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RECENT DEVELOPMENTS IN PYROSHOCK SIMULATION USING FIXTURES WITH TUNABLE RESONANT FREQUENCIES

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Biographies

Neil Davie received a B. A. degree in math and physics from Augustana College in 1976. He received an M. S. degree in Theoretical and Applied Mechanics from the University of Illinois in 1978. Since that time he has been employed by Sandia National Laboratories. He was involved in mechanical modeling, and structural analysis until 1982 when he began working in the area of shock testing and pyroshock simulation. Presently he is a Senior Member of the Technical Staff in the Shock Testing Laboratory.

Vesta Bateman received a B. S. degree from Vanderbuilt University and an M. S. and Ph. D. from the University of Arizona, all in Mechanical Engineering. She taught for four years in the Mechanical Engineering Department at Virginia Tech. Since 1980, she has been at Sandia National Laboratories in Albuquerque, New Mexico where she is a Senior Member of the Technical Staff in the Shock Testing Laboratory.

Abstract

Pyroshock is a potentially severe environment produced by the detonation of explosively actuated components and stage separation hardware. Electronic components exposed to pyroshock events during flight or deployment can be damaged by this high frequency, high G shock. Flight qualification of these components may be accomplished using one of many existing techniques to simulate the pyroshock environment in the laboratory. Two new techniques developed at Sandia National Laboratories allow larger components to be tested to a wide variety of pyroshock environments. The frequency content and amplitude of the simulated pyroshock can be easily controlled in a predictable manner. The pyroshock environment is produced by the resonant response of a test fixture that has been excited by a mechanical impact. The resonant fixture has a dominant frequency that can be continuously adjusted over a frequency range that is typically found in most pyroshock environments. The test apparatus and techniques utilized by each method will be described in this paper. Experimental results will be presented which illustrate the capabilities of each method. A patent application has been submitted by DOE for one of these methods.

Keywords

Pyroshock, resonant fixture, pyrotechnic shock simulation

Background - Pyroshock Simulation

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 maximax 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 system. 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 damping ratio of the SDOF system is normally chosen to be 5% for pyroshock data analysis.

Due to the high cost and complexity of most aerospace systems, component qualification using the actual pyroshock environment on complete assemblies is not reasonable. For these reasons laboratory simulations of pyroshock environments are conducted on individual components and subassemblies. References 1 - 5 give detailed discussions of various methods which have been used to provide laboratory simulation of pyroshock environments. All of these techniques use either explosives or mechanical impact to excite a structure or fixture into resonance which in turn delivers the simulated environment to an attached component. The majority of these techniques rely heavily on trial-and-error procedures to obtain the desired SRS.

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Motivation to Improve Existing Methods

Pyroshock simulation techniques previously developed at Sandia are designed to minimize the use of trial-and-error. As described in detail in References 1, 4, and 5, these methods employ a mechanical impact to excite a fixture into a resonance which produces the desired SRS. The use of trial-and-error is virtually eliminated by fixture design based on the SRS requirement, rather than using an arbitrary fixture as in other methods. Basically, the fixture is designed such that its first mode of vibration corresponds to the dominant or "knee" frequency in the SRS requirement (Ref. 3). These techniques use either a Hopkinson bar excited into longitudinal resonance or a plate fixture excited into bending resonance by a mechanical impact. Figure 1 shows a "typical" SRS test requirement along with an SRS obtained using a 10"x2"x96" long aluminum Hopkinson bar 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.

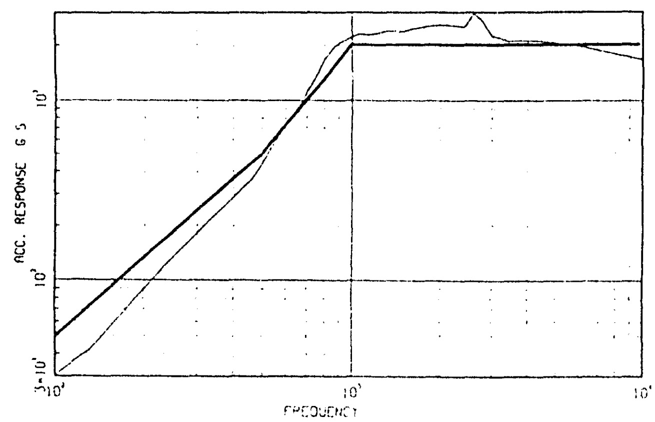

Figure 1 Typical SRS Test Specification with Non-tunable Resonant Bar Simulation

In this example, the SRS requirement is "typical" in the sense that its shape is applicable to many environments found in aerospace and weapon systems encountered at Sandia. The "knee" frequency (in this case 1000 Hz) is usually between 500 Hz and 3000 Hz, with peak SRS amplitudes between 500 G and 20,000 G.

