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Quieting a rib-framed honeycomb core sandwich panel for a rotorcraft roof

Stephen Hambric[1]; Micah Shepherd[1]; Royce Snider[2]; and Carl May[2]

[1] ARL/Penn State University

[2] Bell-Textron

ABSTRACT

A rotorcraft roof sandwich panel has been redesigned to optimize sound power transmission loss (TL) for frequencies between 1 and 4 kHz. The roof section, framed by a grid of ribs, was originally constructed of a single honeycomb core/composite facesheet panel. The original panel, characterized in [1], has coincidence frequencies near 700 Hz, leading to poor TL across the frequency range of 1 to 4 kHz. To quiet the panel, the section was split into two thinner sandwich subpanels separated by an air gap. The air gap was sized to target the fundamental mass-spring-mass resonance of the panel system to less than 500 Hz. The panels were designed to withstand structural loading from normal rotorcraft operation, as well as 'man-on-the-roof' static loads experienced during maintenance operations. Thin layers of VHB 9469 viscoelastomer from 3M were included in the facesheet ply layups, increasing panel damping loss factors from about 0.01 to 0.05. The optimized panel is expected to provide more than 10 dB transmission loss improvement at critical rotorcraft transmission tonal frequencies.

Keywords: Transmission Loss, Sandwich Panel
I-INCE Classification of Subjects Number(s): 13.1.4, 33, 42, 47.2, 75.3

1. INTRODUCTION

Commercial rotorcraft are powered by drive systems comprised of complex transmissions. As the transmission gears rotate at high rates of speed, they induce vibrations and noise at Gear Meshing Frequencies (GMFs) in the transmission cavity above the cabin (see the example in Figure 1). Structural roof panels are driven acoustically and structurally by the tones, radiating sound into the cabin. Composite materials are sometimes used to construct lightweight stiff panels for rotorcraft which meet structural integrity requirements, but also lead to increased interior sound radiation due to their reduced impedances and increased sound radiation efficiencies. Expensive and heavy acoustic treatments are therefore often added to the panels to reduce sound transmission. A more efficient noise control approach, however, is to better design the structural panel itself to minimize noise.

To characterize the structural-acoustic behavior of a typical sandwich roof panel, a notional design was constructed, as shown in Figure 2. A honeycomb core composite face sheet sandwich panel (see Figure 3) is mounted between a rectangular frame of large, stiff aluminum ribs which represent the roof rails (which run forward and backward) and the intercostal beams (which run side to side). The center panel edges taper downward to pure face sheet stacks at the rib mounting points, which extend beyond the ribs. The structural-acoustic behavior of the test panel has been evaluated computationally and experimentally previously [1]. The measured and simulated transmission loss (TL) of the baseline panel, shown in Figure 4, is low (less than 25 dB) between 1-4 kHz, the frequency range most critical for speech communication, and where strong rotorcraft transmission tones typically occur. In particular, our application is most concerned with Bull and Pinion Gear Meshing Frequencies (GMFs) at 1 and 3 kHz emanating from the rotorcraft transmission. Reference (1) also shows that the sound transmission through the center panel, which has a coincidence frequency of about 600 Hz, dominates the TL behavior.

[1] sah19@arl.psu.edu, mrs30@arl.psu.edu
[2] rsnider@bh.com, cmay@bh.com

In this paper, we summarize the design of an optimized roof panel which increases TL through the sandwich panel region between 1 and 4 kHz, and specifically near 1 and 3 kHz (the transmission GMF tones), but also does not violate several important design constraints, including:

- the allowable weight (mass/area) cannot exceed 5.7 kg/m^2,
- the thickness toward the cabin interior cannot exceed – 1.59 cm, and
- the structural materials must withstand worst-case in-flight loads as well as 'man-on-the-roof' loading for maintenance operations.

These constraints preclude the use of methods suggested in the literature to soften the sandwich panel to shift coincidence frequency upward, thereby ensuring the well known TL coincidence dip is higher than the frequency range of interest. Also, adding mass is clearly not an option due to the stringent weight requirement. This constraint is, of course, common in the aerospace community.

Three well known noise control procedures were applied to develop an optimized panel design:

- splitting the panel into two subpanels separated by an air gap,
- including a blanket, also used for insulation and fire protection, in the air gap, and
- embedding very thin viscoelastic layers within the outer and inner face sheet assemblies to increase structural damping.

This paper summarizes the development of the optimized panel, including a brief assessment of the structural integrity calculations.

Figure 1 – View of the inside of the transmission region in a commercial rotorcraft.

