

Noise Control Capability of Structurally Integrated Resonator Arrays in a Foam-Treated Cylinder

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ABSTRACT

Corrugated-core sandwich structures with integrated acoustic resonator arrays have been of recent interest for launch vehicle noise control applications. Previous tests and analyses have demonstrated the ability of this concept to increase sound absorption and reduce sound transmission at low frequencies. However, commercial aircraft manufacturers often require fibrous or foam blanket treatments for broadband noise control and thermal insulation. Consequently, it is of interest to further explore the noise control benefit and trade-offs of structurally integrated resonators when combined with various degrees of blanket noise treatment in an aircraft-representative cylindrical fuselage system.

In this study, numerical models were developed to predict the effect of broadband and multi-tone structurally integrated resonator arrays on the interior noise level of cylindrical vibroacoustic systems. Foam layers with a range of thicknesses were applied near the inside surface of the cylinder to represent different degrees of conventional blanket treatments. Excitations including point force as well as harmonic and random fluctuating pressure fields were considered. The results suggest that structurally integrated resonators can be tuned to address a variety of noise control requirements and effectively used in conjunction with foam blanket noise treatments, but their relative benefit is reduced when thicker foam treatments are used.

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INTRODUCTION

High interior noise levels in aerospace vehicles can damage sensitive equipment, lead to crew hearing loss, and annoy passengers. While high-frequency noise can often be attenuated with conventional fibrous and foam blanket treatments, low-frequency noise is much more difficult to control without thick, heavy treatment. Passive acoustic resonators, however, have the potential to provide effective low frequency noise control. Helmholtz resonator, quarter-wave resonators, and variations thereof have seen widespread use in the field of building acoustics. In practice, they are typically combined into arrays of similarly tuned resonators to reduce noise at particular frequencies, or varied in tuning to target a wider band of frequencies. If installed into or alongside a wall, the resonators achieve noise reduction by increasing the absorption of the adjacent acoustic volume, but can also improve the transmission loss of the wall [1].

A successful example of passive resonant noise control within the aerospace industry is the use of perforate-over-honeycomb acoustic liners along the interior walls of aircraft engine nacelles to reduce fan noise. In simplistic terms, the performance of this type of liner is controlled by honeycomb cell depth and perforated face sheet resistance. Manufacturing constraints have previously limited the

variety of achievable cell depths in production liners, but recent developments have demonstrated the ability, through advanced manufacturing techniques, to create variable depth liners that can meet a wide range of noise control requirements [2]. These developments have influenced the work described here.

There are relatively few examples of the use of passive acoustic resonators for aircraft *interior* noise control. A series of efforts were carried out during the late 1980's and early '90's dealing with passive resonator arrays placed within a double wall (fuselage and trim) configuration. Mason and Fahy applied Helmholtz resonators along the perimeter of a flat double wall assembly to reduce transmission within a narrow frequency range near the double wall mass-air-mass resonance [3]. Their results showed that significant improvements to transmission loss can be made when the resonators were tuned to target a narrow range of frequencies. Shortly thereafter, Prydz et al. developed transfer matrix-based analytical methods to account for arrays of Helmholtz resonators installed in a trim panel assembly [4]. Kuntz et al. later demonstrated a 5-6 dB flight test noise reduction in the interior of a propfan driven aircraft (Gulfstream II) by installing an array of Helmholtz resonators between fuselage and trim [5]. However, the authors also mentioned the limitations of trim-

based treatments due to short-circuit paths from the fuselage structure to the interior via “storage compartments, air-conditioning ducts, racks, and galleys”.

The practical use of supplemental or “add-on” resonators able to provide substantial interior noise control at lower frequencies where foam and fiber treatments perform poorly is limited due to size and weight constraints. However, the integration of long “quarter-wave” resonators into existing primary fuselage structures could alleviate these limitations. Along these lines, researchers from the Air Force Research Laboratory and the University of Pittsburgh have recently studied chamber core sandwich structures with integrated acoustic resonators [6, 7]. They demonstrated that the noise reduction of a chamber core cylinder with integrated resonators could be increased at targeted frequencies. Tuned Chamber Core (TCC) is a similar concept recently studied for application in launch vehicle payload fairings [8]. An exploded view drawing and corresponding photo of a composite test article containing 138 resonators from [8] are shown in Figure 1. Improvements in both transmission loss and absorption were achieved around the targeted frequencies (200 Hz to 315 Hz) during standard ASTM E90 and C423 laboratory measurements. However, it is important to also mention that an increase in transmission was evident at higher frequencies due to the additional transmission path through the acoustically large inlets. This may not be a concern for many applications involving conventional blanket treatments, though, as the performance of conventional blankets often exceeds the noise requirements at higher frequencies.

