Type A Shipping Package Harmonic Assessment Methodologies

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ABSTRACT
Response to incident vibrations is a design aspect assessed during the development of Type A fissile shipping packages. Not only are expected transport induced vibrations evaluated for package structural integrity, but also to ensure the integrity of the un-irradiated fuel assembly. Such incident vibrations include, but are not limited to, expected transport induced vibrations, and single-pulse shock inputs occurring at relatively high frequencies.

The regulatory requirement to ensure there is no deterioration in the effectiveness of any closure devices or in the structural integrity of a Type A fissile shipping package is governed by TS-R-1 [1] as well as 10CFR71.71 [2]. In addition to regulatory requirements, the fissile Type A contents, more specifically the fresh fuel assembly, have design specifications that require shipping and handling acceleration loads remain below a design threshold. Other structural design considerations include supplemental vibration damping system(s) tuned for package design features and transport conveyance interfaces which may induce vibrations.

A test program was developed and executed to fully assess the Westinghouse Traveller fresh fuel shipping package harmonic characteristics considering regulatory and fuel assembly requirements. Road course testing, vibration shaker table testing and port handling simulation/drop tests with production Traveller shipping packages were utilized for evaluation. The testing also provided valuable data to determine the proper material selection for shock mitigating materials, their configuration within the shipping package, and identifying resonance frequencies associated with the shipping package design structure.

Once an in-depth understanding of the packaging and fuel assembly's harmonic response to shipping and handling induced vibrations and single-pulse shocks was established, an optimized suspension system design configuration to dampen those incident vibrations was realized.

1. INTRODUCTION
Shipping package mechanical design requires structural and thermal assessments governed by TS-R-1 [1] as well as 10CFR71 [2]. Regulations describe hypothetical accident conditions as well as normal conditions of transport; both of which require the package demonstrate compliance by analysis or testing. In addition to regulatory requirements, Westinghouse imposes design limits on certain structural components of its fresh fuel assemblies. These design limits must be accounted for as part of the shipping package design.

The new Westinghouse shipping package consists of an aluminum inner structure connected to an Outerpack by elastomer neoprene shock mounts, and a damping foam. A single fuel assembly is restrained inside the inner structure by positive axial actuation, and laterally by soft rubber compression pads. The shock isolation and damping system was tuned to limit tri-axial displacement of the inner container; primary concern was the vertical displacement resulting from (1) transport induced (random) vibrations and (2) higher amplitude single-pulse shocks. Knowing that trailer transport natural frequencies are approximately 4.5-6.5 Hz [3], a design objective was to ensure the inner container resonance was damped within this range during transport, otherwise undesirable oscillations could result. Another design objective was to isolate single-pulse shock accelerations occurring at a relatively high-frequency.

Random Vibrations

Road conveyance typically results in random vibration with varying pulse widths. Figure 1 provides an example of road transport induced vibrations when measured using a piezoelectric accelerometer. Due to the random nature of the frequency spectra during transport, power spectral density (PSD), also referred to as acceleration power density (ASD) in transport applications, better describe the amplitude and frequency profiles.

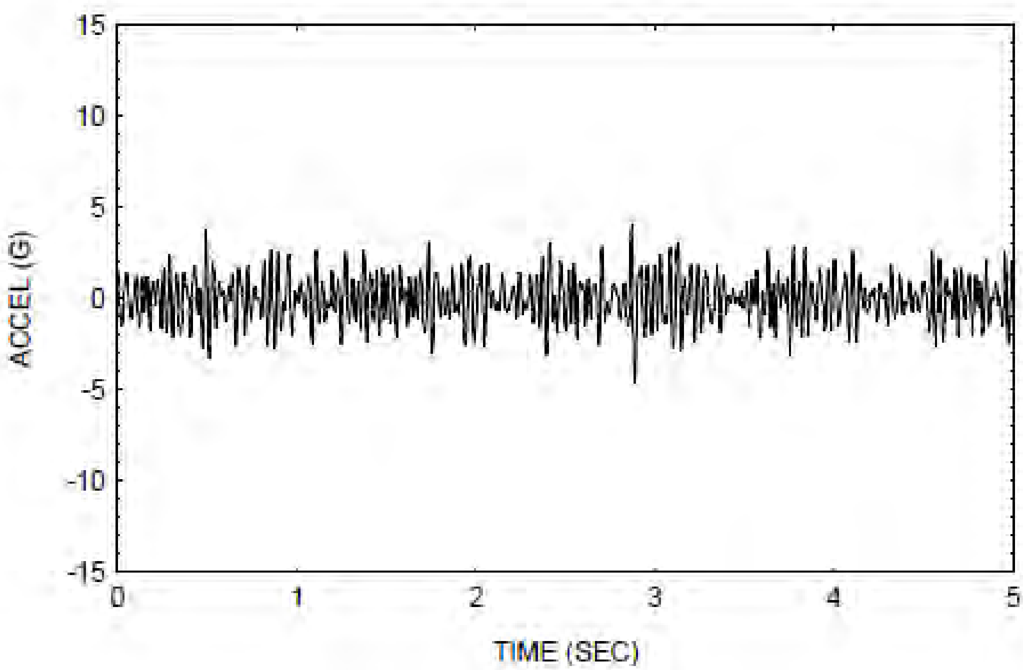


Figure 1 – Example Random Input Frequencies from Road Transport

Random input frequency spectra can be represented by an ASD function with units of acceleration force squared per unit of frequency (G^2/Hz). By filtering the input time-dependent frequencies, the spectral components can be separated by each corresponding time history using band-pass filtering. Using the frequency bandwidth range and associated amplitude value (typically in G_{rms}; where G_{rms} is the square root of the summed squares), the ASD can be calculated simply by dividing the square of the amplitude by the frequency bandwidth:

$$ASD\left(\frac{G^2}{Hz}\right) = \frac{Grms^2}{f\ band}$$

A random frequency signal is plotted by ASD as a function of mid-band frequency as shown in Figure 2, which represents the profile of the random vibration. A spectral G_{rms} value can be calculated by integrating area under the ASD curve. The plot is useful since the largest amplitude (power) can be associated with a frequency (range), and a shock dampener designed accordingly. It is noted that most computer software utilizes the Fourier transform method to obtain ASDs.

