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Sound transmission loss of damped honeycomb sandwich panels

Portia R. Peters[a]
University of Southern California
3651 Watt Way, VHE-602
Los Angeles, CA 90089-0241
USA

Dr. Shankar Rajaram[b]
ATS Consulting
801 S Grand Ave, Suite # 575
Los Angeles CA 90017
USA

Dr. Steven Nutt[c]
University of Southern California
3651 Watt Way, VHE-602
Los Angeles, CA 90089-0241
USA

ABSTRACT

Honeycomb sandwich panels are widely used in aerospace applications due to their high stiffness-to-weight ratios; however, this mechanical benefit is accompanied by poor acoustical performance. The aerospace industry is constantly trying to optimize acoustical and mechanical performances in commercial airplanes. By investigating noise control solutions that reduce sound transmission through the sandwich panels used in aircraft floors, it is possible that quieter panels can be produced without sacrificing strength. Damping has a direct effect on sound transmission loss (TL) of honeycomb sandwich panels at and above coincidence frequency (f_c). The relationship between TL and damping is presented in this paper. The scope of this particular study is to include a materials' perspective in addition to the structural characteristics of honeycomb sandwich panels. This study included the following panel/beam designs: glass-epoxy skin with honeycomb core, carbon skin with honeycomb core, carbon skin with a mid-plane damping layer, and a honeycomb sandwich beam with subsonic wave speed. Damping of beams and TL of panels were measured for all the above configurations and their results analyzed. The experimental results show a strong relationship between TL and loss factor but the specific association is material dependent.

1 INTRODUCTION

Lightweight sandwich panels are used in applications that require high strength and low mass, such as aircraft floors and aircraft components, naval structures, and transportation vehicles. Sandwich panels are composed of face sheets, a core, and adhesive interface layers. The high strength-to-mass ratio of these panels imparts superior noise transmission. However, in most applications, this is a negative attribute. By studying acoustic properties such as transmission loss (TL) and the damping loss factor of these materials, a better understanding of how to mitigate the sound transmitted without adding significant additional weight can be obtained. This knowledge can then be applied towards new designs of acoustically superior sandwich panels.

The TL plot of an orthotropic sandwich panel against frequency is typically bilinear, showing different dependencies below and above the critical coincidence frequency [1,2,3]. The critical coincidence frequency is the frequency at which the wave speeds in the panel match the speed of sound in air. Above this frequency there is an efficient exchange of energy with the surroundings and the TL is low. This value falls between 1000-2000 Hz for typical sandwich designs used as

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[a] Email address: ppeters@usc.edu
[b] Email address: shankar.rajaram@gmail.com
[c] Email address: nutt@usc.edu

airplane floors [4]. The first slope (below the coincidence frequency) is mostly stiffness and mass-controlled, while the second slope is controlled by coincidence frequency. This coincidence frequency is dependent on the core shear wave speed and the damping properties.

Damping of a structural panel has a direct effect on the TL properties of honeycomb sandwich panels at and above coincidence frequency [2, 5]. Kurtze and Watters demonstrated that damping increases transmission loss above coincidence frequency in structural panels [2]. SEA studies by Davis showed the same TL trend for damped honeycomb sandwich panels above coincidence frequency [5]. Li, et al obtained the loss factor for composite sandwich beams using the modal bandwidth method, which can be used to determine the damping for a single mode [6]. The Ross-Ungar-Kerwin model was one of the pioneering theories for studying damping in sandwich structures [7-10].

By characterizing the damping and TL of the panels, the relationship between the two properties can be established. Though such a relationship has been studied in the past, it was mostly confined to the structural characteristics of the test panels. In this paper, the loss factors of five different sandwich panels are measured and its relationship to measured TL is discussed. While earlier papers limit this discussion to mechanical features, the present study attempts to incorporate a materials' perspective as well.

2 EXPERIMENTAL SET-UP

2.1 Loss Factor

The classical method for measuring damping characteristics of materials is the Oberst method [11]. This method consists of a cantilever beam clamped at one end, allowing the beam to be excited without extra weight or damping. Problems with the effects of clamping conditions on the dynamic characteristics of cantilever type composite structures were investigated by Hwang *et al* [12], who attempted to increase the natural frequency and damping of structures.

