SOUND TRANSMISSION THROUGH A CURVED HONEYCOMB COMPOSITE PANEL

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Composite structures are often used in aircraft because of the advantages offered by a high strength to weight ratio. However, the acoustical properties of these light and stiff structures can often be less than desirable resulting in high aircraft interior noise levels. In this paper, measurements and predictions of the transmission loss of a curved honeycomb composite panel are presented. The transmission loss predictions are validated by comparisons to measurements. An assessment of the behavior of the panel is made from the dispersion characteristics of transverse waves propagating in the panel. The speed of transverse waves propagating in the panel is found to be sonic or supersonic over the frequency range from 100 to 5000 Hz. The acoustical benefit of reducing the wave speed for transverse vibration is demonstrated.

Introduction

Panels constructed from face sheets laminated to a honeycomb core are being incorporated into the design of modern aircraft fuselage and trim treatments. The mechanical properties of these panels offer a distinct advantage in weight over other commonly used construction materials. The strength to weight ratio of honeycomb composite panels is high in comparison to rib stiffened aluminum panels used in previous generations of aircraft. However, the high stiffness and low weight can result in supersonic wave propagation at relatively low frequencies, which adversely affects the acoustical performance at these frequencies. Poor acoustical performance of these types of structures can increase the cabin noise levels to which the passengers and crew are exposed.

Others[1-5] have demonstrated, to some extent, the vibro-acoustic properties of these types of constructions. He et. al. studied the response of beams constructed from fiberglass face sheets laminated to honeycomb cores.[1] Supersonic wave speeds for transverse vibrations were observed in the beam samples at relatively low frequencies. Nilsson[2] studied wave motion in flat honeycomb composite panels. Analytical expressions for the response of the panels were developed and validated by comparison to experiment. The importance of addressing the wave speed was discussed in lieu of the acoustic characteristics. Davis[3] studied how changes in the material used to construct honeycomb panels affected the wave speed. The density of the core material was varied, and a decrease in the wave speed was correlated to a decrease in the core density. The acoustical properties of the panel with respect to both acoustical and mechanical excitations were investigated for various core densities. A transmission loss increase and radiation efficiency decrease were demonstrated for a decrease in core density.

The purpose of the current effort is to extend the previous studies of honeycomb composite structures to curved panels and validate a transmission loss model based on numerical analysis of a curved honeycomb composite panel. The panel considered in this paper is fabricated from carbon fiber face sheets laminated to a Nomex honeycomb core. Grosveld et. al. and Buehrle et. al. have developed and validated a finite element and a boundary element model to predict the vibro-acoustic response of this panel.[4,5] The model developed by these authors was used in this effort to predict the sound power transmission loss of the panel resulting from a diffuse acoustic excitation. The sound power transmission loss was measured, and predictions of the transmission loss using the numerical models are validated by comparisons to the measurement. The wavenumber of flexural vibration resulting from a point force excitation were both measured and predicted. The wave speed was shown to be supersonic resulting in poor acoustical properties. A study is presented in which limp mass is added to the panel to slow the wave speed. The acoustical benefit of slowing the wave speed in the honeycomb composite panel is discussed. Conclusions are drawn about future efforts that will focus on improving the acoustical properties of this type of panel construction.

Experimental Setup

The experimental setup used to test the response of the curved honeycomb composite panel is illustrated in Figures 1 through 5. The composite panel was installed in the transmission loss window in the Structural Acoustic Loads and Transmission (SALT) facility

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(Figures 1 and 2) at NASA Langley Research Center [6]. The SALT facility is a transmission loss suite consisting of a reverberation chamber and an anechoic chamber connected by a 54-inch by 54-inch transmission loss window. The volume of the reverberant source room and anechoic receiving room are 9817-ft^3 and 11900-ft^3, respectively. Based on the characteristics of the rooms, the 100 Hz one-third octave band is the lowest frequency yielding a reliable transmission loss estimate.[6] The curved panel was held in the transmission loss window with a high-density fiberboard frame that was 2-inches thick (Figure 2). A plastic tubing gasket was used to seal the junction of the fiberboard frame and composite panel to reduce flanking energy from the source room into the receiving room. The composite panel was 0.82-inches thick with a radius of 41.5-inches measured from the center of the honeycomb core. Each carbon fiber face sheet was nominally 0.023-inches thick. The arc length was 62-inches and the length of the straight edge was 54-inches. The composite panel weighed 19.6-lbs.