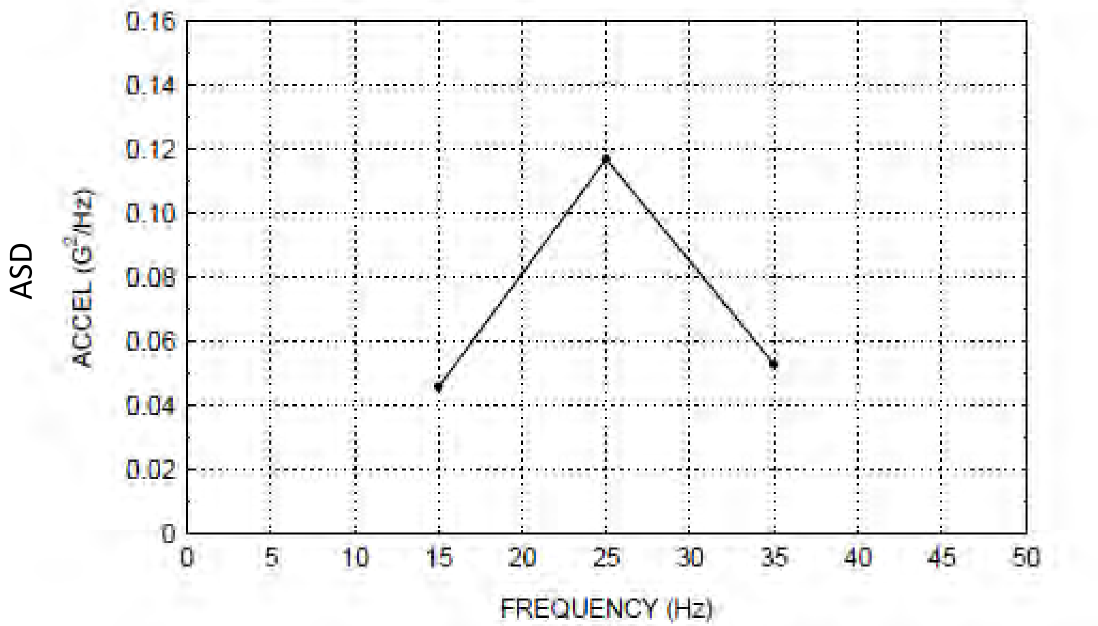


Figure 2 – Example Band-Pass ASD Plot of Random Frequency

The shock isolation system's desired response is shown graphically in Figure 3, which is obtained by the general solution to the mass equation of motion for an under-damped system (x is displacement and T_d is the damped period). Transport induced random vibration amplitudes are sinusoidally reduced which protects the shipping package and its contents from potential mechanical damage.

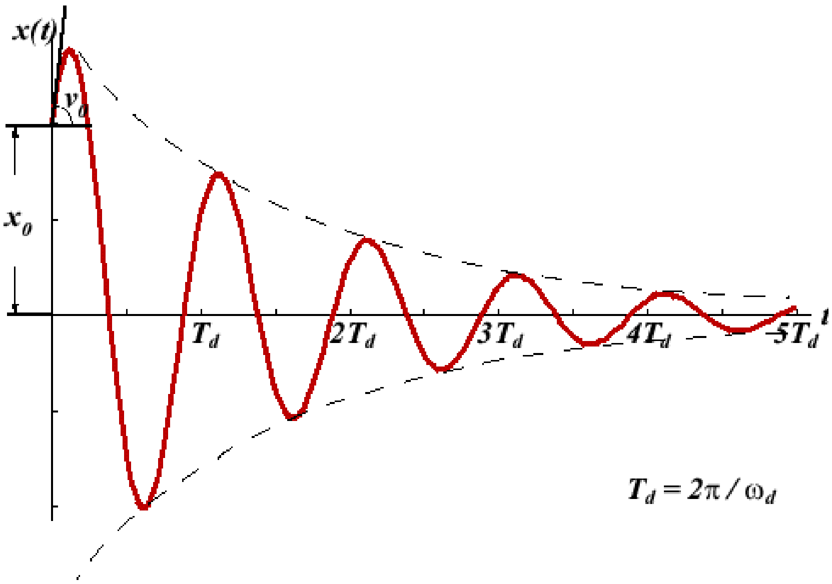


Figure 3 – Damped Sinusoidal Response of a Shock Isolating Element

The performance of an isolation system can be determined by the transmissibility (also referred to as "Q" factor) of the system; defined as the ratio of the energy going into the system to the energy coming from the system. Transmissibility can be expressed in terms of acceleration, force or vibration amplitude. Transmissibility (T) is equal to the following (for a single-degree-of-freedom system):

$$T = \left| \frac{A_{out}}{A_{in}} \right| = \sqrt{\frac{1 + \left(2\varsigma \frac{f_d}{f_n}\right)^2}{\left[1 - \left(\frac{f_d}{f_n}\right)^2\right]^2 + \left[2\varsigma \frac{f_d}{f_n}\right]^2}}$$