For a test requirement with a different "knee" frequency, different resonant fixture dimensions would be required. 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 24"x24". 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 tunable resonant fixtures 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 present 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.

Two new pyroshock simulation techniques have been developed using tunable resonant fixture concepts to overcome the above limitations of previously existing technology.

Technique 1 - Tunable Resonant Beam

Concept

The mechanical system conceived to provide a continuously adjustable resonant frequency was a beam rigidly clamped between two massive blocks as shown in Figure 2. The first bending mode of this system can be roughly predicted from a simple beam with fixed-fixed end conditions (Equation 1). The frequency of the first bending mode can be adjusted by moving the clamping location of the two masses, thus changing the length of the free span of the beam between the masses. 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 its 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 must be of the appropriate duration so that most of the impact energy is delivered to the first mode of the fixture. If the duration is too short, higher bending modes would tend to dominate the response instead of the first mode 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. The following equation gives the first bending frequency for an aluminum beam with fixed-fixed or free-free end conditions:

$$f_1 = 203801\frac{t}{L^2} \qquad \text{(Ref. 6) Equation 1}$$

where:
f_1 = 1st bending frequency, (Hz)
t = beam thickness, (in)
L = length of beam (in)

Is this eq'n good for all beam thickness or just thin beams?

Proof of Concept Using Small Scale Apparatus

In order to prove the tunable resonant fixture concept described above, a small scale apparatus shown in Figure 2 was fabricated. This apparatus consisted of a 20" long x 2" wide x 1/2" thick resonant beam. Each end of the beam was clamped as shown between a pair of steel blocks using 3/8" bolts (not shown). The position of the clamping blocks could be adjusted in order to vary the free length of the beam between the blocks. An accelerometer was attached to the midpoint of the resonant beam to measure the acceleration response of the beam. The opposite side of the beam was then struck with a small hammer such that the first bending mode was excited. Measurements were made for several different distances between the clamping blocks.

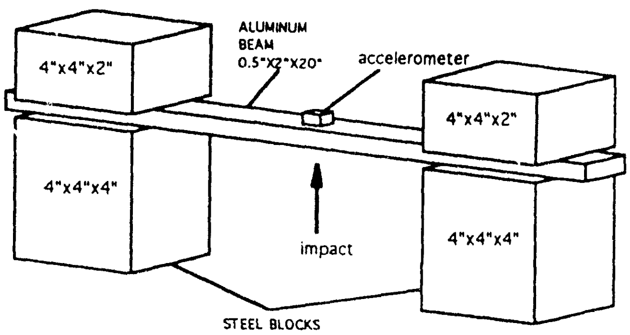

Figure 2 Small Scale Tunable Beam Concept

Figure 3 shows a typical acceleration time history and SRS for this small apparatus. The plots shown are for a 4" spacing between the blocks. Note that the shape of the SRS is desirable for simulating pyroshock environments encountered for many aerospace and weapon applications. Similar results were obtained for larger spacings except that the SRS knee frequency was correspondingly lower. Reference 1 examines these results in more detail. It should be emphasized that the SRS shape remained approximately the same, but was shifted in frequency corresponding to the "knee" frequency. In the set up used for Figure 3, a 1/16" neoprene pad was placed between the resonant beam and the clamping blocks to increase the damping in the response. Without the pads, the response lasted much longer than is typical for most pyroshock environments. Equation 1 predicted resonant frequencies that were higher than the measured values, with agreement getting worse at higher frequencies. This is to be expected since it is increasingly difficult to achieve a fixed end condition at higher frequencies. The use of the neoprene pad also tended to lower the actual resonant frequency. However, the trend of the resonant frequency as a function of clamp spacing in Equation 1 was followed by the measured data. These small scale experiments verified the tunable resonant beam concept and provided confidence for the fabrication of the larger apparatus described below.