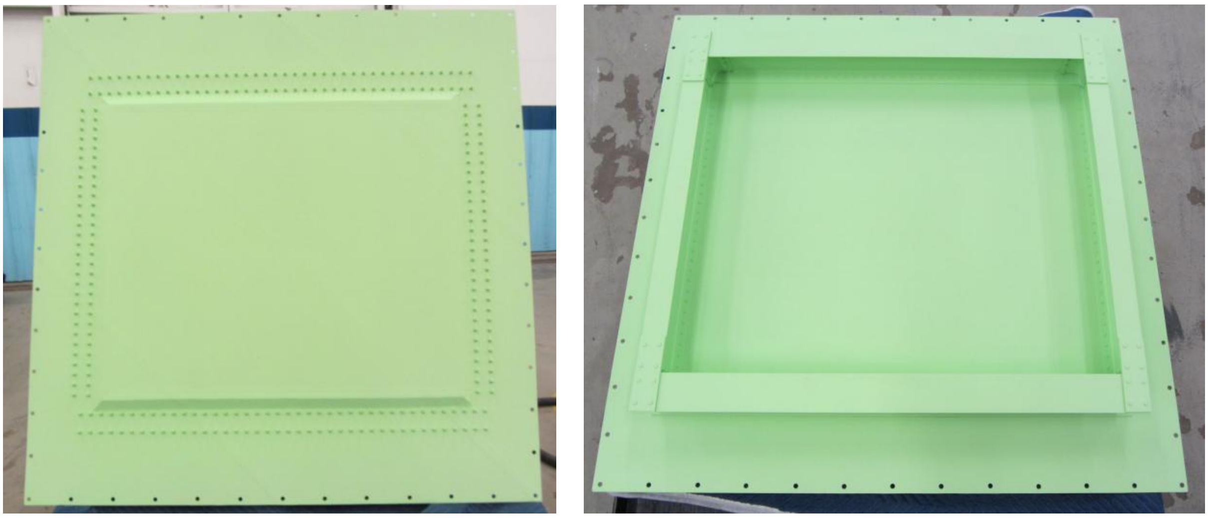

Figure 2 – Inner (left) and outer (right) views of the notional roof panel.

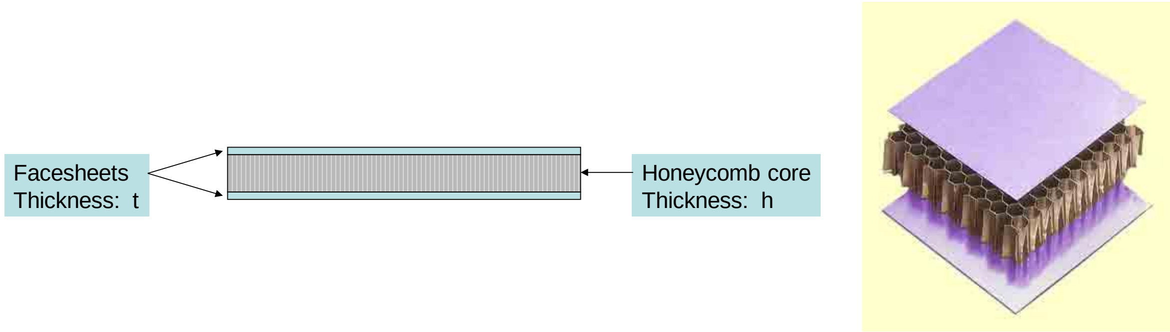

Figure 3 – Schematic of typical honeycomb core/composite facesheet panel.

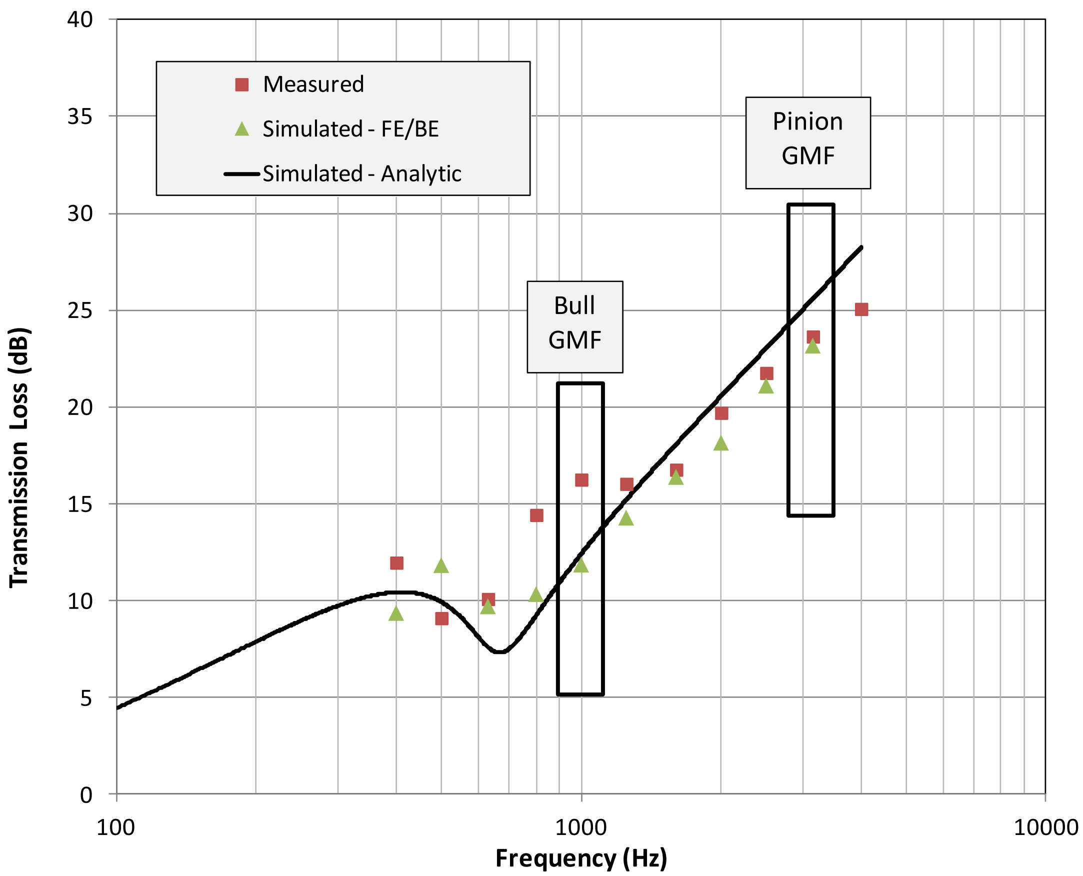

Figure 4 – Measured (at NASA's SALT facility [2]) and simulated TL for diffuse acoustic drive. Simulated and analytic curves are for the center panel region only (ignoring negligible sound transmitted through edge panels)