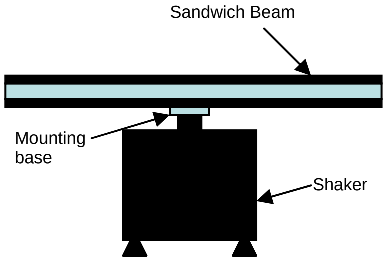

Figure 1: The experimental test apparatus included a shaker table with a center mounted composite beam.

These clamping condition limitations prompted the need for an excitation method that does not interfere with the data collection, so the free-free configuration was chosen. Equation (1) is the ratio of the dynamic response of the beam by the imposed motion, and equation (2) is the solution [11]. This equation mathematically illustrates that the free-free beam excitation method is similar to the traditional cantilever technique and sufficient to measure damping properties.

$$\begin{aligned} H(x,\omega) = {} & \frac{1}{2}\frac{\cosh(\beta L/2)+\cos(\beta L/2)}{1+\cosh(\beta L/2)\cos(\beta L/2)}[\cosh(\beta x)+\cos(\beta x)] \\ & +\frac{1}{2}\frac{\sinh(\beta L/2)-\sin(\beta L/2)}{1+\cosh(\beta L/2)\cos(\beta L/2)}[\sinh(\beta x)+\sin(\beta x)] \end{aligned} \qquad (1)$$

$$\omega_n = (\beta L)_n^2 \sqrt{\frac{Ek^2}{\rho L^4}} \text{ where } (\beta L)_n^2 = (\frac{2n-1}{\pi} * L)^2 \tag{2}$$

The experimental set-up model is shown in Figure 1. The beam is attached at the midpoint to an electro-dynamic shaker by a threaded rod. Lightweight aluminum fixtures with attached screws were mounted on the composite beam samples to affix the sample to the shaker table. The composite beam samples used in this experiment were approximately 914.4 mm long by 50.8 mm wide and 10.0 mm thick. The shaker table was calibrated using an aluminum reference beam with the same dimensions as the sandwich beam samples. The measured nodes of the reference beam were within 1% of the predicted nodes calculated using equation (2). Damping measurements were performed for five honeycomb sandwich beams, listed in Table 1. The sample names correspond to the panels used in Rajaram *et al* [1,13].

A white noise signal propelled an electro-dynamic shaker, which then excited the beam in the center through a line displacement. Both the tip motion and the center motion were measured using a lightweight accelerometer. The data was collected using commercial software (PULSE, B&K, Inc.). The software employs a dual FFT and modal bandwidth measurement in the analysis. All results were the average of three trials for each sample.

Table 1: List of beams tested and their properties. Panel designs described and TL results presented in reference [1].

Sample Name	Skin Type	Core Material	Surface Mass (kg/m^2)	Special Properties
G	Glass/Epoxy	Nomex®	2.24	
H	Glass/Epoxy	Kevlar®	2.46	
C	Carbon	Nomex®	1.90	
MP	Carbon	Nomex®	2.82	Mid-plane damping layer
SSS-2	Carbon	Nomex®	3.14	Subsonic core

2.2 Transmission Loss

TL measurements were performed at a small-scale facility [14] using an intensity technique described by ASTM 2249-02 [15]. The facility consisted of a reverberation source chamber and an anechoic receiver chamber with a window opening between them for the test panel. The source chamber was qualified between 315 Hz and 10 kHz following the standard procedures of ISO 3741 [16]. The irregular shaped reverberation chamber was comprised of 9 non-parallel, ceramic-tiled walls that enclosed a volume of 15 cubic meters. The reverberation time was ~1.5 seconds in most frequency bands of interests. The receiver chamber was a rectangular-shaped anechoic chamber enclosing a volume of 15 cubic meters. The interior of the chamber was lined with 0.15 m polyurethane foam wedges. The test sample, typically 1.067 m x 1.067 m, occupied a window between the two chambers. A metal frame clamped the sample along the edges on the receiver chamber side. The source and receiver chambers were mechanically isolated from one another through a rubber slab interlayer. The incident sound on the panel was measured by space-averaging the sound pressure level in the source chamber using a pressure microphone mounted on a rotating boom. The transmitted sound was measured using an intensity probe in the receiver chamber. The probe was used with a 50 mm spacer from 315 Hz to 1 kHz and a 12 mm spacer from 1.25-10 kHz. The probe was installed on a traverse system, and data was collected from an 11x11 grid on the sample surface. All results were the average of three trials with each spacer.