Where: T = Transmissibility

A_{out} = Energy out of system (transmitted force)

A_{in} = Energy into system (Disturbing force)

z = Damping ratio

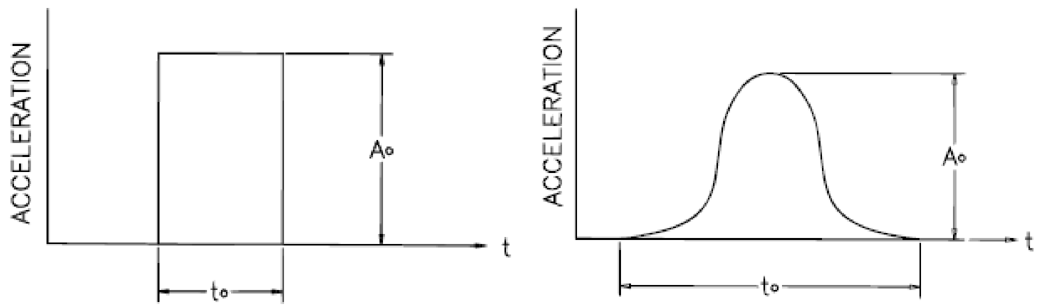
f_d = Driving frequency

f_n = Natural frequency

If the driving frequency equals the natural frequency ($f_d/f_n = 1$), the system operates at resonance. If damping is ignored in the equation for transmissibility, a system that is operating at resonance will have a transmissibility approaching infinity. As damping increases, the transmissibility at resonance decreases. When a vibration isolation element with very little damping is used at or near resonance, the energy amplification can create many problems, ranging from a simple increase in noise levels to catastrophic damage to mechanical equipment.

Single-Pulse Shocks

During handling (and in some cases transport), a transient condition occurs where a single pulse of energy is imparted to the shipping package in a short time period and with a large acceleration as shown in Figure 4. The shipping package shock isolation system is designed to reduce the amplitude of a potentially damaging high-magnitude shock.



$V = gA_0t_0$ $\qquad$ $V = \frac{g}{2}A_0t_0$

Figure 4 –Acceleration Shock Pulse Examples

The shock isolator will undergo displacement as it dissipates the incident mechanical energy; the shock isolation element increases the time to decelerate the mass, thus reducing the magnitude of the input acceleration force imparted to the structure. This desired response is shown graphically in Figure 5.

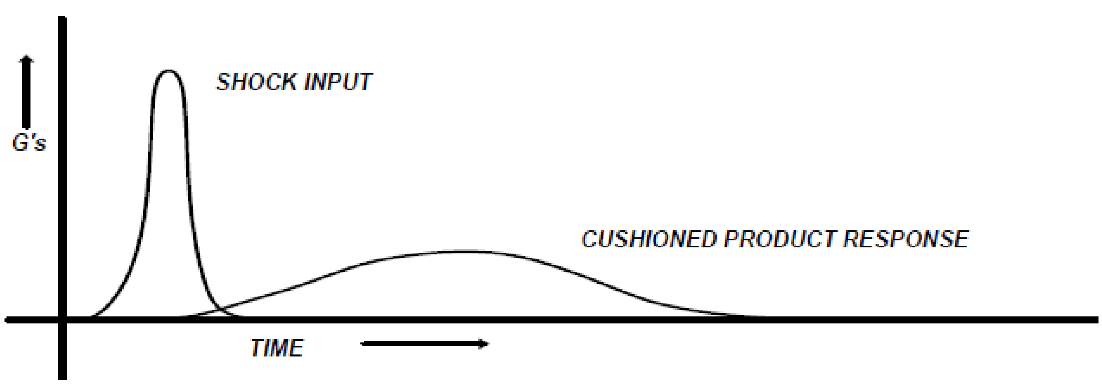


Figure 5 – Cushioned Response of a Shock Isolating Element

2. TEST METHODS and TEST RESULTS

A test program was developed using production shipping packages and replica fuel assemblies to properly simulate actual transport and handling conditions. The test program included road course testing, drop testing, and shaker table testing of a production shipping package. Also evaluated were a lead-filled replica fuel assembly and a mild steel ballast. Data post-processing provided the basis for design selection of materials, package design configuration, and also coupling methods to the transport conveyance.

Road Course Testing and Results

Standard tractor-trailers were used to conduct a series of road course testing, and standard tie-down methods were employed. Trailers were loaded with at least two production shipping packages containing either a replica fuel assembly or a steel ballast. The primary objective was to characterize the shock mount isolation system, and the secondary objective of the testing was to determine if the package responded harmonically different between a steel ballast and a replica fuel assembly since the ballast is relatively rigid as compared to the replica fuel assembly. The remainder of the transport load was empty packaging for load balancing as seen in Figure 6. Approximately 25 road tests were conducted with varied road conditions. The conditions included interstate highways, local roads and a designed course with speed bumps and large curbs.

Figure 6 – Tractor-Trailer Rig with Test Load

The trailer bed accelerations are practically equal to the input shipping package vibrations since the package stack is essentially coupled to the trailer bed. Tri-axial piezoelectric accelerometer/data recorders were mounted on the trailer bed in three locations: driver's axle end, middle, and over the rear axle as shown in Figure 7. The recorders were set to trigger at a relatively low force level (0.5G - 2G depending on test), and then record for several hundred milliseconds ensuring that the pulse is recorded. Inside the shipping package, the inner structure was fitted with piezoelectric accelerometer/data recorders as well. This permitted acceleration measurements of the inner structure for an initiating energy pulse, although the trigger was set approximately 20% lower than the external trigger since damping was expected through the shock mounts and damping foam. Figure 8 shows an example of a mounted piezoelectric accelerometer/data recorder located at the inner structure's top end.

