THE EFFECTS OF POROUS MATERIALS ON THE NOISE INSIDE A BOX CAVITY

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Porous materials are usually attached to the vehicle body metal panels to reduce the radiation of the noise into the vehicle cabin. This study focuses on investigating the effects of porous materials on the noise level due to the structure-borne noise excitation. Because of the complexity of the analysis of noise in a full vehicle cabin, a box cavity is used. The box has five rigid walls and one flexible aluminium plate, which make a coupled plate and cavity enclosure system. The pressure microphones are used for measuring the Sound Pressure Level (SPL) at different measurement locations in the box cavity with and without porous materials. A Computer Aided Engineering (CAE) model is also developed to simulate the experiment setup using the Finite Element Method (FEM). Good agreement has been obtained between the experimental results and the simulation values for the acoustical response at different locations inside the box cavity. Furthermore, the influences of porous materials on the vibration reduction of the plate and the noise reduction of the cavity are also characterised. The experimentally validated developed model is used to characterise the effects of damping and porous treatments in reducing the structure-borne noise.

1. Introduction

The automotive industry is facing challenges to reduce the total weight of a vehicle by various methods including the use of thin metal panels in the vehicle body. However, using thin metal panels creates several Noise Vibration and Harshness (NVH) challenges such as increased structure-borne noise in the vehicle cabin. This is due to the fact that use of thin metal panels results in lower inertial forces, consequently leading to high vehicle body vibration amplitudes and generates more structure-borne noise [1]. On the other hand, for Electric Vehicles (EV), due to the absence of engine combustion noise, the structure-borne road noise is more annoying than for Internal Combustion Engine (ICE) vehicles, which it has created some new NVH challenges [2, 3]. Furthermore, cancelling structure-borne noise is often challenging and it is costly. The countermeasures for structure-borne noise are unlikely to be applied in the late design phase, because they are linked to the shape, thickness, and design of each metal panel of the vehicle.

Porous sound-absorbing materials are usually coupled with vehicle body metal panel and cabin to absorb vehicle interior noise, but the effects on the noise attenuation of structure-borne noise are not well understood. Therefore, fundamental research with experiment and simulation is required. Davy has presented a method for calculating the directivity of the radiation of noise from a vibrating panel [4]. Some researchers have used Green's Function, mixed Finite Element (FE) and classical displacement formulation approaches to predict the sound field inside an enclosed space lined with porous materials [5 - 9]. Optimisation has also been done by Yamamoto [10] based on the concept of the density approach to topology optimisation and transfer matrices. His proposed optimisation method is an effective method that recommends the application of countermeasures. The reduced impedance approach that shortens the calculation time on large-scale models such as a trimmed vehicle body was review by Zhou [11].

The objective of this paper is to deliver a validated Finite Element (FE) model and appropriate experimental methods, for investigating the effects of porous materials on the noise level in a box cavity due to structure-borne noise excitation. Accordingly, an FE model of the box cavity is developed. An experimental setup is then used for the validation of the developed model. In the results section, a Frequency Response Function (FRF) is used to express the system response as a function of frequency at one selected microphone measurement location. Finally, the simulation and experimental results are compared and discussed.

2. Method

2.1 Experiment

Figure 1 shows a box cavity developed in this study. It has the size of L_x = 450 mm, L_y = 300 mm and L_z = 400 mm. The box cavity has five rigid walls, which are made of 25 mm Tempered Perspex. Only one face of the box is not rigid and it is a 5 mm aluminium plate that is carefully bolted to the box. An electromagnetic shaker is used as the vibration excitation. The shaker head is glued to the centre of the aluminium plate. The shaker input force for this experiment is a random signal with a frequency range of 50 to 500 Hz.

The SPL inside the cavity is measured by using a pressure microphone at frequencies between 50 to 500 Hz. In this paper, the microphone is located at x = 95 mm, y = 145 mm, z = 120 mm on the x-y-z coordinate system. In this experiment, the acoustic field in the cavity is influenced by the vibration of the plate so that the cavity is perturbed by the point loading of the plate.

