Loss factors of honeycomb sandwich structures: an experimental approach

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Abstract: Loss factor measurements were performed on different sandwich panels to determine the effects of different skin and core materials on the acoustical properties of sandwich structures. The study included the following panel/beam designs: glass-epoxy skin with a meta-aramid (m-aramid) core, glass-epoxy skin with a para-aramid core, carbon skin with a m-aramid core, carbon skin with a m-aramid core with a mid-plane damping layer, and a carbon skin with a m-aramid core of subsonic wave speed. The loss factors were calculated for all the above configurations. Analysis of the results revealed that inserting a viscoelastic material in the mid-plane of the core resulted in the highest loss factor. Secondly, panels constructed with carbon fiber skins exhibited larger loss factors than glass fiber skins. Panels designed to achieve a subsonic wave speed did not show a significant increase in loss factor above the coincidence frequency. The para-aramid core had a larger loss factor value than the meta-aramid core. The data indicated that judicious choice of sandwich materials can lead to reductions in noise transmission.

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1. Introduction

Lightweight sandwich panels are used in applications that require high specific stiffness and strength, such as aircraft flooring, naval vessels, and transportation vehicles. However, the use of

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sandwich panels frequently results in undesirable levels of noise transmission. As a result, continual efforts are made to improve acoustical performance without compromising mechanical performance of panel systems. While there are multiple approaches to mitigating noise, damped panels transmit less noise near the critical frequency [1]. These traits can be exploited to reduce the noise generated from a vibrating panel. Ultimately, the reduction of noise also depends on the noise source. However, by exploring noise control measures that increase the loss factor of sandwich structures through passive damping controls, it should be possible to design and produce quieter panels without sacrificing strength, thereby reducing noise levels in passenger cabins. This correlation between a higher damping factor and decreased noise generation has been shown in mechanical gears made from composite materials [2] and this current study is aiming to show this correlation for honeycomb sandwich panels.

Understanding the acoustic behavior of the component materials of sandwich panels can lead to approaches that mitigate the sound transmitted without adding significant additional weight. Knowledge of the loss factor of component materials can be used to develop designs of new sandwich panels with superior acoustic properties.

Since the late 1950s, vibro-acoustic properties of sandwich structures have been widely studied. For example, the Ross-Ungar-Kerwin model constituted a pioneering theory for studying damping in sandwich structures [3-6]. Kerwin also determined the effectiveness of viscoelastic material on damping, [3] and expressions for the loss factors of structures with two or three components were presented by Ungar. Ungar later reexamined and refined his previous loss factor equations to account for the vibration energy of highly damped systems [5, 6]. Finally, Mead and Markus furthered the study of damping by formulating an approximate method for determining both loss factors and resonant frequencies for honeycomb structures [7] and formulating a 6th-order

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differential equation of motion derived in terms of transverse displacement for a sandwich beam with a viscoelastic core [8].

More recent advances include experiments to determine loss factors for honeycomb sandwich panels with composite face sheets [9], combining honeycomb and viscoelastic material to affect damping and vibration control [10], and determining the effects of thickness and delamination on damping in honeycomb [11]. The effects of different honeycomb core geometries have also been explored for possible noise control and structural improvements [12, 13]. Li, et al determined the loss factor for composite sandwich beams using the modal bandwidth method, which can be used to determine the damping for a single mode, [14] while Nilsson conducted a study of wave propagation, loss factors, and transmission loss for sandwich plates. [15] Finally, He and Rao presented an analytical model for the transverse and longitudinal vibrations in sandwich beams [16], and Shi et al were able to use inverse methods to determine material parameters in beams [17].

However, these studies, like most others, were based on theoretical models or experiments that focused solely on the mechanics of the structure. Shorter18 performed an analytical study on wave propagation and damping for honeycomb sandwich structures using viscoelastic laminates, and Ghinet, Atalia, and Osman [19] studied the effects of materials on the transmission loss of curved honeycomb sandwich panels. However, few attempts have been made to quantify the impact of material choice on the loss factors of honeycomb sandwich structures. In the present work, we investigate the influence of material choice on the damping of honeycomb sandwich panels through experimental data. As an extension of this experiment, one of the samples chosen uses an unconventional method of increasing the loss factor. A panel was designed and produced with a damping layer in the mid-plane rather than placing the damping layer on the skin-core interface. The

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damping layer was placed in the center of the core as a proof of concept experiment demonstrating that exposing the damping material to maximum shear force would result in maximum damping.