Figure 7 – Accelerometer Monitor and Recording Device on Trailer Bed

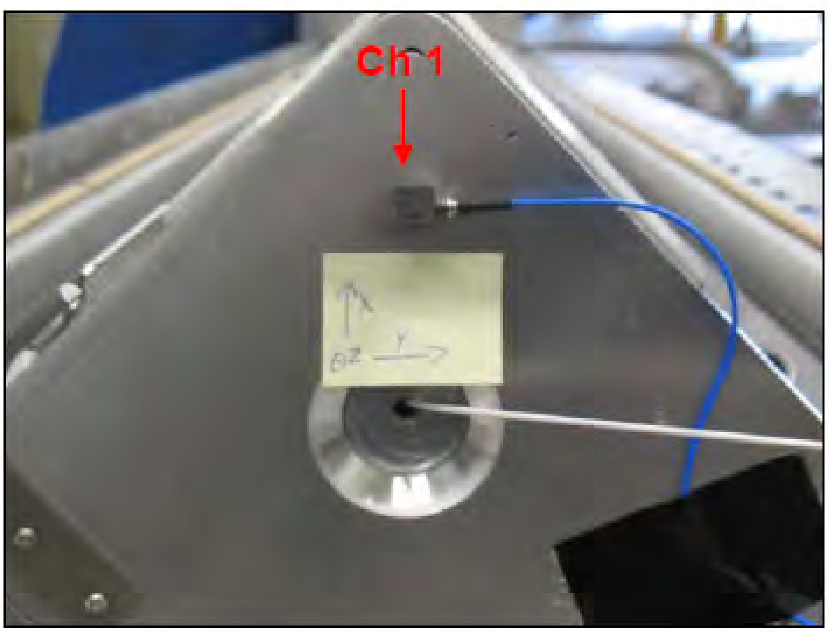


Figure 8 – Accelerometer Monitor and Recording Device on Inner Structure

A typical road test involves a speed bump test at varying speeds. Testing indicated that at speeds less than 15 MPH, and greater than 25 MPH, the shock pulse did not possess enough energy to generate significant forces at the inner structure [due to small amplitude at low frequency or large frequency at high amplitude]. Figure 9 provides an example of a speed bump road course. Including random shocks from transport and speed bumps, a typical test of 1000 miles would result in approximately one hundred recorded vibration events.

Figure 9 – Speed Bump Test Road Course

A typical power spectral density for an entire test trip is shown in Figure 10. The PSD graph shows that the region of highest energy input is 7-9 Hz, which identifies a frequency range that could result in undesirable acceleration forces on the inner structure or fuel assembly and possibly a resonance condition of the inner structure. Although there is an energy spike at higher frequencies, the input frequencies typically pass through the mechanical structure without effect. Testing indicated that the replica fuel assembly and mild steel ballast resulted in statistically similar harmonic responses, with respect to the precision of the test equipment.

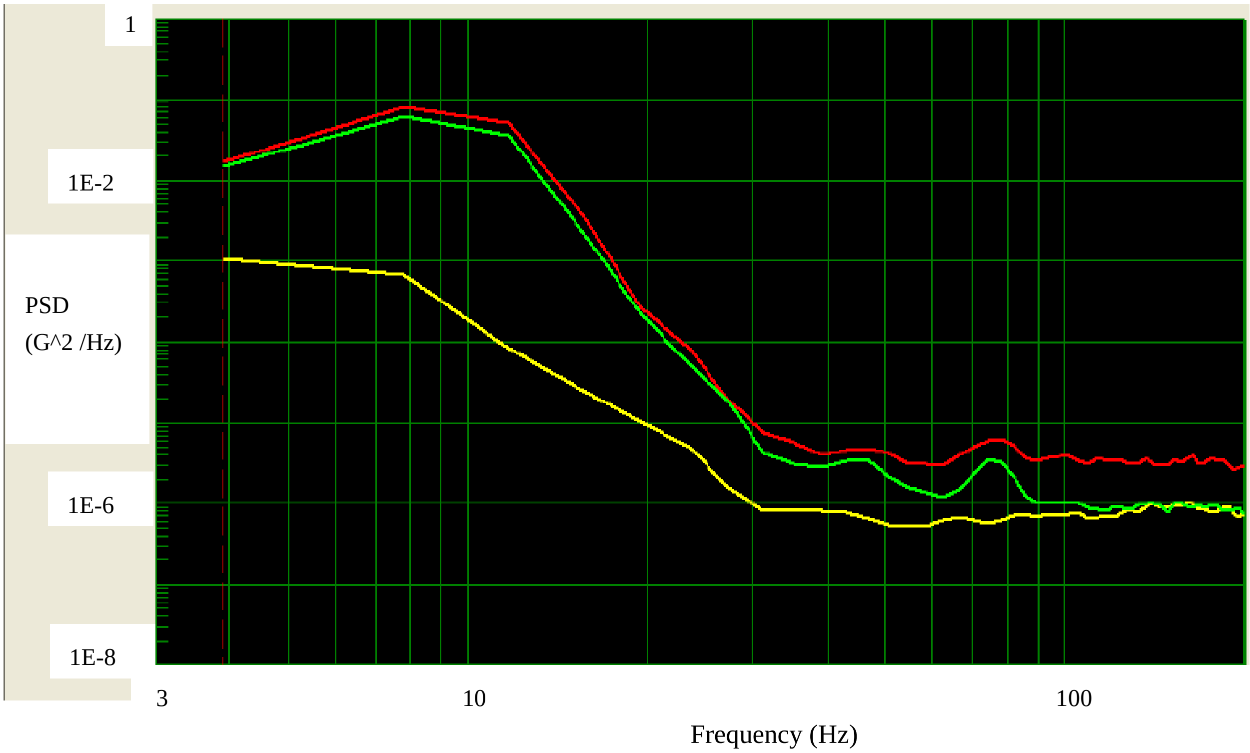


Figure 10 – Road Course Test PSD

Shock (Drop) Testing and Results

Using the same test configuration as the road course test, transport shock testing was performed entailing a curb drop-off. The tractor-trailer was driven over the curb as shown in Figure 11 at relatively low speeds to induce a low-frequency, high amplitude energy pulse.