2. PANEL DESIGNS

Schematics of the cross-sections of the baseline and optimized panel are shown in Figure 5. The face sheets are made of layers of Cytec G30-500/5276-1 Carbon/Epoxy fabric, 0.2 mm (0.0079 in) thick, and the honeycomb cores are Hexcel Kevlar (1/8" cell size, 3.3 lb/ft^3). The cores are sandwiched by inner and outer face sheets. The baseline panel uses 3-layer fabric plies with [0/45/0] orientations, where the '0' indicates fibers in the 0 and 90 degree directions (aligned with the edges of the overall panel) and the '45' indicates fibers in the + and – 45 degree directions. The edges, which do not include honeycomb core, are stacks of 10 plies with orientation [0/45/45/0/0]S, where 'S' indicates symmetry about the center ply for the remaining angles. Selected properties of the panel materials are shown in Table 1. Note that the honeycomb core shear modulus is stiffer in the ribbon direction.

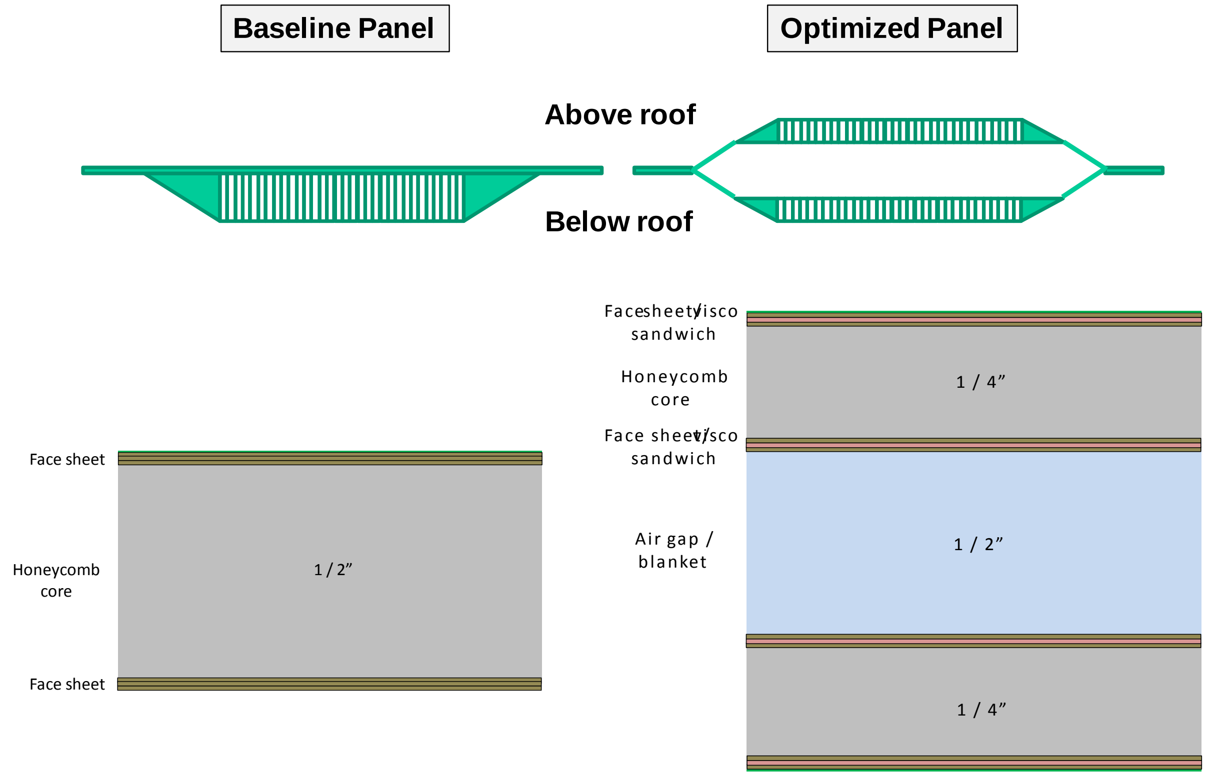

Figure 5 – Schematics and cross sections of baseline (left) and optimized (right) panels (not to scale).

Table 1 – Selected center panel material properties

Property	Face Sheets	Kevlar Core
E_{11}, E_{22} (GPa/Msi)	57 / 8.3	-
ν_{12}	0.21	-
G_{13} (MPa/ksi)	-	139 / 20.1 (ribbon)
G_{23} (MPa/ksi)	-	68 / 9.8 (warp)
ρ (kg/m^3 / lb/in^3)	1550 / 0.0560	47 / 0.0017

2.1 Embedded Viscoelastomer

Thin layers of viscoelastomer may be embedded within the facesheet sections to increase structural damping, and therefore random incidence TL at and above the panel critical frequency. The face sheets used in these panels, however, are extremely thin and lightweight. Each sheet is comprised of only three layers of 0.2 mm thick carbon fiber (+-90/+-45/+-90 degree orientations). Replacing the center layer with viscoelastomer requires similarly thin and light damping material. We use 3M’s VHB 9469 adhesive, which is 0.13 mm thick, of comparable mass density, and is formulated to have high damping properties near room temperature and at frequencies between 1 and 4 kHz. Replacing the center layer of each face sheet with a layer of the VHB material leaves outer and inner carbon layers with 0 and 90 degree ply orientations, leading to reduced face sheet net stiffness.