2. Experimental Set-up

In this study, data was acquired from Oberst beam [20] measurements (for lightly damped materials) and from suspended beam tests (for highly damped materials). Loss factors were subsequently calculated using these data.

2.1 Oberst Beam Method

The classical method for measuring damping characteristics of materials is the Oberst method which involves exciting a cantilever beam clamped at one end. Problems arise from the effects of clamping conditions on the dynamic characteristics of cantilever type composite structures (Hwang et al [21]). These clamping condition limitations prompted us to seek an excitation method less sensitive to boundary conditions, which led to the adoption of a free-free configuration illustrated in Fig. 1. Eqn (1) is the ratio of the dynamic response of the free-free beam by the imposed motion, Eqn (2) is the definition of the beta term, and Eqn (3) is the definition of the omega term used to predict the modes. These equations were derived using the compact model of the beam equation [20], and illustrate that the free-free beam excitation method is similar to the traditional cantilever technique. In fact, a cantilever beam has the same dynamical behavior as a free-free beam with twice the length, excited in the center, Thus, the equations are suitable for measurement of structural damping properties. For the free-free beam, only the even modes will be excited, although the slope and relative displacement

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to the imposed motion are void at the center. Therefore, the modal behavior will be similar to a clamped beam [20].

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

$$\beta^4 = \frac{\rho A \omega^2}{EI} \tag{2}$$

$$\omega_n = (\beta L)_n^2 \sqrt{\frac{Ek^2}{\rho L^4}} \quad \text{where} \quad (\beta L)_n^2 = (\frac{2n-1}{\pi} L)^2 \tag{3}$$

In the above equations, H is the ratio of the dynamic response of the beam divided by the imposed motion, x is the position, ω is the excitation frequency, L is the length of the beam, ρ is the mass density of the beam, A is the cross-section area, E is the elastic modulus, I is the second moment of area of the beam, k is the wave number, and n is mode number.

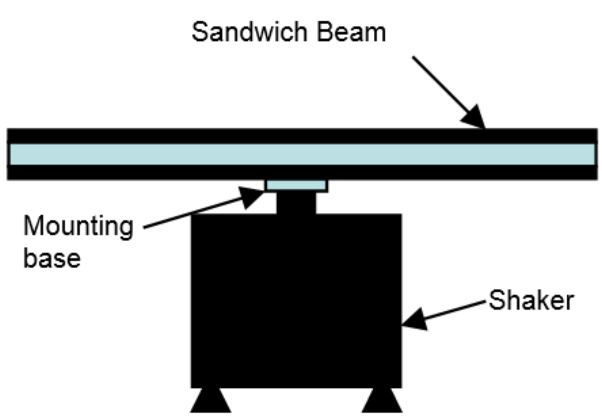

***Figure 1**: The experimental test apparatus included a shaker table with a center mounted composite beam.*

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The experimental set-up is shown in Fig.1. Lightweight aluminum mounting bobbins were bonded to the midpoint of the composite beam samples. The beam was then attached to an electro-dynamic shaker by a threaded rod. The composite beam samples used in this experiment were approximately 900 mm × 50 mm × 10 mm. The apparatus was first tested using an aluminum reference beam with these same dimensions. The measured modes of the reference beam were within 1% of the predicted modes calculated using equation (3). Damping measurements were performed for five honeycomb sandwich beams, listed in Table 1. The layup for each of the samples was 2 plies unidirectional, 0/90, and the volume fraction of the composite material of the skin for all samples was ~50%. The sample notation corresponds to the panels used by Rajaram et al [22, 23]. The samples were tested at room temperature (22°C) to simulate aircraft interiors, the primary application for these materials.

Table 1: *List of beams tested and their properties. Panel designs described and TL results presented in reference[19].*

Sample Name	Skin Type	Core Material	Special Properties	Layer Thickness skin/core/skin (mm)	Young's Modulus (Nm^{-2})
G	Glass/Epoxy	Nomex®		0.5/9/0.5	20x10^9
H	Glass/Epoxy	Kevlar®		0.5/9/0.5	20x10^9
C	Carbon	Nomex®		0.5/9/0.5	100x10^9
MP	Carbon	Nomex®	Mid-plane damping layer	0.5/9/0.5 Damping layer = 0.5	100x10^9
SSS-2	Carbon	Nomex®	Subsonic core	0.5/9/0.5	100x10^9

A white noise signal was used to drive the shaker, which then excited the midpoint of the beam through a line displacement. Both the tip motion and the center motion were measured using lightweight (0.65 gram) accelerometers placed at the midpoint and one end of the beam. Data was collected using commercial software (PULSE, B&K, Inc.) using a 10 kHz frequency span and a resolution of 1600 lines. The software employs a dual FFT and modal bandwidth measurement in the analysis. The loss factor values were obtained using commercial software (PULSE, B&K) which used the half bandwidth method for calculations. All results were the average of three trials for each

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sample, and each trial varied by less than 1.5%. This method was used for all samples with the exception of Sample MP, for which the suspended beam method was used.

