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Experimental demonstration of a tunable acoustoelastic system

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Abstract

Acoustoelastic coupling occurs when a hollow structure's in-vacuo mode aligns with an acoustic mode of the internal cavity. The impact of this coupling on the total dynamic response of the structure can be quite severe depending on the similarity of the modal frequencies and shapes. Typically, acoustoelastic coupling is not a design feature, but rather an unfortunate result that must be remedied as modal tests are often used to correlate or validate finite element models of the uncoupled structure. Here, however, a test structure is intentionally designed such that multiple structural and acoustic modes are well-aligned, resulting in a coupled system that allows for an experimental investigation. Coupling in the system is first identified using a measure termed the magnification factor and the structural-acoustic interaction for a target mode is then measured. Modifications to the system demonstrate the dependency of the coupling with respect to changes in the mode shape and frequency proximity. This includes an investigation of several practical techniques used to decouple the system by altering the internal acoustic cavity, as well as the structure itself. Furthermore, acoustic absorption material effectively decoupled the structure while structural modifications, in their current form, proved unsuccessful. The most effective acoustic absorption method consisted of randomly distributing typical household paper towels in the acoustic cavity; a method that introduces negligible mass to the structural system with the additional advantages of being inexpensive and readily available.

Keywords - Structural-acoustic interaction, Coupled modes, Acoustic absorption, Acoustic modes, Modal testing

1 Introduction

Acoustoelasticity is the phenomena that describes the coupling between the modal responses of a structure and an enclosed acoustic volume, such as a fluid-filled cavity [1]. Such situations commonly arise in structures within the automotive and aerospace industries [2,3]. If any modes of the structure and acoustic volume are similar in both shape and frequency, the modal responses can couple and behave similar to a vibration absorber [4]. This coupling leads to an increased number of resonance peaks in the structural response as compared to the structural response when neglecting the effects of the fluid. Furthermore, the structural mode shapes will be the same at these additional coupled peaks [5]. If the existence of this coupling is not known a priori, this can lead to confusion when analyzing the results of a structural modal test used for analytical model correlation. As the analytical models typically neglect the interaction with the surrounding fluid leading to the structure-only (in-vacuo) response, performing a modal test on a coupled structure will lead to discrepancies when compared to the model.

One method of mitigating this issue is by including the fluid loading effects in the analytical model; however, this approach significantly increases model complexity and computational expense. Instead it is desirable if the issue can be mitigated in the experimental realm. In the automotive industry, structural modifications have shown success in reducing the structural-acoustic coupling by altering the structural mode shapes and frequencies and, consequently, reducing noise levels within the passenger compartment [2, 6-8]. Although these methods reduce sound levels, the coupling is not completely eliminated and undesirable peaks in the structural response may still persist. Instead, inclusion of an acoustic absorption material may be required to adequately damp out the acoustic response and fully eliminate this coupling, resulting in a decoupled structural response [2]. This paper seeks to build upon this knowledge by providing a more thorough experimental investigation to determine methods of both quickly identifying when acoustoelastic coupling occurs and measuring the associated structural-acoustic interaction. This paper also examines several methods of modifying either the acoustic volume, or the structure itself, to remove this coupling and achieve the decoupled structural response.

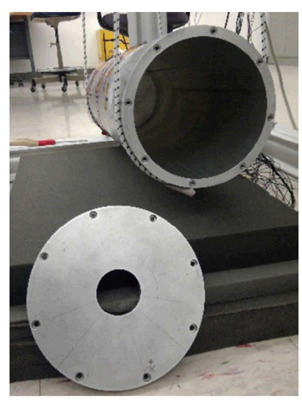
The remainder of the paper is composed as follows: Section 2 introduces the hollow, cylindrical shell deliberately designed to exhibit coupling that was used in this experimental study. Also provided is a schematic of the accelerometer and microphone locations used to determine the structural and acoustic mode shapes. Section 3 shows the results for a typical hammer test and the methods and procedures used to both identify and measure the structural-acoustic interaction. Section 4 shows the results of several mitigation strategies employed to decouple the structural response, including acoustic volume modifications and structural modifications.

2 Hardware and Test Set-up

The process for designing a hollow shell structure which exhibits acoustoelastic coupling began by looking up standard size, off the shelf, aluminum tubing. A relatively thick wall was desired, so shells with walls ranging from 6.35 to 12.7 mm were considered. Next, several finite element models were built with wall thickness and radius per the standard tubing sizes and several lengths were used for each radius and wall thickness combination. Modes were computed using these models and the low-order ovaling mode frequencies recorded in a table. Next, rigid walled acoustic modes were computed analytically for the interior dimensions of each of the shell finite element models; these acoustic mode frequencies were also recorded in a table. With the structure and acoustic mode frequencies for several modes for several geometries, the next step was to find the geometry which provided close matching between at least one pair of structure and acoustic modes (in terms of both shape and frequency). This was represented as a percent different in structure vs. acoustic mode frequency for ovaling modes of the same circumferential and axial orders. It was found that an aluminum shell with 12.7 mm wall thickness, 203 mm outer diameter and 610 mm length provided good matching at the 2,1 and 3,0 modes. The test hardware was then built with these specifications and did in fact exhibit acoustoelastic coupling. The cylinder was then suspended from two bungee cords to approximate a free-free boundary condition. Both end caps were connected to the cylinder using eight bolts torqued. Figure 1a shows the test with material properties and dimensions recorded in Table 1. Figure 1b shows one end cap with a small hole that enabled a rod holding the microphone array to move and rotate within the structure.