Both point force and diffuse acoustical excitations were studied. In the case of a point force excitation, a shaker was attached to the panel and driven by pseudo-random noise. The normal surface velocity response of the panel was measured using a scanning laser vibrometer. The force and acceleration at the drive point were measured using an impedance head. The excitation point was located 22.25-inches in the arc direction and 15.25-inches in the axial direction from the bottom right corner of the panel (Figure 2). The frequency response functions between the surface velocity, measured with the vibrometer, and the excitation force were recorded. The data were taken over a fine grid of measurement points on the panel surface to yield a spatial velocity distribution of the panel. The spatial velocity distribution was used to calculate the dispersion behavior of transverse waves propagating in the panel using wavenumber decomposition techniques.

In the case of an acoustical excitation, the reverberation room was excited by four speakers driven by white noise (Figure 3a). The sound power incident on the panel was calculated from the response of the source room measured with six half-inch condenser microphones randomly distributed throughout the reverberation chamber.[7] A traverse mechanism (Figure 3b) was used to measure the intensity transmitted through the composite panel into the receiving room.[7] The intensity radiated into the receiving room was measured using 4 two-microphone acoustic intensity probes positioned by the traverse on a measurement surface that was offset 3-inches from the surface of the panel (Figure 4). The intensity was sampled at total of 924 equally spaced stationary points. The transmitted sound power was found from the average of these discrete intensity measurements multiplied by the area of the measurement surface. The transmission loss of the composite panel was computed as the ratio of the incident sound power to the transmitted sound power.[8]

Limp lead vinyl was applied to the panel to increase the mass of the system (Figure 5). The lead vinyl was attached using double-sided tape. The lead vinyl completely covered the surface of the panel. The weight of the panel was 19.6-lbs and the weight of the additional lead vinyl and tape was 19.1-lbs. Thus, the treatment almost doubled the surface density of the composite panel. The insertion loss of the lead vinyl was found from the ratio of the measured transmission loss with and without the lead vinyl attached to the panel.

Numerical Modeling

A NASTRAN finite element and a Comet Acoustics boundary element model of the curved composite panel were developed and validated by Grosveld et. al. and Buehrle et. al.[4,5] The honeycomb core of the composite panel was modeled with NASTRAN solid CHEX-8 elements using the bulk properties of the honeycomb material. The face sheets were modeled with NASTRAN plate CQUAD-4 elements that were attached to the elements of the core at each grid point. Full allowance was made for anisotropy of the core material and orthotropy of the face sheets. There were 43 element in the arc direction, 43 elements in the axial direction, and the core was modeled with a single solid element through the thickness. Each face sheet was also modeled using a single plate element through the thickness with the equivalent laminate properties. The element size ensured adequate convergence of the velocity response up to a frequency of 1000 Hz. A clamped boundary condition was applied to each node at the free edges of the face sheets. The velocity response of the finite element model was predicted for a prescribed excitation mechanism. In the study presented here, two excitation mechanisms were studied: a unit point force excitation and an approximation of a diffuse acoustic excitation. The point force excitation was used to predict the dispersion characteristics of the panel and the acoustic excitation was used to study the sound transmission characteristics of the panel. The point force excitation was applied at the same location as the shaker was placed in the experiment. The diffuse acoustic excitation will be detailed later in this section.

A boundary element model was used to predict the sound power radiated from the panel as a result of a velocity response predicted using the finite element model. The radiating surface of the composite panel was modeled using 24 QUAD-4 elements in both the axial and arc directions. A total of 576 boundary elements were used to represent the surface velocity of the panel. This resolution resulted in adequate convergence of the radiated sound power up to 1000 Hz while still yielding reasonable computation times. The frame of the transmission loss window, the blue portion shown in Figure 3b, and the fiberboard fixture were also included in the boundary element model. To simulate the same acoustic boundary conditions present in the experiment, the elements used to model the frame and the fixture had a zero velocity boundary condition imposed. The sound power radiated by the panel was predicted for a free field condition.