2.2 Suspended Beam Method

In some cases (particularly those with high-frequency/high damping), resonant modes were ill-defined when using the Oberst beam method. For such cases, an alternate testing apparatus was designed and built, as shown in Fig. 2. This method generally yields accurate results for $\eta k^L l > 10$ where η is the loss factor, k^L is the real part of the wave number, and l is the length of the beam24. The $\eta k^L l$ value for Sample MP reached this minimum threshold because the end of the beam had good sand termination. For test articles with lower loss factors, the half-value bandwidth method used in the Oberst beam method yields accurate results (when $\eta k^L l$ is less than approximately two). The sample sizes remained the same as the Oberst beam method.

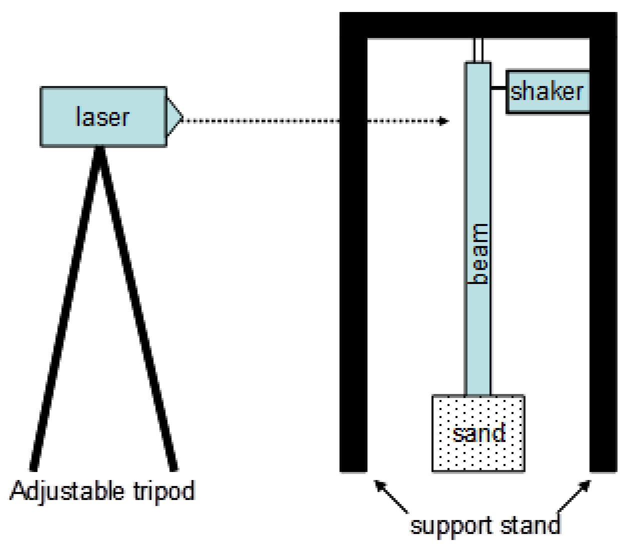

Figure 2: *A modified version of the suspended beam set-up for highly damped samples.*

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The suspended beam method was used for the sandwich beam with a viscoelastic layer inserted in the middle (Sample MP). The suspended beam apparatus (Fig. 2) was built to capture the velocities at predetermined positions based on Eqn (4)24.

$$D' = \frac{10\log\left[\left|v_o^2\right| / \left|v_1^2\right|\right]}{(x_1 - x_o)} \tag{4}$$

$$\eta = \frac{D'\lambda}{13.6} \tag{5}$$

In these equations η is the loss factor , D' is the reduction in vibration level per unit length, λ is the bending wavelength, v_o and v_1 are the initial and final velocities respectively, and xo and x1 are the initial and final positions respectively. Vibration was measured using a non-contact laser vibrometer. The top of the beam was loosely connected to the supporting frame, and the end of the beam was embedded in sand to help minimize the reflection of waves. The surface velocity was measured at one centimeter increments along the beam. These measurements were then used in equations (4)-(5) to calculate the loss factor. All results were the average of three trials for each sample, with each trial within 3% of the other results.

The two methods used in this study differ in the fact that the Oberst beam method provided a direct measurement of the loss factors, while the suspended beam method relied on distances and velocities to calculate the loss factor. There was not a direct correlation between the two methods because the suspended beam method was constructed specifically for highly damped samples, which could not be measured using the Oberst beam method. Therefore, a reference sample for both methods was not feasible, since a damping treatment would be required in order to test the sample using the suspended beam method.

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3. Results and discussion

The loss factor values for the five samples are shown in Fig. 3. Four of the samples showed similar loss factors of 0.01 – 0.03, which were large relative to Aluminum (0.003). The sample beams also showed a weak dependence on frequency, with mild perturbations around 1500 Hz, near the coincidence frequency, which is the frequency at which the wavelength of the panel equaled the wavelength of the speed of sound in air. The coincidence frequency values were taken from a previous study on transmission loss, in which identical materials were used [22]. Sample MP, which featured a mid-plane damping layer, yielded the largest loss factor values.