3 RESULTS & DISCUSSION

3.1 Loss Factors

There are four main measures of damping: damping ratio, loss factor, quality factor, and the imaginary part of complex modulus. In this paper, we use the loss factor, which is twice the damping ratio. It is a dimensionless number, because since the damping ratio is a dimensionless ratio that defines the amount of damping in a system. Similar to TL, larger values of loss factor indicate superior sound insulation.

The loss factor results for the five samples are shown in Figure 2. Sample MP has the highest loss factor values which are expected since the mid-plane dyad layer was inserted to improve damping of the structure. The increased damping effect only allowed loss factor data to be measured up to a frequency of 1885 Hz after which the sample failed to have nodes which could fit the 3dB measurement. This limitation is due to the highly damped material meeting with the empirical limitations of the current experimental set-up. However, based on extrapolation and the trends of the other samples, it is expected that the loss factor at frequencies beyond the coincidence frequency will not decrease.

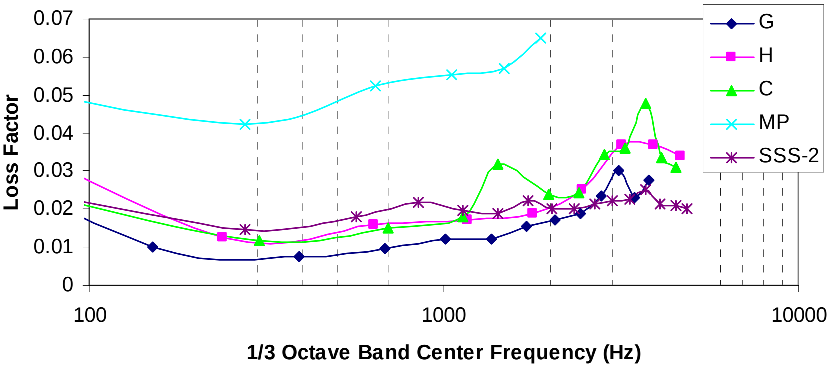

Figure 2: The loss factors for the five composite sandwich beam samples.

3.2 Transmission Loss

Transmission loss is the difference between the incident energy and the transmitted energy. The TL results for the five samples are shown in Figure 3. The results for samples G,C,H, and SSS-2 were previously published. Sample MP is new data, and it has the second highest TL of the five samples. For all panels except SSS-2 the slope of TL above coincidence frequency indicate that the values are well below mass law predictions. Mass law predictions give a measure of acoustical performance of a panel with a given mass. Since the test panels are of different masses, the mass effect on the TL performance was evaluated using the mass law deviation (MLD). The MLD was defined as the difference between the measured transmission loss and the mass law predicted transmission loss. A positive or higher value for MLD indicates superior acoustical performance and a negative or lower value for MLD indicates inferior acoustical performance [1].

There are different motions in a sandwich panel which are frequency dependent. Panel bending is the dominant motion at low frequencies, core shear motion at middle frequencies, and face sheet bending at high frequencies. A higher stiffness core leads to lower TL and greater noise radiation.

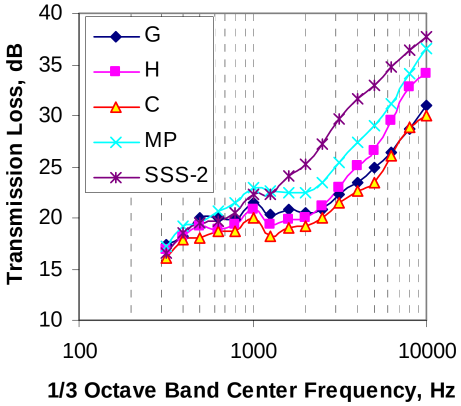

Figure 3: The transmission loss (TL) data for the five composite sandwich beam samples.