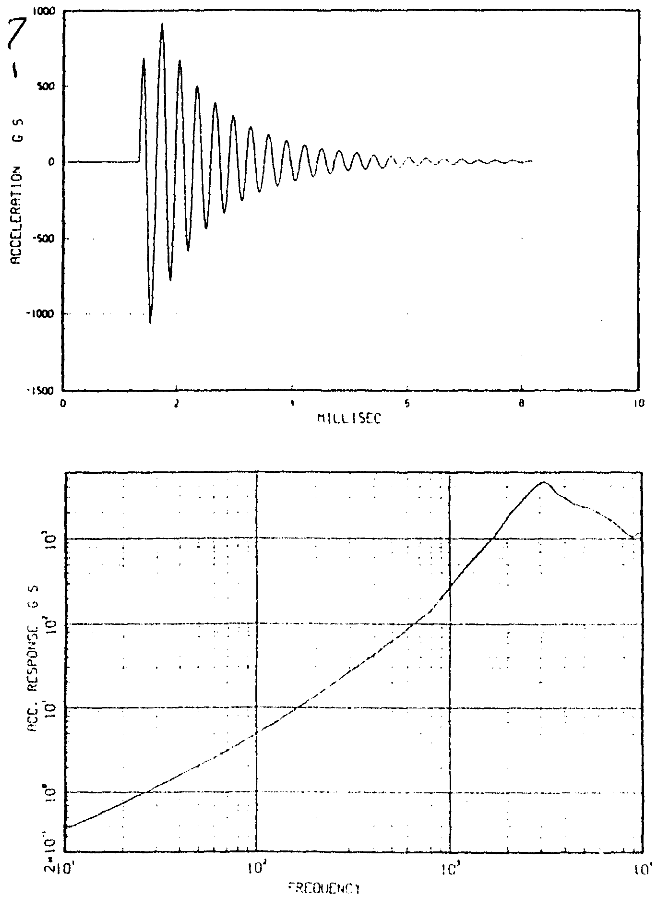

Figure 3 Time History and SRS, Small Scale Beam with 4" Spacing

Design and Analysis of Tunable Resonant Beam Apparatus

The small scale apparatus was modeled using ALGOR[1] finite element code. This model was developed so that a predictive tool would be available to aid in the design of a larger tunable resonant beam apparatus. The ALGOR model predicted resonant frequencies within about 10% of those measured in the experiments described above. The positive results from the small scale testing and analysis gave us confidence to design the larger resonant beam.

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1 ALGOR is a registered trademark of Algor Inc., Pittsburgh, PA.

We then designed the basic elements of the tunable resonant beam apparatus, and modeled it with ALGOR. Using the modal analysis features of ALGOR, we verified the tunability of the dominant 1st bending mode of the resonant beam. These analyses also showed several lower amplitude modes at frequencies below the dominant bending mode of the beam. These lower frequency modes were asociated with the concrete and steel base. The existence of these lower frequency modes caused some concern that these lower modes could be excited and interfere with the intended response of the resonant beam. To determine if this might be true, we used ALGOR to calculate the transient response of the beam when subjected to a force pulse. The resulting transient response was completely dominated by the desired mode, with no significant influence from the lower modes observed in the modal analysis. These results gave us confidence to design and fabricate the tunable resonant beam apparatus described below.

Apparatus Description

A photo of the completed apparatus is shown in Figure 4. A simple enlargement of the small scale apparatus would have resulted in two rather large masses that would need some elaborate mechanism to position them at various points along the resonant beam. Instead, the apparatus shown in Figure 4 uses a single large mass as a platform to which the resonant beam assembly can be attached using smaller and easily movable clamping plates. This base consists of a 4" thick steel plate which is integrally cast onto the top of a large concrete block. Each end of the 4"x10"x60" aluminum resonant beam is held between a pair of steel plates which are clamped to the steel and concrete base with a set of 1" diameter threaded rods. The ends of the threaded rods are anchored in the base with special nuts that slide in "T" slots machined in the steel plate (similar to a milling machine table). When the upper nuts on the threaded rods are loosened, each pair of clamping plates can be easily repositioned using hand wheel and ball screw assemblies. When the nuts are tightened, each clamping assembly approximates a fixed end condition on the resonant beam. The two sets of clamping plates are normally positioned symmetrically about the center (and impact point) of the beam, but the design allows for independent positioning which will provide the opportunity to investigate non-symmetric configurations in the future. The clamping plates are fitted with pneumatic piston and roller bearing assemblies that, when actuated (with the clamping nuts loose), lift and separate the clamping plates and resonant beam. This roller mechanism allows the clamping plates to be easily moved, and also provides spacing for the insertion of rubber or other damping pads.

