Chapter 7
An Undamped Dynamic Vibration Absorber on a Resonant Plate Shock Test

David E. Soine, Adam J. Bouma, and Forrest J. Arnold

Abstract Dynamic vibration absorbers are widely used for vibration reduction on vehicles, buildings, industrial equipment, and many other applications. If an undamped or very lightly damped absorber is tuned to suppress the resonant response of a structure, two new resonances are created at frequencies above and below the suppressed resonance peak. An experimental investigation was conducted to explore the effect of such a vibration absorber on shock response spectra obtained during resonant plate shock testing. The shock response spectrum at the original plate resonant frequency was reduced. The response of the two new resonant frequencies appeared equal in a Fourier spectrum, but only the response of the higher frequency resonance had a strong effect on the shock response spectrum, creating a new "knee frequency" at a higher frequency than that of the original unmodified plate.

Keywords Resonant · Fixture · Plate · Pyroshock · Absorber

Introduction

Resonant fixture shock tests are a means to perform mid-field and far-field pyroshock simulation in the test laboratory. The method can be adapted to simulate aspects of stage separation, ejection shock, blast events, flight shock, and margin tests for other shock environments ordinarily performed on a vibration shaker but outside the shaker test capability. Resonant fixtures in use include square, rectangular, or round plates, beams, and bars of various desirable responding frequencies. Shock practitioners have developed means to adjust the natural frequency and damping of these resonant shock test systems and are on the lookout for new tools to adjust resonant shock test equipment to better meet test specifications.

In the discussion after a presentation on resonant bar shock testing at IMAC-XL, it was mentioned that dynamic vibration absorbers (also called tuned mass dampers or tuned absorbers) can be applied to resonant shock test fixtures. For the resonant bar application, shock practitioners in some cases might desire to suppress or alter the bending modes of the resonant bar. The authors recognized that some existing experimental test items at Sandia National Laboratories could be fashioned into a dynamic vibration absorber and decided to perform initial work on a resonant plate. This decision allowed the team to use existing hardware and finite element models to get familiar with the application of dynamic absorbers as quickly as possible. A short investigation was conducted into the effects of such a dynamic absorber on a resonant plate bending mode.

Background

Resonant plate is a type of resonant fixture laboratory test that exploits the transverse bending mode of a plate, whose frequency is pre-determined by selecting appropriate plate dimensions. The test article is usually attached to the center of the plate surface, and the plate is struck with a projectile on the opposite side. The resulting pulse and ringdown event, measured with an accelerometer and analyzed via the Shock Response Spectrum (SRS), is very repeatable. To facilitate a wide range of shock event simulations, a test laboratory will modularize this technique, resulting in several resonant plates of different dimensions in the lab inventory, each designed to respond at different frequencies. To keep costs low, simple square plates of aluminum are most often procured and adapted for use [1].

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Simple aluminum plates exhibit low damping, leading to long shock test ringdown times on the order of 150 msec or greater. To better simulate field environments, a shorter ringdown time is often desired. Damping devices such as "damping bars" and constrained layer approaches have been developed in response to this need [2]. In addition, the force pulse applied to the resonant plate by the projectile is modified by selection of the projectile mass, velocity, and "programmer material." This activity to optimize the force input to the resonant plate is critical to adequately excite the bending mode of the plate while limiting the response at higher test assembly natural frequencies. Aerospace component qualification test specifications are tailored to these test methods and are usually specified by a simple straight-line SRS profile and a ±6 dB tolerance band. The typical SRS specification has a +12 dB/octave slope up to the "knee frequency," where it flattens to zero slope at a specified SRS acceleration level. In the test lab, a plate bending frequency that corresponds to the knee frequency is chosen to perform the shock test [3].

Though these methods have generally proven successful for single-axis testing, there is a desire to expand resonant fixture shock techniques to provide a multi-axis response with higher fidelity to the field environment. For a resonant fixture to provide a controlled shock response in more than one axis requires a test assembly designed to have fundamental responding modes with more sophisticated, multi-axis motion [4,5]. These test assemblies must be designed and optimized via computer modeling and simulation. Sometimes, one of several responding modes will drive an undesirable response at the test article. A dynamic vibration absorber could be a design feature useful in disrupting that undesirable modal response [6].