Figure 2 shows a schematic of the twenty-one uniaxial 10 mV/g accelerometers, at various locations on the cylinder surface, which were used to capture the structural modes of interest. Figure 3 shows the axial locations traversed by the array of 10 mV/Pa microphones used to capture the acoustic modes of interest of the internal air cavity, as well as the angular location of each microphone taken from the +X axis. The custom microphone holder held the microphones fixed with a separation angle of 45 degrees between each microphone. Two measurements taken at each of the seven axial locations within the cavity, with the microphone holder and rotated 180 degrees between measurements, captured the full acoustic mode shapes of the cavity with a total of 14 measurements.

a) Cylinder suspended from soft bungee cords with the accelerometers bonded on the outer surface, and the microphones distributed in an array.



b) Hollow cylinder cavity with end cap removed.

Fig. 1 Experimental test setup

Table 1: Cylinder properties and dimensions

Material	Aluminum
Length	610 mm
Inner Diameter	190 mm
Outer Diameter	203 mm
Wall Thickness	13 mm

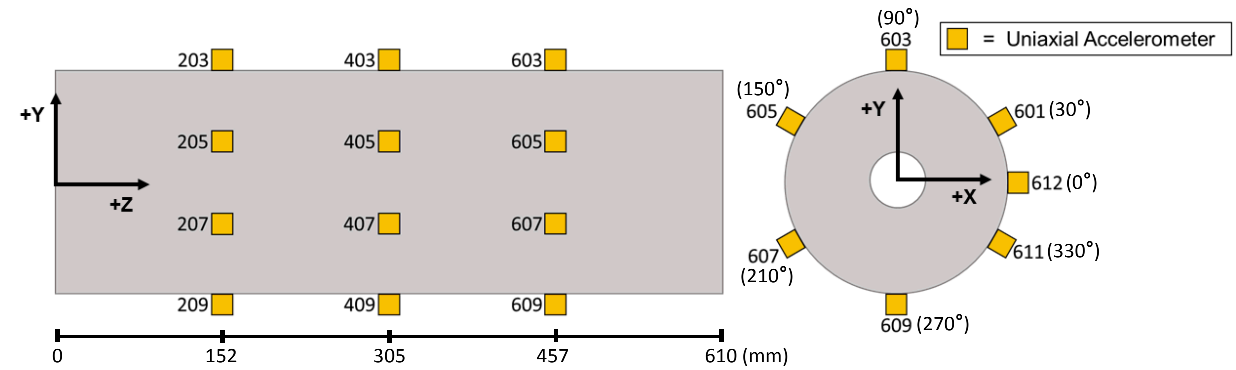


Fig. 2 Accelerometer measurement locations

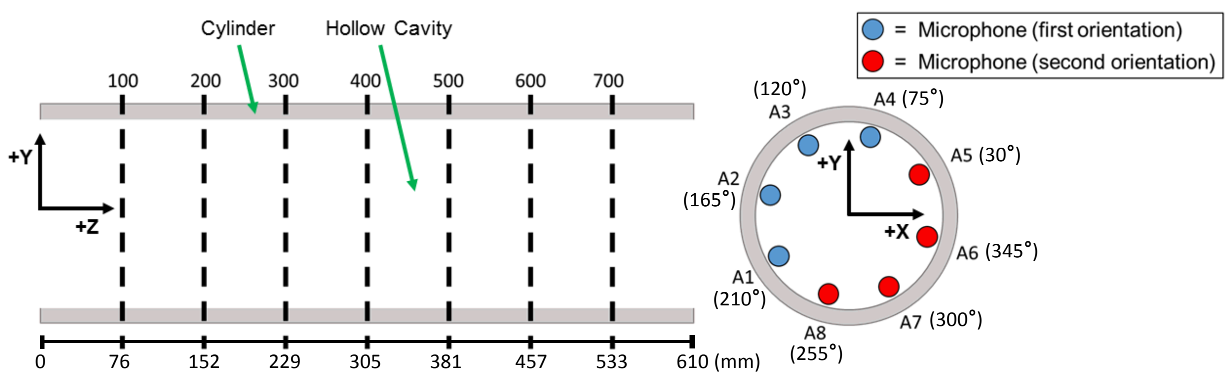


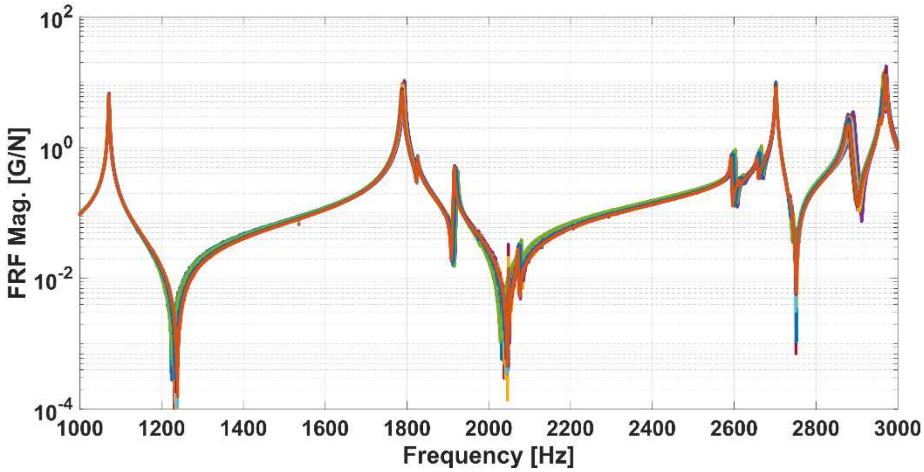
Fig. 3 Microphone measurement locations within the cylinder cavity

3 Identification and Measurements of Acoustoelastic Coupling

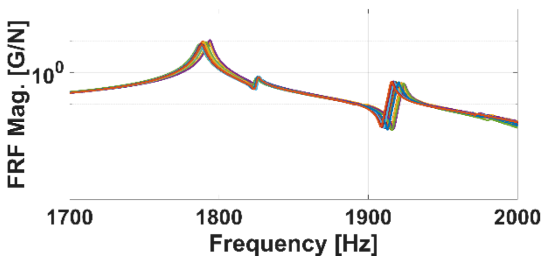
This section introduces methods for both identifying and measuring the acoustoelastic coupling in the system. A modal impact hammer provided the excitation to excite both the structure and consequently, the coupled acoustic response of the internal cavity. The first part of this section covers repeatability tests that provided the uncertainty in the response from day-to-day testing. The second part introduces a method of identifying the frequency ranges where the system seems to exhibit acoustoelastic coupling when both microphone and accelerometer data is available. The third part measures this structural-acoustic interaction and discusses the procedure used to extract the coupled structural-acoustic mode shapes.

3.1 Repeatability Testing

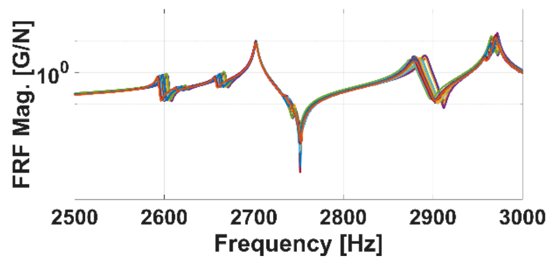