A 3" ID air gun barrel is housed in a cylindrical space in the center of the concrete mass. The air gun breech, main valve, and reservoir are within the space under the center of the concrete mass. Other valving and controls are contained in an enclosure on the back side of the concrete mass (see photo in Figure 4). The air gun operates on compressed air or nitrogen, and has a Maximum Allowable Working Pressure of 300 psi. The gun is loaded through the breech which allows the resonant plate assembly to remain in place for this operation. The projectile is a 3" diameter flat nosed aluminum or steel cylinder up to 12 " long. It is fired vertically upward to impact the center of the beam, which is then excited into resonance. Alignment of the air gun barrel is not required since it is built into the apparatus design. The gun design is such that the projectile only partially exits the barrel upon impact , and thus rebounds back to the bottom of the barrel where it is in position for the next test. The impact duration can be easily adjusted by using various thicknesses of felt pads at the point of impact. The weight of the projectile will also affect the impact duration. The amplitude of the beam's resonant response can be adjusted with the impact speed (i.e. air gun firing pressure).

Experimental Results from Tunable Resonant Beam Apparatus

Tests were conducted for six different configurations which are summarized in the table below:

Table I

distance between clamps (in.)	neoprene damping pads	measured resonant freq. (Hz)	Resonant Freq. (Hz) from Eq. 1
30	no	630	900
24	no	1000	1400
18	no	1400	2500
30	yes	570	900
24	yes	750	1400
18	yes	1200	2500

The main conclusion to be seen from Table I, is that the resonant frequency is indeed tunable with the resonant beam apparatus. The measured resonant frequencies, are 30% to 50% lower than the frequencies predicted for a perfectly fixed-fixed beam of the same length. This trend was expected based on the small scale results, however, the difference was expected to be smaller. Figure 5 shows the time history and shock spectrum from the test on line 2 in Table I. The SRS plot shows the desired general shape for pyroshock simulation. Similar results were obtained for the other test conditions, except with different knee frequencies corresponding to each change in the beam length. The addition of neoprene pads appeared to do