3.3 Mid-Plane Damping

One prototype sandwich structure featured a viscoelastic sheet inserted at the mid-plane of the honeycomb core (sample MP). The effect of this mid-plane damping layer was assessed by comparing the damping properties with a conventional sandwich beam featuring identical skin and core materials (sample C). The comparison between samples C and MP is shown in Figures 4a and 4b, which show loss factor and MLD as functions of 1/3 octave band center frequency. The mid-plane damping layer greatly improves the loss factor of the beam. An extrapolation of the loss factor for MP panel shows 300% improvement in damping over its conventional counterpart. This is corroborated in the TL results for MP panel. The MLD plot in Figure 4b shows that TL gradually improves beyond coincidence frequency for MP panel and the enhancement is up to 3 dB. The TL results are consistent with previous works [1, 2, 4] which found that damping improves TL beyond coincidence frequency. The MLD plot negates the difference in masses between samples C and MP, thus showing that the TL difference is not a function of mass but rather a function of the mid-plane damping layer.

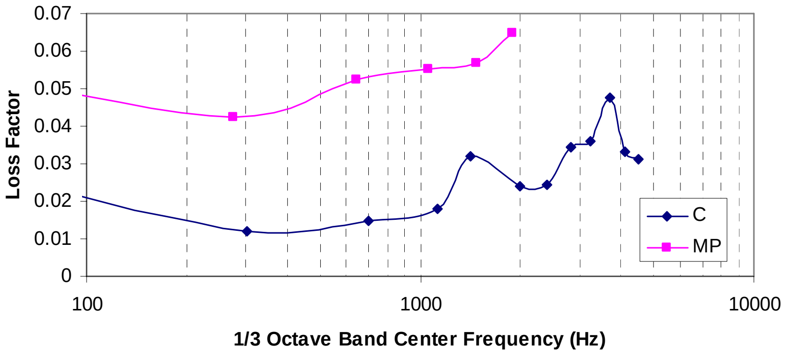

Figure 4a: The loss factor comparison between samples C and MP.

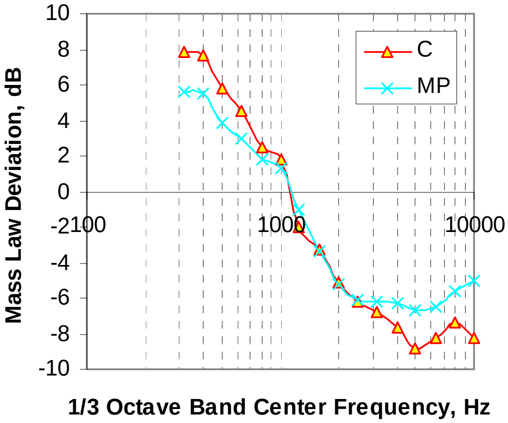

Figure 4b: The MLD comparison between samples C and MP.

3.4 Skin Damping

Comparison of samples G and C reveals the effect of skin material on damping and TL, as shown in Figures 5a and 5b. Sample G has glass/epoxy face sheets, while sample C features carbon face sheets. Sample C has a consistently higher loss factor than panel G with glass-epoxy skin over the entire frequency range. The higher loss factor for the carbon-phenolic skin of sample C correlates to its higher TL at high frequencies as indicated by MLD trends (Figure 5b). For frequencies below the coincidence frequency there does not appear to be a strong correlation between loss factor and TL for samples with different skins, as this frequency regime would be expected to be mass dominated for sandwich panels [1, 2, 4].

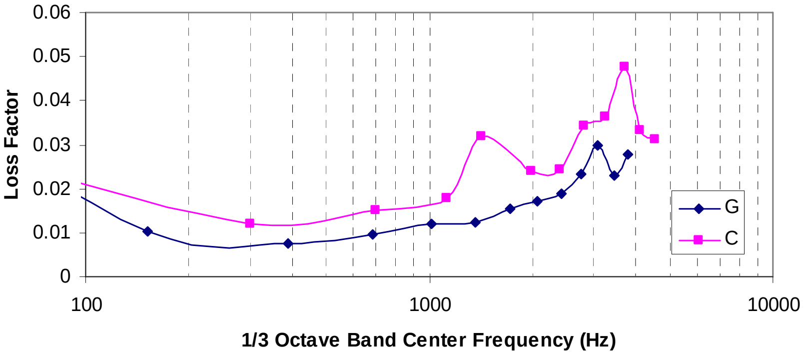

Figure 5a: The loss factor comparison between samples G and C.