To use the finite element and boundary element models to predict the transmission loss of the composite panel, a simulation of the excitation mechanism present in the experiment was needed. A diffuse acoustic excitation of the finite element models was developed based on plane wave propagation. A large number, N, of plane waves having random angles of incidence, random magnitudes, and random temporal phase angles were summed together to simulate a diffuse field excitation. A plane wave incident on the surface of the composite panel at angles θ_n and ψ_n is shown in Figure 6. The angles θ_n and ψ_n are uniformly distributed random numbers on the intervals $[0,\pi]$ and $[0,2\pi]$ respectively and represent the angles of propagation in spherical coordinates. The n^{th} plane wave has a magnitude of $P_n\cos(\theta_n)$, where P_n is a uniformly distributed random number on the interval [0,1]. Thus the steady state pressure of a single plane wave is

$$P_n(x,y,z,t) = P_n\cos(\theta_n)e^{-ik_x x}e^{-ik_y y}e^{-ik_z z}e^{i(\omega t+\phi_n)} \quad (1)$$

where ω is the angular frequency, ϕ_n is a random temporal phase angle uniformly distributed on the interval of $[0,2\pi]$, and k_x, k_y and k_z are the wavenumber in the x, y and z directions, respectively, found from

$$k_x = k\sin(\theta_n)\cos(\psi_n) \quad (2a)$$

$$k_y = k\sin(\theta_n)\sin(\psi_n) \quad (2b)$$

$$k_z = k\cos(\theta_n) \quad (2c)$$

where k is the wavenumber in air at a particular analysis frequency. The random temporal phase angle is introduced to prevent the N plane waves from having the same phase angle at the origin. A weighting function of $\cos(\theta_n)$ is included in the pressure magnitude to correct for the probability distribution of incident plane waves likely present in the experimental excitation.[9] The random variables θ_n, ψ_n, P_n and ϕ_n are unique for each of the N plane waves. Assuming steady state simple harmonic motion and a nearly rigid boundary condition at the interface between the composite panel and the acoustic space, the spatial pressure distribution exciting the composite panel is

$$\hat{P}_n(x,y,z,\omega) = 2P_n\cos(\theta_n)e^{i(\phi_n - k_x x - k_y y - k_z z)} \quad (3)$$

where x, y and z are evaluated on the surface and k_x, k_y and k_z are evaluated at a particular angular frequency ω. The pressure acting on the surface of each element, e, in the finite element model, due to the N plane waves, was computed using the x, y and z coordinates of the element center x_e, y_e and z_e. The total pressure at the center of each element, P_e, due to N incident plane waves is

$$\hat{P}_e(\omega) = \sum_{n=1}^{N} 2P_n\cos(\theta_n)e^{i(\phi_n - k_x x_e - k_y y_e - k_z z_e)} \quad (4)$$

where N is the number of plane waves used to approximate the diffuse field. This pressure distribution acting on the surface elements of the finite element model was used as an excitation and was assumed to be uniformly distributed over the element. It should be noted that this pressure acts on only one face sheet of the composite panel. The velocity response of the finite element model was predicted due to the pressure excitation. The predicted velocities were imported into the boundary element model of the composite panel and the transmitted sound power, Π_t, was predicted.