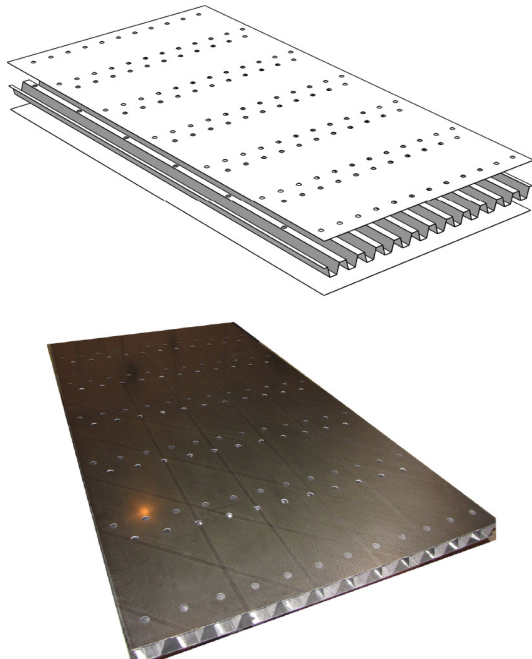


Figure 1. Exploded view drawing and photo of the 1.219 m by 2.438 m TCC test panel from [8].

Generally speaking, TCC and related concepts require that the primary fuselage or fairing structure be of a corrugated-core sandwich design (or similar fluted-core, Z-core, box-core, etc. design). Inlets are machined at particular locations on the interior face sheet and end-caps are inserted (or co-bonded) into the core chambers

at specific locations to create an array of tuned acoustic resonators. For a given application, it is envisioned that the chamber and inlet geometries would be designed to provide a desired noise reduction spectrum while also maintaining load bearing capabilities. While the ability of these structures to contain arrays of tunable, low frequency performing resonators without an appreciable increase in weight or reduced interior volume is of particular interest here, one can also imagine additional nonacoustic possibilities, such as fuselage integrated system wiring, environmental control system ducting, or thermal treatments.

This paper examines the benefits and relative trade-offs of conventional foam blanket and structurally integrated resonator array noise treatments in a corrugated-core cylinder using numerical modeling methods. The cylinder considered is representative in diameter of a regional airliner fuselage section. A description of the approach used when creating the numerical models is given. Details pertaining to how the corrugated-core structure and resonator arrays were represented in the model are also described. The noise control capability of the resonator array is a function of the input impedance of each resonator in the array. Consequently, a model of the resonator impedance is also described along with experimental evaluations. Finally, the model is exercised while considering different foam blanket treatments, resonator array configurations, and excitations and the resulting noise reductions (NR) attributed to each treatment are discussed.

MODELING

The following sections describe the finite element model of the cylindrical vibroacoustic system, and include details pertaining to how the corrugated-core structure and resonator arrays were represented in the model.

Cylindrical Vibroacoustic System

Finite element (FE) models consisting of a cylinder structure with interior and exterior acoustic systems and various thicknesses of foam blanket treatments were developed for analyses using a commercial FE software [9]. The cylinder is 2.6 m long with a diameter of 2.734 m. This diameter is representative of a regional airliner fuselage. An example of the FE model is shown in Figure 2*. While an FE modeling approach is more computationally expensive relative to other prediction methods, it was found to be tractable within the frequency range of interest (below ~500 Hz) through the use of parallelization.

The interior acoustic system was comprised of AC3D20 quadratic hexahedral elements with standard condition properties ($\rho = 1.21 \text{ kg/m}^3$, $c = 343 \text{ m/s}$). An additional foam blanket region near the inner surface of the cylinder was also modeled using AC3D20 elements, but frequency dependent complex effective densities and bulk moduli were used in this case. The foam properties were determined using Miki's equivalent fluid model and a lightweight 6 kg/m^3 blend of melamine with a flow resistivity of $9400 \text{ Pa}\cdot\text{s/m}^2$ was assumed [10, 11]. Hard wall boundaries were applied to the acoustic volume at the top and bottom of the cylinder. The cylinder was modeled with thick shell S8R quadratic quadrilateral elements and had an uniaxial stress-strain

approximately 162 kg. No structural boundary conditions were applied to the cylinder, i.e., it was modeled as “free hanging”. To account for the mass of the foam, additional nonstructural masses were applied to all foam-coupled shell elements. Energy lost due to exterior sound radiation was accounted for with an infinite radiation boundary condition, which was realized by wrapping the cylinder with two layers of acoustic elements and an outer layer of ACIN3D8 quadratic infinite acoustic elements. This abbreviated exterior boundary condition was used to approximate the radiation impedance imposed upon the outer surface of the cylinder. Because detailed knowledge of the exterior sound field was not needed, a more precise representation involving a spherical exterior volume was deemed unnecessary.