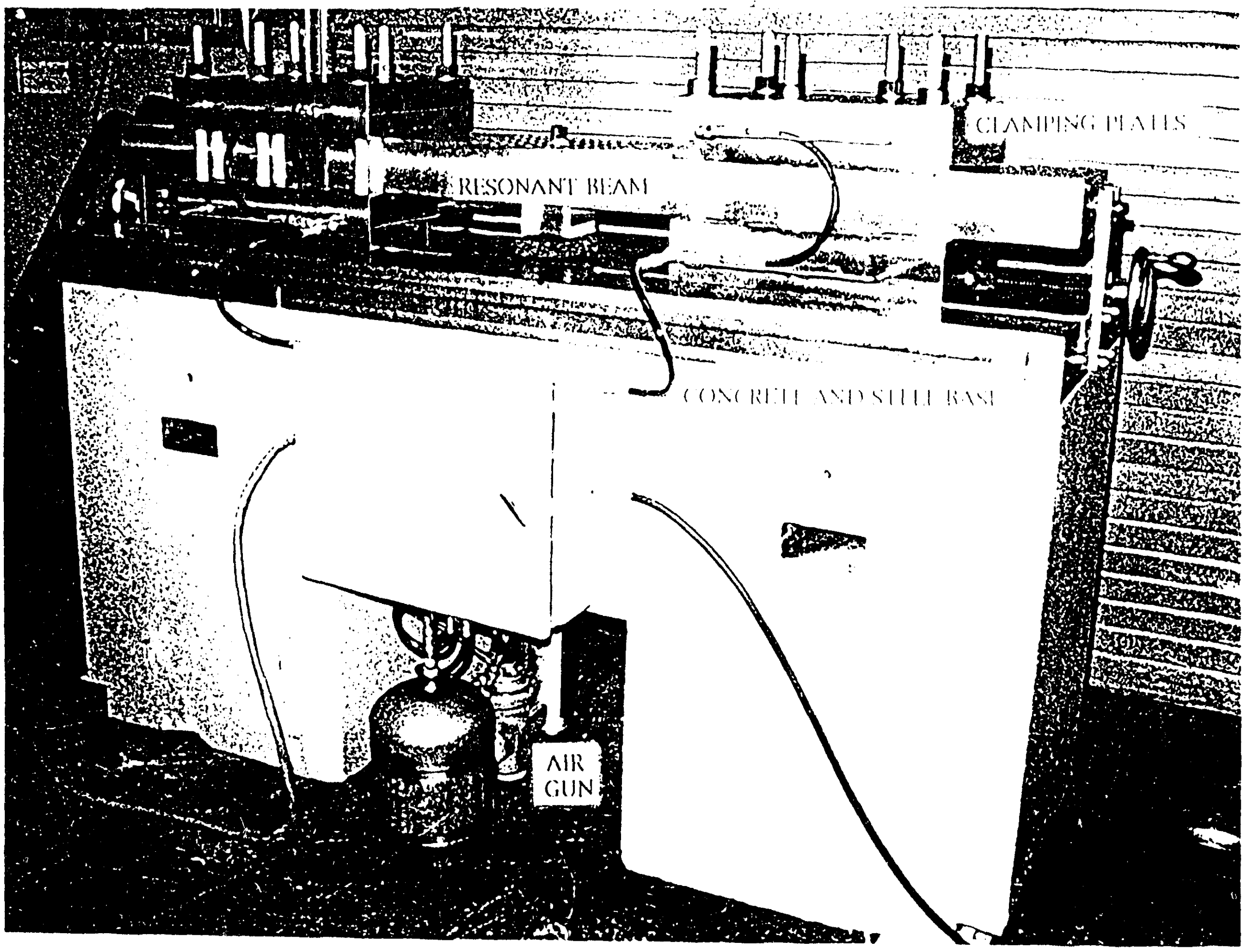

Figure 4 Tunable Resonant Beam Apparatus

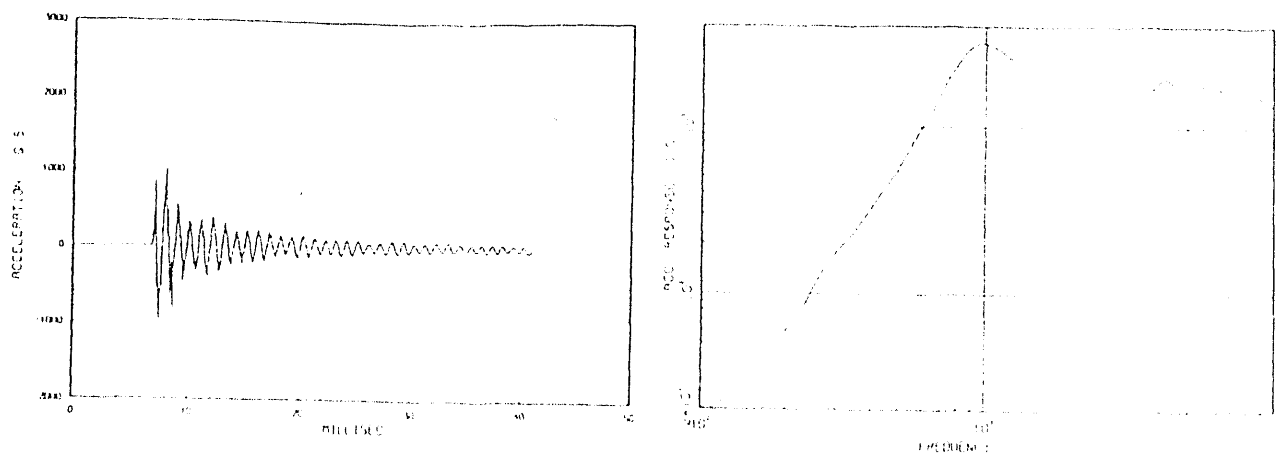

Figure 5 Time History and SRS, Tunable Resonant Beam with 24" Spacing

more to lower the resonant frequency than it did to increase the damping. Fortunately, the resonant beam apparatus in general is more damped than the small scale apparatus.

The apparatus was also modeled with the ALGOR code for the 24" clamp spacing. The modal analysis yielded a dominant mode at 1146 Hz. (compared to 1000 Hz measured). An ALGOR transient analysis also compared favorably with the experimental results.

Technique 2 - Tunable Resonant Bar

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). Recall that Equation 1 tended to overestimate the knee frequency by around 50% at frequencies approaching 3000 Hz (Table I). 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 6, 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. 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 II 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 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 above.