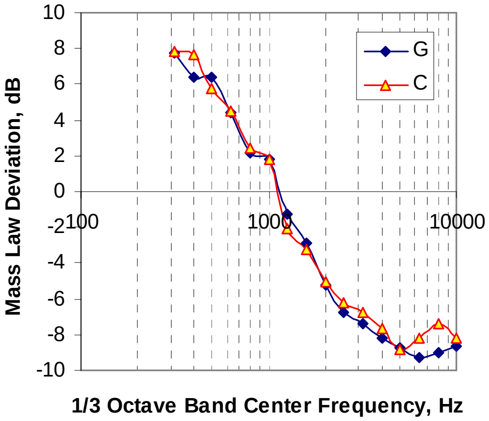

Figure 5b: The MLD comparison between samples G and C.

3.5 Wave Speed

Sample SSS-2 featured a thinner core and thicker face sheets than conventional sandwich designs. This unconventional design was selected to achieve a subsonic shear wave speed of approximately two-thirds the speed of sound. A subsonic core such as the core for sample SSS-2 delays the onset of the coincidence frequency and should thus, in principle, impart superior acoustical performance. Indeed, this is the case, as shown in Figure 6b, which indicates that the TL of sample SSS-2 is 5-7 dB greater than that of sample C for frequencies above the coincidence frequency of ~1500 Hz [1]. However, the loss factor of SSS-2 above the coincidence frequency was consistently lower than the reference panel, as shown in Figure 6a. This shows that a subsonic design is more effective than a damped design at high frequencies, which is further proved by the MLD comparison of SSS-2 and the highly damped MP in Figure 6c.

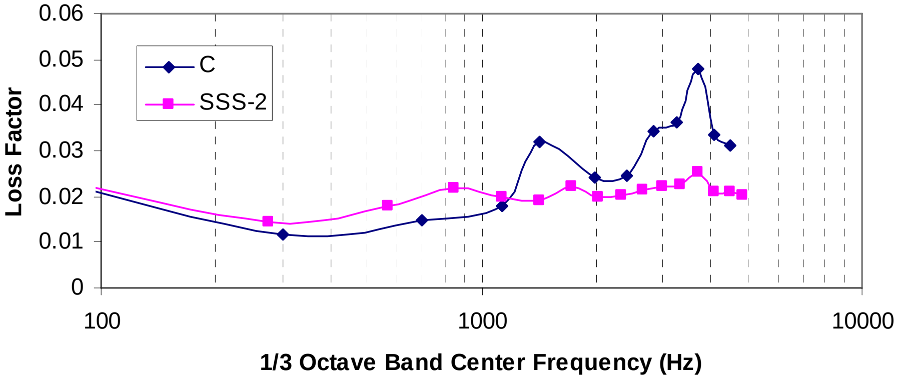

Figure 6a: The loss factor comparison between samples C and SSS-2.

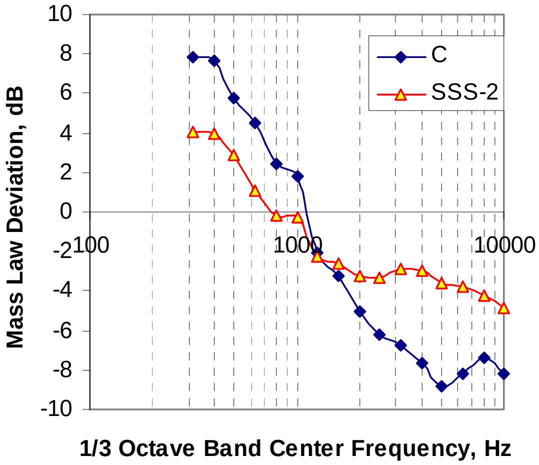

Figure 6b: The MLD comparison between samples C and SSS-2.

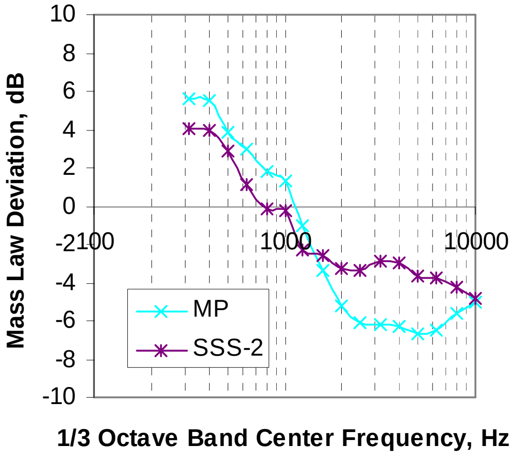

Figure 6c: The MLD comparison between samples MP and SSS-2.