Figure 11 – Example Curb Fabricated for Shock Test

Figure 12 presents the truck-bed measured acceleration data as a function of time recorded at the rear axle. As the rear portion of the trailer passes over the curb, the rear piezoelectric accelerometer begins to oscillate between 2G and -2G; approximately 0.5 seconds later the rear axle passes over the curb and responds with an approximate 5.5G shock at approximately 40-50Hz.

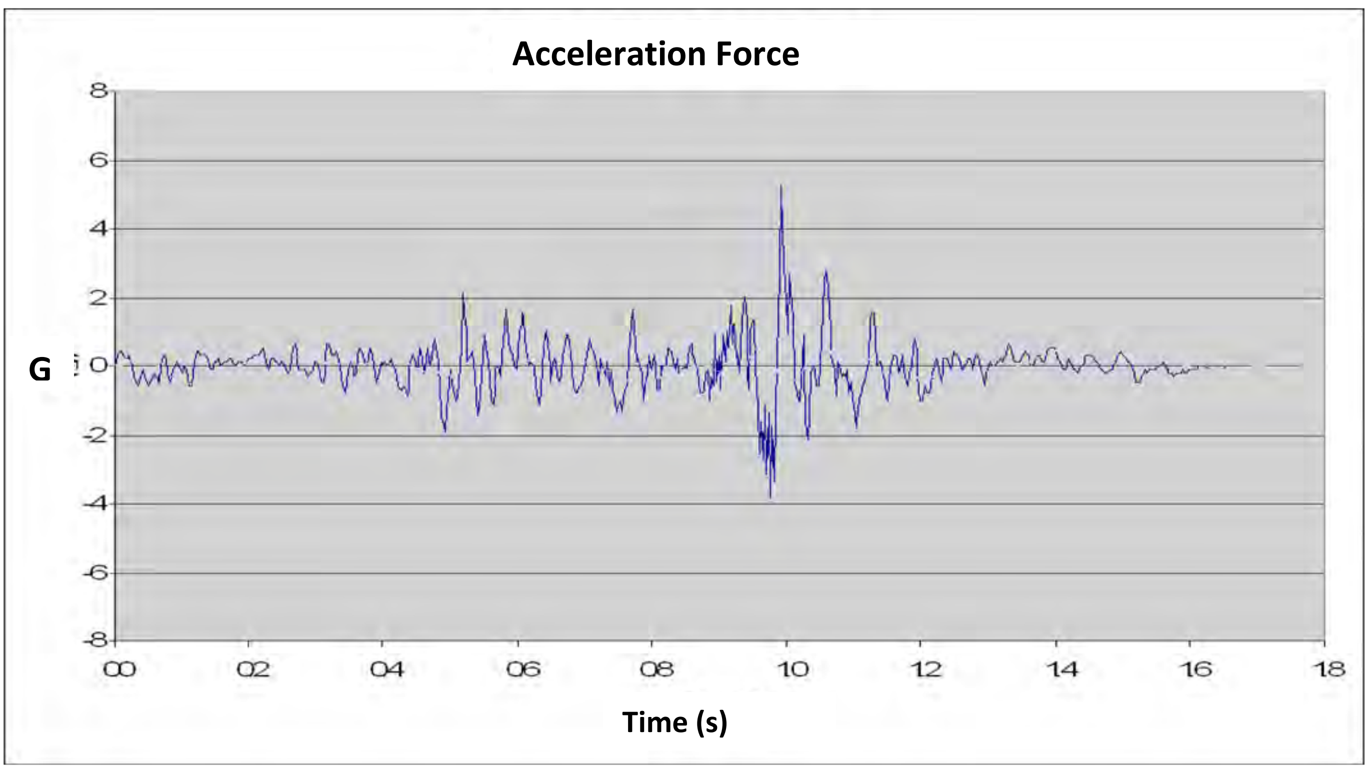


Figure 12 - Incident Vertical Accelerations from Curb Shock Test Measured on Trailer Bed

The shipping package outer and inner structure response is shown in Figure 13 by blue and red lines, respectively. Since the outer structure is coupled to the trailer, its response is rigid and nearly identical to the incident vibration profile. The inner structure is isolated from the outer structure with a rubber shock mount system; its response shows the initial shocks amplified with the more energetic, higher frequency pulse damped to an acceptable level. These measurements indicate that resonance could occur for longer pulses with large amplitudes.

