Designing Hardware for the Boundary Condition Round Robin Challenge

David E. Soine and Richard J. Jones, Jr., Kansas City National Security Campus*, 14520 Botts Road, Kansas City, Missouri, USA
Julie M. Harvie, Troy J. Skousen and Tyler F. Schoenherr, Sandia National Laboratories** P. O. Box 5800-MS 0840, Albuquerque, NM 87185

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Qualification of products to their vibration and shock requirements in a laboratory setting consists of two basic steps. The first is the quantification of the product's mechanical environment in the field. The second is the process of testing the product in the laboratory to ensure it is robust enough to survive the field environment. The latter part is the subject of the "Boundary Condition for Component Qualification" challenge problem. This paper describes the challenges in determining the appropriate boundary conditions and input stimulus required to qualify the product. This paper also describes the step sand analyses that were taken to design a set of hardware that demonstrates the issue and can be used by round robin challenge participants to investigate the problem.

Fixture design strategies for many component level shock and vibration tests remain as they were in the middle of the last century. The traditional approach attempts to provide rigid boundary conditions (or "infinite impedance") at the component interface locations. The literature refers to two general ideas for rigid fixture guidance. The first, more strict, guidance is to design a fixture to have a first participating mode at a frequency at least four times the maximum test frequency. A second more general guidance is for the frequency of the first participating mode of the fixture to be higher than the maximum test frequency. Analogous guidance based on pulse duration can be found for shock testing. Unfortunately environment specifications, component attachment interfaces, and environmental test fixtures are generally not developed as a cohesive package; they are often independently developed by different engineering groups. Component shock and vibration tests can be specified to (higher) frequencies or (shorter) durations, without considering the component fixture design at all, making it essentially impossible to design a rigid fixture according to the guidance-the dynamic modes of the fixture (and likely the test system to which it is attached) will be excited.

The structural dynamics and environmental testing communities have long recognized the shortcomings of the rigid fixture approach. Once the test envelope has moved out of the rigid body motion regime, it has been shown that dramatic under- and over-testing can occur due to the structural dynamics of both the next assembly and the lab test system itself. Scharton[1] documented potential approaches to address the issue, including consideration of modal density of the vehicle assembly as a tool to guide fixture and test design, followed later by Edwards[2] who outlined the potential of partial next-assembly style test fixtures. Various military and space testing standards have evolved to deal with the issue, focusing on full and sub-assembly level or large component test articles. They mention dynamically representative test fixtures, and give guidance on other techniques such as force limiting to manage the risk of overtest, but specific design guidance for component level fixtures is minimal. Daborn[3] has proposed techniques that utilize a large portion of the next assembly as the test fixture, and Mayes[4] has been developing substructuring methods that could be useful to evaluate test fixture design.

The component qualification process

The basic goal of the challenge problem is to design an environmental shock and vibration test fixture for a component. The term component implies that it is a part of whole-components are built up into sub-assemblies and sub-assemblies are combined to create a full vehicle assembly. The means by which the components are attached to the next assembly varies and includes weldments, bolted joints, potting, clamping systems, adhesive joints, and other methods. Historically, some of the most challenging components to test have multiple connection points to the next assembly.

Shock and vibration testing at the component level is carried out to demonstrate 1.) that an individual component will survive and function correctly throughout the expected life of the component, and 2.) that the manufacturing process is being carried out correctly on an ongoing basis. Component shock and vibration specifications in many aerospace applications are derived from actual flight tests of some full vehicle assembly. The derivation of the specifications can involve a substantial amount of data analysis, modeling, test time compression techniques, ground tests of sub-assemblies, and other engineering processes. The goal is to deliver an environment that can be applied in the test lab that reflects the expected field loading of the component and exercises the same failure modes.

