SAND2016-10785C

Simultaneous Qualification Testing of Multiple Components and the Influence of Closely Spaced Vibration Modes

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* Sandia National Laboratories is a multi-mission laboratory managed and operated by Sandia Corporation, a wholly owned subsidiary of Lockheed Martin Corporation, for the U.S. Department of Energy's National Nuclear Security Administration under contract DE-AC04-94AL85000.

ABSTRACT

Vibration and shock qualification testing of components can be an expensive and time-consuming process. If the component is small, often two or more units can be mounted on a fixture and tested simultaneously to reduce test time. There is an inherent danger in simultaneously testing two or more identical components as the fundamental natural frequencies and mode shapes of the individual components will be nearly identical with some slight variation due to manufacturing variability. Testing in this manner can create a situation where closely spaced vibration modes produce unwanted interference between the two units under test. This phenomenon could result in a case where one unit is over-tested while the other is under-tested. This paper presents some experimental results from simultaneously testing pairs of components which show distinct interference between the units. Some analysis will also be presented showing how variations in the components can alter the intended test response, potentially impacting component qualification.

Keywords: Vibration, Shock, Multiple Components, Tuned Absorber, Beating

INTRODUCTION

Laboratory testing time is often expensive and difficult to obtain. One popular solution to this problem is to try and test as many components as possible in the limited time available. A common solution for small components is to install multiple components on the shaker table or shock machine simultaneously when the tests specifications are the same. This is an obvious time saver since two, four, or more components can be tested in the same time as it would normally take to test one component. However, given that each identical component is likely not truly identical due to slight manufacturing variability, the situation can easily arise with several closely spaced resonant frequencies in a test. Exciting multiple components with nearly the same resonant frequency can create unwanted interference between the individual units under test. This could result in a situation where one unit is over-tested while others are under-tested.

The situation described in this work was discovered while analyzing test data from shock and vibration tests of two identical test articles mounted on the same fixture, as depicted in **Fig. 1**. The test data clearly shows that at some points during the test series the two component responses interfered in a way significantly different from what was expected. Experimental evidence of beating was recorded as well as the appearance of responses analogous to a tuned vibration absorber in the frequency response functions.

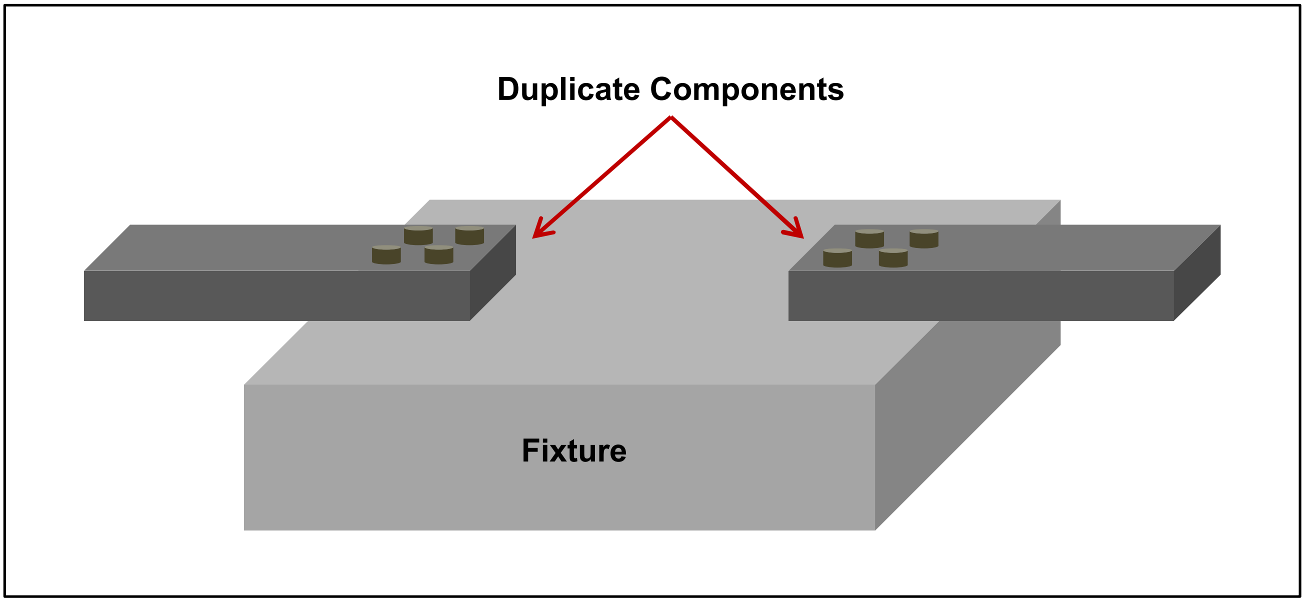

Fig. 1 Schematic of Multiple Component Test Setup for Shock and Vibration Testing

A common method for protecting a system from a steady-state harmonic disturbance is with a tuned vibration absorber. Vibration absorbers are designed with a second spring-mass system added to the primary device which protects it from vibrating. The effect is to change the original single-degree-of-freedom system into a two-degree-of-freedom system. The stiffness and mass of the absorber system are chosen to minimize the motion of the original system at the expense of substantial absorber mass motion [1].

