

Acoustically Absorbing Lightweight Thermoplastic Honeycomb Panels

James M. Jonza, Thomas Herdtle, Jeffrey Kalish, Ronald Gerdes, and Taewook Yoo
 3M Company

Georg Eichhorn
 3M Deutschland GmbH

ABSTRACT

The aerospace industry has employed sandwich composite panels (stiff skins and lightweight cores) for over fifty years. It is a very efficient structure for rigidity per unit weight. For the automobile industry, we have developed novel thermoplastic composite panels that may be heated and shaped by compression molding or thermoforming with cycle times commensurate with automotive manufacturing line build rates. These panels are also readily recycled at the end of their service life. As vehicles become lighter to meet carbon dioxide emission targets, it becomes more challenging to maintain the same level of quietness in the vehicle interior.

Panels with interconnected honeycomb cells and perforations in one skin have been developed to absorb specific noise frequencies. The absorption results from a combination and interaction of Helmholtz and quarter wave resonators. Computer modeling was used to design panels that absorb one of the problematic frequency ranges (800-1,250 Hz), caused by tires on the roadway. This was achieved in thin (7-8 mm) panels of low density (0.25-0.35 g/cc). Experimental results of acoustic absorption and transmission loss of polypropylene (PP) panels will be presented along with their mechanical performance.

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INTRODUCTION

There is work underway worldwide to reduce carbon dioxide emissions following the Paris agreement, signed by 195 nations in December of 2015 [1]. The transportation industry is working to reduce the weight of vehicles, since this has a direct correlation with CO₂ emissions for internal combustion engines. For electric vehicles, weight reduction increases vehicle range per charge.

Sandwich panels, stiff skins with cores of sufficient compression and shear strength for the application, are an area of technology heretofore under-utilized in the ground transportation industry. To make this technology viable, continuous production with recyclable materials are important, since the percentage by weight at the end of a vehicle's life continues to increase [2].

As vehicles become lighter, the acoustical mass law dictates that more noise will be transmitted. It becomes imperative that acoustical engineering solutions be developed to maintain the level of "quietness quality" in the passenger cabin. Furthermore, the limits on drive-by noise are coming down in many nations [3], necessitating the absorption of sound to reduce it in the neighboring environment.

Fibrous absorbers are commonly used to absorb sound. The energy of the sound wave is absorbed by the air velocity between the fibers, with viscosity causing energy loss in the form of heat. The efficiency

of absorption for frequencies below 2,000 Hz is poor, leading to fibrous absorption layers that are 25-50 mm thick to cover 500-1,000 Hz [4]. These mats typically have no mechanical function, and add some weight to vehicles.

Honeycomb panels have seen widespread use in the aerospace industry because of their excellent stiffness/weight ratio. The mechanical properties and design criteria are covered in textbooks [5, 6]. The vibroacoustics of sandwich panels were recently reviewed by D'Alessandro et al [7]. Because of their decreased mass, sandwich panels are inferior to solid panels in their sound transmission loss (STL), and a drop in STL occurs when the wave speed in the panel equals that in air (coincidence frequencies for panel wave propagation modes). Several approaches may be used to improve the STL of sandwich panels: 1) decrease the shear modulus and increase the viscoelasticity of the core 2) reduce the stiffness/weight ratio and/or 3) use damping materials. For the latter approach, a damping layer in the middle of the (stiffer) core is most effective. Wang [8] did a thorough assessment of sandwich models in the prediction of STL and handles the topic of coincidence frequencies (see also Moore [9]).

Because the STL improvements typically compromise the stiffness and/or the weight, there has been considerable work to make honeycomb panels that absorb sound. The principle of Helmholtz resonance has been used in honeycomb panel sound absorption. A volume of air in the cavities of the core communicates with the

atmosphere via perforations in the skin or a porous skin. The volume in the cavities is the key variable to tune the absorption. This approach leads to a narrow absorption band [10].

Another approach is the microperforated panel, where the skins have sub-millimeter holes to provide acoustic resistance and low acoustic mass reactance. Maa [11] provided an approximate analytical solution to panels with microperforated skins some distance from the back skin (or solid surface) and showed that for perforate constants less than one, 3 to 4 octaves of absorption breadth are possible for relative acoustic resistances of 3-5. This work did not consider any structure in the cavity between the perforate and solid skins.

More recently, Pan et al [12] studied the sound absorption of honeycomb panels with a microperforated skin, with an experimentally measured absorption band from 700 to 1,700 Hz for 19 mm thick panels with 12.7 mm hexagon width, 0.3 mm skin thickness and 0.8 mm diameter holes.