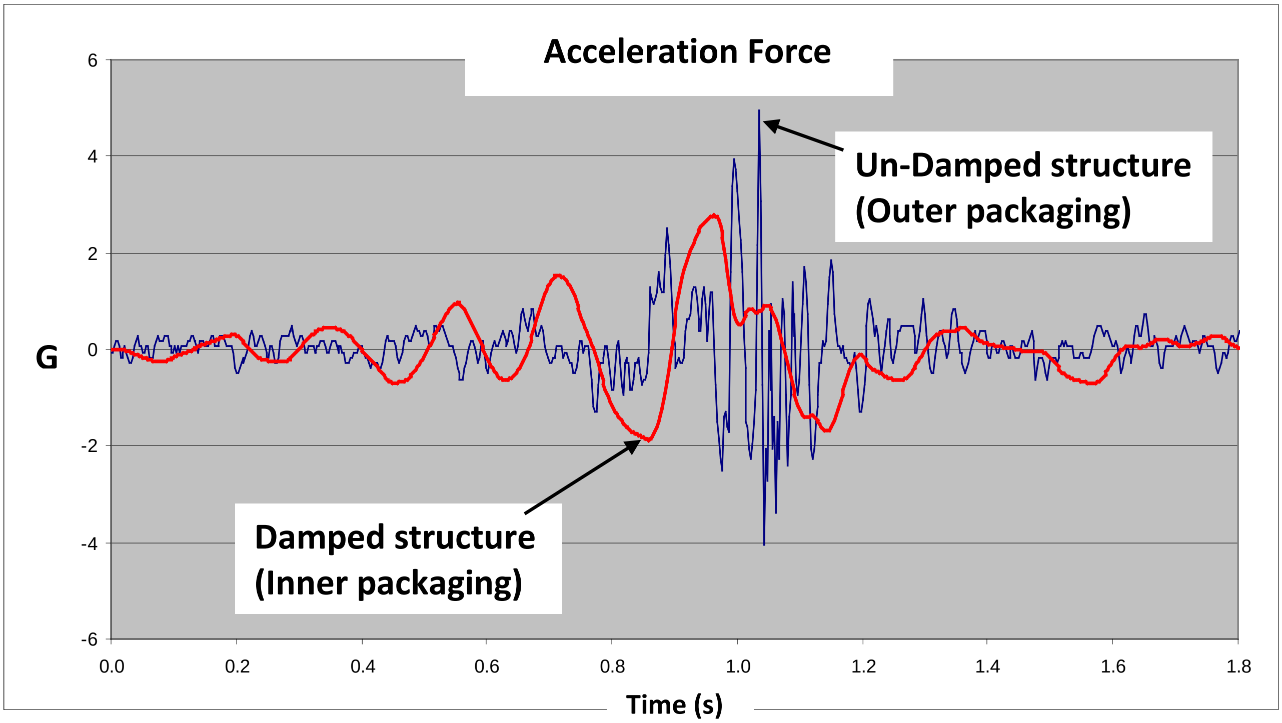


Figure 13 - Outer and Inner Structure Shipping Package Response to Shock Acceleration

From the inner structure harmonic responses, the elastomer neoprene shock mount system design has potential for amplification in the 7-10 Hz range, and also 70-80 Hz from the PSD in Figure 10. In order to choose a different design configuration, including but not limited to different materials, shaker table testing was selected since the input vibration profile can be controlled and replicated.

Shaker Table Testing and Results

Road course and shock testing indicated an acceleration spectrum of undesirable amplification in the 7-10 Hz range. Due to the shipping package aspect ratio and elastomer neoprene shock mounts, the inner structure may respond with relatively large displacement and thus undesirable acceleration forces. Two prototype suspension systems were designed for testing and evaluation: silicon rubber shock mounts and the base elastomer neoprene shock mounts with supplemental shock-mitigating foam. These designs were compared to the base elastomer neoprene shock mounts system to determine the effective damping (transmissibility).

An effective method to assess suspension system design is a controlled vibration testing on a shaker table. By measuring transmissibility, modified shock system designs may be compared to the current shock system design. Shaker table inputs are typically a repeatable sine sweep vibration input, governed by ASTM D4728 standard. Both a replica fuel assembly and a mild steel ballast were tested on the vibration shaker table to accurately represent shipping package mass properties. The shaker table transmissibility testing consisted of:

- Sine sweep per ASTM D4728 with constant 0.5g acceleration. The frequency was increased 2 octaves/minutes from 3 Hz to 200 Hz.

The test set-up is shown Figure 14. The left image shows the package closed with the wiring installed and the truck tie-down simulated. The right image shows the new silicone shock mounts installed. Figure 15 shows the test set-up with the base elastomer shock mounts and the Ethafoam addition.