Young’s Moduli (computed assuming a Poisson’s ratio of 0.499) and loss factors for VHB 9469 are compared at 20 and 30 degrees C in Figure 6. The loss factors are quite high, ranging between 0.7 and 1.1 between 1 and 4 kHz. The net damping benefits of the VHB material were checked by performing experimental modal analyses on two test coupons. The coupon dimensions (48 cm x 58 cm) were chosen to avoid modal degeneracy, so that each structural mode is distinct in frequency and easily identified. Hexcel Kevlar core (1.27 cm thick) was used for the test coupon cores. The two coupons were constructed using different approaches. In the first panel, the carbon fiber and VHB were cocured, such that part of the VHB fused with the epoxy in the carbon fiber sheets. This formed a hybrid structure with uncertain properties. A second panel with pre-cured carbon fiber sheets post-bonded with the VHB was also constructed.

Complex modes were extracted from experimental modal analysis data, and loss factors and resonance frequencies were compared for the two panels. Figure 7 compares the modal loss factors for the two panels for frequencies up to about 5 kHz. The post-bond approach consistently yields higher damping, and both construction approaches lead to strong damping improvements at 1 and 3 kHz, where the dominant transmission tones occur. Based on these data, the post-bond approach was used for the optimized panel, and a nominal panel structural damping value of 0.05 is assumed for acoustic performance simulations.

Replacing the center carbon layer reduces the face sheet stiffness, thereby reducing the flexural wavespeeds. The measured mode shapes were used to estimate modal wavenumbers, which combined with modal frequencies determine modal wavespeeds. The modal wavespeeds were then used to infer an overall panel flexural rigidity 42% lower than that of the baseline panel rigidity. The reduced stiffness and wavespeed lead to slightly higher acoustic coincidence frequencies.

The test coupons also provide an opportunity to verify the FE modeling procedure for sandwich panels with layers of VHB material. Figure 8 shows a schematic of the cross-sectional modeling of the panels. The coupons were modeled with 4,370 quadratic solid elements. Each ply layer was modeled with one element through its thickness, and four elements represent the Kevlar core. The ribbon direction was modeled along the length of the panel. The adhesive layers between the inner plies and the core were not modeled explicitly, but the layer masses were simulated instead by increasing the adjacent ply surface mass densities. The final modeled and measured weights match almost exactly.

The viscoelastomer Young's Moduli were varied over several center frequencies per the data shown in Figure 6. Complex modes were extracted using NASTRAN for each property set, and modal frequencies were determined based on proximity to the center frequency of each set. Figure 9 compares the measured and simulated resonance frequencies, which agree to within +-4%. Figure 10 compares measured and simulated structural loss factors, which agree well for frequencies above 1 kHz. Below 1 kHz, the simulated loss factors are higher than the measured ones. However, since this project focuses on frequencies between 1 and 4 kHz, we will not pursue the cause of this discrepancy. Overall, the good agreement between measured and predicted resonance frequencies and loss factors confirm the modeling procedure and the underlying material properties.

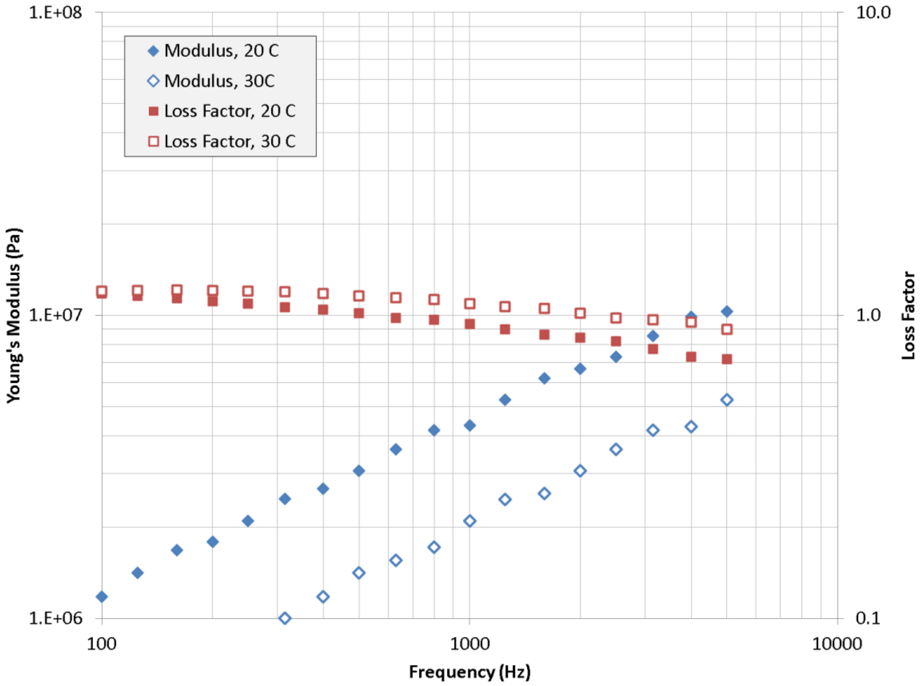

Figure 6 – Young's modulus and loss factor for 3M VHB 9469 at 20 C and 30 C.