The theoretical developments for a tuned vibration absorber are always based on the fundamental premise of a steady-state sinusoidal excitation at constant frequency. Their application is usually limited to machinery running at a constant speed. Inman presents a mathematical development for deriving the appropriate absorber mass and stiffness to eliminate motion of the primary system [1]. To generate the optimum response, it is necessary to design the mass and stiffness such that the natural frequency of the vibration absorber equals the natural frequency of the system to be controlled. This in turn is the exact definition of closely spaced modes.

Closely spaced modes are defined as two vibration modes whose frequencies are close to a common mean. Typically, the modes are within about ten percent of a common mean. Closely spaced modes can become especially problematic when their modal effective masses are significant and approximately the same order of magnitude [2]. If the modal effective mass is not significant then there is usually no notably different response. However, if the modal mass is significant in both of the closely spaced modes, beating can occur and the component responses can be significantly altered.

SHOCK TEST RESULTS

Shaker shock testing on the two component set-up was performed in each of the three axes at increasing severity levels. **Fig. 2** shows a plot of the normalized acceleration input to the fixture base. The input appears to be a straightforward, rapidly decaying shock event.

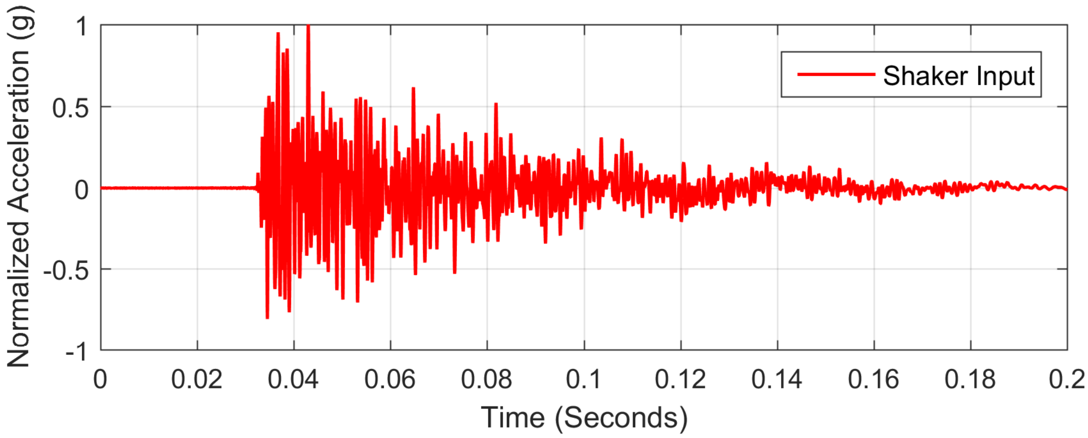

Fig. 2 Normalized Shaker Shock Input Time History

Fig. 3 and **Fig. 4** present the response on the cantilever part of test articles A and B, respectively. Since the test articles are cantilevered in the fixture, it would be reasonable to expect a response profile to be similar to the input with some amplification. Component B shows a time history plot similar to what is expected with a normalized peak response of approximately 2.83g. In contrast, Component A shows a time history plot with an obvious 0.03 second beat period. The peak normalized acceleration from the Component A time history is also 3.18g, approximately 12.4 percent greater than the Component B response, although both parts were tested side-by-side.