3.6 Core Material

Sample H featured a Kevlar® core, which was ostensibly stiffer and stronger than conventional core materials. Somewhat unexpectedly, this core resulted in values of loss factor and TL that were greater than those of reference sample G, which featured a conventional (Nomex®) core (Figures 7a and 7b). While the TL difference between panels G and H is negligible, the MLD shows that for panels of equivalent masses Kevlar would have 2-3 dB higher TL at higher frequencies (Figure 7b). The difference between the loss factors reveals the possible reason (Figure 7a). Panel H has a higher loss factor than panel G particularly beyond 3 KHz. The larger variance in the loss factors compared to the TL values may derive from the different inherent stiffness of the cores for Nomex® and Kevlar®. Kevlar® derives its inherent higher stiffness from the inflexible "para-aramid" structure compared to the more flexible "meta-aramid" structure of Nomex®. In a sandwich design, the stiffer Kevlar® cores are believed to impart lower modal density to the panel beyond coincidence frequency compared to less stiff Nomex® cores of similar density [4,13]. This lower modal density beyond coincidence frequency could possibly dampen the panel with high stiffness core. This will be investigated in the future.

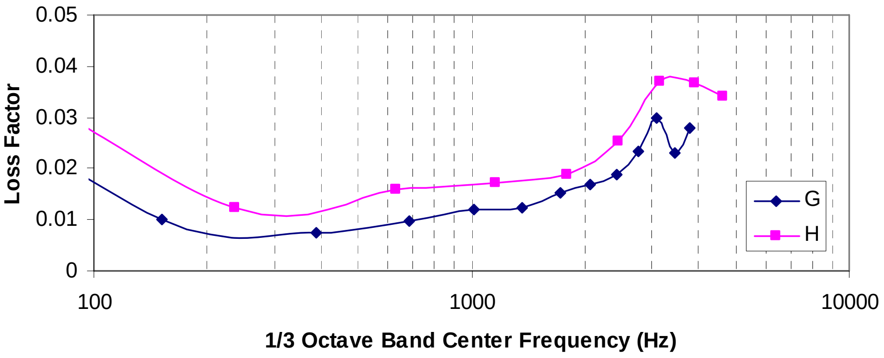

Figure 7a: The loss factor comparison between samples G and H.

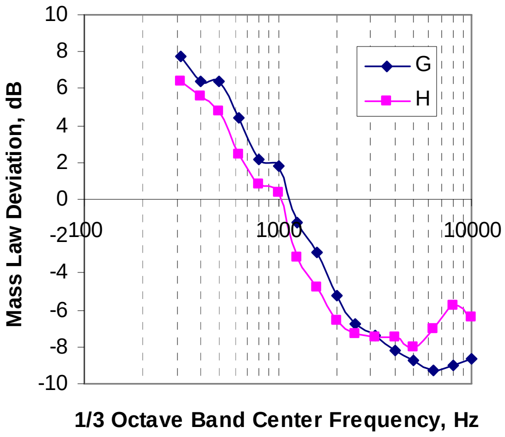

Figure 7b: The MLD comparison between samples G and H.

4 CONCLUSIONS

A suite of prototype sandwich structures was fabricated and tested in order to assess the effects of simple materials and structural parameters on the acoustic properties of loss factors and TL values. (1) Sandwich beams with a mid-plane damping layer exhibited greater loss factor and considerable TL improvement compared to the control reference sample without a mid-plane damping layer. (2) There was a positive correlation between TL and loss factor for skin materials above the coincidence frequency. Below the coincidence frequency there does not appear to be a relationship between TL and loss factor. (3) There is an inverse relationship between TL and loss factor for subsonic panels. Both Kevlar® and Nomex® cores show that higher loss factor values correlate to higher TL values. Overall, the experimental results show that there is a strong relationship between TL and loss factor and the specific association depends on the material.