To compute transmission loss, the ratio of the incident to transmitted sound power is needed. The incident sound power is computed from the intensity vector of each of the N plane waves. The intensity vector, $\vec{I}_n$, of the n^{th} plane wave is

$$\vec{I}_n = -\frac{[P_n\cos(\theta_n)]^2}{\rho c}[\sin(\theta_n)\cos(\psi_n)\vec{i} + \sin(\theta_n)\sin(\psi_n)\vec{j} + \cos(\theta_n)\vec{k}] \quad (5)$$

The intensity normal to the surface of each element of the finite element model is found from the dot product of the intensity vector and the element normal, $\vec{r}_e$. The sound power incident on an element e, of area A_e, due to the n^{th} plane wave is

$$\Pi_{i,n,e} = A_e\vec{I}_n \cdot \vec{r}_e \quad (6)$$

where the element normal $\vec{r}_e$ is computed for each element using the finite element model geometry. The total sound power incident on the panel for all N plane waves and E elements is

$$\Pi_i = \sum_{e=1}^{E}\sum_{n=1}^{N}\Pi_{i,n,e} \quad (7)$$

The transmitted sound power is computed as outlined above. The predicted transmission loss of the composite panel is computed from the ratio of the predicted incident and transmitted sound power

$$TL = 10\log_{10}\left(\frac{\Pi_i}{\Pi_t}\right) \quad (8)$$

Results and Discussion

The predicted and measured axial and circumferential dispersion curves[9] of the composite panel are compared for a point force excitation (Figure 7). There is good agreement between the predicted and measured transverse wave behavior. The dispersion curves of the structural waves shown in Figure 7 are compared to the wavenumber of air (Figure 7, green line). From these data, the wave speed of the structural waves is shown to be supersonic in the axial direction and sonic in the circumferential direction. The close matching of the wave speed in the composite panel and the wave speed in air results in highly efficient coupling between the two media. The measured transmission loss of the composite panel is shown in Figure 8 (blue line) and is compared to the transmission loss of two flat limp masses weighing 16-lbs and 64-lbs (red and green line respectively). The transmission loss of the composite panel, which weighed 19.6-lbs, was substantially lower than would be expected based on mass law behavior. The poor acoustical performance is due to sonic/supersonic wave propagation in the panel.

To study the effects of slowing the wave speed of the composite panel, limp lead vinyl was attached to the surface (Figure 5). Addition of the lead vinyl doubled the surface density of the composite panel but did not significantly increase the stiffness. This is not the preferred method of slowing wave speed in aircraft panels, but will be used here for illustrative purposes. The dispersion behavior of the composite panel with and without the lead vinyl attached is shown in Figure 9 at 1200 and 2500 Hz. The wavenumber of air at these frequencies is overlaid (Figure 9, green line). The wave speed of the transverse vibration is pushed subsonic by the addition of the lead vinyl. Thus, the transmission loss of the panel with the added limp mass should be mass controlled in this frequency region. The measured transmission loss of the composite panel with and without the lead vinyl attached is shown in Figure 10a and again is compared to the transmission loss of two limp masses. The change in transmission loss of the composite panel due to the addition of the lead vinyl is shown in Figure 10b. An increase of 14 dB was obtained at frequencies above 1500 Hz (Figure 10b) for a doubling of the surface density. The 8 dB increase beyond the 6 dB expected from doubling of the surface density[9] is a consequence of slowing the wave speed of the structural vibration to subsonic speeds and causing the composite panel to exhibit mass law behavior. Future efforts at NASA Langley Research Center will investigate design changes to the honeycomb core that will slow the wave speed but not adversely affect the strength or weight of the composite panel.

A model was needed to study the effects of changes in the panel design on the sound transmission properties. A diffuse acoustical excitation was applied to the finite element model of the composite panel developed by Grosveld et. al. by summing a large number of incident plane waves.[4,5] The resultant pressure distribution on the surface of the panel was used to excite the finite element model. The transmission loss was found from the ratio of the incident and transmitted sound power. A comparison of predicted and measured transmission loss is shown in Figure 11. There is good agreement between the predicted and measured trends in the narrow band transmission loss (Figure 11a). When averaged into one-third octave bands, the prediction and experiment are within 3 dB (Figure 11b), which is an acceptable level of error. The numerical model developed in this paper offers significant advantages over analytical methods for evaluating the transmission loss of composite structures. Complex models having spatially variable properties and discontinuities can be easily evaluated using numerical approaches. The model presented in this paper will be used in future efforts to evaluate the effects of honeycomb core design changes on the acoustical properties of the composite panel.