Inception of the challenge problem

Faced with an expanding test environments envelope and an aggressive production schedule, engineers at the Kansas City National Security Campus proposed a research project to harness emerging additive manufacturing technologies for environmental test fixture design and fabrication. It was expected that topology optimization would play a significant role in the effort, and the team realized that understanding the criteria for success-the objective function for the optimization-might be non-trivial. This work aligned with ongoing research at Sandia National Lab and teams from both organizations began collaborating in 2016. The combined team recognized that input from the wider aerospace testing industry as well as the structural dynamics community would be needed to make lasting and significant progress. This led to the idea of a common test bed or round-robin challenge problem to make it easy for researchers and small teams to engage the issue.

Hardware goals, concepts, and first demonstration

As the research team considered the scope of the process and envisioned a hardware model, several goals were formed:

1. The model would include a full assembly, to enable direct formulation of an environment and to provide a "truth" test.
2. The model would have a simple component that can be easily detached from the assembly and tested individually.
3. The full assembly will be low cost and simple to fabricate or acquire.
4. The component would be small enough to test on a small environmental shaker system or shock machine.
5. The component response should be non-trivial. Single axis inputs result in multi-axis motions.
6. The component and assembly interact dynamically.
7. The under-and over-test problem would manifest if a rigid fixture approach was used. After a brief review of objects already in use for research and round-robin testing, the team brainstormed on potential structures for the

full assembly and the component. Use of an assembly that could be sourced locally or purchased at retail was strongly considered but the team could not locate an assembly with the required simplicity. Review of materials from the product list of McMaster-Carr was carried out and several schemes for bolted assemblies were proposed. The square tube-section was conceptualized and the team recognized it might have the right combination of simplicity and structural dynamics for the project.

Regarding component design, the desire for non-trivial component response drove the team to consider a three leg design.

The challenge with the three-leg design was in how to attach it to the assembly. A competing proposal was to use a simple beam along with some short sections of C-channel. A finite element model and additively manufactured plastic prototypes were created for both proposals, and because of its immediate availability, hardware was ordered for the C-channel design.

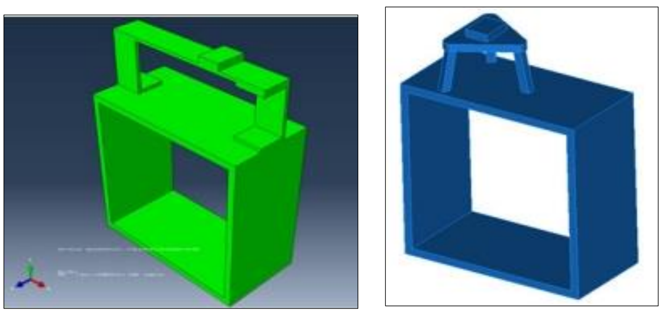

Figure 1. Component concepts: C-channel vs. three-leg designs.

Prototype development

The first prototype was constructed of steel, and six inch square steel tube was only available with rounded corners. The model was updated to reflect the rounded corners, and a presentation demonstrating the component boundary condition problem was generated, elements of which are shown in Figures 2, 3, and 4, and Table 1. Figure 2 shows, from left to right, the assembly stress field with a flat random vibration Acceleration Spectral Density (ASD) of 0.001 g2/Hz from 100 Hz to 2000 Hz on the bottom surface of the assembly. In this assembly, the vertical direction ASD was measured at one of the feet of the component and used as the input spectrum in the component test configuration (middle graphic). Note that the stress fields of the component differ between the assembly and the component test configuration. To exercise the same damage mechanism between the two configurations, the stress field should be similar to each other. The graphic on the right shows the vertical ASD response of a point on the top of the component in the assembly and the component test configuration. The differences in the responses are concerning. Figure 3 shows the differences in the modes of the assembly and the component test configuration graphically by mode. Figure 4 shows the MAC of the two configurations at select points on the component, which indicates how closely related different modes between the configurations are. Table 1 lists the modes of the two configurations.