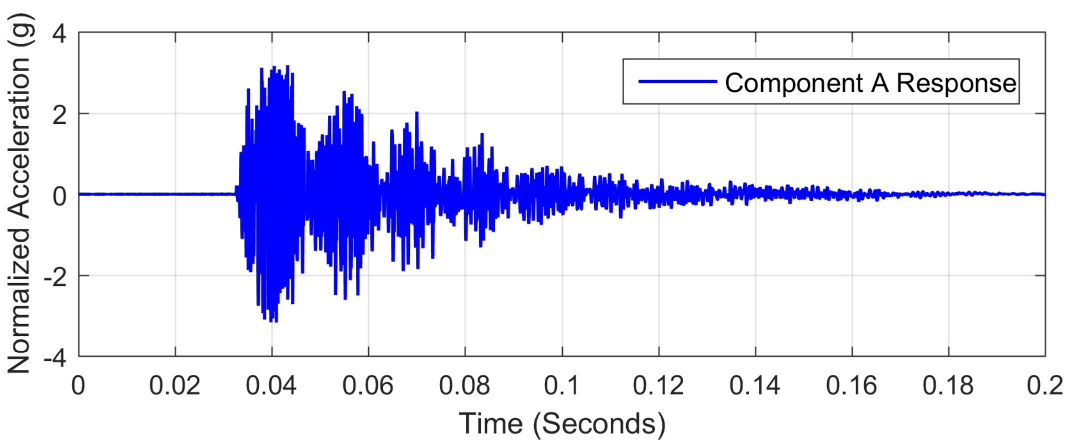

Fig. 3 Component A Normalized Shaker Shock Response Time History

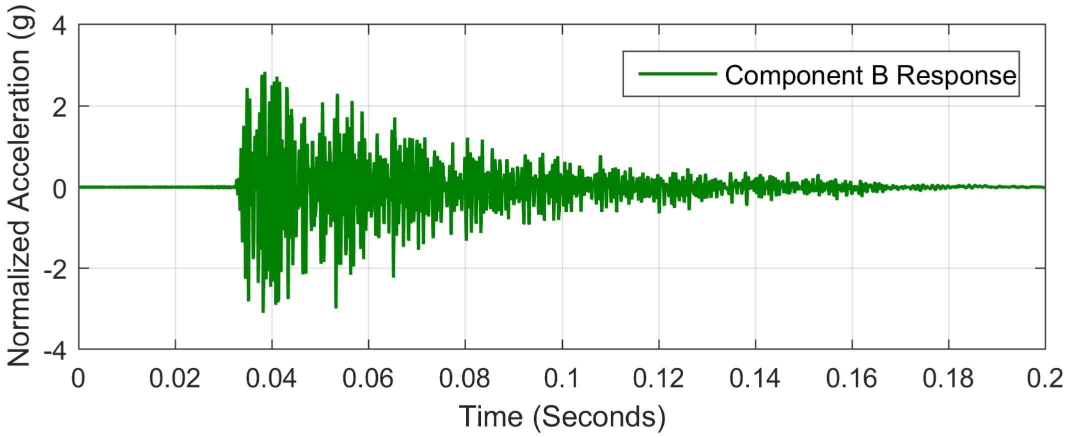

Fig. 4 Component B Normalized Shaker Shock Response Time History

Fig. 5 and **Fig. 6** present the responses on the cantilever part of test articles C and D, respectively. For this second test, components A and B were replaced with components C and D and a nearly identical input time history was applied. However, for this test, the beating response was not apparent in either component. Also of note is that the maximum normalized peak acceleration of Component C was 3.11g and 2.90g for Component D, a difference of about seven percent. This is a more reasonable difference than the previous test where beating occurred.