Baseline tests with an empty internal cavity were recorded on different days and times throughout the six week testing period. Figure 4 shows the drive-point FRFs for location 603 for each of the baseline measurements. The baseline drive-point FRFs changed in magnitude and shifted in frequency across many of the peaks. Identified causes of these variations include slight changes in bungee cord tension and location, cylinder end cap removal and reattachment, and variations in the enclosed air properties such as temperature and pressure. Air properties had a significant effect on frequency shifts because it results in a change in the speed of sound and a corresponding shift in acoustic frequencies, affecting the acoustoelastic coupling interactions. In Figures 4b and 4c, the close up of key frequency ranges show that the structural peak at 2700 Hz has very minimal frequency shifts between days, whereas the structural peak at 1790 Hz exhibits frequency shifts of the same magnitude of the coupled acoustic peaks. These large day-to-day variations show that it is important to only compare data from the same day to ensure differences observed are a result of test setup alterations and not changes from other factors



a) FRF, 1000–3000 Hz.



b) Zoomed in FRF, 1700–2000 Hz.



c) Zoomed in FRF, 2500–3000 Hz.

Fig. 4 Baseline test comparisons on different days and times

3.2 Coupling Identification

Coupling is identified through a simultaneous analysis of acoustic measurements and structural measurements. Figure 5 shows both the acoustic FRF from microphone A3 located at line 600. Also shown is the average structural FRF from all the accelerometer measurements. The hammer impact excites an increased number of acoustic modes compared to the structure; these additional acoustic peaks indicate that, although coupling does exist, it is not large enough to appreciably effect the structural response. In order to quickly identify when an appreciable amount of coupling exists to affect the structural response, a quantitative measure termed the magnification factor *MF* was developed:

$$MF = \left| \frac{H_{Ac}}{H_{St,RMS}} \right| \tag{1}$$

where H_{Ac} is the acoustic FRF at a point inside the cavity and $H_{St,RMS}$ is the RMS value of the average structural FRF across the entire frequency range of interest. As the structure drives the air inside the cavity, this measure is in terms of pressure / acceleration. Figure 5 also shows the *MF* at every frequency line and the sharp peaks indicate large coupling. In this frequency range of interest, three frequency bands exhibit large coupling: f = 1700-2000 Hz, f = 2550-2700 Hz, and f = 2900-3000 Hz.