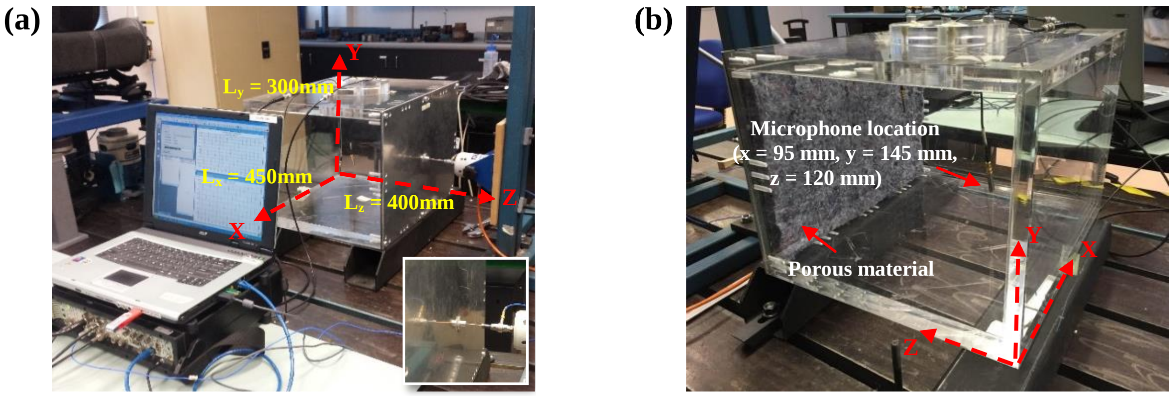


Figure 1. Box cavity experiment in laboratory, showing (a) the dimensions of the box cavity and the shaker head glued to the aluminium plate and (b) the microphone location and the porous material glued on the aluminium plate.

Two porous materials, with 5 and 20 mm thicknesses, are used in this study. They are glued on the aluminium plate. The porous materials are mainly constructed from Polyester Staple Fibers and Polypropylene Terephthalate (PET). The two-microphone impedance tube test is used to obtain the normal Sound Absorption Coefficient (SAC) for the different porous materials according to the ASTM E1050-08 standard [12]. An inverse method is then used to obtain the physical properties such as tortuosity, porosity, airflow resistivity, and characteristic length based on the measured SAC

[13]. These physical properties are then used as input parameters for the numerical simulation. A database of material properties for the available samples has already been developed [14]. Table 1 shows the physical properties of the selected porous sound absorbing materials used in this study.

Table 1. Physical properties of the selected porous sound-absorbing material.

Physical Properties	Porous Material 1 (5 mm)	Porous Material 2 (20 mm)
density - ρ_t (kg/m^3)	140.07	52.80
airflow resistivity - σ (Ns/m^4)	181000	37330
porosity - ϕ	0.98	0.92
tortuosity - τ	2.90	1.10
viscous characteristic length - Λ (m)	0.000112	0.000192
thermal characteristic length - Λ' (m)	0.000224	0.000384

2.2 Numerical Simulation

2.2.1 FE Model

An FE model is developed and is used to predict the effects of porous sound-absorbing materials on the noise level in the cavity. Figure 2 shows the geometry and the FE model of the box cavity used in this study. In this model, simply supported edges are used; the rigid air cavity is weakly coupled to a thin aluminium plate and porous materials. For modelling the thin aluminium plate, two-dimensional shell elements are used. The material properties for the plate are assumed to be Young's Modulus E = 70 GPa, Poisson's ratio v = 0.3, and density ρ = 2800 kg/m^3. The excitation is modelled as a point load applied on the plate at the same location as the experiment setup. Three-dimensional solid elements are used for modelling the porous materials and the air cavity. In this study, the air properties are density ρ_a = 1.21 kg/m^3, and speed of sound c = 340 m/s. A constant plate damping equal to 3% and a constant damping of 1% for the air cavity are used.

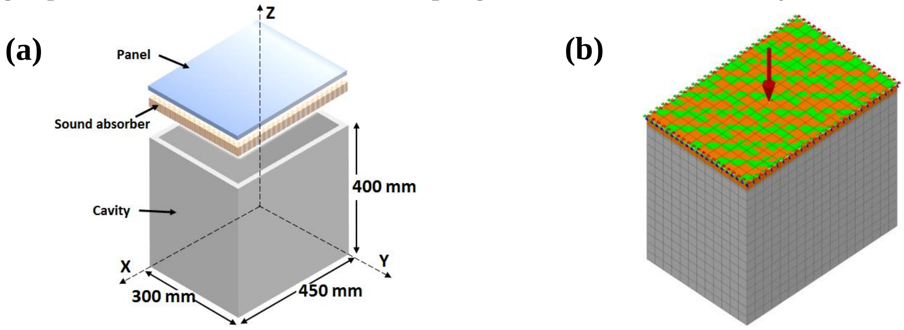


Figure 2. (a) Geometry of the box cavity, (b) FE model of the box cavity used in this study.