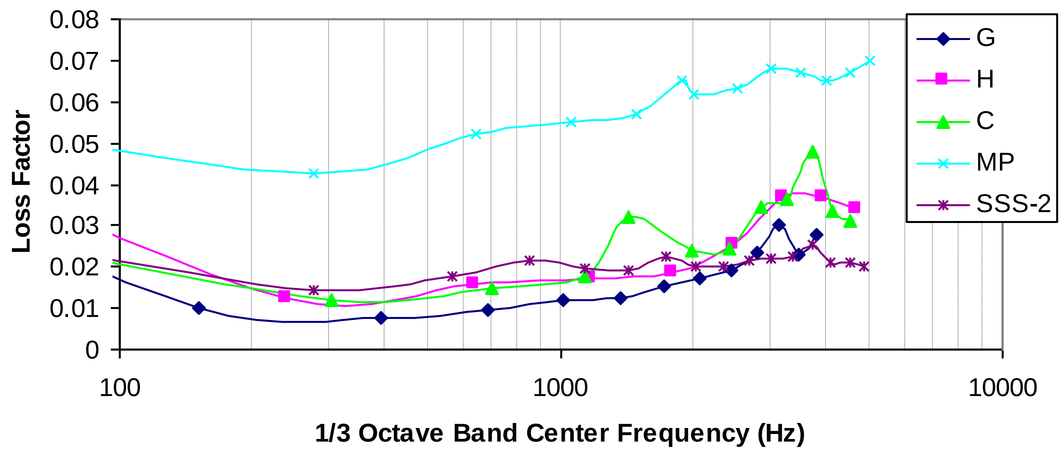

Figure 3: *The loss factors for the five composite sandwich beam samples.*

3.1 Mid-plane damping

Sample MP featured a viscoelastic sheet inserted at the mid-plane of the honeycomb core to increase damping. The panel was constructed by slicing a panel identical to sample C through the mid-plane, then bonding the damping sheet between the halves using brush-on epoxy. The addition of the viscoelastic sheet added 0.5 mm to the thickness of the sample. The mid-plane coincides with the

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maximum shear force in the beam when a bending load is applied. The effect of the 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 Fig. 4, a plot of the loss factor as a function of frequency. The mid-plane damping layer increased the loss factor of the beam by up to 233%. Fig. 4 also shows that the damping curve for sample MP is similarly parallel to the control panel cover the entire frequency spectrum, demonstrating that the damping from the mid-plane viscoelastic layer is independent of frequency. This behavior indicates that the operative damping mechanism is unlike constrained layer damping (CLD), in which the usual frequency dependent behavior of the viscoelastic core [25] would be apparent. However, the data shows that the loss factor is independent of frequency, thus indicating free layer damping.

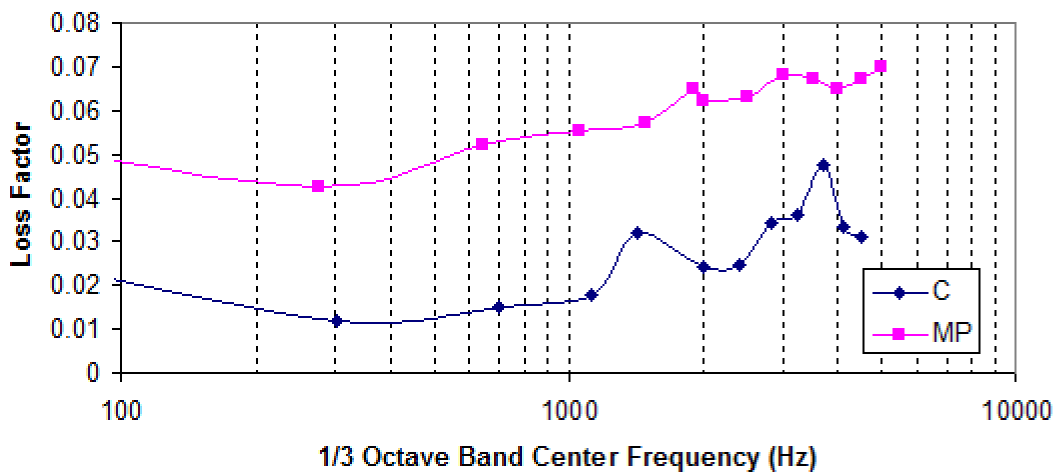

Figure 4: *The loss factor comparison between samples C and MP.*

The moments for shear stress are zero at the surface of the skin and maximum at the interface of the skin and core. The shear stress is constant through the thickness of the core, which carries the entire shear load when a bending moment is applied26. Because the core carries the entire shear load, it is less likely that the core will constrain the viscoelastic layer. Thus, the data supports the