Concluding Remarks

When designing honeycomb composite structures for use in aircraft, it is necessary to incorporate acoustic benchmarks into the design cycle. Thus, tools to predict and interpret the vibro-acoustic properties of these types of structures are needed. The panel presented in this study exhibited transverse vibration that was supersonic, resulting in poor acoustical properties. This wave behavior was analyzed both experimentally and numerically and the benefit of slowing the wave speed of the structural waves was demonstrated. A finite element and boundary element based approach to predict the transmission loss of the composite panel was presented and validated. Future efforts at NASA Langley Research Center will focus on using the tools presented in this paper to analyze the effects of design changes on the vibro-acoustic properties of honeycomb composite panels.

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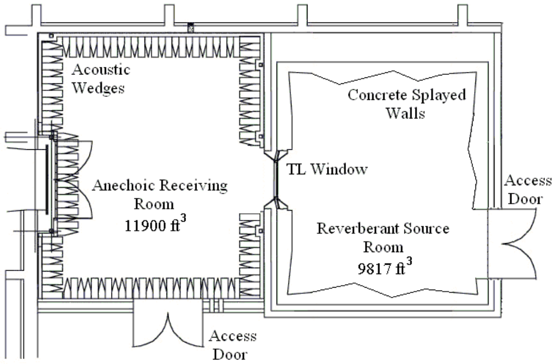

Figure 1: Schematic of the Structural Acoustic Loads and Transmission facility.

Figure 2: The curved honeycomb composite panel mounted in the transmission loss window of the SALT facility.

a)

b)

Figure 3: Setup used to measure the transmission loss of the panel: a) the excitation of the source room with 4 speakers (only 3 are shown) and b) the traverse mechanism used to measure radiated acoustic intensity.

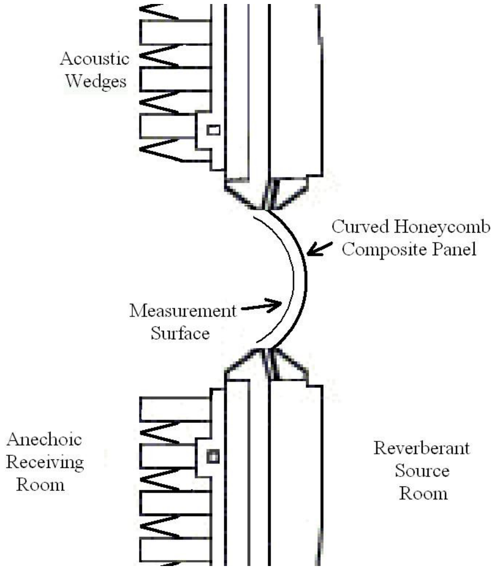

Figure 4: Schematic of the transmission loss window with the composite panel installed illustrating the location of the measurement surface.

Figure 5: Limp lead vinyl attached to the curved honeycomb composite panel.

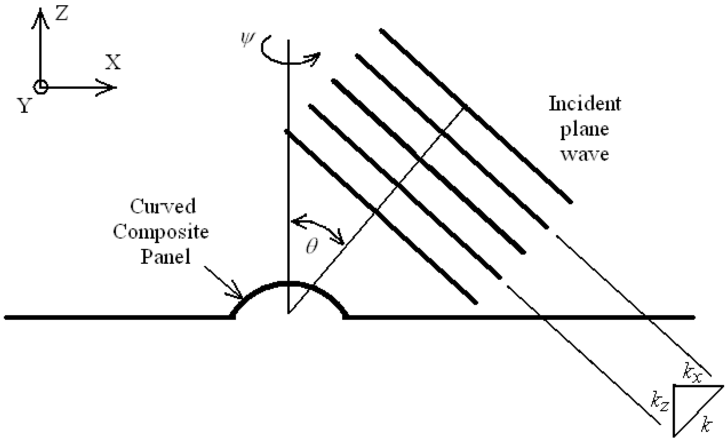

Figure 6: Plane wave incident on the composite panel. The plane wave is shown propagating in the x-z plane, the y axis is into the page. The angle ψ represents a rotation of the heading of the plane wave about the z axis.