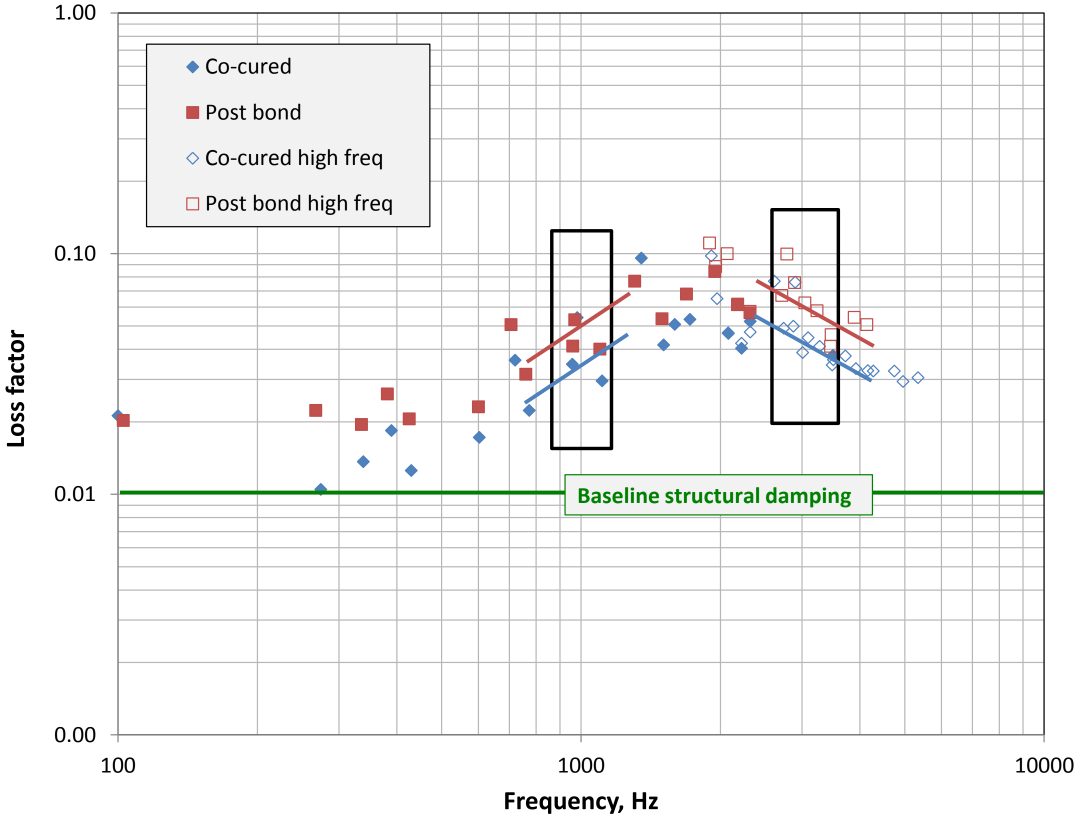

Figure 7 – Measured structural loss factors for two test coupons with embedded VHB 9469.

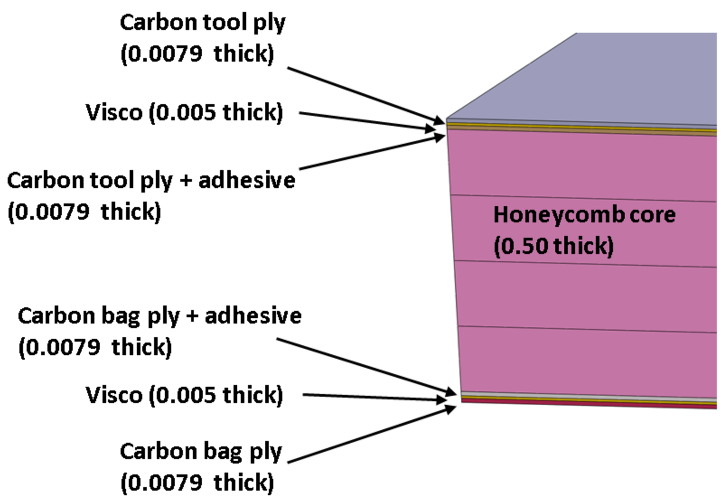

Figure 8 – FEM model for test coupons with embedded VHB material.