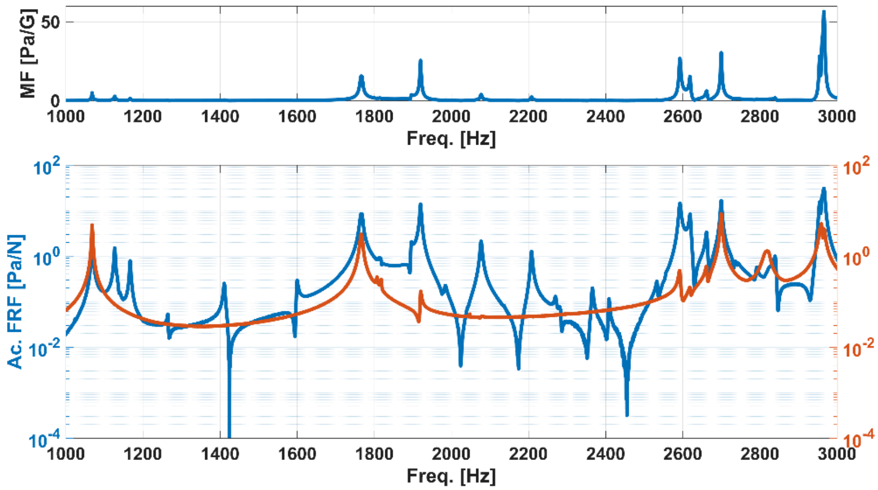


Fig. 5 Top: Magnification factor as a function of frequency. Bottom: Acoustic FRF for a single microphone location (left axis) and structural FRF averaged over all locations (right axis)

3.3 Coupled Mode Shapes

This section outlines the procedure used to measure the coupled structural-acoustic modes. The cylindrical test article showed symmetry requiring two hammer impacts at locations 603 and 605 (60º apart) to separate the repeated roots. An initial test provided the acceleration responses with the microphones removed from the cavity. To obtain the modal parameter estimates (frequencies, damping values, and mode shapes)**,** we utilized the Polymax curve fitting routine built-in with the LMS Test.Lab software. The results presented here focus on the f = 1700-2000 Hz frequency range where the first set of coupled modes exist, as indicated in Fig. 5. Figure 6 shows the structural drive-point FRF at location 603 over this range as well as the structural mode shapes shown with the hollow surface elements. The first peak near 1790 Hz is the location of two repeated (2,1) ovalling modes where the anti-node of the first mode was orientated with the +Y-axis of the cylinder and the second mode showed a 45˚ rotation; the second peak near 1820 Hz is two repeated bending modes but did not exhibit coupling with any acoustic modes; the third peak near 1920 Hz is the location of the coupled (2,1) ovalling modes.

Measuring the coupled acoustic mode shapes was not as straight forward. The microphones are not acoustically transparent so they add a perturbation to the acoustic cavity at each measurement location, and since the microphones were roved throughout the cavity, the boundary conditions are slightly different for each run. Additionally, the cap required an access hole for maneuvering the microphone array. Consequently, there was a slight frequency shift in the modes near 1920 Hz as the microphone array moved along the axial direction of the cavity. Such a shift required modal parameter estimations at each microphone location to obtain the local mode shapes which were then stitched together to produce the full cavity mode shape. Similar to the structural mode shapes, the acoustic mode shapes also showed repeated roots due to the symmetry of the internal cylindrical cavity. Figure 6 also shows the acoustic mode shapes for each mode of interest as filled volumes. Outward deflections indicate high pressure while inward deflections indicate low pressure. Furthermore, the first set of modes near 1790 Hz show that both the structural and acoustic modes oscillate in-phase, while the coupled pair near 1920 Hz show out-of-phase motion. These results are consistent with the findings in a previous numerical study that showed the lower frequency mode of a coupled pair exhibited this in-phase motion while the higher frequency mode exhibiting out-of-phase motion [5]. This result also makes sense when comparing this coupling behavior to a vibration absorber where the two coupled masses show in-phase motion at the lower modal frequency, and out-of-phase motion at the higher modal frequency [4].

