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SOUND TRANSMISSON LOSS OF FOAM-FILLED HONEYCOMB-CORE SANDWICH PANELS

Ran Zhou, Malcolm Crocker*, Zhaohui Yu and Zhuang Li

Department of Mechanical Engineering, Auburn University
201 Ross Hall, Auburn University, AL 36849-5341, USA
zhouran@auburn.edu(e-mail address of lead author)

Abstract
Composite panel structures are often used in industry, automobile, ships and aircraft, because of the advantages they offer of high strength to weight ratios. However, the acoustical properties of these light and stiff structures can be less desirable than equivalent metal structures. This results in high interior noise levels. A number of researchers have studied the acoustical properties of honeycomb-core and foam-core sandwich panels. Not much work, however, has been carried out on foam-filled, honeycomb-core (FFHC) composite panels. In this paper, measurements of the sound transmission loss of foam-filled, honeycomb-core sandwich panels with different configurations and thicknesses are presented. Some theoretical approaches to predict sound transmission loss in sandwich structures are reviewed. The effect of the thickness of core and face sheets is investigated. The measured results are compared with those predicted by statistical energy analysis (SEA) approach.

INTRODUCTION

The sound transmission loss (TL) is an important consideration in the design of many structures, building walls and floors, ship hulls, aircraft sidewalls and walls of machinery enclosures. Over the years, two main methods have been used to predict the TL in sandwich structures. One is a classical analysis, based on wave impedance; the other is statistical energy analysis, based on energy flow relationships.

Kurtze and Watters [1] derived a simple expression to describe the wave impedance of a sandwich panel from an equivalent circuit analogy. The core acts as a spacer that does not transmit shear, and the skins respond as elementary bent plates. Ford *et al*[2] presented a free vibration analysis of sandwich panels with rigid polyurethane foam cores, and noted a relation between dips in experimental transmission loss curves and the resonance frequencies in both bending and dilatation motions. Smolenski and Krokosky[3] corrected the errors in Ref.[2], and also explained that the coincidence effect differs from bending coincidence .

The first effort at calculating the transmission loss of sandwich panels is attributed to Dym and Lang [4-5]. They presented a formula for transmission loss in terms of wave impedances and incidence angle. They also pointed out that for symmetric configurations, the symmetric and antisymmetric components are uncoupled naturally. Moore and Lyon [6] presented an analytical model for the random incidence TL for symmetrical panels with isotropic and orthotropic cores materials.

Statistical energy analysis (SEA) was developed by Lyon[7-8], Scharton [8] and Ungar[9] in the early 60's. Since then, there have been many papers written on the subject. Some of them clarified theoretical assumptions and calculated or evaluated values for necessary parameters. Maidanik [10] formulated the radiation resistance of a flat panel in a reverberant acoustic field. Crocker and Price [11] developed expressions for the transmission loss and the radiation resistance of a flat panel due to mechanical excitation. Cremer *et al*[12] developed expressions for transmission coefficients at various junctions.

Due to the demands for higher load capacity for aircraft, reduced fuel consumption for passenger cars, the number of applications for composite sandwich structures consisting of stiff face sheets and a lightweight core is steadily increasing. Compared to plain honeycomb cores, foam-filled honeycomb cores have a much larger bonding surface and improve the impact damage resistance capacity of composite panels [13]. In this particular study, the face sheets were made of graphite plain weave fabric and the cores were made of paper honeycomb filled with polyurethane (PUR) foam.

TL MODELS

Classical analysis is the most straightforward way to calculate transmission loss. It can be used to predict the transmission loss at any incidence angle, but it needs prior knowledge of the dynamics of the panels. Statistical energy analysis considers a steady state situation. The power balance of the subsystems forms the basis for SEA calculations. SEA is most suitable for broadband excitation over a bandwidth encompassing many uncoupled-system natural frequencies. SEA does not account for symmetric vibration modes. So it does not predict coincidence caused by skin-core-skin vibration.