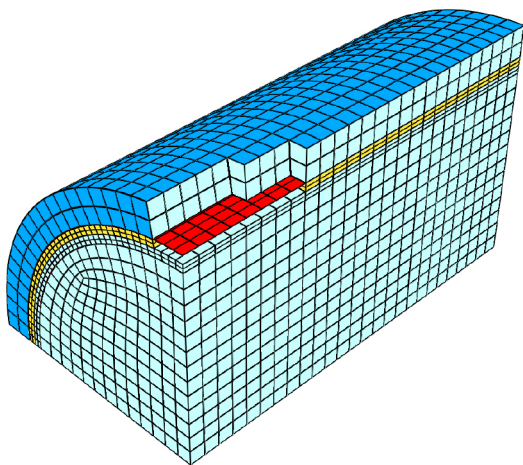


Figure 2. Cutaway view of cylinder FE model with 51 mm thick foam treatment. Acoustic elements shown in light blue, equivalent fluid foam elements in yellow, infinite acoustic elements in dark blue, and cylinder shell elements in red.

Structural-acoustic and acoustic-acoustic “tie” constraints were specified at the surfaces between shell and acoustic regions and between the interior acoustic volume and an annular region near the foam treatment. The acoustic-acoustic tie was utilized simply for rapid mesh refinement near the cylinder wall. This improved the convergence rate of the model by better resolving the relatively high wavenumbers within the foam and direct acoustic field of the cylinder. The structural-acoustic tie, on the other hand, coupled the normal velocities of the vibrating structure to pressure at the interfaces between acoustic medium and structure.

The analysis was partitioned into multiple bands from the 40 Hz to 500 Hz 1/3 octaves. The element sizing was specified to provide at least 8 quadratic elements per wavelength in the acoustic regions at the highest frequency in each band of analysis.

Corrugated-Core Sandwich Structure

A notional corrugated-core configuration was assumed for the purposes of this study. The structure is comprised of aluminum ($E = 70$ GPa, $\rho = 2700$ kg/m³, $\nu = 0.33$) with uniform 1 mm thick elements as depicted in the cross-section schematic of Figure 3. The area density of this structure is 7.2 kg/m².

To reduce model size, homogenized shell properties were used in lieu of modeling the detailed corrugated-core geometry. In-plane elastic constants $E1$, $E2$, $\nu12$, and $G12$ were defined for the homogenized shell while assuming orthotropic elastic behavior under plane stress. Transverse shear moduli $G13$ and $G23$ were also included. While a starting point for the estimation of these properties was informed from previous derivations by Lok [12], they were ultimately determined through trial and error by comparing eigensolution results of homogenized shell cylinders with explicitly modeled corrugated-core cylinders as shown in Figure 4.[†] The homogenized shell properties listed in Table 1 were found to adequately represent the stiffness characteristics of the corrugated-core structure selected for this study.

Table 1. Corrugated-core homogenized shell properties.

thickness	30 mm	$\nu12$	0.33
ρ	240 kg/m ³	$G12$	5.8 GPa
$E1$	4.9 GPa	$G13$	0.9 GPa
$E2$	7.8 GPa	$G23$	1.9 MPa

Resonator Impedance

As previously described, the core of the sandwich structure is partitioned into long cavities that can be used as acoustic resonators. A schematic of a single resonator is depicted in Figure 5. An inlet is created for each resonator by drilling a hole in the inside facesheet near one end of the cavity. Since the inlets are compact relative to the acoustic wavelength (within the frequency range of interest), the resonators are considered locally reacting and their effect on the interior acoustic field can be modeled as a surface impedance or admittance boundary condition in the vicinity of the resonator inlet. This assumption eliminates the need to explicitly model the core and cavity within a finite element modeling context.

The impedance model used in this work includes a term that accounts for the cavity and a separate term that accounts for the inlet. The specific acoustic impedance of the cavity below the frequency at which cross-section duct modes occur can be found using a one-dimensional plane-wave analysis as

$$z_c = \frac{-j\rho c S_i}{S_c} \cot(k_c L_c), \quad (1)$$

where ρ and c are the ambient density and speed of sound, respectively, S_i is the area of the inlet, S_c is the cross-section area of the cavity, L_c is the length of the cavity, and $k_c = \omega/c + (1-j)\alpha_c$ is the complex wavenumber. The attenuation constant, α_c , accounts for thermal and viscous losses along the cavity walls and is defined using a wide-duct boundary layer absorption model as

$$\alpha_c = \frac{1}{r_c} \sqrt{\frac{\omega\mu}{2\rho c^2}} \left(1 + \frac{\gamma-1}{\sqrt{\text{Pr}}}\right). \quad (2)$$