2.2.2 Modal Analysis

The prerequisite for studying a coupled plate and cavity system is firstly the ability to predict the mode shapes and natural frequencies of the plate and cavity. The mode shapes and resonant frequencies of the plate can be calculated from [15]:

(1) $$w_{mn}(x,y) = C_{mn}\sin\left(\frac{m\pi x}{L_x}\right)\sin\left(\frac{n\pi y}{L_y}\right),$$

(2) $$\omega_{mn} = \sqrt{\frac{D}{\rho t}}\left(\left(\frac{m\pi}{L_x}\right)^2 + \left(\frac{n\pi}{L_y}\right)^2\right),$$

where C_{mn} is an arbitrary constant determined by initial conditions, m is the modal order along the x-axis, n is the modal order along the y-axis, D is the bending stiffness of the plate, ρ is the material density and t is the thickness of the plate.

The mode shapes and resonant frequencies of the cavity can be calculated from the following equations [16]:

(3) $$\psi_{ijk} = \cos\frac{i\pi x}{L_x}\cos\frac{j\pi y}{L_y}\cos\frac{k\pi z}{L_z},$$

(4) $$\omega_{ijk} = \frac{c}{2}\sqrt{\left(\frac{i}{L_x}\right)^2 + \left(\frac{j}{L_y}\right)^2 + \left(\frac{k}{L_z}\right)^2},$$

where c is the sound speed and i, j and k are the indices of the normal modes of the cavity.

2.2.3 Coupled System

A coupled FE formulation for analysing the noise field in the box cavity with a porous material is used [7]. This method uses classical elastic and fluid elements to simulate the plate and the air cavity. It uses a displacement formulation to describe the porous materials, which is based on the Biot theory [17] and further derivation by Panneton and Atalla [6]. In the case of a plate, porous material, and cavity configuration, the following matrices may be used [7]:

(5) $$\begin{pmatrix} -\omega^2[M_s]+[K_s] & [0] & [0] \\ [0] & -\omega^2[M_P]+j\omega[C_P]+[K_P] & -[C] \\ [0] & -[C]^T & \frac{[M_c]}{\rho\omega^2}-\frac{[K_c]}{\rho c^2} \end{pmatrix} \begin{Bmatrix} u_e \\ u_p \\ P_c \end{Bmatrix} = \begin{Bmatrix} f_e \\ f_p \\ u_c \end{Bmatrix}$$

where $[M_s]$, $[K_s]$ are the mass and stiffness matrices of the metal plate, $[M_p]$, $[C_p]$ and $[K_p]$ are the equivalent mass, damping and stiffness matrices of the porous material. $[M_c]$, $[K_c]$ are the mass and stiffness of the acoustic cavity. $[C]$ is the coupling matrix. $\{u_e\}$, $\{u_p\}$ are the vector of nodal displacement for the plate and the porous material. $\{f_e\}$ and $\{f_p\}$ are the nodal loading vector. $\{P_c\}$ and $\{u_c\}$ are the nodal acoustic pressure vector and related volume displacement for air cavity.

3. Results and Discussions

3.1 Resonant Frequencies and Mode Shapes (Coupled Plate and Cavity)

Table 2 shows the comparison of resonant frequencies of the aluminium plate and air cavity obtained from the analytical method, the FE modelling, and the measurements (without porous materials) at frequencies below 500 Hz. Good agreements between the analytical method, the FE modelling, and the measurement results were obtained for both the aluminium plate and the air cavity.

Table 2. Comparison of resonant frequencies of the aluminium plate and the air cavity obtained from the analytical method, the FE modelling, and the measurements (frequency range of 50 to 500 Hz).

Plate natural frequencies (Hz)				
Aluminium Plate Mode No.	Modes	Analytical Method	FE Modelling	Measurement
#1	(1,1)	190.49	189.91	195.00
#2	(2,1)	366.33	365.55	361.25
Cavity natural frequencies (Hz)				
Air Cavity Mode No.	Modes	Analytical Method	FE Modelling	Measurement
#1	(1,1,1)	377.78	378.26	381.00
#2	(0,0,1)	425.00	425.68	431.00

