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Sound Transmission Through Multilayered Shells. Experiments and Modelling

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Summary

The use of porous materials to reduce sound transmission through vibrating structures was investigated in this study. Measurements carried out at LMA showed that adding a layer of melamine foam to a steel plate increases the sound transmission of the treated plate in some frequency bands. A model was therefore developed to predict this increase of the transmitted sound level, induced by covering a vibrating structure with a porous material. The existence of compressional waves, where the quarter wavelength is equal to the thickness of the added layer, was found responsible for the increase of sound transmission at the corresponding frequencies. Biot theory and the equivalent fluid model were used to model the acoustical behavior of the porous material. With the model based on Biot theory, which involves heavy data processing, accurate predictions were obtained. To overcome the drawbacks of the Biot model, a simpler procedure was developed for predicting the frequency range in which transmission increases. Based on the surface admittance at normal incidence, a value which can be easily determined experimentally using an impedance tube, the proposed procedure efficiently predicts the frequency range in which higher sound transmission occurs.

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1. Introduction

One of the main factors contributing to noise pollution in urban areas and transport vehicles is sound transmission. The characteristics of the sound sources are often not very well known, nor is area to which the sound is transmitted. This makes active sound control [1] difficult to apply and to sustain. One of the most feasible solutions consists of using absorbing materials to reduce the sound levels transmitted.

The most commonly applied technique for reducing the sound levels transmitted by walls consists in adding one or more sound absorbing layers, for instance consisting of viscoelastic or porous materials or synthetic foams. To be able to apply these additional materials as efficiently as possible in terms of the cost and added mass, it is necessary to know how these layers will affect the sound transmission. Detailed information about their effects makes it possible to predict the amounts of absorbing material required and where to place it, and even to determine the optimum characteristics of the material in advance.

In the present study, we investigated the effects of absorbing materials made of porous media on the sound transmission. Many previous studies have dealt with materials of this kind, which are frequently used in practical applications, and most of them have focused on sound absorption aspects. Here we studied the influence of a layer

of porous medium attached to a vibrating metal plate on the sound transmission of such a multilayered shell. It has been observed experimentally that adding an absorbing layer can increase the transmitted sound level. The aim of this study was therefore to develop a model predicting the frequency bands in which this undesirable effect occurs. This predicting model is based on the results of an impedance tube experiment and can therefore be used in industrial contexts.

This paper is organized as follows. In section two, the results of the experiments carried out using two different porous materials are presented. A method of predicting regions of higher sound transmission is presented in section three. This method was tested numerically and by comparing the results obtained using the prediction model with the experimental data.

2. Measurements

Sound transmission through a metal plate lined with a layer of porous material was measured at the Laboratoire de Mécanique et d'Acoustique (LMA) in Marseille. A large anechoic chamber and a smaller hemi-anechoic chamber are connected by a window that can be used to clamp the object under investigation. The experimental set-up used for this purpose is shown in Fig.(1). A loudspeaker emitting white noise was used as an acoustical sound source in the anechoic chamber. The transmitted sound field was recorded in the hemi-anechoic chamber using two microphones, placed at different distances

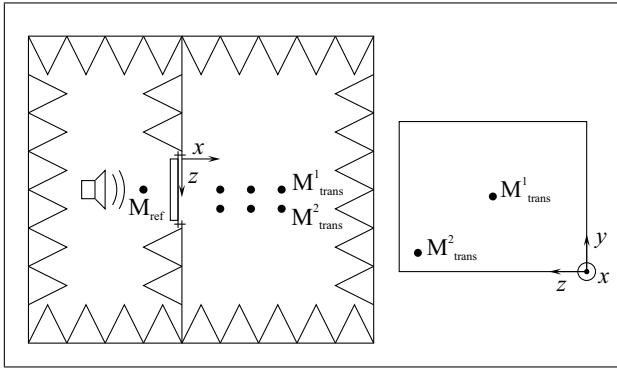


Figure 1. Experimental set-up (top) and position of microphones (bottom). Microphone positions are: $z = 450$ mm and $y = 400$ mm for M^1_{trans} , $z = 610$ mm and $y = 210$ mm for M^2_{trans} .