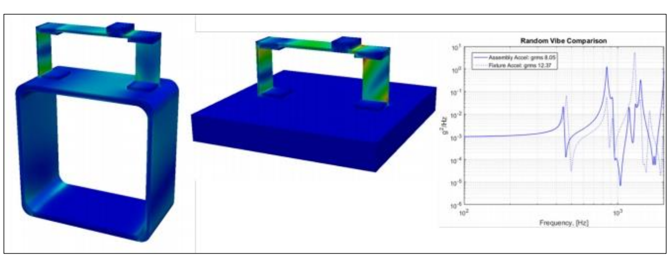

Figure 2. Component testing boundary condition problem, random vibration (simulation) example. Stress field from random input to the base of the assembly (l.). Component stress field from acceleration input at the base of the component, derived from the assembly response (center.) Assembly response vs component response on fixture, derived using traditional techniques (r.) From the presentation at IMAC 35.

Upon fabrication of the steel hardware, some problems became apparent. The flatness of the walls of the steel tube section was inadequate. The faces of the steel c-channel were not square (this could have been due to fabrication), and the internal faces of the c-channel were not parallel to the outer faces. As a result of these dimensional issues, the flat bar stock on the top of the component showed a substantial curvature when the parts were assembled, and if the assembly was bolted together, the bolts would have to provide a clamping load on surfaces that were not parallel.

The team decided to try an assembly made of aluminum, and ordered raw material of the same nominal dimensions as the steel prototype. The hardware was better dimensionally and the team decided to use aluminum instead of steel. Fabrication of the first aluminum prototype was completed prior to the IMAC 35 Structural Dynamics Conference, and the team brought the aluminum prototypes and a presentation that demonstrated the component testing boundary condition problem to the conference.

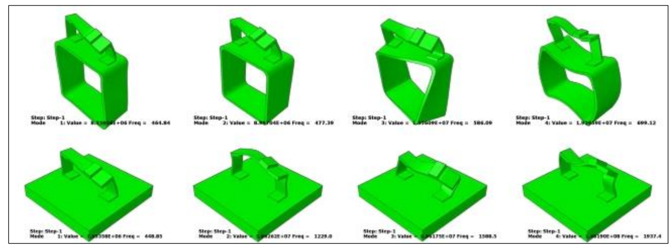

Figure 3. First four flexural modes from Finite Element Models of the component-in-assembly and component-on-fixture configurations, steel, free boundary conditions.

Table 1. List of Modes-Steel models.

Mode Number	Mode Frequency (Hz) Steel Component+Assembly Model	Steel Component+Fixture Model
1	465*	449*
2	477*	1229*
3	586	1588*
4	699	1937*
5	1131	2415
6	1284*	2878*
7	1537	3404*
8	1812*	3681
9	1924	4476

* modes dominated by local flexure of the component

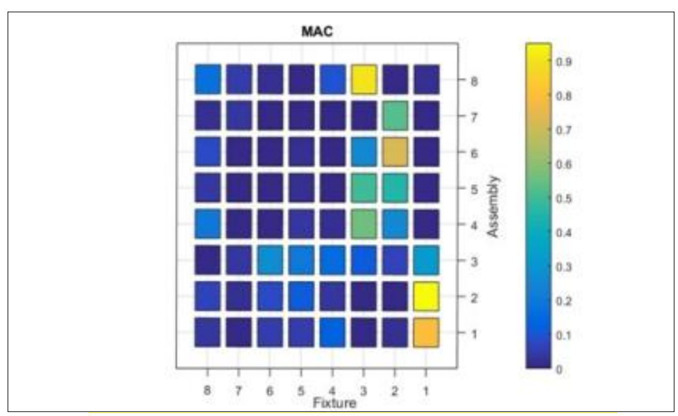

Figure 4. MAC from selected points on the component.

Component Attachment

The goal of simple assembly and disassembly drove a demand for bolted joints in the hardware. The original C-channel design had one bolt at each interface. In discussions following a meeting on the boundary condition problem at the IMAC 35 conference, an experienced dynamicist suggested four bolts at each interface. Reluctant to increase the system complexity and cost, the team decided to try a four-bolt approach at the component interface.