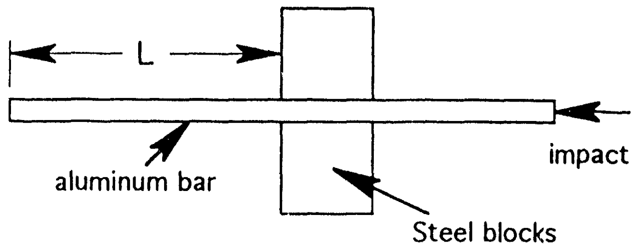

Figure 6 Tunable Resonant Bar - Concept 1

Table II

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

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 II. Details of these experiments will not be discussed in this paper. 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.

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.

$$f_1 \text{ (free-free)} = \frac{c}{2L} \qquad \text{(Ref. 3) Equation 2A}$$

$$f_1 \text{ (fixed-free)} = \frac{c}{4L} \qquad \text{(Ref. 3) Equation 2B}$$

where:
c = wave speed in bar (≈ 199,000 in/sec for aluminum)
L = bar length, (in)

Figure 7 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.

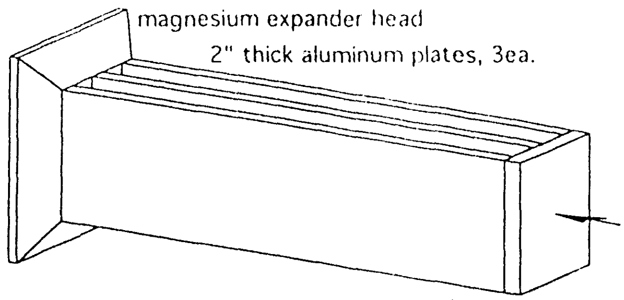

Figure 7 60" Tunable Resonant Bar - Concept 2

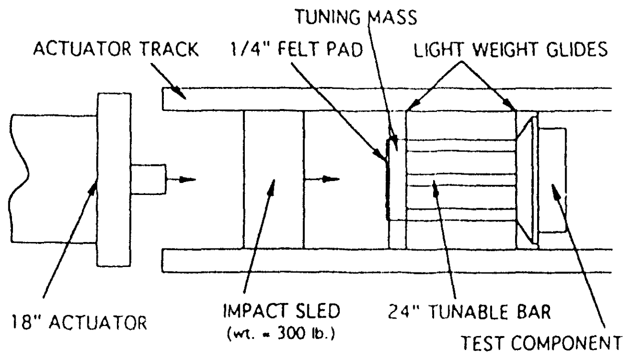

Figure 8 Tunable Resonant Bar Test Set Up

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

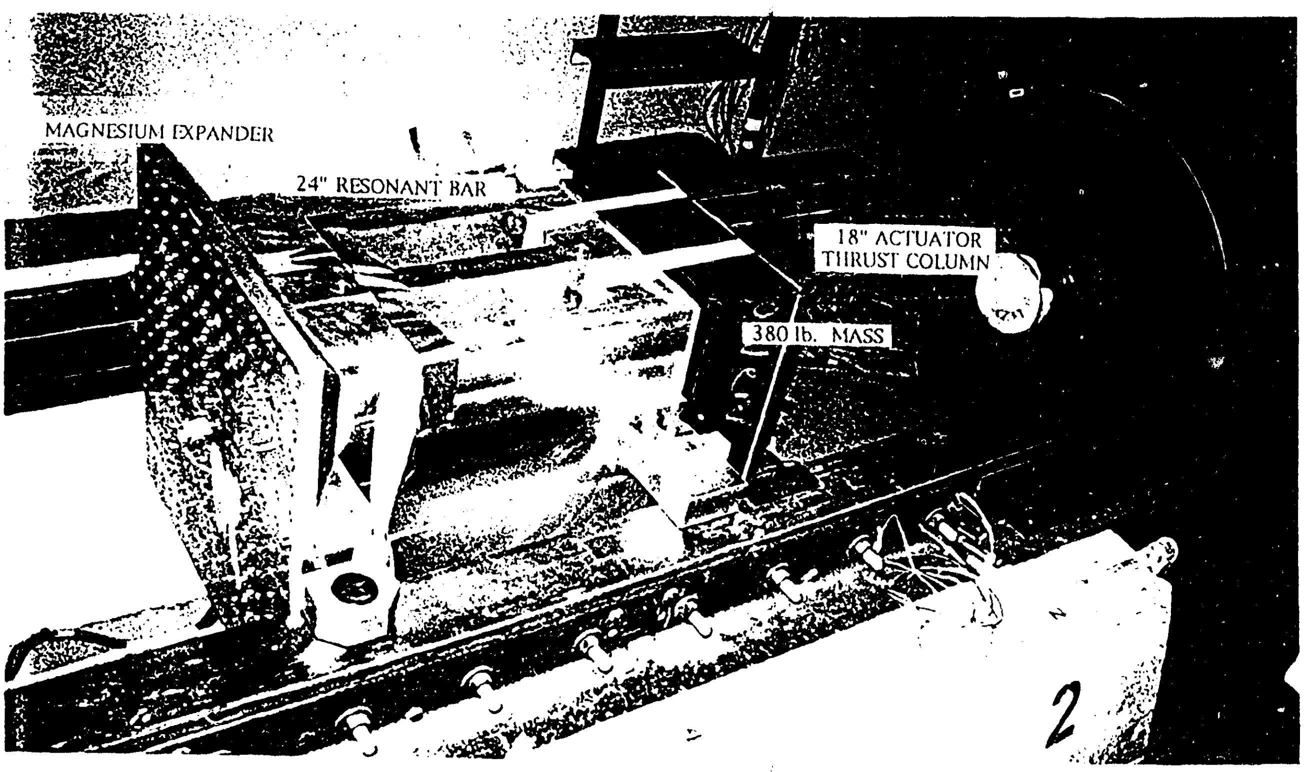

Figure 9 24" Tunable Resonant Bar on Actuator Track resonant bar fixtures were fitted with guides that allowed free motion along the actuator track. An impact from a small sled was used to excite the resonant bar fixture into longitudinal resonance. The resonant