Classical Approach

For a sandwich panel, the displacements of the face sheets are shown in Fig.1. The transverse and in-plane motions of the face sheets can be decomposed into antisymmetric and symmetric terms as follows:

$$w_a = (w_1 + w_2)/2\,,\ w_s = (w_2 - w_1)/2\,,$$
$$u_a = (u_1 - u_2)/2\,,\ u_s = (u_1 + u_2)/2\,. \qquad (1)$$

The actual pressure fields acting on the top and bottom face sheets are decomposed into asymmetric and symmetric pressure terms as follows:

$$p_a = (p_1 - p_2) \ , \ p_s = -(p_1 + p_2)\,. \tag{2}$$

For symmetric configurations, the wave impedances are[4],

$$\begin{bmatrix} p_a \\ p_s \end{bmatrix} = \begin{bmatrix} Z_a & 0 \\ 0 & Z_s \end{bmatrix} \begin{bmatrix} i\omega w_a \\ i\omega w_s \end{bmatrix}. \tag{3}$$

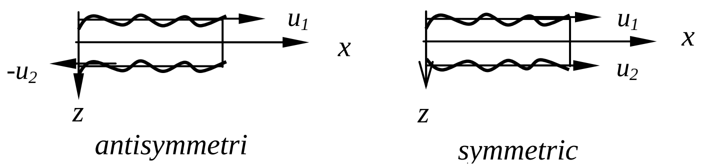

Fig. 1. Symmetric and antisymmetric skin displacements.

Assume that the incident sound pressure p_i is incident on skin 1, the reflected pressure p_r is equal in magnitude to p_i, and there is neither an incident nor reflected pressure on skin 2. The pressure on the skin sheets can be described as follows:

$$p_1 = p_i + p_r + p_{rr} = 2p_i - R_{air}(i\omega w_1)\,,$$
$$p_2 = p_t = R_{air}(i\omega w_2)\,, \qquad R_{air} = \rho_{air} c_{air} / \cos\theta\,, \tag{4}$$

where p_{rr} and p_t denote the pressures radiated by skins 1 and 2, respectively, R_{air} is the modified impedance in air, and θ is the incident angle.
Eliminating w_s or w_a from Eqs. (1)-(4), yields the acoustic transmission coefficient,

$$\tau(\theta,\phi) = \left| p_i^2 \big/ p_t^2 \right| = \left| \frac{2R_{air}(Z_s - Z_a)}{(2R_{air} + Z_a)(2R_{air} + Z_s)} \right|^2 = \left| \frac{Z_s / 2R_{air} - Z_a / 2R_{air}}{(1 + Z_a / 2R_{air})(1 + Z_s / 2R_{air})} \right|^2 . \tag{5}$$

If only antisymmetric motion is considered in the frequency of interest, which implies that Z_s is infinite, the transmission coefficient reduces to,

$$\tau(\theta,\phi) = \left| 1 \big/ (1 + Z_a / 2R_{air}) \right|^2 . \tag{6}$$

Then the diffuse field acoustic transmission coefficient is,

$$\tau_d = \Gamma \int_0^{\pi/2} \int_0^{\theta_{\lim}} \tau(\theta,\phi) \sin\theta \cos\theta d\theta d\phi\,, \quad \Gamma = \int_0^{\pi/2} \int_0^{\theta_{\lim}} \sin\theta \cos\theta d\theta d\phi, \ \text{ where } \ \theta_{\lim} = 78^o . \tag{7}$$

The transmission loss is given by, $TL= -10log_{10}(\tau_d)$ dB.

Statistical Energy Analysis

Consider the transmission suite and the corresponding SEA model shown in Figs. 2 (a) and (b), respectively. The steady-state power balance equations for the three subsystems are [11,14],

$$\Pi_{1,in} = \Pi_{1,diss} + \Pi_{12} + \Pi_{13} = \omega\eta_1 E_1 + \omega\eta_{12} n_1 (E_1/n_1 - E_2/n_2) + \omega\eta_{13} n_1 (E_1/n_1 - E_3/n_3), \quad (8)$$

$$\Pi_{2,in} = \Pi_{2,diss} - \Pi_{12} + \Pi_{23} = \omega\eta_2 E_2 - \omega\eta_{12} n_1 (E_1/n_1 - E_2/n_2) + \omega\eta_{23} n_2 (E_2/n_2 - E_3/n_3), \quad (9)$$

$$\Pi_{3,in} = \Pi_{3,diss} - \Pi_{13} - \Pi_{23} = \omega\eta_3 E_3 - \omega\eta_{13} n_1 (E_1/n_1 - E_3/n_3) - \omega\eta_{23} n_2 (E_2/n_2 - E_3/n_3), \quad (10)$$