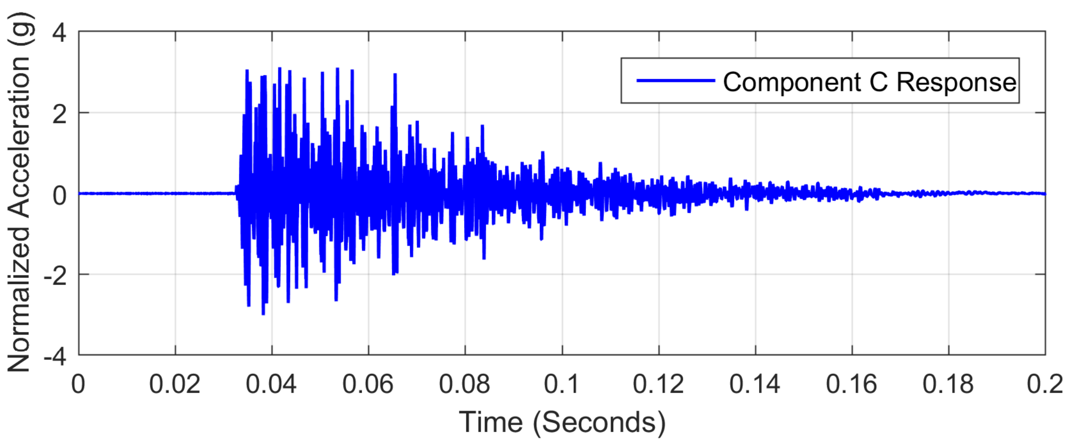

Fig. 5 Component C Normalized Shaker Shock Response Time History

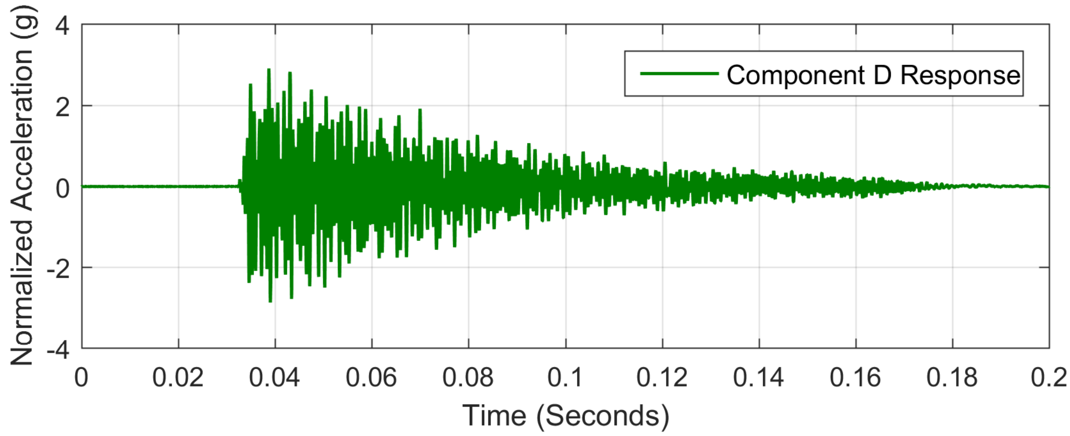

Fig. 6 Component D Normalized Shaker Shock Response Time History

The results of the two shaker shock tests, which were intended to be identical, were actually quite different from a time history perspective. **Fig. 7** shows the corresponding acceleration Shock Response Spectra (SRS) plots for the four components overlaid. The SRS plot shows that the response of components B, C, and D only differ slightly. However, the response of component A shows significantly increased energy in the 60 – 350Hz range as well as somewhat higher energy above 5kHz. Since the inputs to the two test were essentially identical and each test consisted of two identical components mounted side-by-side, the resulting difference in the SRS must be attributed to the beating response measured on component A.