Prototype parts were fabricated and a simple impact survey done to determine the effect the four bolt interface would have on test-to-test variability when the component was detached and re-attached to the assembly. The survey showed that the peaks of the acceleration Frequency Response Functions (FRFs) of the four-bolt configuration exhibited less variability than those of the single-bolt configuration. The team accepted the survey results and adopted the four-bolt configuration to attach the component to the fixture. Photos from the survey and a comparison of FRFs are shown in Figures 5 and 6.

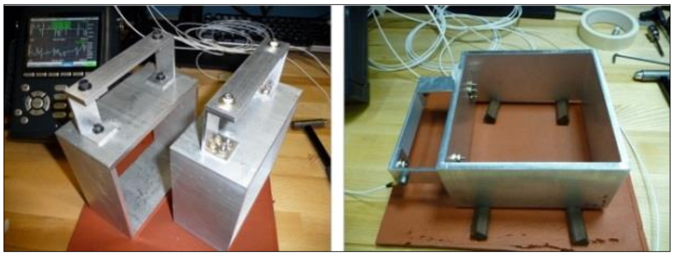

Figure 5. Impact survey on Single vs Four bolt attachment.

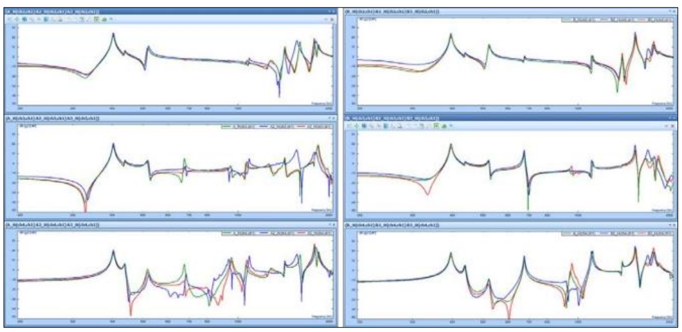

Figure 6. Single bolt attachment (l.) vs. Four bolt attachment (r.) at a driving point (top plot) and two additional response locations. Once the initial data set was taken, the component was removed and re-attached to the assembly twice, with the same measurements repeated after each re-attachment.

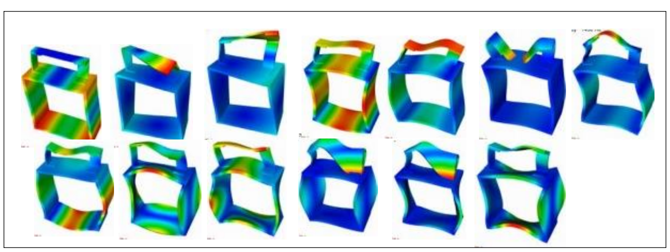

Figure 7. Uncut Aluminum Box Assembly flexural mode shapes from modeling.

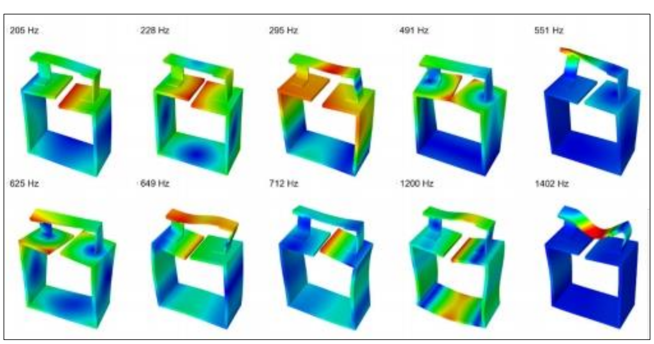

Figure 8. Flexural mode shapes of the final design, from a finite element model.

Table 2. Natural Frequencies from modeling and testing.