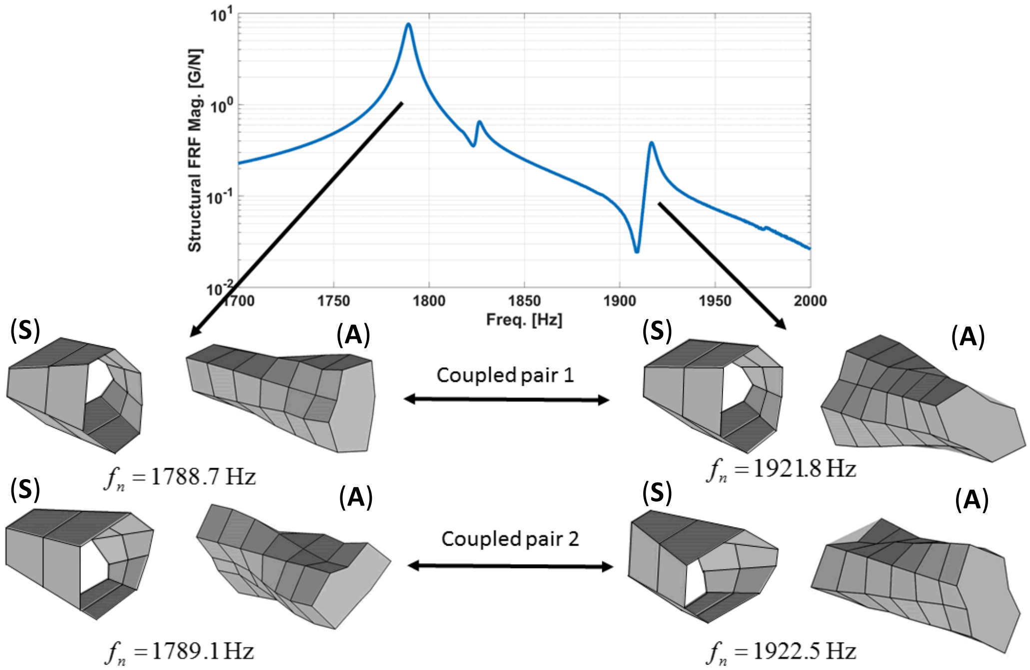


Fig. 6 Structural drive-point FRF with the associated coupled structural-acoustic modes shown. Hollow surfaces represent structural modes (**S**), and filled volumes represent acoustic modes (**A**) (positive and negative displacements correspond to positive and negative pressures)

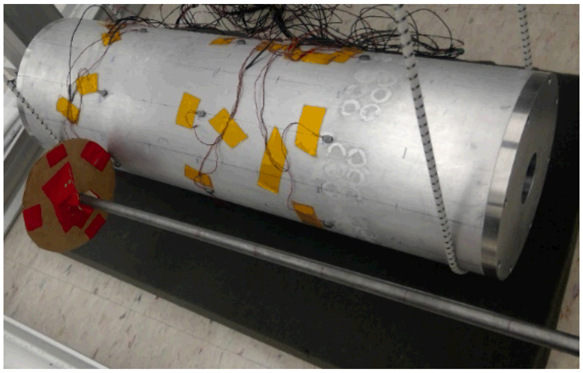
4 Coupling Mitigation Strategies

This section details the strategies employed to mitigate the acoustoelastic coupling and is separated into two approach methodologies: acoustic cavity modifications and structural modifications. Although there are three frequency ranges that exhibit coupling, this analysis focuses on the 1700-2000 Hz frequency range for brevity, although similar results were seen for the coupled modes in the other frequency ranges.

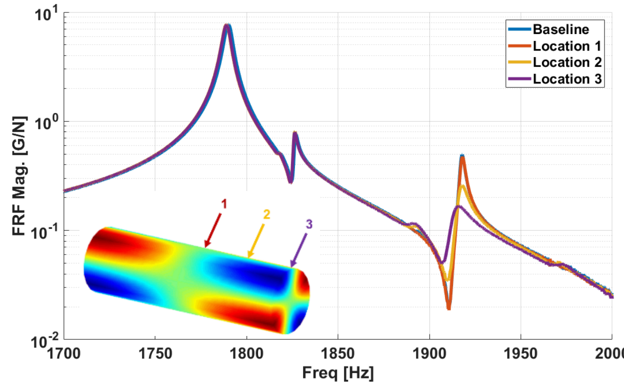
4.1 Acoustic Modifications

4.1.1 Partition Inclusion

As an initial attempt to disrupt the acoustic mode shapes and to reduce coupling, Fig. 7a shows a cardboard partition inserted into the cavity using a suspended rod to avoid contact with the structure. The partition is included to alter the boundary conditions and corresponding acoustic mode shapes and frequencies. Figure 7b shows the three axial locations of the partition as well as the target acoustic (2,1,1) mode, as well as the resulting effects on the structural drive-point FRF. With the partition placed at location 1, the axial node line of the acoustic shape, there was minimal effect on the coupling. As the partition travelled closer to the end of the cylinder where the pressure magnitudes increased, the coupling decreased, though did not completely disappear. This result is surprising as the expectation was that locating the partition closer to the edge would approximate an empty cylinder with no partition. The reason for this discrepancy may be that the partition was not fully rigid, and that there were air gaps around the periphery.



a) Cardboard disk partition connected to a rod to vary axial location.



b) FRF variation as a function of partition location.