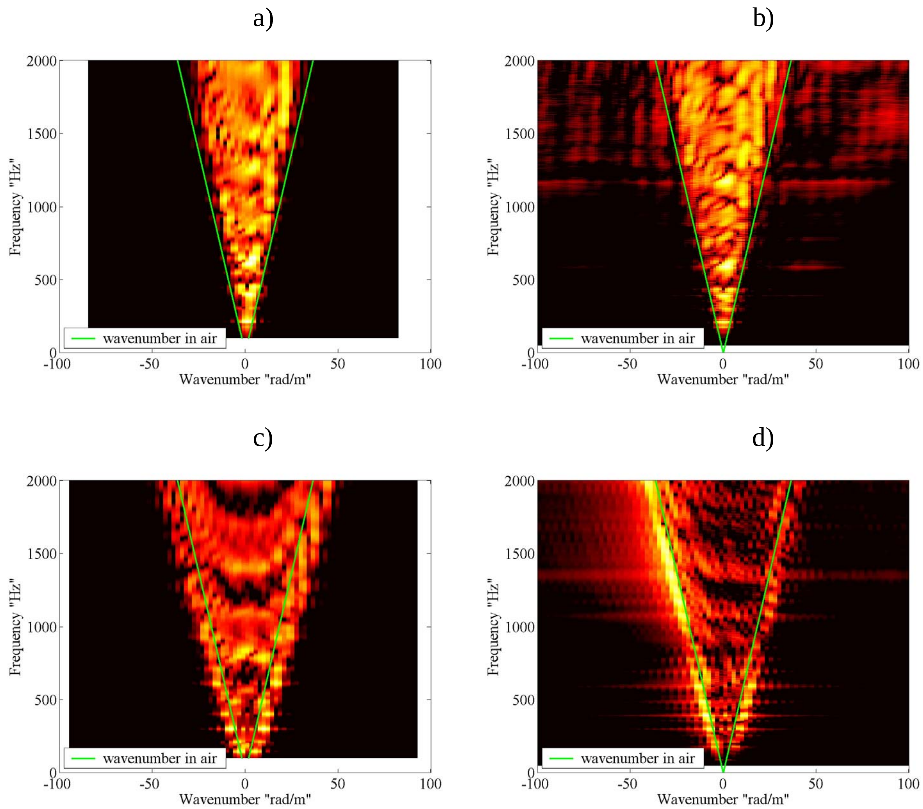

Figure 7: Comparison of the predicted and measured dispersion behavior of the panel to the wavenumber in air: a) the axial wavenumber predicted by the NASTRAN model, b) the axial wavenumber calculated from measured surface velocities, c) the circumferential wavenumber predicted by the NASTRAN model, d) the circumferential wavenumber calculated from measured surface velocities.

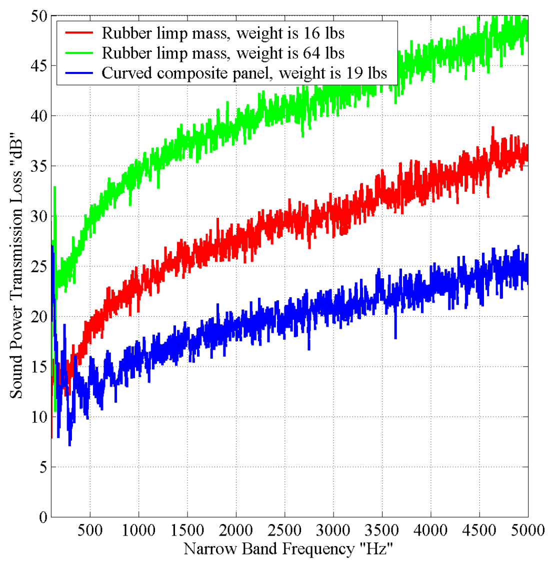

Figure 8: The transmission loss of the honeycomb composite panel compared to the transmission loss of two limp masses.