$x_i \in [.02, .04, .1, .2, .3, .5, 1.0, 2.0]$ m in front of the plate (see Figure 1). Three different plates were used:

1. a 2 mm thick steel plate,
2. a 2 mm thick steel plate covered with a 30 mm melamine layer and
3. a 2 mm thick steel plate covered with a 20 mm felt layer, glued with epoxy.

Plates two and three were mounted so that the porous layer faced the sound source.

The steel plates measured 890 mm×755 mm. The absorbing layers were slightly smaller so that they would not be clamped too. The frequency range in which the sound transmission analysis carried out, which was set at 100 to 3200 Hz, was split into two bands, one from 100 to 800 Hz with a frequency resolution of 0.25 Hz and the second from 800 to 3200 Hz with a frequency resolution of 1 Hz.

The sound transmission through the steel plate covered with a 20 mm layer of felt was as expected. The transmitted sound power measured at distance $x_i = 4$ cm at the microphone position M^1_{trans} is shown in Figure 2. At low frequencies (<200 Hz) the porous layer had no effect on the transmitted noise level. The resonance frequencies decreased slightly, due to the added mass. From 200 Hz onward, the resonance peaks were greatly attenuated, and hence the transmitted sound level was also attenuated. The transmission behavior of the steel plate covered with a 30 mm melamine layer was very different. The results of the measurements, at the same microphone position as for the felt layer, are given in Figure 3. Apart from the resonance peak at 390 Hz, the added layer of melamine was found to have no effect on the transmitted sound power up to a frequency of 600 Hz. The material was simply too light and did not introduce enough damping to affect the vibration response of the plate. At frequencies above 600 Hz, a slight decrease in the transmitted sound power was observed. But in the 2200 Hz to 2700 Hz range, greater sound transmission was found to occur with the porous layer rather than without it (see Figure 3). This effect was observed at all distances x_i in front of the plate and is especially shown for the central microphone positions M^1_{trans} . Adding the melamine layer to the vibrating

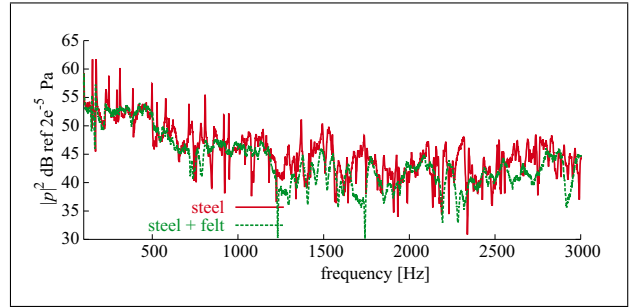


Figure 2. Sound power transmitted through a steel plate as compared to a steel plate lined with a 20 mm felt layer (experimental data). Frequency range: 100–3000 Hz.

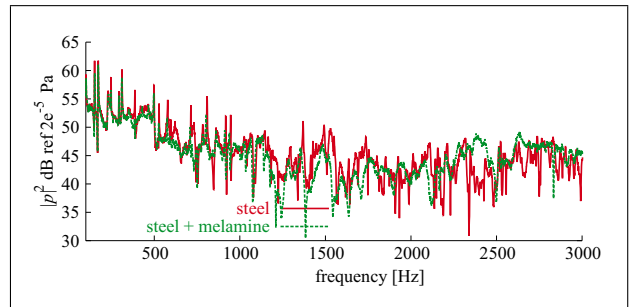


Figure 3. Sound power transmitted through a steel plate as compared to a steel plate lined with a 30 mm of melamine layer (experimental data). Frequency range: 100–3000 Hz.

structure therefore resulted in a higher sound transmission in a specific frequency band. Although the observed increase (≈ 3 dB) in the sound transmission of the lined plate as compared to the untreated plate may not be considered much, but, if comparing the measured transmitted sound power of the two plates lined with different materials, a reduction of the transmitted sound power of about 5 dB when using a layer of felt instead of melamine can be obtained (see Figure 4 and Figure 5). A possible 5 dB reduction of the transmitted noise makes it worth choosing the most suitable absorbing material. In the following section, we will present a tool that makes it possible to predict the frequency bands in which higher sound transmission will occur when a vibrating structure is lined with a layer of porous material.