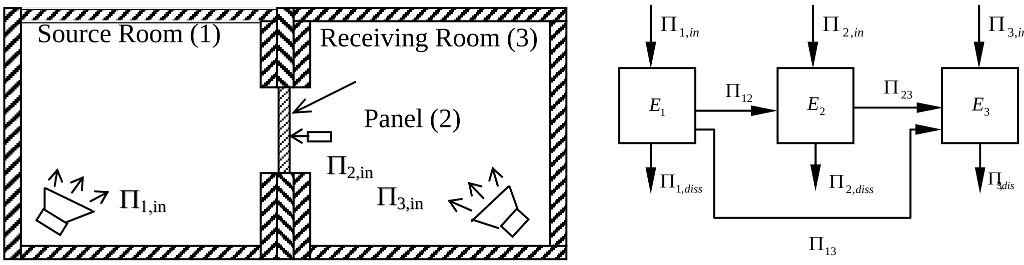

Fig. 2. (a) The transmission suite, (b) a three subsystem SEA model.

where $\Pi_{j,in}$ is the power flow into system j , $\Pi_{j,diss}$ is the internal power dissipated in system j , Π_{ij} is the power flow from system i to j, and Ej is the total energy of system j, η_j is the damping loss factor of system j, n_j is the modal density of system j, η_{ij} is the coupling loss factor when the power flow from system i to j.

Assume the panel is clamped between the transmission suite, and excited by a loudspeaker, $\Pi_{2,in} = \Pi_{3,in}=0$. Since the sound pressure level in the source room is significantly greater than that in the receiving room, $E_1/n_1 >> E_3/n_3$, the systems are coupled, and $\eta_{23}= \eta_{21}= \eta_{rad}$, where η_{rad} is the acoustic radiation loss factor of the panel. Thus, using the reciprocity relationship $\eta_{ij}n_i= \eta_{ji}n_j$, Eqs. (9) and (10) yield,

$$E_2/n_2 == (E_1/n_1)\left[\eta_{rad}/(\eta_2 + 2\eta_{rad})\right], \quad (11)$$

$$E_3 = \left[E_1\eta_{13} + E_2\eta_{23}\right]/[\eta_3 + \eta_{31} + \eta_{32}]. \quad (12)$$

Substituting Eqs. (11) into (12), gives the ratio of energy between two rooms [11] ,

$$\frac{E_1}{E_3} = \frac{\eta_3 + (n_1/n_3)\eta_{13} + (n_2/n_3)\eta_{rad}}{\eta_{13} + (n_2/n_3)\eta_{rad}^2 /(\eta_2 + 2\eta_{rad})}. \quad (13)$$

Assuming the two rooms are large enough, n_1, $n_3 >> n_2$, and ignoring high order of loss terms, Eqs (13) reduces to,

$$E_1/E_3 = n_1/n_3 + \eta_3/\eta_{13}\,, \qquad n_i = V_i\omega^2/(2\pi^3 c_{air}^3)\,, \tag{14}$$

where V_i is the room volume.
The coupling loss factor η_{13} due to non-resonant transmission is obtained from [11],

$$10\log_{10}\eta_{13} = -TL + 10\log_{10}[A_p c_{air}/(4V_1\omega)]\,. \tag{15}$$

where A_p denotes the effective area of the panel.
The transmission loss is then,

$$TL = 10\log_{10}\left[\frac{A_p c_{air}}{4V_1\omega\eta_3}\left(\frac{E_1}{E_3} - \frac{n_1}{n_3}\right)\right]. \tag{16}$$