	Natural Frequency (Hz)		
Mode Number	Uncut Aluminum Box Assembly FEM Model	Uncut Aluminum Box Assembly Test	Cut Aluminum Box Assembly FEM Model
7	416	398	205
8	521	443	228
9	590	528	295
10	714	679	491
11	1150	963	551
12	1350	1116	625
13	1450	1381	649
14	1603	1517	712
15	1879	1635	1200
16	2021	1736	1402
17	2158	1799	1619
18	2245		1732
19	2347	1944	1947
20			2111

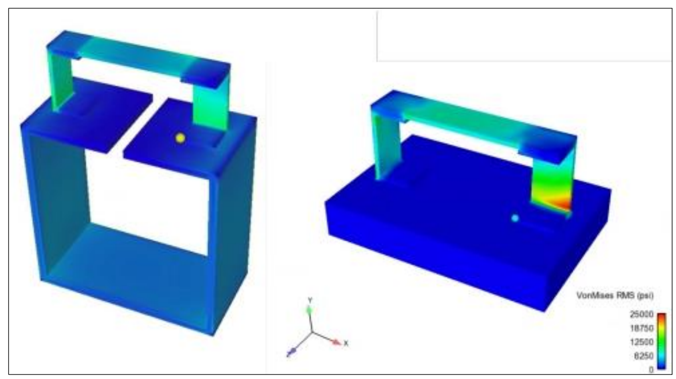

Figure 9. Component testing boundary condition problem simulation for random vibration. Random acceleration input was applied to the base of the assembly. The stress field (color contours) and response at the component base (indicated by a yellow dot) was calculated (l.). Component stress field from acceleration input at the component base (indicated by a green dot), derived from the assembly response (r.).

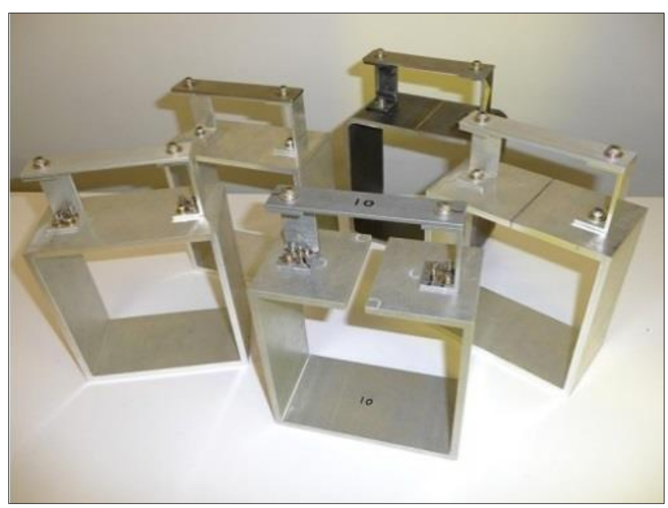

Figure 10. Development prototypes, and the final version (front) of the Box and Bench test bed for the component test boundary condition.Problem.

Progression to the Final Design

The last changes to the design were motivated by a desire to have the component and the rest of the assembly interact dynamically, and to have more assembly modes present below 2000 Hz. Modes of the original aluminum assembly model are shown in Figure 7. The team felt the component and assembly did not interact enough. A cut, between the component attachment locations and through the box section was modeled, resulting in an assembly in which the component plays a much larger role in the system dynamics. The natural frequency of the first flexural mode was lowered dramatically. The team decided to keep the cut in the box section, anticipating that the fixture design problem will directly address the issue of mismatched dynamic impediance between the test fixture and the next level assembly. Modes of the cut system are shown in Figure 8, and tables of modes and frequencies of various models and tests are in Table 2.

Summary

The hardware for the boundary condition challenge has matured. The research team looks forward to more interaction with structural dynamics modeling and test communities, as we strive to develop performance criteria and design guidance for creation of high fidelity component test fixtures and testing methods.

References

1. Scharton T. D., "Impedance Simulation Vibration Test Fixtures for Spacecraft Tests," Shock and Vibration Bulletin, #40, 1969.

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The author can be reached at: harvie.julie@gmail.com.