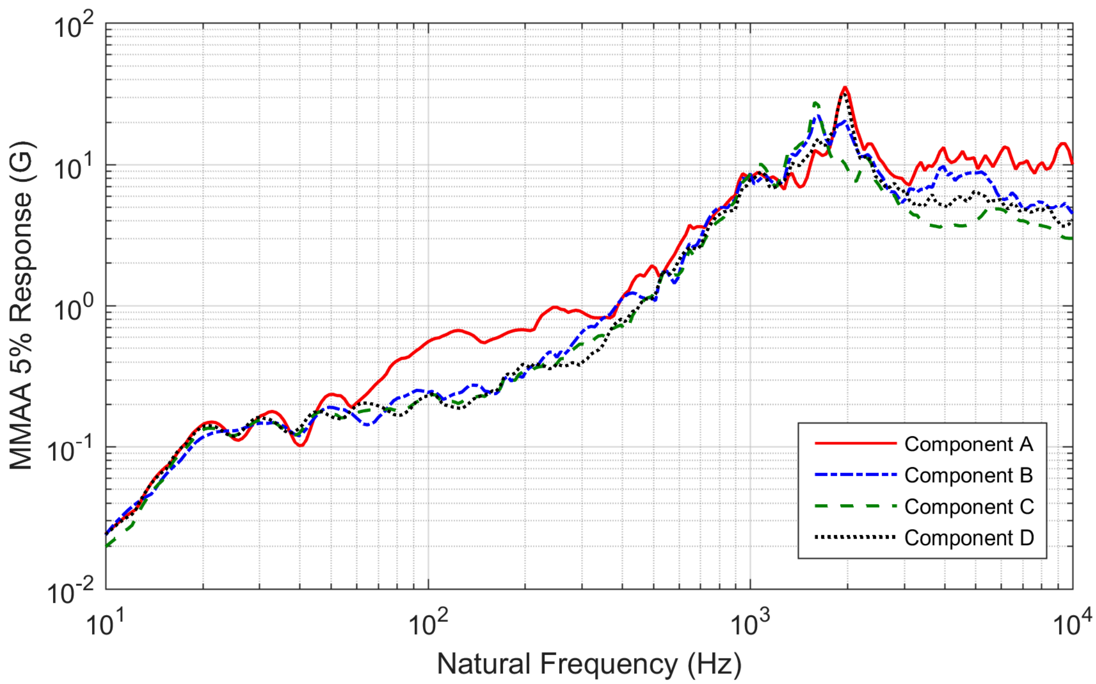

Fig. 7 Comparison of Response SRS for Components A, B, C, and D

VIBRATION TEST RESULTS

In addition to the shaker shock testing discussed previously, random vibration testing was also conducted on the components mounted side-by-side in the test fixture. **Fig. 8** shows the resulting Frequency Response Functions (FRF) from the single-axis random vibration testing of Components A and B mounted side-by-side in the same test fixture. As can be seen here, the FRFs from the accelerometers mounted on the cantilever portion of the component show that the natural frequency of Component A is about 2050Hz where the natural frequency of Component B is around 1900Hz. The difference between these modes is about 7.5 percent which is less than the approximate 10 percent threshold for closely spaced modes discussed earlier. As a result, Component B shows a significantly higher response than Component A and this difference in natural frequency is likely the reason that the beating response showed up so clearly in the shaker shock testing.

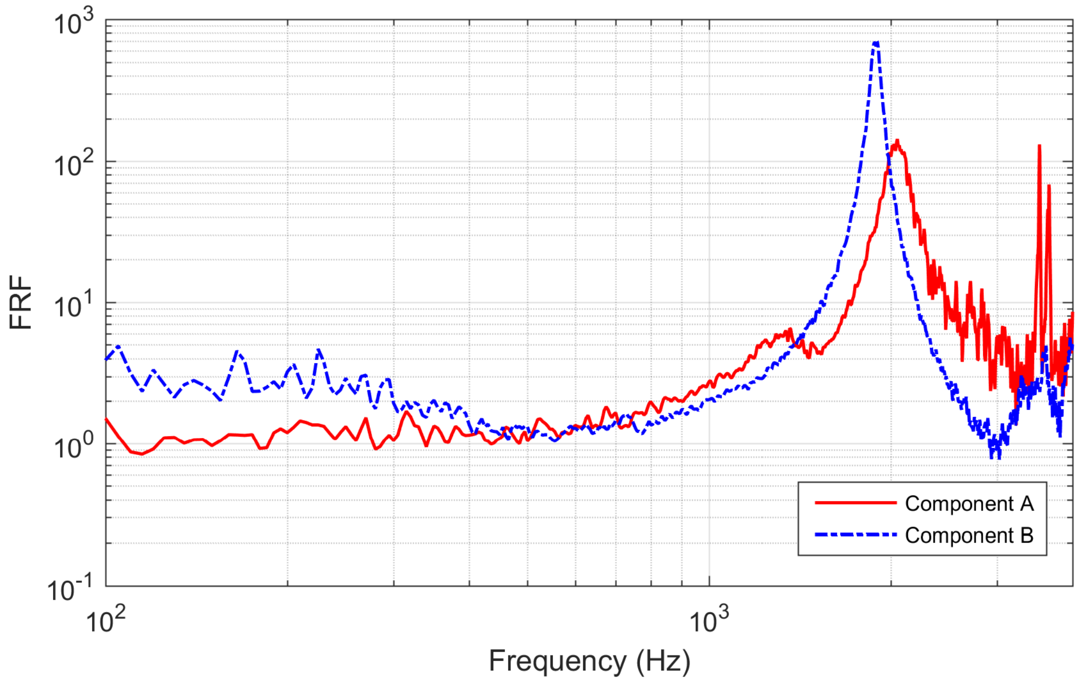

Fig. 8 Components A and B Frequency Response Functions to Random Vibration Input

Fig. 9 shows the FRFs calculated from the testing of Components C and D in the same side-by-side configuration. The results shown here are strikingly different than those shown in **Fig. 8**. These results show a strong natural frequency at 1900Hz for component C, basically identical to that shown previously for Component A. However, Component D shows what appears to be resonant frequencies at 1610Hz and 2300Hz which is almost evenly split left and right of the Component C 1900Hz frequency. Since Components C and D are physically the same, within normal manufacturing tolerances, the results here are highly suspicious. It appears from the FRF that at 1900Hz, Component C is vibrating significantly while Component D is essentially behaving as a rigid body. This is precisely the desired outcome of the tuned vibration absorber described previously. Likewise, the consequence of adding a tuned vibration absorber is the addition of frequencies to the left and right of the desired operating frequency. This is readily apparent in the FRF shown here.

These results are most unusual in that the theory of tune vibration absorbers always assumes a steady state sinusoidal input motion, not a random vibration input.