TRANSMISSON LOSS RESULTS

The conventional two-room method was used to determine the sound transmission loss in the Sound and Vibration Laboratory at Auburn University. The transmission suite consists of two reverberation rooms. Each room has two walls made of wood with fiberglass filled in between them, and they are separated from each other by fiberglass, and mounted on air bags. The panels under investigation were mounted in the window in the walls between the two rooms. The volumes of the source room and receiving room are both equal to 51.15 m^3. The dimensions of the panels under test are 0.84 and 0.42 m. By assuming that through the panel under test is the only path by the sound travels and the sound in the two rooms is diffuse, the sound transmission loss is given by,

$$TL = L_{p1} - L_{p2} + 10\log_{10}\frac{A_p}{\tau A_p + S_2\alpha_{2avg}} = NR + 10\log_{10}\frac{A_p T_2 c}{24V_2\ln 10}\,, \tag{17}$$

where L_{p1} and L_{p2} are space-averaged the sound pressure levels measured in the two rooms, and T_2 and V_2 are the reverberation time and volume of the receiving room [14].

Experimental Results

In this particular study, the authors investigated the *TLs* of four different composite sandwich panels. The face sheets are all made of graphite plain fabrics, but the surface density varies with thickness and the amount of epoxy used in the bonding. The cores are foam-filled honeycomb. The original honeycombs are the same, while the foam in panels *A* and *B* is more rigid than that used in panels *C* and *D*. The physical parameter values for the panels are given in Table 1.

All the measurements were conducted according to ASTM E90-99. The results are presented in one-third octave bands, as shown in Figs. 3 and 4.

Table 1 Parameters of panels under study.

Panel	Face sheet (Graphite Plain weave)		Core (Foam-filled Honeycomb)	
	Thickness (m)	Surface Density(kg/m^2)	Thickness (m)	Surface Density(kg/m^2)
A	0.00024	0.5217	0.00635	1.016
B	0.00048	0.7920	0.00635	1.016
C	0.00048	0.7920	0.0127	0.914
D	0.00048	0.7920	0.0243	1.750

The *TL* curves demonstrate the strong orthotropic behavior of the four panels.

The first resonance frequency of panel *B* is higher than that of panel *A*. The critical frequencies of both panels *A* and *B* are between 1000~2000Hz. The *TL* curve of panel *B* approaches that of panel *A* above the coincidence frequencies.

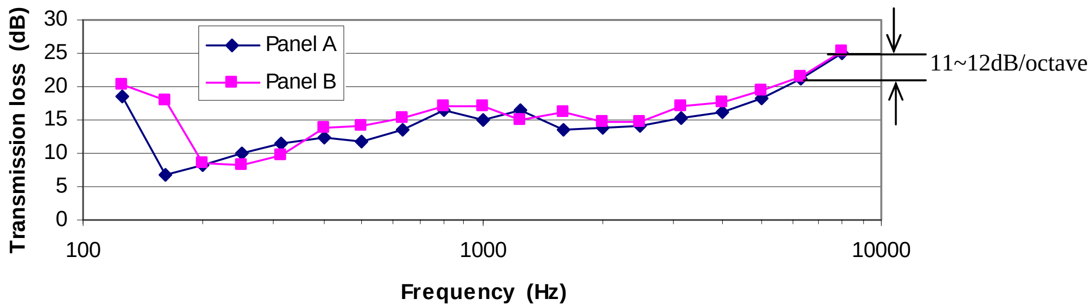

Fig. 3. The measured transmission loss for panels A and B

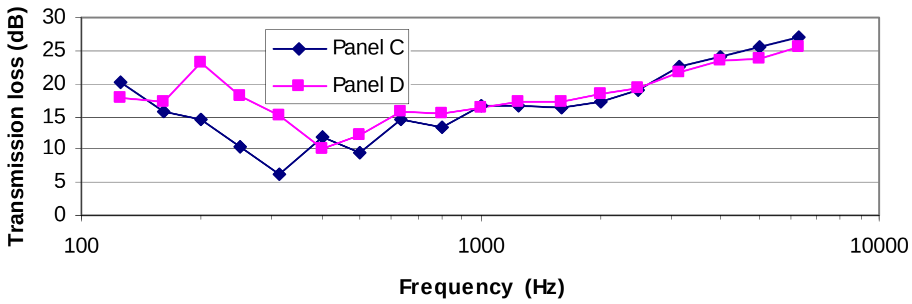

Fig. 4. The measured transmission loss for panels C and D

The first resonance frequency of panel *D* is higher than that of panel *C*. The coincidence frequencies of panel *C* or *D* are not as obvious as those of panels *A* or *B*.