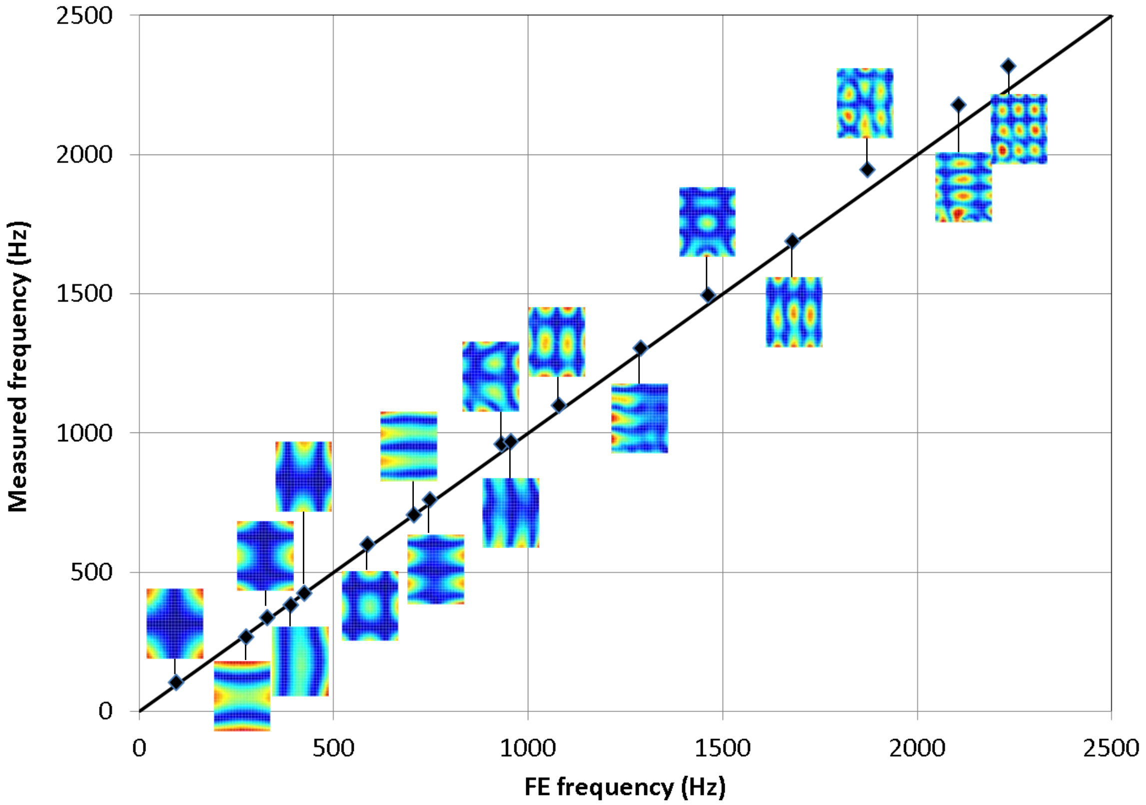

Figure 9 – FE vs. measured resonance frequencies for several mode shapes for test coupon with embedded VHB 9469.

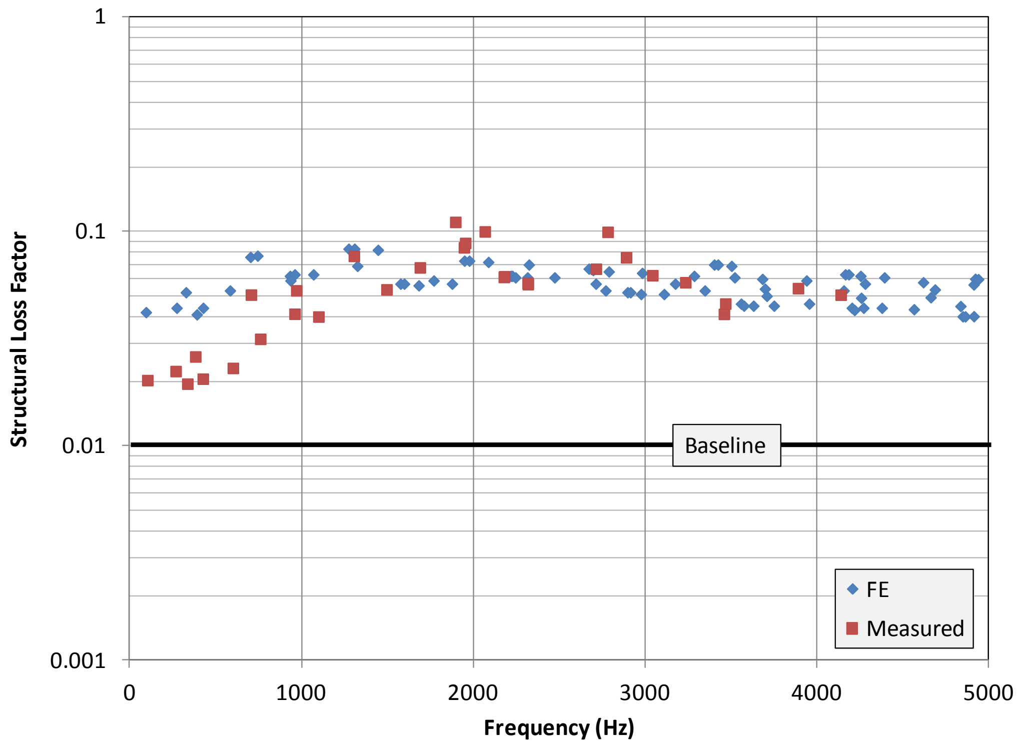

Figure 10 – FE vs. measured damping loss factors for test coupon with embedded VHB 9469.

2.2 Air gap sizing and fill

The 12.7 mm (0.5 inch) gap between subpanels is chosen to ensure that sound transmission degradation associated with the well-known mass-spring-mass resonance of a double panel system is well below 1 kHz. The resonance frequency, where each panel acts as a lumped mass connected by the stiffness of the air gap, is:

$$f_o = \frac{1}{2\pi}\sqrt{\frac{\rho c^2 / d}{m_1 m_2 / \;\; m_1 + m_2}}$$

where ρc^2 is the bulk Modulus of air, d is the gap thickness, and m_1 and m_2 are the two outer panel area densities. In the equation, the numerator represents the air gap stiffness per unit area, and the denominator represents the effective total panel mass per unit area. This resonance amplifies the sound transmission through the double panel system at and around its resonance frequency. The effects of the gap thickness on the mass-spring-mass resonance, and on the overall panel thickness, are summarized in Table 2. A 12.7 mm gap shifts the resonance below 500 Hz, which is sufficiently low so that TL degradation should not occur above 1 kHz.