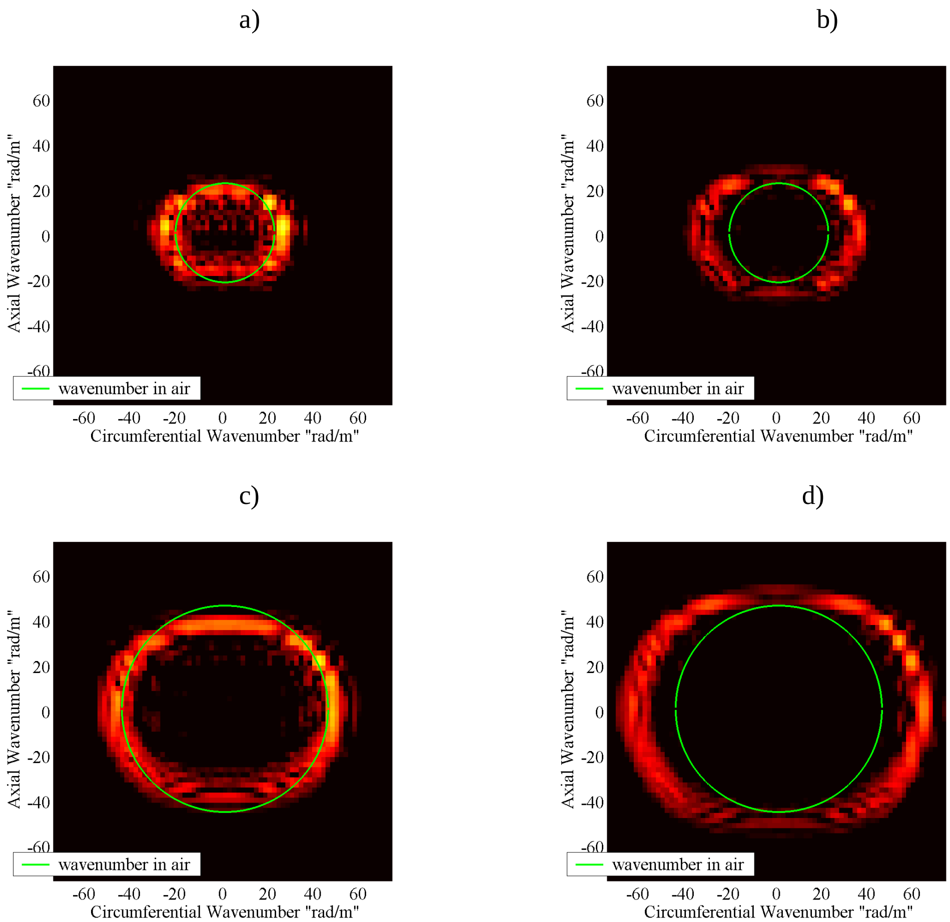

Figure 9: Dispersion behavior at two different frequencies with and without lead vinyl attached to the panel, the green line is the wavenumber in air: a) structural wavenumber at 1200 Hz without lead vinyl attached to the panel, b) structural wavenumber at 1200 Hz with lead vinyl attached to the panel, c) structural wavenumber at 2500 Hz without lead vinyl attached to the panel and d) structural wavenumber at 2500 Hz with lead vinyl attached to the panel.

a) b)