This paper will briefly go over the design criteria for sandwich panels, then cover the absorption behavior of panels with interconnected cells within the core. The absorption behavior as a function of the panel thickness, cell sizes, passageway percentage and skin hole sizes will be demonstrated via modeling and experimental measurements. The absorption band from 800-1,250 Hz that corresponds to road noise frequencies will be exemplified with this structure. Sample preparation, acoustical measurements and mechanical property measurements are also described.

DESCRIPTION OF THE CONCEPT

Sandwich Panels for Stiffness with Minimal Weight

The sandwich principle, two skins separated by a lightweight core is found in nature in the structure of bones. Most commonly, dense bone skins are surrounding spongy, cancellous bone. There are numerous commercial uses of sandwich panels with foam cores, including structural thermally insulating panels that are gaining favor in building construction. Most of the foams used in sandwich panels have micron to millimeter-scale closed cells, and therefore have no mechanism for acoustical absorption.

Honeycomb cores have macrocellular structure, with walls arranged in a pattern within the plane of the panel and perpendicular to the skins. The density is determined by the wall thickness and geometry of the pattern. The most common honeycomb is hexagonal in shape, often with two of the six walls having twice the thickness of the other four walls because those walls were glued or fused together, then the structure was pulled open to form the hexagons. Commercially available hexagonal honeycomb panels are available with hexagon widths from 3 to 25 mm, thicknesses from 3 to 150 mm and densities from 20 to 300 kg/m³.

For those not familiar with the flexural property increases from separating the skins with a lightweight core, an example calculation is shown in Figure 1.

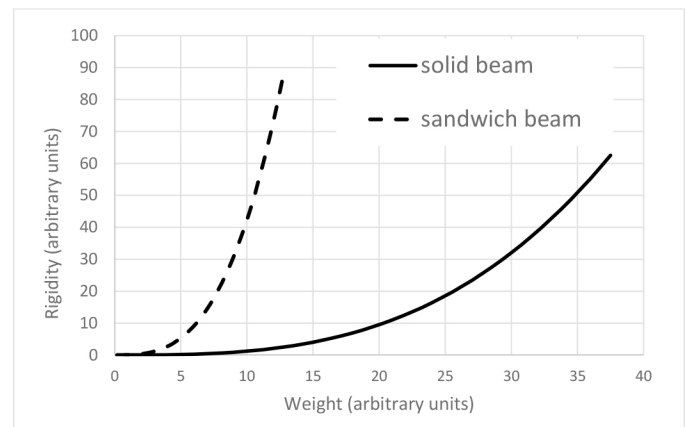


Figure 1. Flexural rigidity from equation (1) plotted vs. weight for a solid beam compared to a sandwich panel. Each skin is made of the same material as the solid beam and is 10% of the total panel thickness. The core is set at 5% of the solid material's density and 1% of the solid material's modulus.

The flexural rigidity is given by equation (1), where the first term comes from the stiffness of the skins, the second from the skins separated by the core and the third from the core.

$$D = E_s \frac{bt^3}{6} + E_s \frac{btd^2}{2} + E_c \frac{bc^3}{12} \quad (1)$$

where D = Rigidity ($N \cdot m^2$), E_s = Elastic Modulus of the skin (Pa), E_c = Elastic Modulus of the core (Pa), b = width (m), t = skin thicknesses (m), c = core thickness (m), and d = skin separation distance (m) = $c + t$.

Gibson and Ashby [13] provide some methodology to optimize sandwich panel constructions for all the variables in equation (1) for stiffness, strength, and weight for several loading conditions.

Sandwich panel flexural strength is usually limited by the adhesion strength of the core to the skins or the core shear strength when stiff skins are used. Compression of honeycomb cores requires considerable energy, as the walls need to buckle progressively with increasing strain. Dynamic energy absorption usually exceeds the energy to crush a honeycomb panel in a slow compression [14] and the specific energy absorption is high.

Increasingly, a higher percentage of the weight of vehicles needs to be recyclable at the end of the vehicle's life, so we have prepared acoustically absorbing honeycomb panels out of thermoplastic resins with either no fiber reinforcement, or a single fiber type.

Interconnected Cells for Lower Frequency Absorption

Typical honeycomb panels, with a solid, continuous skin on either side, have very little inherent absorptive properties. If one skin is created with microperforations, as suggested by Maa [15], then the panels can have significant absorption. However, the peak absorption will be at a frequency dependent on the depth of the honeycomb,

related to the quarter-wave resonance; for example, an 8 mm deep honeycomb with a typical microperforated skin would have peak absorption over the range of 2,500-4,500 Hz.

Alternatively, if a much smaller perforation ratio is used with, for example, one small hole per chamber, then each honeycomb volume can act as an independent Helmholtz resonator. However, the absorption is often narrow-band and the peak performance drops quickly when the peak frequency is below 1,000 Hz.