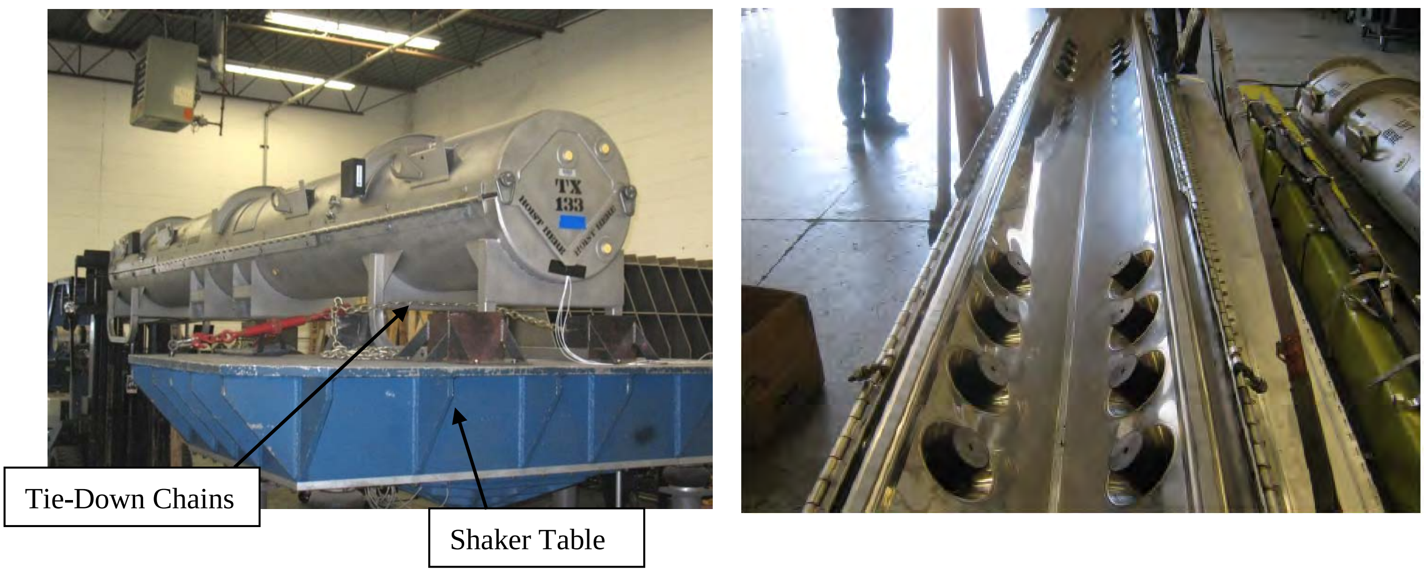


Figure 14 – Shaker Table Testing Set-up. Picture on Right Shows Silicone Shock Mounts

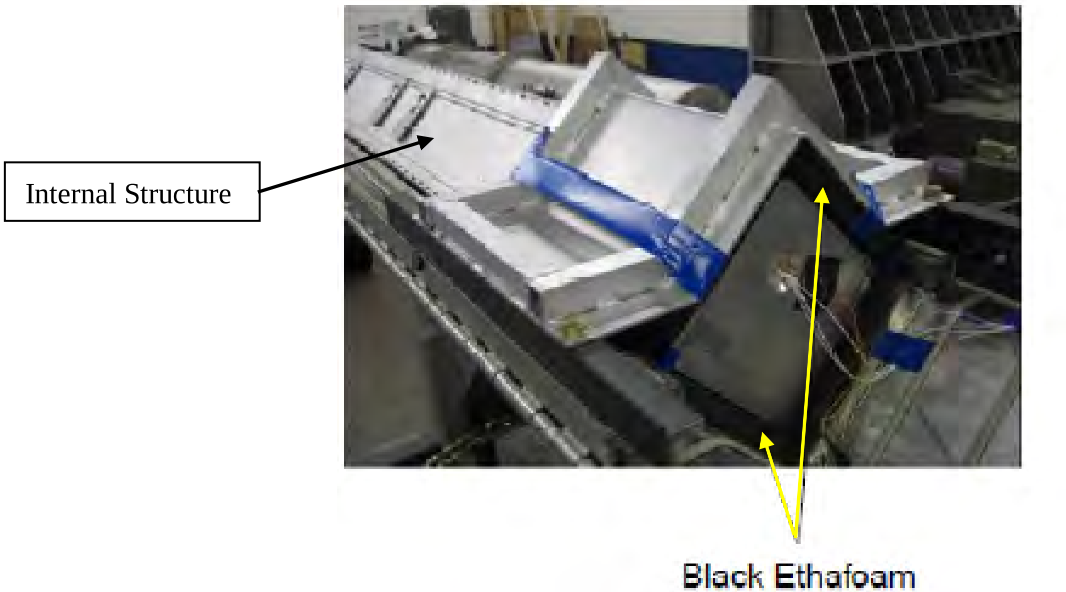


Figure 15 – Modified Suspension System with Ethafoam and Elastomer Neoprene Shock Mounts

Piezoelectric accelerometers were positioned on the inner structure’s top end, as this section has the greatest potential for relatively high amplitude response accelerations that are associated with greater vertical displacement. The measured accelerations were divided by the input accelerations to determine the transmissibility (Q), and the software subsequently plotted the transmissibility as a function of frequency. This process was performed for each suspension system design and repeated to ensure data reliability. Figure 16 provides the three suspension system designs’ typical transmissibility response to the sine sweep. Note that above 100Hz all suspension systems are effective dampers as the transmissibility is below 1.0. The sine sweep testing indicates that all three suspension systems posses two transmissibility peaks; in the 7-10Hz range and one at approximately 60Hz.