Rather than leave the air gap empty, it is filled with a 9.5 mm (0.375 inch) thick layer of Amber Microlite AA insulation (24 kg/m^3) from Johns Manville. The insulation provides thermal insulation, as well as reduced sound transmission through its added mass. It is common to add an extra layer of Microlite contained within a thin plastic covering on the inside surfaces of current rotorcraft roof panels. However, the layers are costly, and must often be removed when servicing the panels. Including the insulation inside the panel is preferable. The added acoustic transmission loss benefits are modest, and due to the added mass of the material, as shown in Table 3.

Table 2 – Effects of air gap thickness on mass-spring resonance frequency and overall panel thickness.

Air gap thickness (mm)	Resonance Frequency (Hz)	Overall panel thickness (mm)
3.18	911	18.0
6.35	644	21.2
12.7	456	27.5

Table 3 – Measured and mass-law based sound transmission loss improvements due to use of Microlite.

Frequency (Hz)	dB, Measured by Vendor	dB, Mass Law
500	2.4	2.4
1000	4.6	4.7
2000	6.6	6.8
4000	8.8	10.5

3. Analytic TL estimates

An analytically based estimate of the performance benefits of the split panel damped design, using infinite panel TL theory (validated against the baseline panel measurements shown in Figure 4) is shown in Figure 11. The optimized panel coincidence dip is higher in frequency, since the split panel cores are half the thickness of that of the baseline panel. Also, the face sheet rigidities are 42% lower due to the softer center VHB layers. However, the coincidence dip was targeted to lie between the two transmission GMF tones at 1 and 3 kHz. The double panel concept nearly doubles TL at both 1 and 3 kHz. Adding the Microlite may also improve TL, but it remains to be seen how well it works above the coincidence dip. The performance improvement of a few dB near 1 kHz is fairly certain, but the 3 kHz benefit must be proven by measurement in the NASA SALT facility [2] when the optimized panel is constructed.

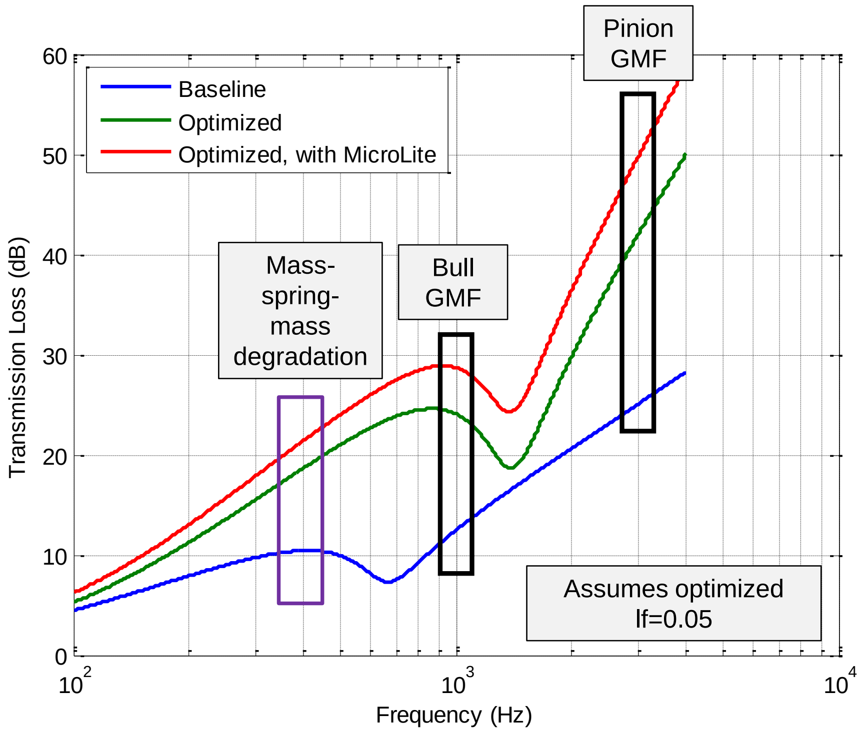

Figure 11 – Analytic estimates of transmission loss benefits of split panel concept.

4. FE Modeling and Structural Analysis

The optimized panel was modeled using finite elements, as shown in Figure 12. Each ply of fabric and each layer of VHB were discretely modeled with a single layer of elements through the thickness. Each core was modeled with two elements through the thickness. Adhesive plies were not included in the model, as they have negligible effect on the structural performance of the panel. Beams, straps and angle brackets were used to represent the support structure of a prototypic roof frame. The brackets, straps, beams, and panel are connected with fasteners. These elements are connected together in the FE model using spring elements at the fastener locations. The nominal smeared material properties are listed in Table 1.

The panel dynamic response is very sensitive to the total panel weight. The weights of the adhesive (between face sheets and honeycomb core), surface paint, and fasteners are accounted for in the model. Point mass elements are added at each fastener location. The adhesive and paint weights are included by adjusting the density of the elements adjacent to them.