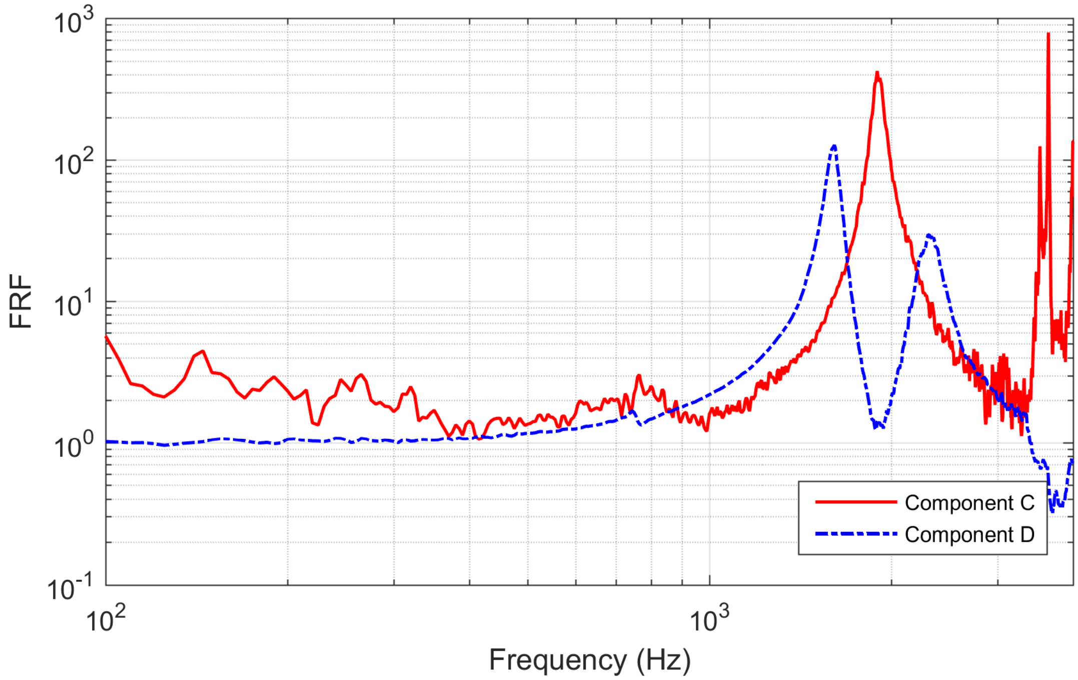

Fig. 9 Components C and D Frequency Response Functions to Random Vibration Input

SIMULATION RESULTS

To gain insight into the phenomena seen in the laboratory, a simple finite element analysis was performed. The finite element model used 18 beam elements, eight elements for each of the two identical cantilever beam components and two elements connecting the two components and simulating a stiff fixture. The element cross-sectional properties were arbitrarily chosen such that the two components would have a nominal first resonant frequency in the 2.2 – 2.3kHz range using aluminum material properties. The two elements representing the fixture were modeled as steel with section properties substantially greater than the component properties. This was done so that the fixture would be extremely stiff compared to the components but not mathematically rigid.

For this finite element study, a simple half-sine high-frequency shock type displacement was simulated at the center of the test fixture and a transient simulation was performed to predict the responses at the free ends of the cantilevered components. Several simulations were performed by varying the modulus of elasticity for the two cantilever components.

The first simulation, as shown in **Fig. 10**, used identical material properties for the two components. The next three simulations, shown in **Fig. 11**, **Fig. 12**, and **Fig. 13**, represent variations in the modulus of elasticity of one, five, and ten percent, respectively. The resulting frequency shifts, noted in the legends, are all well within normal tolerance ranges for components; however, the variation in response is dramatic due to the development of a closely spaced mode response phenomena.