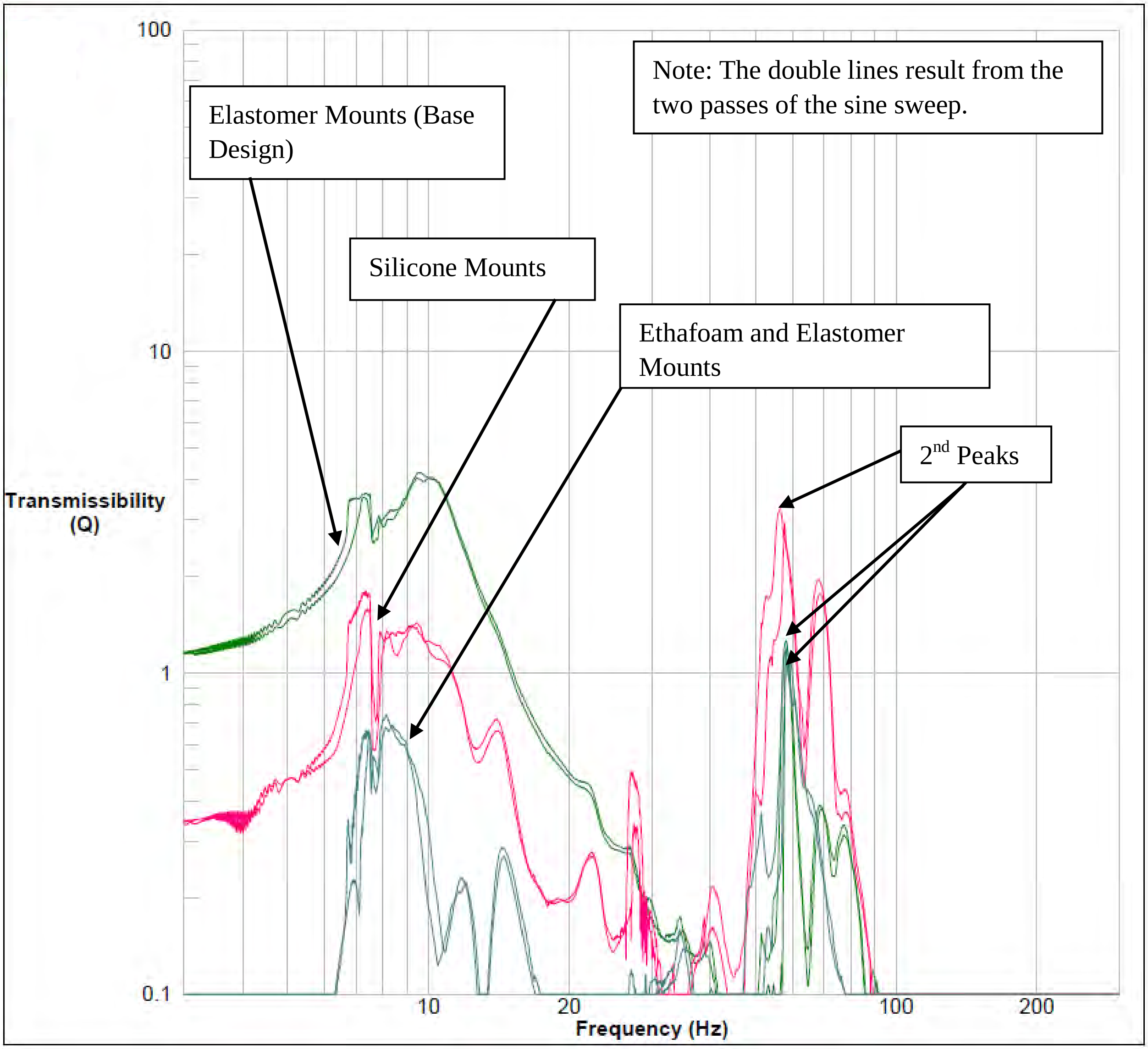


Figure 16 – Transmissibility Response of Three Suspension Systems

The transmissibility data demonstrates that the Ethafoam/elastomer neoprene shock system provided the optimum damping for the sine sweep. Testing indicated that the elastomer neoprene shock mount system could result in resonance in the 7-10 Hz range. Figure 16 shows the transmissibility is approximately 0.75 thus effectively damping the incident accelerations in the 7-10 Hz range. The resultant data is presented in Table 1. It can be seen the base design (elastomer neoprene mounts) tends to amplify the input accelerations at low frequencies and is in agreement with road test PSD data. Both silicone and Ethafoam/elastomer neoprene mount designs are effective dampers at low frequencies. The second peak response demonstrated essentially no amplification by the elastomer neoprene and Ethafoam/elastomer neoprene mount designs since the transmissibility value was approximately 1.0. The silicone mount suspension system demonstrated amplification between 55 and 60Hz. This shock mount system is not an optimum design since there is the possibility that the system may produce unacceptably high accelerations at higher frequencies.

Table 1 – Measured Suspension System Transmissibility and Associated Frequencies

Shock System Design	1st Peak Q	1st Peak Frequency	2nd Peak Q	2nd Peak Frequency
Elastomer Mounts	4.20	9.2 Hz	1.12	58 Hz
Silicone Mounts	1.85	7.3 Hz	3.15	56 Hz
Ethafoam & Elastomer Mounts	0.73	8.1 Hz	1.03	58 Hz

3. CONCLUSIONS

A test program was developed to assess the harmonic characteristics of the new Westinghouse Traveller shipping package. A production shipping package was evaluated to identify acceleration responses to transport and shock induced vibrations. Road course testing and shock (drop) testing provided harmonic responses to expected transport conditions. Data from these various transport tests demonstrated a potential for inner structure undesirable accelerations in the 7-10 Hz range. Vibration shaker table testing was then utilized to further evaluate specific frequency spectral responses and provided a controlled, repeatable test condition. It is noted that ASTM D4728 random testing (ISTA 3H Air Ride) may also be performed on a vibration table to determine the amplitude and frequency profiles in terms of acceleration power density (ASD) in lieu of over-the-road testing.

The testing provided comparative data to assess different shipping package suspension system material and configuration designs. Two performance objectives were achieved by this testing program: 1) The suspension system demonstrated effective reduction of shock impulses in cases of rough handling such as speed bumps, and 2) The test program resulted in an optimized suspension system design configuration to dampen random incident accelerations for a frequency range of 3-200 Hz.

4. REFERENCES

1. IAEA Safety Standards Series TS-R-1, Regulations for the Safe Transport of Radioactive Material, 2009 Edition, Page 89.
2. United States Nuclear Regulatory Commission Regulations, Title 10, Code of Federal Regulations, Part 71, Subpart F.
3. Winkler, C.B., UH-HSRI-PF-74-9, “An Examination of Ride Problems of the 6500 Gallon Liquid Carbon Monoxide Trailer”, September 1974.