Testing and Analysis

In this investigation, the effect of a cantilevered-mass dynamic absorber on the bending mode of the resonant plate was observed. The cantilevered mass was made up of a cylindrical pedestal and square plate mass, which when assembled was known to have a fixed base natural frequency near 500 Hz and a weight of 9.4 pounds; see Figure 1. The resonant plate dimensions were 23.75 x 23.75 x 1.25 inches, with an approximate natural frequency of 500 Hz. The plate was suspended by ropes in front of an air gun, with two rods bolted to the plate to enable the plate to be leveled with cords when loaded with the dynamic absorber; see Figure 2 below. No damping bars were installed. To measure the shock event, an accelerometer was placed in the center of the plate, where a test article would ordinarily be installed for a uniaxial shock test. The cantilevered mass was bolted to the plate with precision spacers between the pedestal and the plate, since earlier investigations demonstrated that the precision spacers linearize the interface at the base of the pedestal and improve correlation with a finite element model. Setup testing was performed to select the projectile weight (35 pounds,) projectile velocity (17 feet/second,) and programmer material (two $1/4$ inch pieces of felt) for the rest of the testing.

Fig. 1 Cylindrical pedestal and square plate.

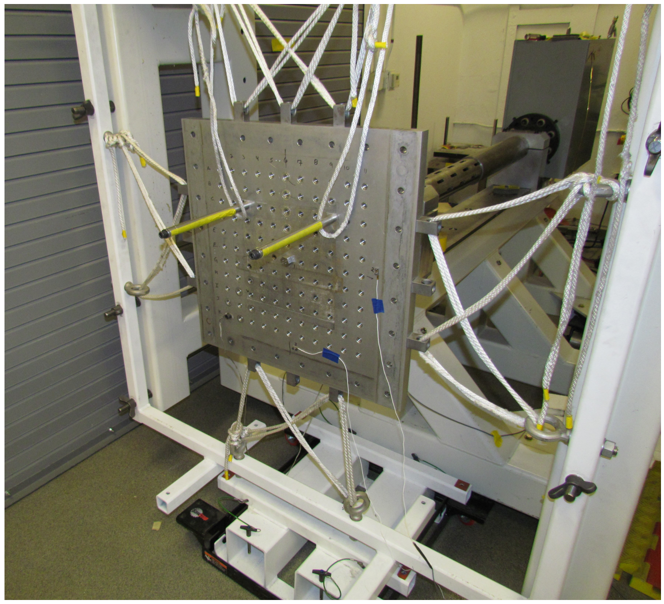

Fig. 2 Bare resonant plate assembly and air gun.

When the center of the plate was struck with the projectile, the bending natural frequency of the unloaded resonant plate in the test configuration was 483 Hz, based on the peak of the acceleration Fourier transform (FFT) magnitude.

To develop the cantilevered mass into a tuned dynamic absorber, the cantilevered mass was attached to a drop table to approximate a fixed base boundary condition. The mass was then struck with a mallet, and acceleration was measured. The cantilevered bending natural frequency was 564 Hz in its original state. Additional masses in the form of steel screws, nuts, and spacers were added to each side of the cantilevered mass to tune the frequency to match the plate frequency of 483 Hz; see Figure 3. Acceleration response of the bare plate struck by a projectile and of the cantilevered mass dynamic absorber on a fixed base is compared in Figure 4 below, which also includes an illustration of the bare-plate bending mode shape estimated from a finite element model of the plate.

The cantilevered mass was initially installed on the resonant plate, 4.5 inches to the left and 4.5 inches down (i.e., Y = –4.5, Z = –4.5) from the plate center; see Figure 5. At that off-center location, based on the bending mode shape, a reasonable amount of plate rotation was expected to interact with the bending of the dynamic absorber. A shock test was performed, and the FFT and Shock Response Spectrum (SRS) of the acceleration in the center of the plate was recorded and is shown in Figure 6. Also shown is the response of the bare resonant plate for comparison. Based on the FFT response, it appeared

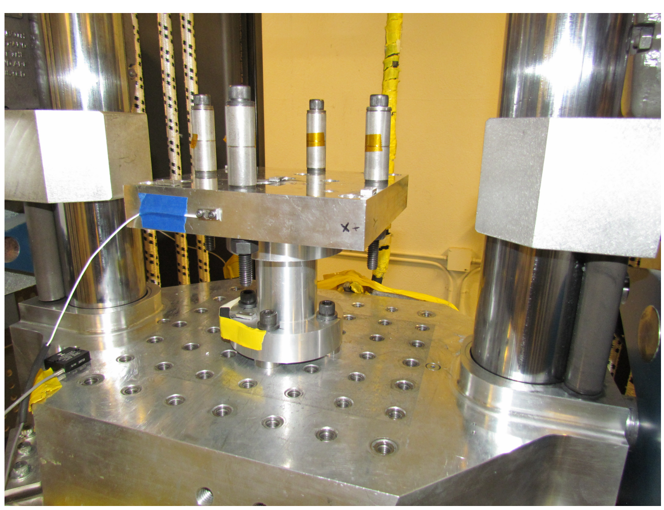

Fig. 3 Cantilevered mass, tuned to 483 Hz by adding spacers and bolts.