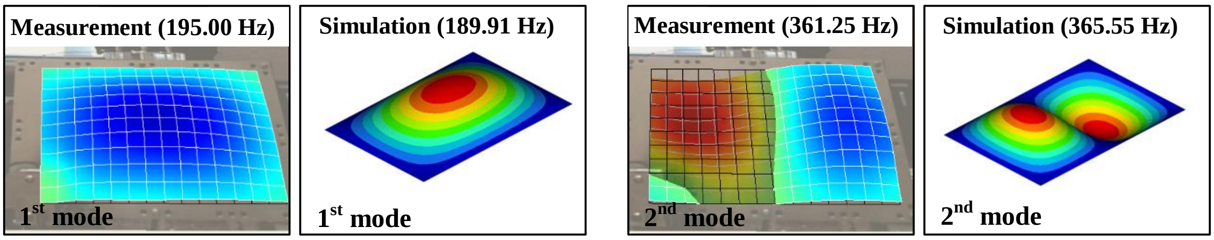


Figure 3. The mode shapes of the aluminium plate obtained from the FE modelling and the measurements made using a laser scanning vibro-meter in the laboratory.

The comparison of the plate mode shapes between the FE modelling and the measurement is shown in Figure 3. Very good agreement was observed between the FE modelling and the measurements. This agreement is required to ensure that the accuracy of the FE vibro-acoustic model is not degraded by discrepancies in the FE model of the metal plate. In other words, it was concluded that the FE model of the plate developed in this study was reasonable.

3.2 Acoustic Response (Coupled Plate and Cavity)

Figure 4 shows the FE modelling results of the SPL as a function of frequency for the coupled plate and air cavity system. The frequencies of the peaks of the calculated SPL graph were consistent with the metal plate and the air cavity resonant frequencies. This suggests that the peaks of the SPL graph were caused by either the air cavity modes or by the metal plate vibration modes. Hence, the vibration modes of the plate and air cavity have significant effects on the radiated sound inside of the box. The peaks observed at 189.91 Hz and 365.55 Hz were caused by the resonances of the aluminium plate, and the peaks at 378.26 Hz and 425.68 Hz by the resonances of the acoustic air cavity as shown in Figure 4.

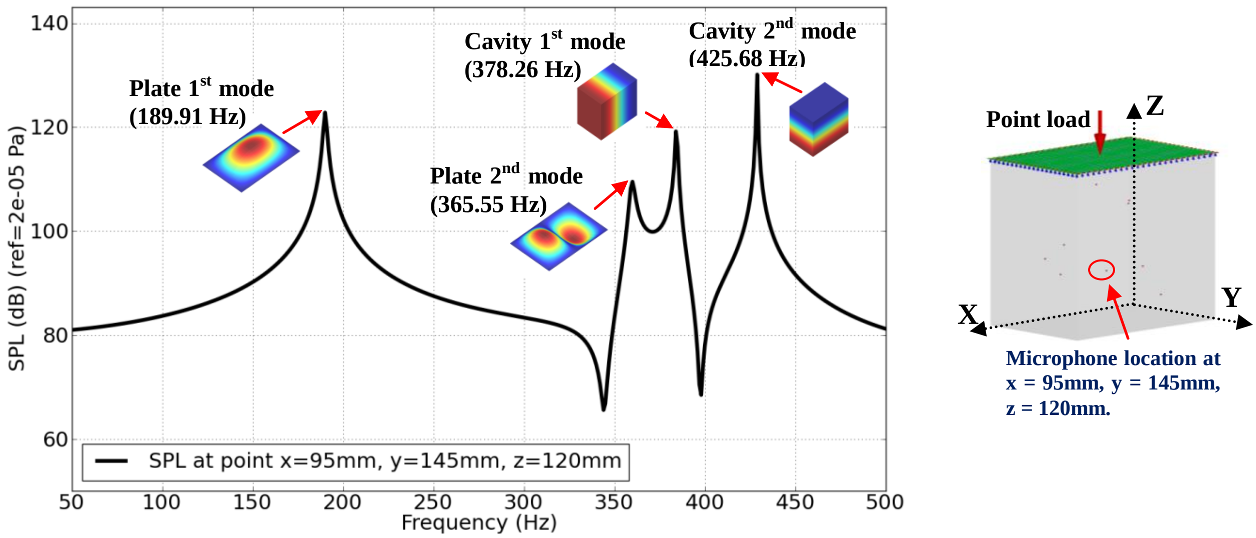


Figure 4. FE modelling results of SPL in terms of frequencies for the coupled system. A time-harmonic point load with unit amplitude (1N) was used. The output is for the point at x = 95 mm, y = 145 mm, z = 120 mm, which is the same microphone measurement location as in the experimental setup.