Fig. 7 Inclusion of cardboard partition within cylinder cavity

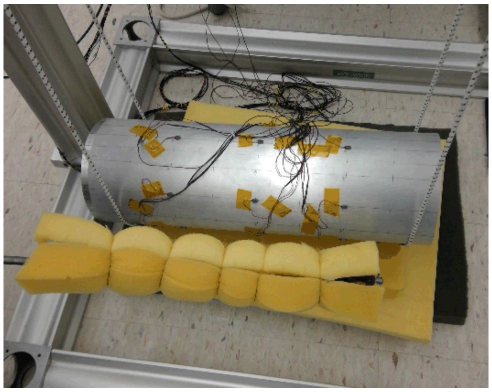
4.1.2 Foam Inclusion

Rather than disrupting the acoustic mode shapes, foam was inserted into the cavity to provide a source of acoustic damping to remove the acoustic energy from the system in a similar manner to [4]. Two different methods were utilized to insert the foam: Fig. 8a shows the first method with foam wrapped around the rod suspended in the cavity such that it was not contacting the structure and Fig. 8b shows the effects on the structural FRF. Filling the cavity with only 27% of foam resulted in complete removal of the coupled peak. For the structural peaks affected by the coupling behavior, there was a notable increase in damping as the foam increased in the cavity to a certain point corresponding to 66% of the cavity filled with foam. After which, further increases in foam decreased the damping for the structural peak as well as induced a slight frequency shift. This is a similar phenomenon seen with the vibration absorber model for the coupled system used in [4].

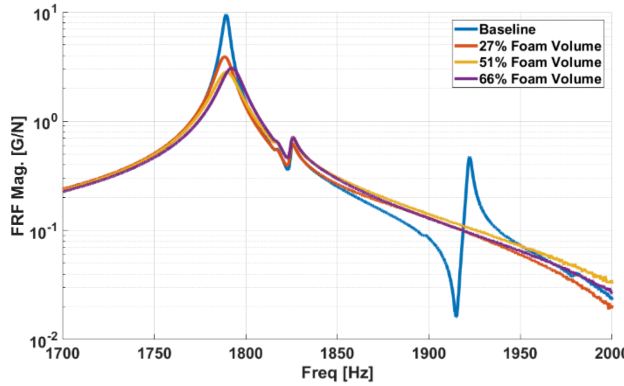
The second method involved incrementally inserting foam cubes, each with dimensions 50x50x50 mm^3, into the cavity. Figure 9a shows the foam cubes located within the cavity and Fig. 9b shows that for the same volume of foam, the foam cubes provided more acoustic damping and offers better reduction in coupling. One reason may be that wrapping the foam around the rod caused some compression in the foam, reducing the acoustic absorption potential. Although using foam cubes does require some contact with the structure, the mass-loading effect of the foam on the structure was negligible. Furthermore, inclusion of the foam eliminated the coupling from the 1.8 kHz and 2.7 kHz structural peaks with approximately 23% and 5% of the cavity filled with foam, respectively. The structural peaks at these frequencies also exhibited small shifts in frequency and decreases in damping after increasing the volume of foam past a certain point.

4.1.3 Randomly distributed paper towels

Other possibilities of introducing acoustic absorption while adding negligible mass to the system included randomly distributing common household paper towels within the cavity. Figure 10a shows the paper towels randomly distributed in the cavity and Fig. 10b compares the structural FRF from the best-case foam configuration with the paper towels. Surprisingly, including the paper towels resulted in the most effective removal of coupling, as evidenced by the slightly larger frequency shift and amplitude increase in the structural peak. Paper towels are the most advantageous approach by being more affordable, readily available, and quicker to implement than any of the previous mitigation strategies investigated. However, since the paper towel installation was random, it would not be easily repeatable or standardized.

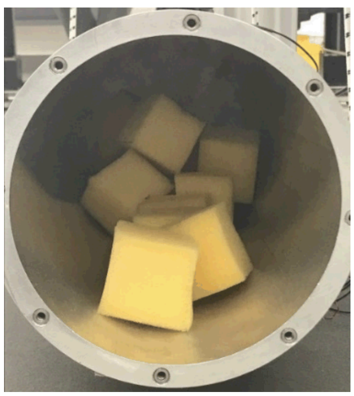


a) Foam wrapped around rod to suspend in cavity.

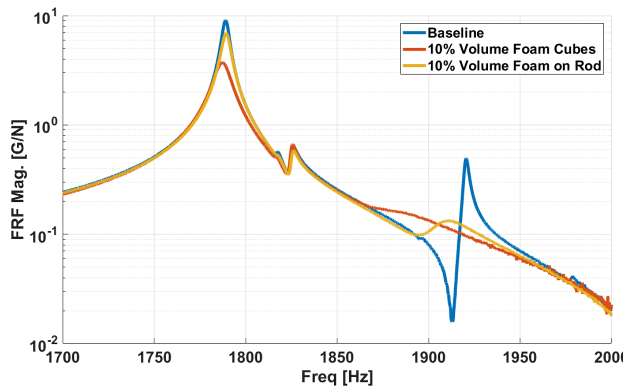


b) FRF variation with increasing foam volume.