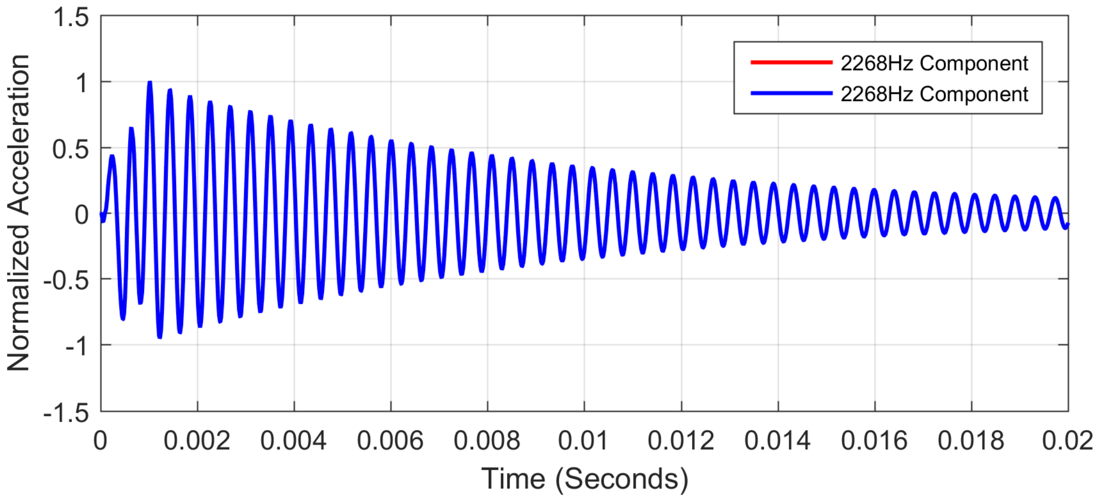

Fig. 10 Simulated Two Component Free End Responses with Identical Material Properties

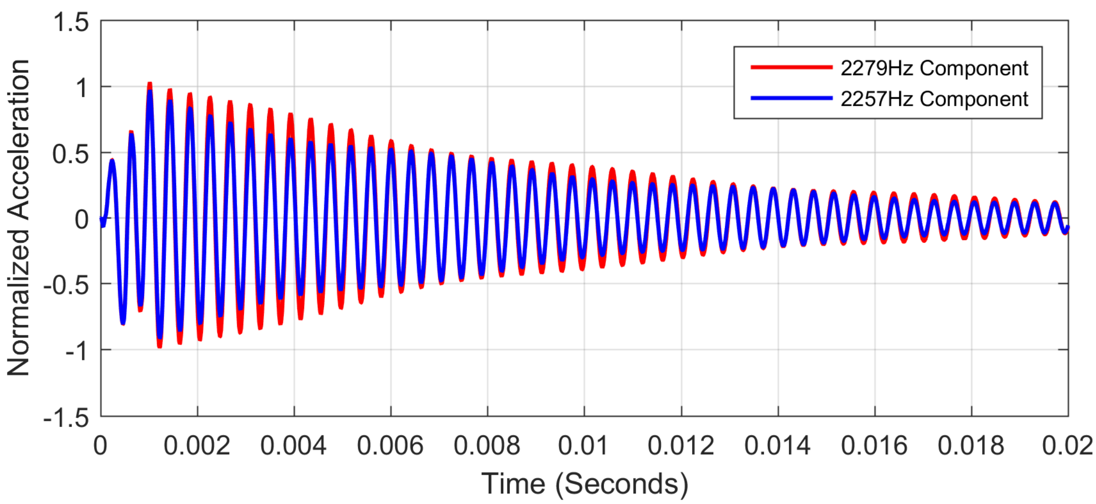

Fig. 11 Simulated Two Component Free End Responses with ±0.5% First Mode Variation

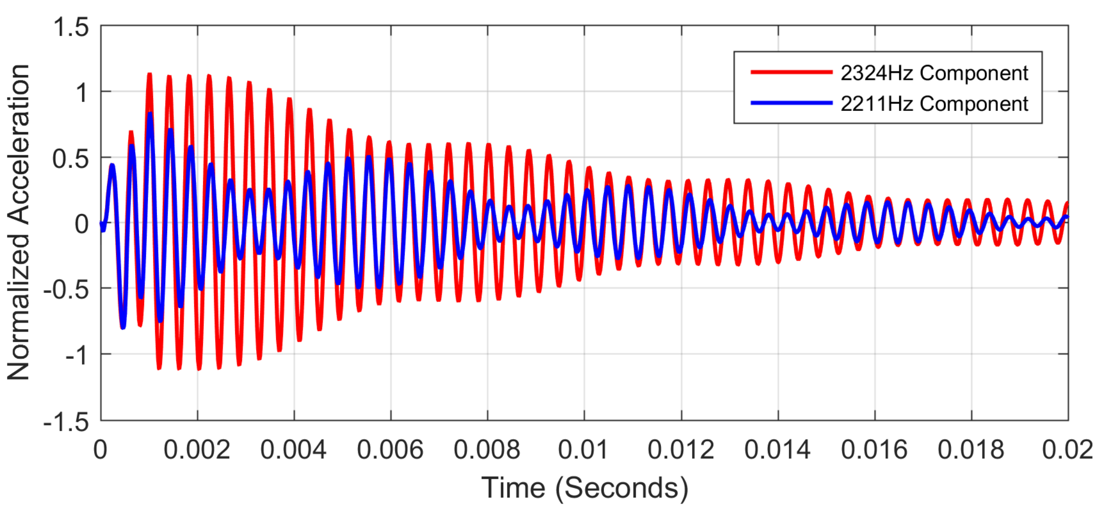

Fig. 12 Simulated Two Component Free End Responses with ±2.5% First Mode Variation