3.3 Effects of Porous Materials

The influence of porous materials on the sound pressure level of the cavity is characterised in this section. Figure 5 shows the comparison of FE modelling results of acoustic response at point x = 95 mm, y = 145 mm, z = 120 mm for the coupled plate and cavity system with and without porous materials. Figure 6 shows the comparison of measurement results of acoustic response at the same point (*x* = 95 mm, *y* = 145 mm, *z* = 120 mm) for the coupled plate and cavity system with and without porous materials. Both FE modelling and measurement results show that the porous materials provide a significant reduction of the radiated sound energy compared to when the system is not treated. The overall noise attenuation due to the porous materials in the frequency range from 50 to 500 Hz was nearly 10 dB. It should be noted that porous material 2 has a better acoustic performance at the resonant frequencies where it absorbs more sound energy. Incidentally, it can be concluded that porous material damps the plate vibration at the resonant frequencies and also reduces the SPL in the box cavity.

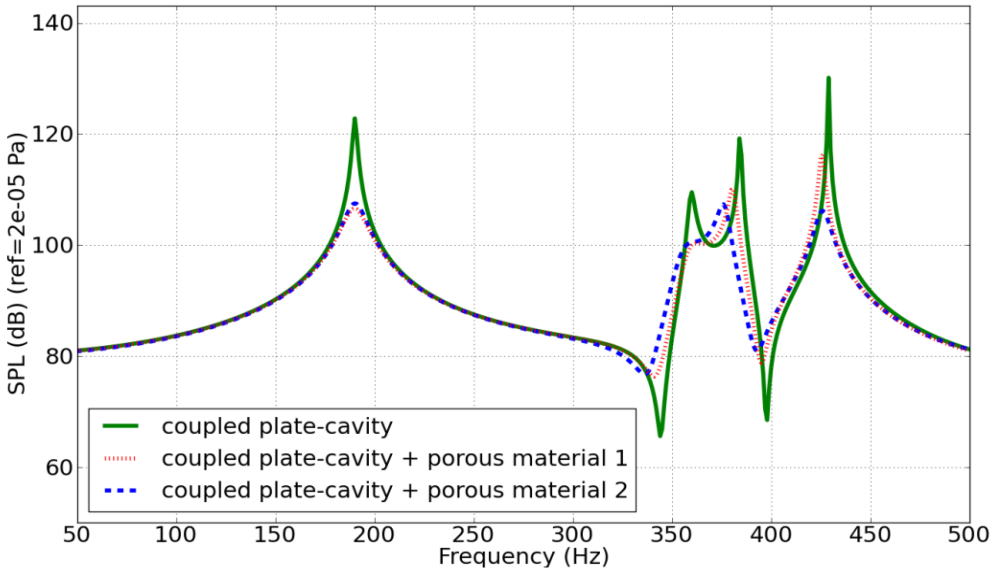


Figure 5. Comparison of FE modelling results of SPL as a function of frequency at the point *x* = 95 mm, *y* = 145 mm, *z* = 120 mm, for the coupled system, the coupled system with porous material 1 and the coupled system with porous material 2. A time-harmonic point load with unit amplitude (1N) was used.