frequency of the bar can be reduced by bolting steel plates on the impact end. Figure 8 shows the test configuration, and Figure 9 shows a photo of the 24" bar fixture positioned on the actuator track.

Several tests as depicted by Figure 8 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 10 shows the acceleration time history and SRS for one of the tests on the 60" 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 III.

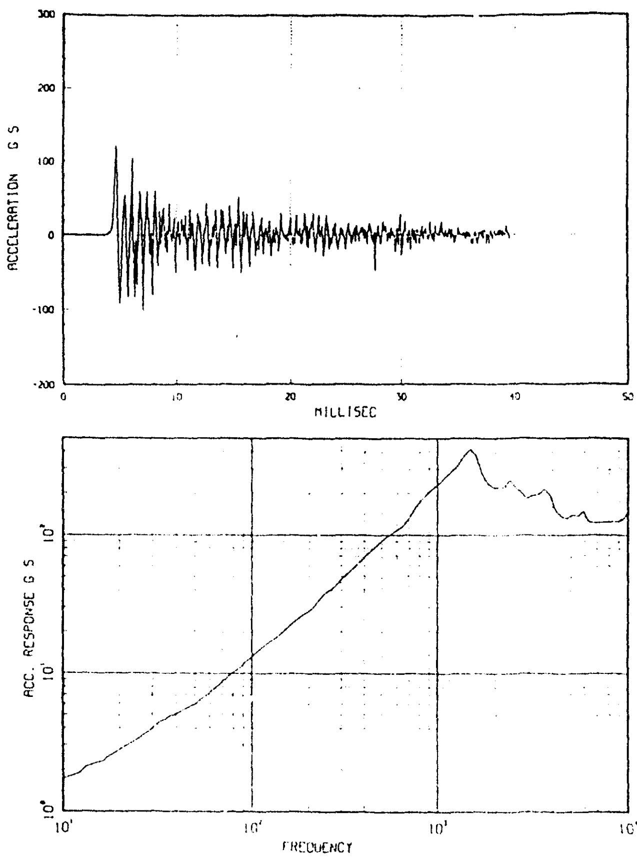

Figure 10 Time History and SRS, 60" Tunable Resonant Bar, without Added Mass

The results shown in Table III 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. An ALGOR finite element model could easily be developed to further aid in the test set up. 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 III

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 22"x22" mounting base can be tested. The first technique uses a tunable resonant beam excited into bending resonance, while the second 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, i.e. no knowledge of what resonant fixture geometry to start with, and no knowledge of the effect of experimental changes.

Acknowledgments

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