While we are focused mostly on the acoustic performance of the panel, it is still necessary to analyze the panel structural integrity using critical design loads for representative rotorcraft roof panels. Skin panel strength, ramp strength (the transition region between the center panel and the frame), edgeband fiber and bearing strength, panel stability and step load (man-on-the-roof) response were analyzed. Upper skin applied ultimate loads were based on 150% of limit flight loads and were used to analyze the critical skin region using an elevated temperature wet open hole compression allowable. The edgeband fiber strength analysis is similar to the skin panel strength analysis, except that the loading moment is applied directly to the edgeband. The edgeband bearing strength was assessed for 'jump takeoff' load conditions. To assess buckling, the panel was held fixed at the frame and critical loads and moments were applied. The resulting critical buckling eigenvalues are both substantially greater than 1.0, demonstrating that the plies in the ramp will provide adequate stability under worst case operating conditions. Finally, a 600 lb man-on-the-roof load was applied to the center of the panel. A nonlinear static analysis was run which shows that the top panel will contact the bottom panel in this case, sharing the load between the panels. Assuming a worst-case deflection (shown in Figure 13), maximum stresses were computed and found to be within allowable limits. The maximum edge forces were then used to compute critical strains in the upper face sheet, which were also found to be well within allowable limits.

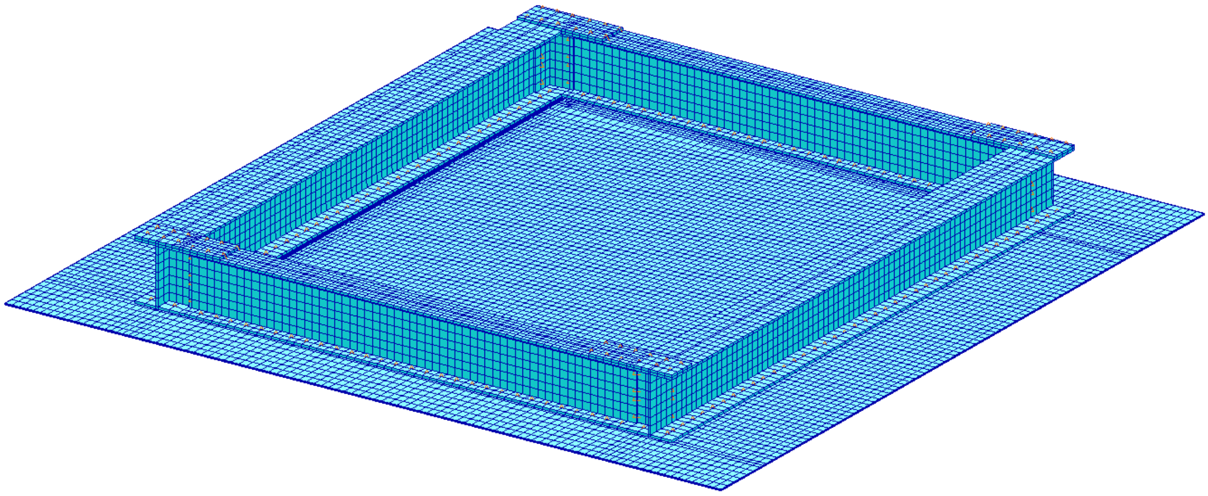

Figure 12 – FE model of optimized panel.

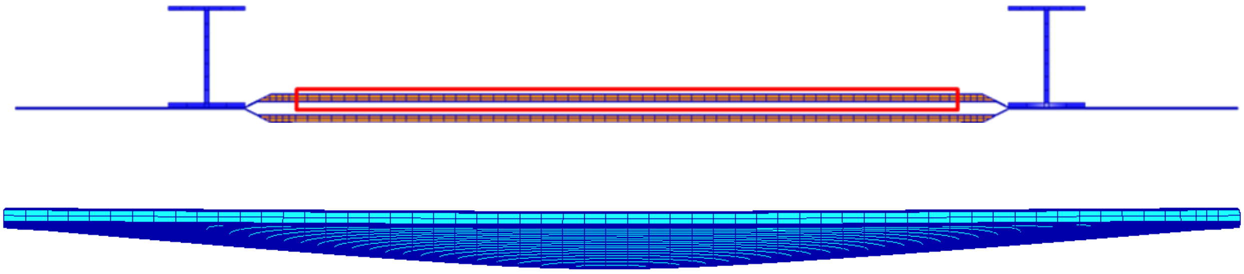

Figure 13 – Top: Panel section for step load analysis highlighted in red box; Bottom: Non-linear analysis deflection results.

5. SUMMARY AND CONCLUSIONS

An optimized rotorcraft framed roof sandwich panel has been designed to improve sound power TL between 1 and 4 kHz, and for two transmission GMF tones at 1 and 3 kHz in particular. The final optimized panel balances acoustic performance with structural integrity constraints, as well as meeting weight and space goals. The split panel concept is augmented with damped face sheets which include embedded VHB viscoelastic material, and filled with MicroLite blankets. The air gap is sufficiently thick so that the mass-spring-mass panel resonance is well below the lower frequency range of interest. Although the optimized panel is thicker than the baseline panel, the excess thickness is shifted to outside the fuselage, and will not affect the transmission or other electrical, mechanical, or hydraulic elements in the roof cavity region. The optimized panel design will be constructed, and then tested in NASA's SALT facility to confirm the simulated TL improvements.

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