The subsonic panel showed superior performance for TL but only average performance for loss factors. Conversely, the mid-plane damping panel had consistently high loss factor values but the TL was not as prominent. Combining the panels by inserting a mid-plane damping layer into the subsonic core could achieve both high TL and loss factor values. However, merging these designs would sacrifice the overall mechanical stability of the panel leaving opportunity for optimizing both strength and noise mitigation properties. There are applications such as fuselage linings where lightweight acoustic insulation is called for without strong mechanical properties. Furthermore, the results of the Kevlar® core sample show that TL and loss factor values can be enhanced simultaneously demonstrating that enhanced acoustics can be achieved without sacrificing mechanical performance. The remaining panels (samples G & C) represent conventional designs and showed inferior acoustic insulation. This behavior highlights the opportunity for significant improvements in vibroacoustics.

The combined effect of subsonic wavespeed and high damping on sandwich panels will be studied in the future. The relationship between damping, modal density and core stiffness will also be explored in future to get a better understanding of sandwich panel designs.

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6 REFERENCES

[1] Shankar Rajaram, Tongan Wang, and Steven Nutt, “Sound transmission loss of honeycomb sandwich panels,” Journal of Noise Control Engineering, **54**(2), 46-55 (2005).

[2] G. Kurtze and B.G. Watters, “New wall design for high transmission loss or high damping,” Journal of the Acoustical Society of America, **31**(6), 739-748 (1959).

[3] Frank Fahy, *Sound and Structural Vibration Radiation, Transmission and Response*, (Academic Press, New York, 1985).

[4] A. Paolozzi and I. Peroni, “Response of aerospace sandwich panels to launch acoustic environment,” Journal of Sound and Vibration, **196**(1), 1-18 (1996).

[5] E.B. Davis, “Designing honeycomb panels for noise control”, Journal of the American Institute of Aeronautics and Astronautics, **99**, (1917).

[6] Zhuang Li, and Malcolm Crocker, “Effects of thickness and delamination on the damping in honeycomb-foam sandwich beams,” Journal of Sound and Vibration, **294**, 473-485 (2006).

[7] E.M. Kerwin, “Damping of flexural waves by a constrained viscoelastic layer,” Journal of the Acoustical Society of America, **31**(7), 952-962 (1959).

[8] D. Ross, E.E. Ungar, and E.M. Kerwin, “Damping of plate flexural vibrations by means of viscoelastic laminate,” *Structural Damping* (ASME, New Jersey, 1959) pp. 49-88.

[9] E.E. Ungar, “Loss factors of viscoelastically damped beam structures,” Journal of the Acoustical Society of America, **34**(8), 1082-1089 (1962).

[10] E.E. Ungar and E.M. Kerwin, “Loss factors of viscoelastic systems in terms of energy concepts,” Journal of the Acoustical Society of America, **34**(7), 954-958 (1962).

[11] Jean-Luc Wojtowicki, Luc Jaouen, and Raymond Panneton, “A new approach for the measurement of damping properties of materials using the oberst beam,” Review of Scientific Instruments, **75**(8), 2569-2574 (2004).

[12] H.Y. Hwang, G.L. Hak, and D.G. Lee, “Clamping effects on the dynamic characteristics of composite machine tool structures,” Composite Structures, **66**, 399-407 (2004).

[13] Shankar Rajaram, Portia Peters, Tongan Wang and Steven Nutt, “Comparison of acoustical performance of Kevlar vs Nomex honeycomb cores in sandwich panels,” *Proceedings of the Society for the Advancement of Material and Process Engineering 2006.*

[14] Shankar Rajaram, Tongan Wang, Yunung Chui, Garrett Allen, Puliyampati Thirunarayanan, and Steven Nutt, “Measurement and prediction of sound transmission loss for airplane floors,” American Institute of Aeronautics and Astronautics, Jan 10-13, 2005, Reno, Nevada.

[15] *Acoustics-ASTM standard method for laboratory measurement of airborne sound transmission loss of building partitions and elements using sound intensity,* American Standard ASTM E 2249 02:2002 (American Standards for Testing and Materials, West Conshohocken, USA, 2002).

[16] *Acoustics – Determination of sound power levels of noise sources – Precision methods for broad band sources in reverberation,* International Standard ISO 3741-88 (E): 1988 (International Organization for Standardization, Geneva, Switzerland, 1988).