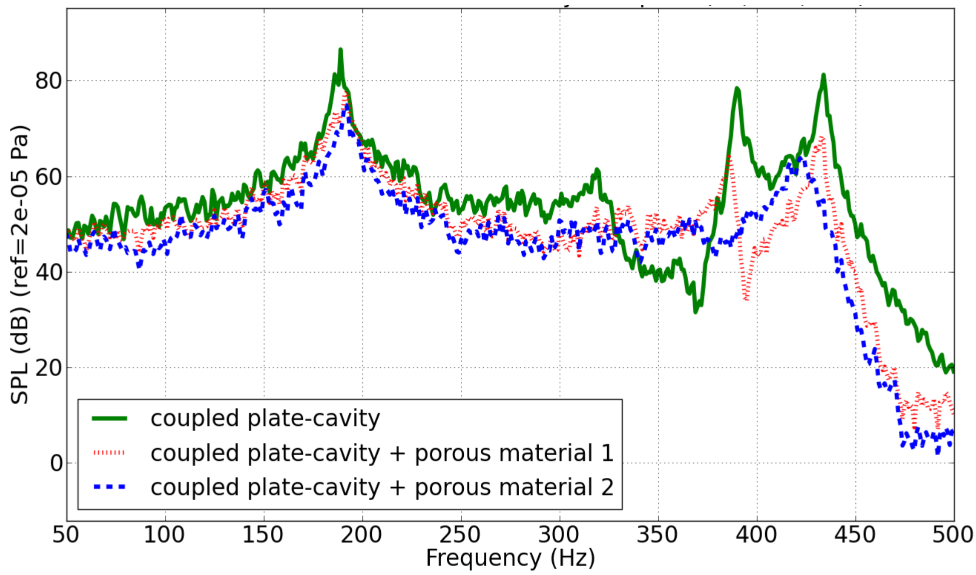
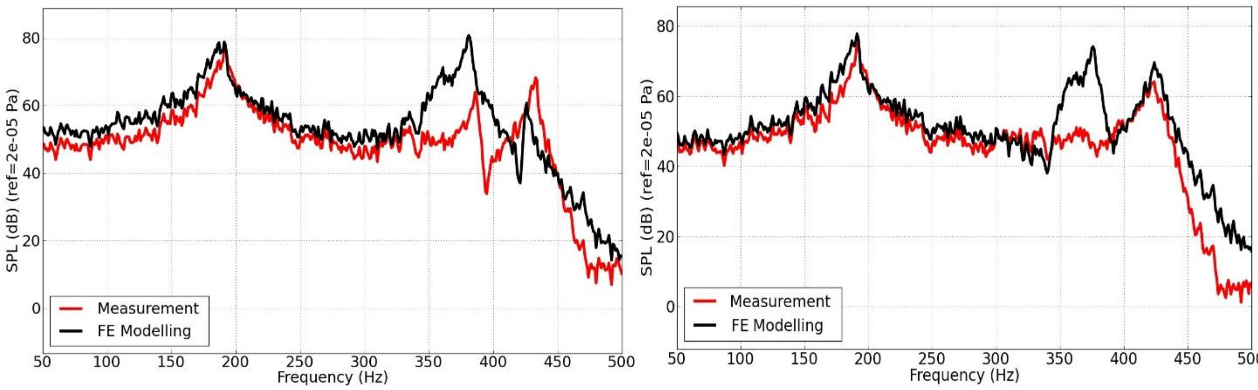


Figure 6. Comparison of measurement results of SPL as a function of frequency at the point *x* = 95 mm, *y* = 145 mm, *z* = 120 mm, for the coupled system, the coupled system with porous material 1 and the coupled system with porous material 2.

3.4 Correlation Results

The FE modelling results revealed that the acoustic FRF's with porous materials were well correlated in the range from 50 to 500 Hz. Figure 7 shows the comparison between the FE modelling and the measurement SPL results when porous material 1 and porous material 2 are attached to the plate. The time-harmonic point load force used for the FE modelling was the same data that was measured experimentally. Some discrepancies between the FE modelling results and the measured data were found at some frequencies. However, good agreement was obtained across the frequency range from 50 to 500 Hz. As a general trend, the FE modelling results can be used to predict the damping effect and noise attenuation of porous materials in the box cavity. However, at frequencies between 350 to 400 Hz, there are some differences between the measured and the FE modelling results. The second plate resonant mode was not captured by the measurement. Perhaps, this is because the detailed local interaction between the plate and the porous material was taken into account only in an approximate way. Therefore, there may be an under or an overestimation of the energy exchange between the aluminium plate and the porous material. On the other hand, measuring the physical properties of the porous material is difficult, because the porous materials are anisotropic. A more accurate and detailed investigation is underway.



(a) **(b)**

Figure 7. Comparison of the FE modelling results and the measurement results at point x = 95 mm, y = 145 mm, z = 120 mm before and after (a) porous material 1, (b) porous material 2 is added to the plate.

4. Conclusions

The main objective of this research was to investigate the effects of porous sound absorbing materials on the noise inside a box cavity. A Finite Element Method (FEM) for analysing the noise field in the box cavity with porous sound absorbing materials was used. A rectangular box cavity experiment was developed to validate the accuracy of the developed FE model. Good agreement has been obtained between the frequency responses, and resonant frequencies of the FE model and the corresponding experimental data. It can be concluded that the sound pressure level inside a box cavity containing a porous sound-absorbing material can be predicted with good accuracy by using the developed FE model. It was observed that attached porous sound-absorbing material also had some influence in shifting the plate resonances and reducing the level of vibration (FRF) in the plate. At low frequencies, the reduction of the plate vibration leads to a reduction in the sound pressure level in the cavity. As an important application of this research, the developed method can be used to optimise the vehicle interior trim in terms of vibration and acoustic performances, weight, and cost.

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