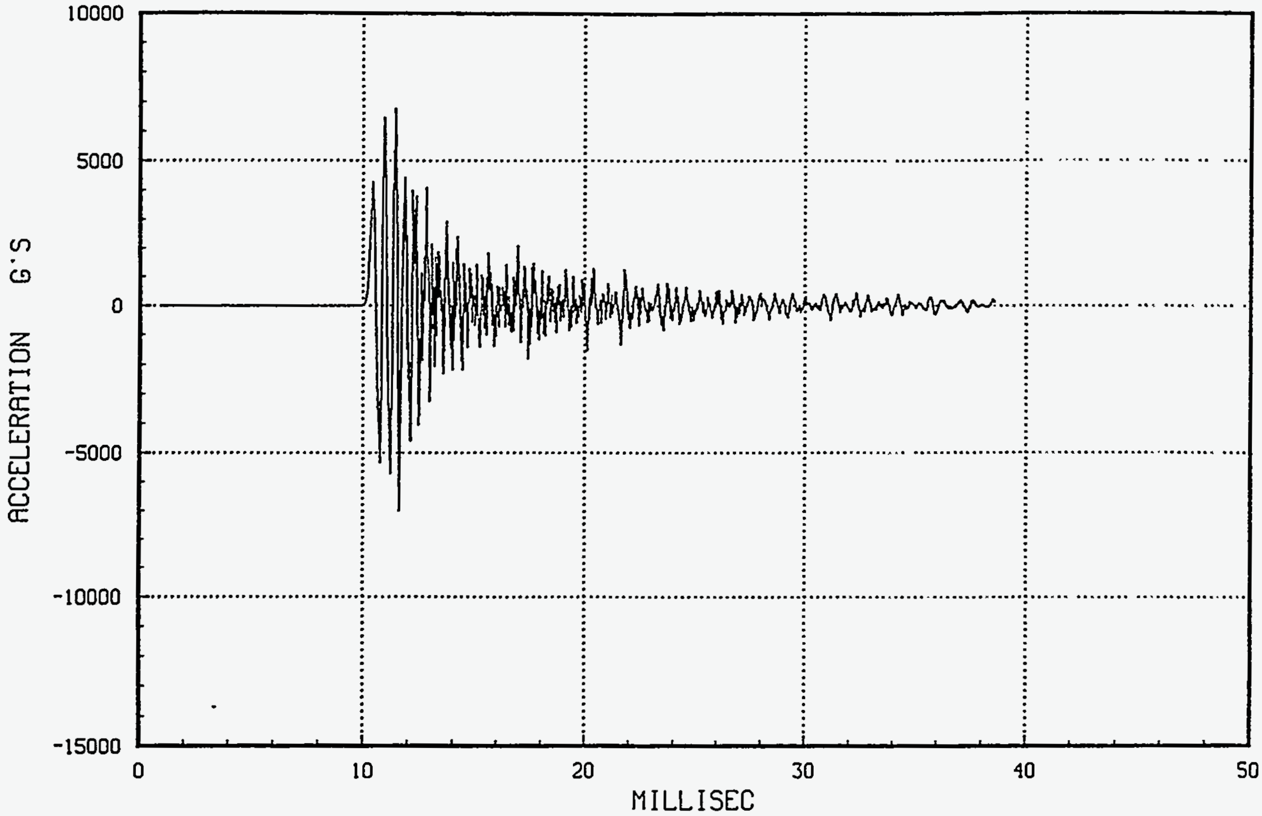


Figure 9 Time History and SRS for 60" Tunable Resonant Bar, without Added Mass



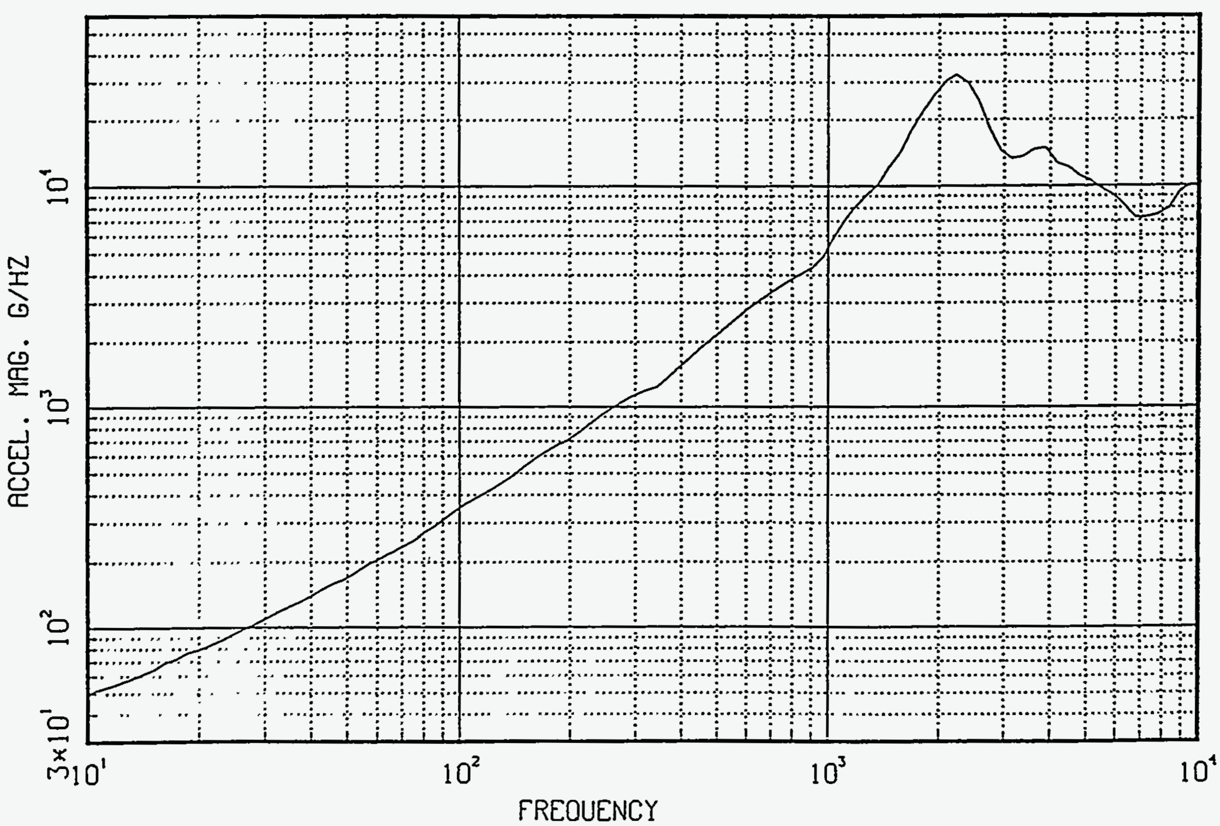


Figure 10 Time History and SRS for 24" Tunable Resonant Bar, with 220 lb. Added Mass

Distribution:

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