Fig. 8 Inclusion of foam as an acoustic damper inside the cylinder cavity

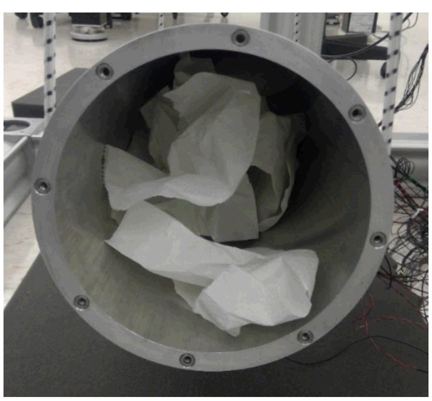


a) Foam cubes placed within cavity.

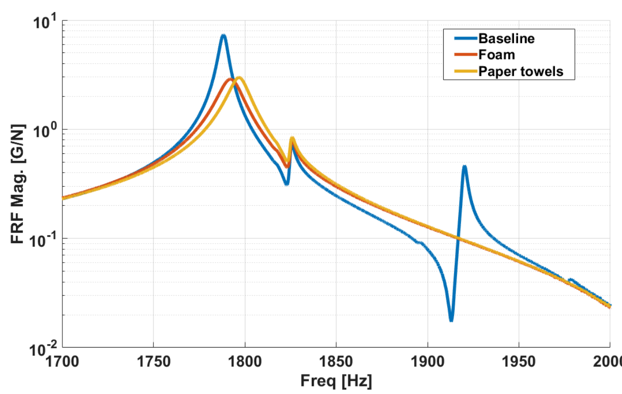


b) FRF showing the two foam approaches.

Fig. 9 Comparison between the two foam approaches: foam placed within the cavity as cubes (contacting), and foam wrapped around rod suspended in cavity (non-contacting)



a) Paper towels within the cavity.



b) FRF comparison between the best-case foam result and the paper towels.

Fig. 10 Randomly distributing house-hold paper towels within the cavity as an acoustic absorber

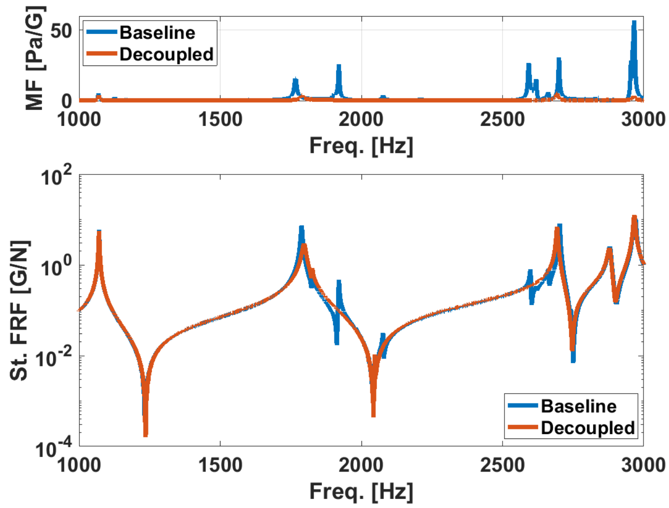


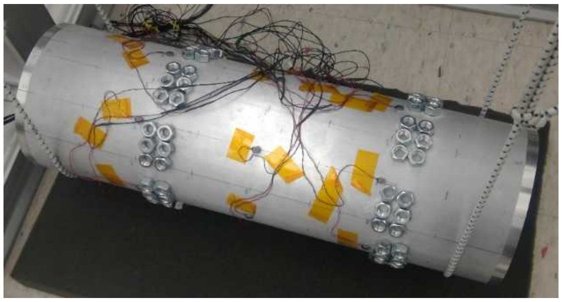
Fig. 11 Comparison of the decoupled and coupled structural FRFs and magnification factor with the decoupled response obtained by including paper towels within the cavity

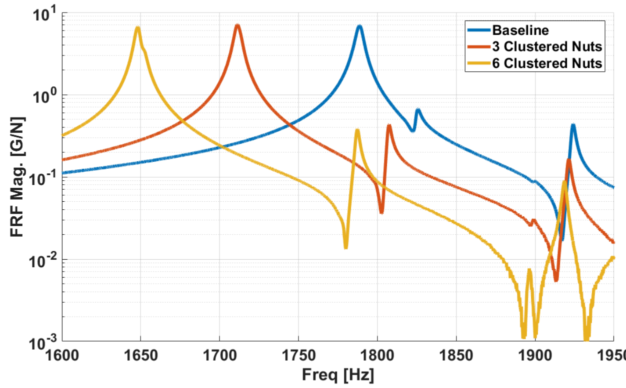
Figure 11 shows the decoupling of the full frequency range tested using the paper towels. The extra peaks associated with the coupled acoustic modes are no longer present in the structural FRF, clearly indicating the effectiveness of this approach. As a second measure of the effectiveness of this approach, the sharp peaks present in the magnification factor corresponding with the coupled modes show a significant reduction.