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assertion that the damping effect shown by the MP sample is free layer damping. The enhancement in damping could also be attributed to the beam acting as two smaller sandwich structures sharing the mid-plane layer as a face sheet. While the damping is significantly increased with Sample MP, the bending stiffness was markedly reduced because of the lower shear stiffness of the core [27].

3.2 Skin damping

Comparison of samples G and C revealed the effect of skin material on damping, as shown in Fig. 5. Sample G featured glass/epoxy face sheets, while sample C featured carbon face sheets. Sample C showed a consistently higher loss factor than panel G over the entire frequency range. However, it is commonly accepted that glass fibers have higher damping than carbon fibers [28]. The apparently contradictory findings shown here can be attributed to two possible causes. First, the fibers used in the construction of the panel as well as the fiber treatment and type of adhesives used in the sandwich panel can affect acoustic properties [29]. In the case of transmission loss, details such as these are minor relative to the overwhelming structural effects. However, in loss factor measurements, material properties play a larger role, because the dynamic behavior of the sandwich structure is governed by the individual components of the panel at mid-to-high frequencies [30]. Second, the material structure of the carbon skin could be responsible for the higher loss factor than the glass-epoxy skin. The turbostratic structure of carbon fibers features well-aligned basal planes which are only weakly bonded, allowing for interplanar sliding. Similar enhanced damping is observed in other C-based structures, such as carbon nanotube-reinforced epoxies, and certain cast irons containing graphitic flakes.

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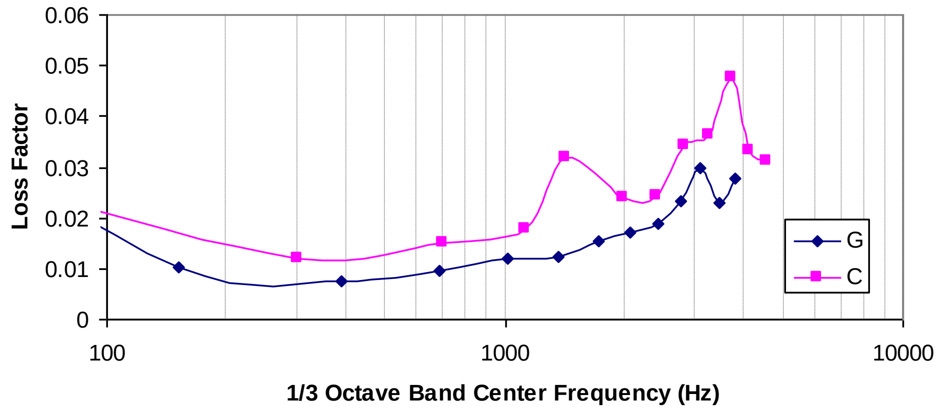

Figure 5: *The loss factor comparison between samples G and C.*

3.3 Wave speed

Sample SSS-2 featured a thinner core and thicker face sheets than conventional sandwich designs, thus achieving a subsonic shear wave speed of approximately two-thirds the speed of sound. Although this is a mechanical modification rather than a material modification, the results provide insight into the effect of wave speed on loss factors. This insight can be used to design materials with varying wave speeds to reduce noise. Testing the subsonic panel for loss factor was an extension of previous work [22, 23] that showed subsonic cores can increase the transmission loss of a panel by several decibels. A subsonic core increases the coincidence frequency and should thus, in principle, impart superior acoustical performance. However, the subsonic panel exhibited a lower loss factor than the control panel, as shown in Fig. 6. This finding can be understood by noting that wave speed and damping are distinct parameters and do not necessarily follow similar trends.