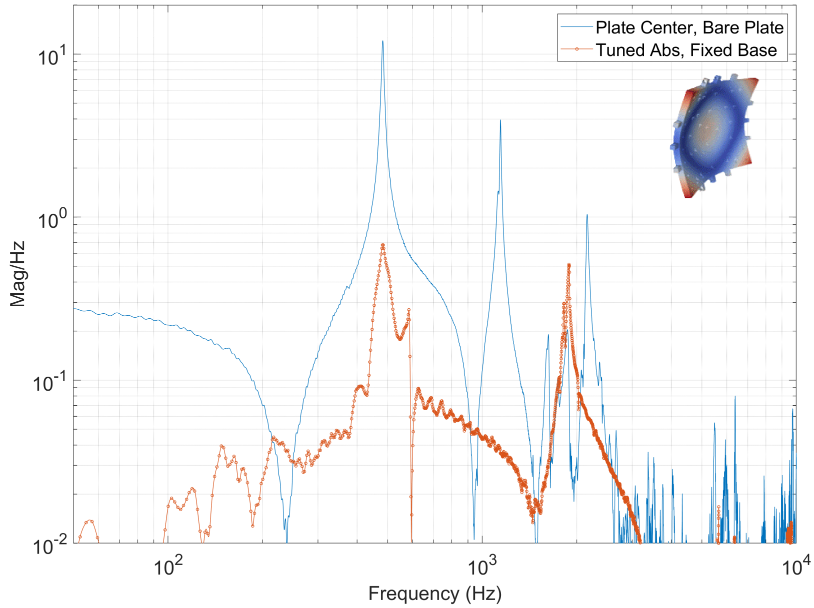

Fig. 4 Fixed base cantilevered mass (tuned absorber) frequency aligns with the resonant plate frequency at 483 Hz.

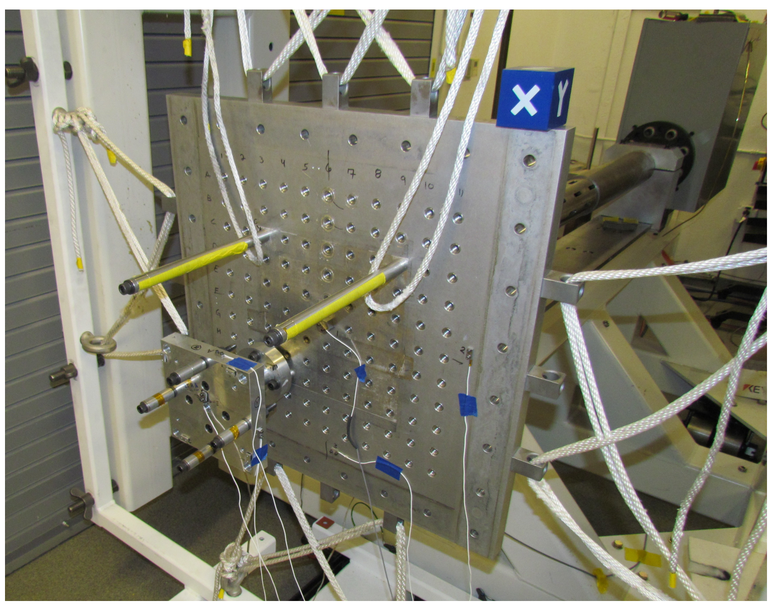

Fig. 5 Cantilevered mass offset from the center of the plate.

that the tuned absorber effect created two resonances in the vicinity of the original bending mode, at 357 and 557 Hz, with the 557 Hz mode dominating at approximately twice the level of the 357 Hz response. The relative amplitude of each peak is of interest when considering the SRS. The SRS of the assembly with the tuned absorber shows a "knee frequency" at 557 Hz, with a peak value 3.9 dB lower than that of the bare plate. Above approximately 400Hz overall SRS levels of the plate with dynamic absorber are about 2–3 dB lower than the bare-plate response.

To further investigate the cantilevered mass dynamic absorber assembly, the dynamic absorber was repositioned to (Y = –3, Z = –3) and (Y = –1.5, Z = –1.5). The resulting FFT and SRS of all four configurations is shown in Figure 7. It was expected that the dynamic absorber effect would be reduced as the absorber assembly was repositioned closer to the center of the plate, where the plate bending mode exhibits less rotation. The test results did not support those expectations, since the lowest knee frequency response amplitude occurred at when the dynamic absorber was at location (Y = –3, Z = –3).