4.2 Structural Modifications

In some cases, the internal cavity of the system may not be accessible and the previously described acoustic modifications will not be available. Instead, the test structure was modified to shift the structural frequencies away from the acoustic frequencies and break up the similarity between the coupled mode shapes. Figure 12a shows the stainless steel nuts that provided the mass-loading on the structure. Bonding clusters of these masses at the antinodes of (2,1) structural mode with even spacing every 45° around the structure ensured adequate mass-loading for both of the repeated roots. Figure 12b shows that the largest applied mass induced over a 150 Hz frequency shift for the structural peak. A much smaller frequency shift occurred for the coupled peak with a corresponding reduction in the peak amplitude; the coupling, however, was not fully removed. Also, the added mass caused a second acoustic mode near $f = 1895$ Hz to begin coupling with the structure, an unintended and seemingly unpredictable consequence caused by the shifting of the structural frequencies. Although not entirely effective in its current form, this method should be further explored by adding significantly more mass to induce more dramatic structural frequency shifts and mode shape perturbations. Structural dynamic modification techniques could then be used to computationally remove the added mass and calculate the structural response.

A second structural modification involved wrapping hose clamps around the cylinder in an attempt to restrict radial motion and add stiffness to the system. Figure 13a shows one configuration of the hose clamped wrapped around the cylinder; Fig. 13b shows that this approach was unsuccessful as there was no significant effect on reducing the magnitude of the coupled peak. In fact, the structural peak showed a slight downward shift in frequency indicating that this method provided more mass than stiffness, and induced a splitting of the repeated roots of the structural peak. However, only a small regime of possible stiffness modifications was tested and the effectiveness of other methods was not explored. It is also worth noting that had these approaches been successful at decoupling the structural response, the FRFs would represent the cylinder with the corresponding structural modifications, thus requiring a method to remove the effects of the added modifications. As such, modifying the structure with discrete masses are more preferable to stiffness modifications as their effects are easier to remove.

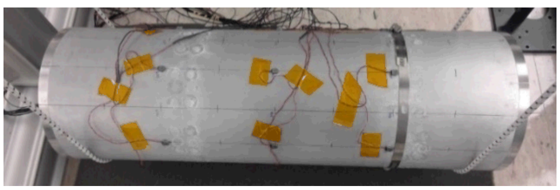


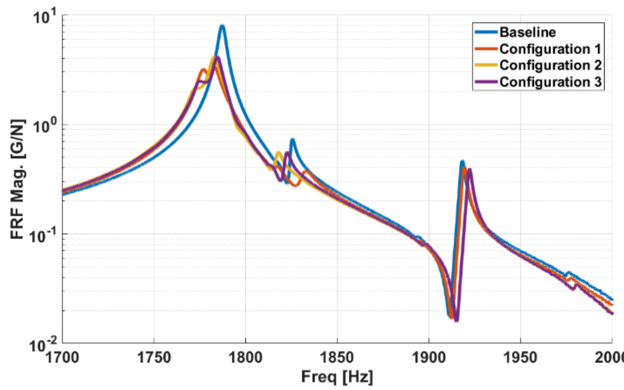


a) Aluminum nuts clustered around cylinder to add mass.

b) FRF variation due to increasing mass.

Fig. 12 Mass added at anti-node locations of the target mode to shift structural frequencies





a) Hose clamp wrapped around cylinder.

b) FRF variation for various configurations.

Fig. 13 Hose clamps wrapped around cylinder to increase stiffness by restricting radial motion

5 Conclusions and Future Work

This paper explored the phenomena of acoustoelastic coupling and developed ways of identifying, measuring, and mitigating the coupling effects. Typical structural impact tests on a hollow aluminum cylinder allowed for measurements of the coupled response FRFs. Development of a quantitative measure termed the magnification factor led to quick identification of the modes exhibiting large coupling. Measurement of the structural-acoustic interaction for one of the coupled modes revealed in-phase motion for the lower frequency coupled pair, and out-of-phase motion for the higher frequency.

Several mitigation strategies were also investigated. When the acoustic cavity is accessible, acoustic modifications showed the most promise in removing the coupling behavior. Selectively locating flexible partitions within the acoustic chamber did reduce, but not satisfactorily mitigate, the coupling behavior. However, including acoustic absorption material within the cavity successfully decoupled the structural response and typical house-hold paper towels proved to be the most effective mitigation method. For situations where the cavity may be inaccessible, the structural modifications employed so far have been unsuccessful.

Although some of the mitigation strategies outlined above provided a decoupled structural response, methods for determining whether this is the true in-vacuo response still require investigation. One method may be to include the structural-acoustic interaction in the analytical system model and tune the model parameters to match both the coupled numerical response to the experimental response; using the tuned structural parameters in a typical finite element model could then provide the decoupled response for comparison. A second method could be to isolate the acoustic modes by fixing or heavily damping the structure and then utilize a substructuring approach to remove their effects from the experimental response.

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