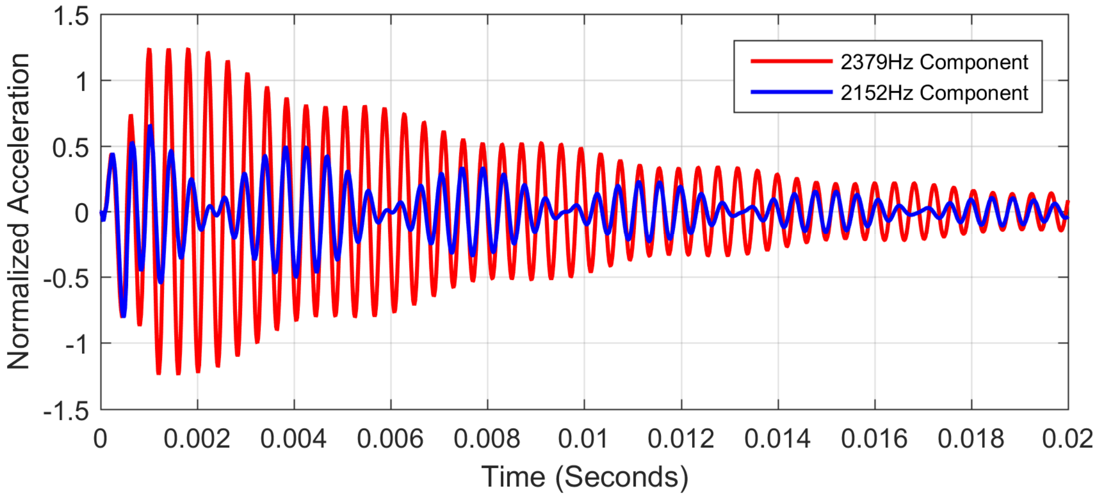

Fig. 13 Simulated Two Component Free End Responses with ±5.0% First Mode Variation

When the components were completely identical, the responses were also identical. However, as the variation in fundamental frequency increased, the responses of the two components became dramatically different. The beating phenomenon became more prevalent, and the difference in peak response also increased. The peak response and fundamental frequency details for all cases are outlined in **Table 1.**

It is also interesting to note that the component with the higher peak response was always the component with the higher frequency. This observation aligns with the tuned absorber theory. Higher frequencies are attributed to increased stiffness and/or decreased mass, and tuned absorbers are typically designed to be much smaller and lighter than the primary structures they are attached to.

Table 1 Comparison of Peak Response for Components with Varying Fundamental Frequency

Variation in Fundamental Frequency	Component 1 Fundamental Frequency	Component 1 Peak Normalized Response	Component 2 Fundamental Frequency	Component 2 Peak Normalized Response
0%	2268 Hz	1.000 g	2268 Hz	1.000 g
0.5%	2279 Hz	1.030 g	2257 Hz	0.969 g
2.5%	2324 Hz	1.137 g	2211 Hz	0.836 g
5.0%	2379 Hz	1.244 g	2152 Hz	0.805 g

CONCLUSIONS

When components are fabricated, there will inherently be some variation in the dynamic characteristics due to manufacturing variability. The amount of frequency variation can have a substantial effect on the response of the components when multiple components are tested simultaneously for qualification testing. It is almost inevitable that some components will be over-tested and others will be under-tested relative to the intended environment exposure. The actual component responses will vary from test to test, as seen in the experimental results, because the dynamic characteristics are slightly different for each unit. If multiple components must be tested simultaneously, it is recommended to fully analyze the test setup and responses and potentially alter the inputs to ensure that all components are seeing the required exposure at a minimum.

Additionally, while these studies were focused on simultaneous testing of only two components, the complexity of response is expected to increase as more components are added. It is not unusual to simultaneously test four or more components. The addition of more components may be investigated in future work.

REFERENCES

1. Inman, D. J., *Engineering Vibration*, Prentice-Hall, Inc., New Jersey, 1994, Pages 250-257
2. Scavuzzo, R. J. and Pusey, H. C., *Naval Shock Analysis and Design*, SVM-17, The Shock and Vibration Information Analysis Center, Booz-Allen and Hamilton, Inc., Falls Church, Virginia, 2000