The *TL* curve of panel *D* diverges from that of panel *C* above the coincidence frequencies. The *TL* curve of panel *D* is smoother than that of panel *C* in the region of resonance frequencies, which implies that the damping increases with the thickness of the core.

Comparison of Measured and Predicted TL

All the sandwich panels studied are symmetric, and there is no dilatation resonance below 20 kHz [2], so the authors used SEA to estimate the random incidence *TL* of the sandwich panels. Since plain weave fabrics have uniform strength in both directions, the face sheets are considered to be isotropic material. The material properties of panel *A* are as follows, face sheet, E_f = 40G*Pa*, *v*=0.2, and FFHC, E_c = 100M*Pa*, G_{xz}=60M*Pa*, G_{yz}=40M*Pa*.

It has been shown that the thickness (T) direction and out-plane shear modulus of FFHC remained as in the unfilled honeycomb core [12].The authors assume that the T direction and the out-plane shear modulus of the two FFHCs are the same, and the Young's modulus of the face sheets are the same. Then for panel *C*, the assumed material properties are, E_f = 40G*Pa*, *v*=0.2, E_c = 100M*Pa*, G_{xz}=60M*Pa*, G_{yz}=40M*Pa*.

The measured *TL*s of panels *A* and *C*, with the estimated results from SEA and the mass law, are given in Figs. 5 and 6, respectively.

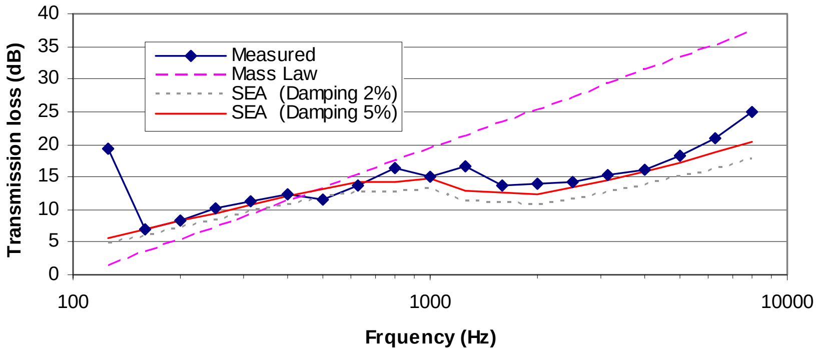

Fig. 5. The measured and estimated transmission loss for panel A.

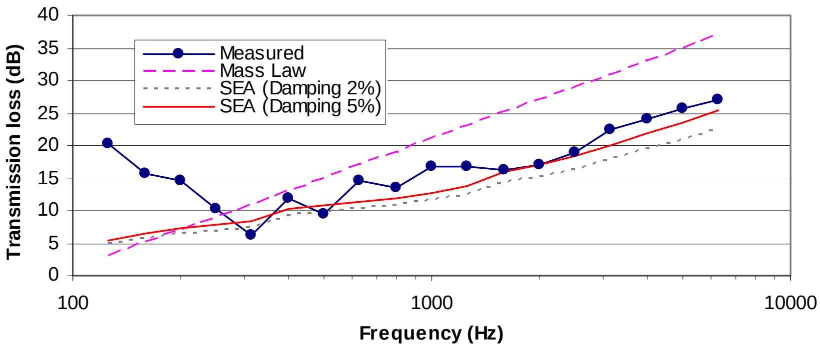

Fig. 6. The measured and estimated transmission loss for panel C.

The estimated results from SEA agree well with experimental values of *TL*. The results indict that sandwich panels with FFHC have high damping, greater than 2%.

SUMMARY

The four sandwich panels with the same original honeycomb have similar acoustical performances in the frequency range 125~8000Hz. Increasing the thickness either of the face sheets or core increases the first resonance frequency. Increasing the thickness in the core improves the damping. The estimated *TLs* from SEA agree well with the experimental values. The FFHC sandwich panel has high damping, greater than 2%.

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