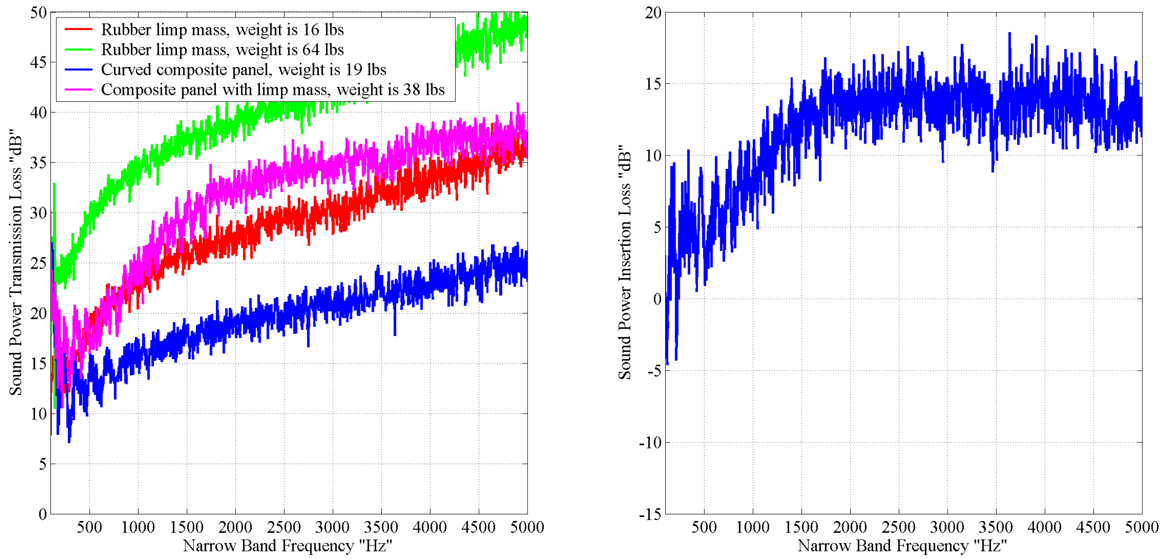

Figure 10: measurement of the transmission loss of the composite panel with lead vinyl attached to the surface: a) transmission loss of the composite panel compared to the transmission loss of limp masses and the transmission loss of the composite panel with lead vinyl attached and b) insertion loss of the lead vinyl.

a) b)

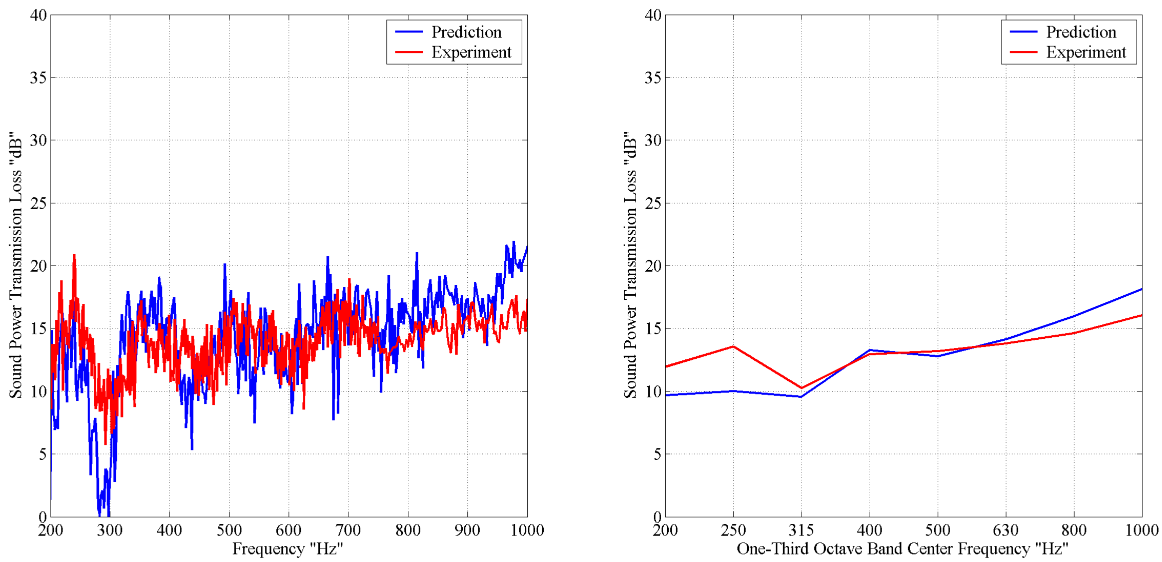

Figure 11: Comparison of the measured and predicted sound power transmission loss of the curved honeycomb composite panel: a) narrow band and b) one-third octave band.