Therefore, in this work, panels were developed where multiple cells of the honeycomb structure are interconnected; with only one opening to the outside from one of the end-cells, these form cavities of various lengths in the plane of the honeycomb. This concept is similar to the works by Warnaka [16] and Hua [17], but applied to a honeycomb structure. The longer channels result in absorption at lower frequencies, since now the whole channel length, consisting of multiple cells, needs to be considered as the “depth”. The opening from the end-cell to the outside can be either a single, relatively large, hole or multiple smaller holes. A schematic of a potential channel layout, comprising 37 hexagons with channels of length 5, 6, 7, and 8 hexagon widths, is shown in Figure 2. This larger hexagon can be repeated to any width or length and maintain nearly isotropic panel properties. The periodic area shown is about 39 cm².

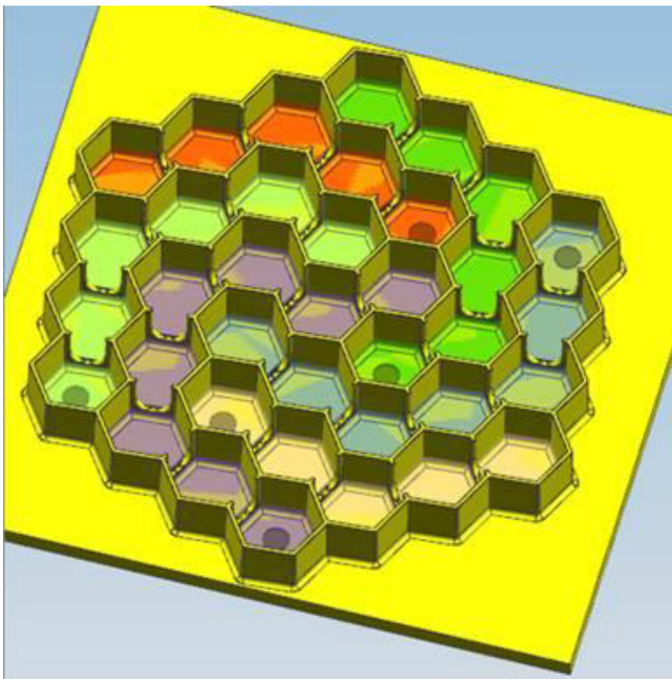


Figure 2. Schematic of a potential layout of connected channels comprising lengths of 5, 6, 7, and 8 cells, shown in differing colors. The potential location of the openings to the channels is indicated by the gray circles. This pattern can be repeated to cover a larger, complete panel.

MODELING

The development of these designs would not have been possible without the extensive use of modeling. In particular, the thermo-viscous model in the Acoustics module of COMSOL Multiphysics® [18] was used. This tool allowed us to determine the viscous losses

created by various channel and hole designs and thus predict the normal incidence acoustic absorption as well as the surface normal impedance as a function of frequency. Random incidence was computed from the impedance under the assumption that the structures are locally reacting.

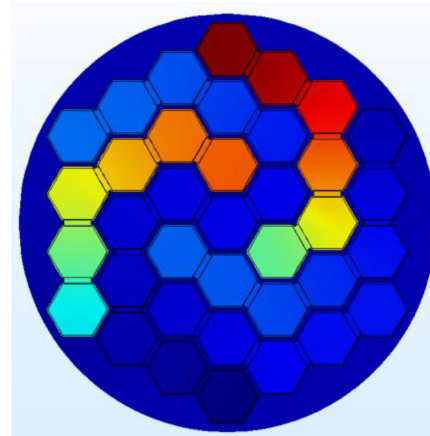


Figure 3. Pressure field at 850 Hz for the design from Figure 2. At this particular frequency, the two length-six channels are near resonance.

Figure 3 shows the acoustic pressure field for an example model with two channels near resonance at this particular frequency; being in resonance results in particularly large viscous losses and thus high absorption at such frequencies.

Modeling showed that the actual channel layout to be substantially irrelevant. Therefore for the experiments described below the most convenient layout for each particular experiment was used. The experiments thus ended up using three differing layouts which will be described later in the experimental section.

Tuning the Absorption Band(s)

There are a large number of design choices that can be made, all of which modify the frequency-dependent absorption. The choices include the number of connected cells creating channels, the opening hole size to the exterior, the height and width of the hexagons, and the opening size connecting the hexagons. All results in this section are for normal incidence absorption.

In this section, we highlight the effects of individual design parameters. The individual examples shown may vary in design (e.g. opening sizes, channel lengths, etc.) from one plot to another, but within each individual plot the only change from curve to curve is the parameter under discussion.

Effect of Number of Connected Hexagonal Cells

The strongest effect on absorption is from the number of connected cells. As can be seen in Figure 4, as the number is increased, the peak absorption moves to lower frequencies. However, since a panel with many connected cells (i.e. long channels) has a fewer total number of channels, the absorption band gets narrower as the number of connected cells is increased.