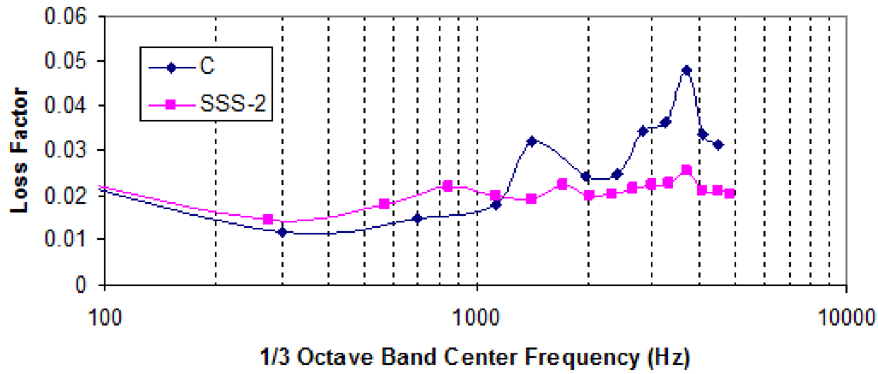

Figure 6: *The loss factor comparison between samples C and SSS-2.*

3.4 Core material

Sample H featured a para-aramid (Kevlar@) core, which was stiffer and stronger than conventional core materials. This panel yielded loss factor values that were greater than those of reference sample G, which featured a conventional meta-aramid (Nomex@) core (Fig. 7) particularly beyond 3 KHz. The difference in the loss factors stems from the different inherent stiffness of the meta- and para-aramid cores. The higher stiffness of Sample H arises from the use of para-aramid fiber paper, as opposed to meta-aramid fiber paper. In the sandwich structure, the stiffer para-aramid cores impart a lower modal density to the panel beyond the coincidence frequency, compared to meta-aramid cores of similar density [22, 31]. This lower modal density enhances the panel damping [22]. The magnitude of the increase is comparable to the difference in mechanical properties of the constituent materials since the para-aramid structure can be molecularly aligned and gives high strength as opposed to the meta-aramid structure which cannot be molecularly aligned and has poor strength.

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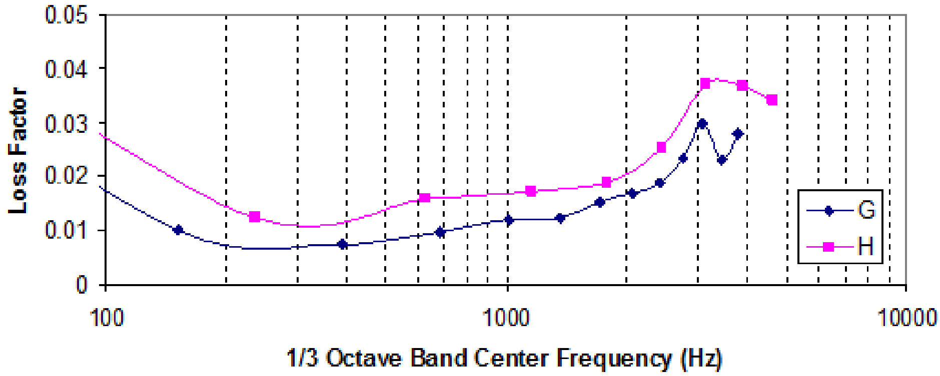

Figure 7: *The loss factor comparison between samples G and H.*

4. Conclusions

Prototype sandwich structures were fabricated and tested to assess the effects of simple materials and structural parameters on the loss factor. Sandwich beams that featured a mid-plane damping layer exhibited substantially greater loss factors compared to the control reference sample, while panels with carbon fiber skins had the greatest loss factor. Structures designed for subsonic shear wave speeds did not exhibit increased loss factors. Of the panels with aramid cores, the panel with the para-aramid core consistently showed the greatest loss factor.

From a materials perspective, noise control of sandwich panels can be strongly influenced by both additive materials, such as the mid-plane viscoelastic material, and by the different skin materials. While the mid-plane damping layer and the para-aramid core increased the loss factor, the subsonic core did not. However, in past work the subsonic core resulted in substantially increased transmission loss. Inserting the mid-plane layer demonstrated the potential of judiciously placed damping materials. This approach could be extended to applications such as automotive frames and panels and airplane fuselages. While the focus of the present work was on frequencies above the

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coincidence frequency, increased damping at lower frequencies is expected to reduce transmission loss in this mass-controlled frequency range.

Overall, noise control requires a comprehensive perspective and solutions that encompass both transmission loss and damping. The present study confirms that the acoustical performance of sandwich structures is influenced by a variety of components that are independently linked. Weight penalties must also be considered when designing acoustically superior panels as a component that improves the loss factor could decrease the mechanical performance of the sandwich structure. Investigating materials and their combination of properties can also be manipulated to produce the optimal material for aerospace applications.

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