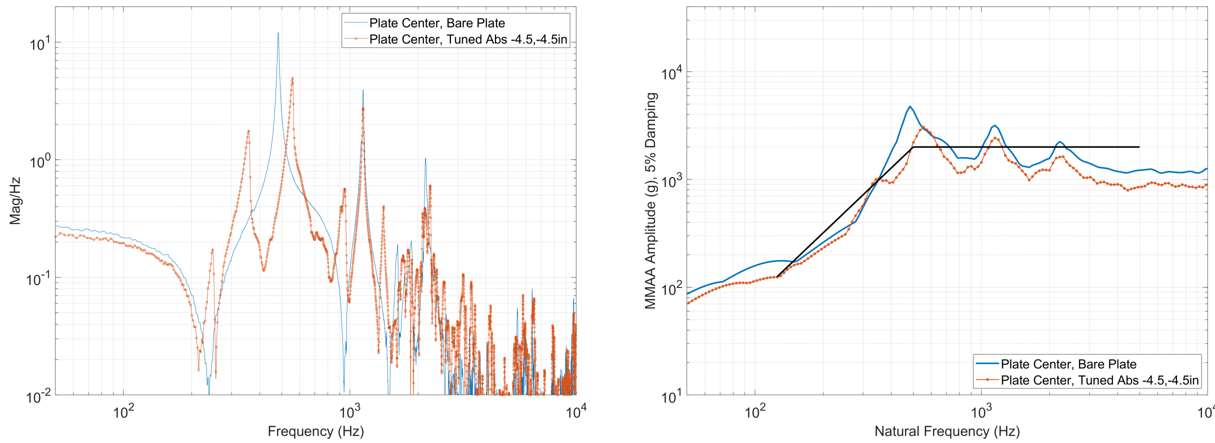

Fig. 6 Plate with tuned dynamic absorber compared to the bare-plate response.

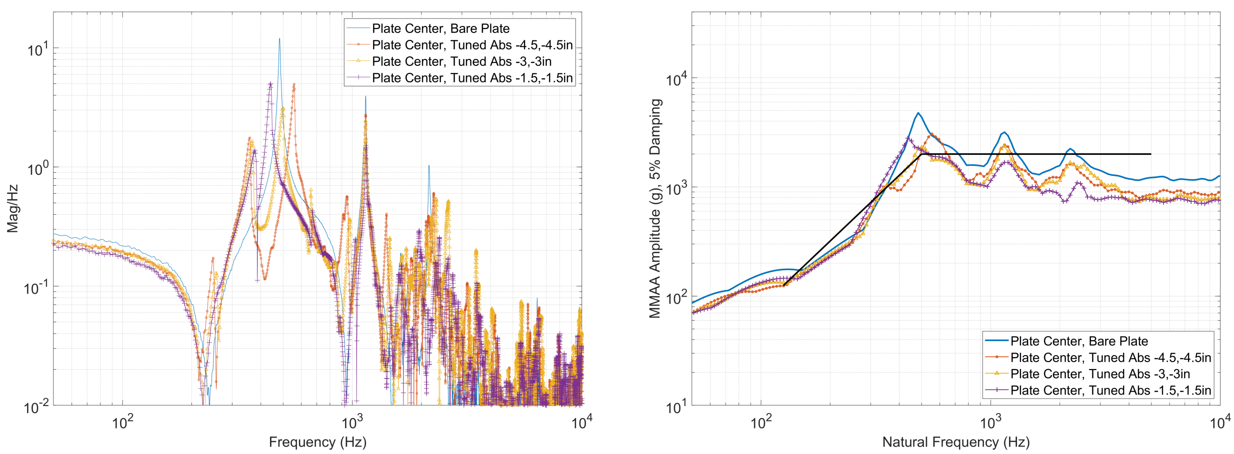

Fig. 7 Relocation of the tuned dynamic absorber on the plate.

Conclusion

The undamped cantilevered mass dynamic absorber modestly reduced the shock response when installed on an undamped resonant plate. In accordance with dynamic absorber theory, the assembly had natural frequencies above and below the target bare-plate mode when the dynamic absorber bending was well activated away from the center of the plate. The higher frequency mode responded with a greater amplitude than the lower frequency mode, and this effect was significant in the SRS analysis. As the dynamic absorber was moved closer to the center of the plate, the frequency of the dominant responding mode dropped and eventually was lower than the original bare-plate resonance, likely due to the absorber assembly mass moving closer to the original bare-plate bending antinode.

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