3. Numerical simulation

In this section, we respond to the effect of a higher sound transmission through a steel plate lined with a porous layer. A similar effect has previously been observed in the case of absorption [2]. The authors observed a fast variation of the surface impedance around certain frequencies. These changes can be explained by applying the Biot theory [3] to describe the porous material. This theory predicts the existence of two compressional waves and one shear wave propagating in a porous material. Whenever the quarter wavelength of one of the compressional waves is equal to the thickness of the porous layer, resonance

Table I. Material parameters of the four porous materials used: Shear module N [N/m²], Poisson ratio ν , density ρ [kg/m³], thickness d [mm], flow resistivity σ [Ns/m⁴], porosity Φ , tortuosity α_∞ , characteristic dimensions Λ and Λ' [m].

Material	N	ν	ρ	d	σ	Φ	α_∞	Λ	Λ'
steel	6.92e+10	.30	8.4e3	2					
melamine	5.55e+4	.44	8.35	30	1.2e4	.99	1.0	0.80e-4	0.28e-3
felt	1.85e+4	.35	50.0	20	0.8e5	.80	1.0	0.26e-4	0.28e-4
foam A	1.8e+5	.40	30	30	5.5e4	.93	3.2	0.27e-4	0.31e-3
foam B	5.5e+4	.35	30	30	5.5e4	.93	3.2	0.27e-4	0.31e-3

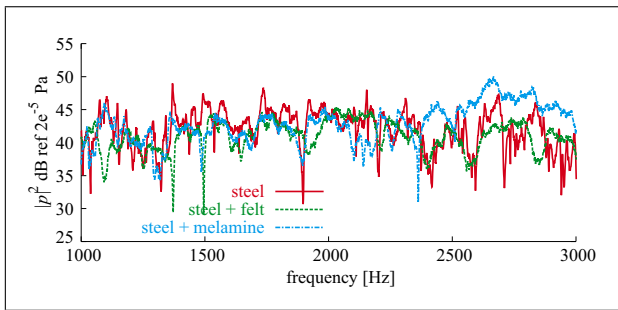


Figure 4. Sound power transmitted through a steel plate as compared to a steel plate lined with a 30 mm melamine layer and a steel plate lined with a 20 mm felt layer (experimental data). Frequency range: 1000–3200 Hz.

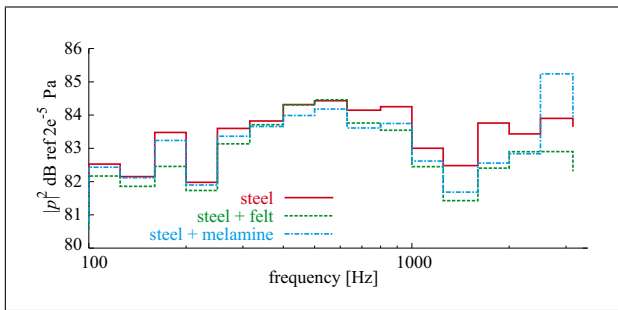


Figure 5. Sound power transmitted through a steel plate as compared to a steel plate lined with a 30 mm melamine layer and a steel plate lined with a 20 mm felt layer (experimental data). Frequency range: 100–3200 Hz (third octaves).

effects will occur which causes the variation of surface impedance. This effect can be used to advantage to design impedance matching layers giving optimum coupling between a transducer and the surrounding medium. This optimum coupling, or energy exchange, between the transducer, in our case the vibrating plate, and the surrounding fluid results in the observed higher sound transmission.

The application of Biot theory requires having a fairly detailed information about the porous material in question, especially about the parameters that need to be fed into the numerical model describing the material. Several experiments have been proposed so far to determine these parameters of the Biot model [4, 5, 6, 7, 8, 9], but most of them are difficult both to set-up and to carry out, especially under industrial conditions. We therefore looked for a way of predicting the regions of higher sound trans-

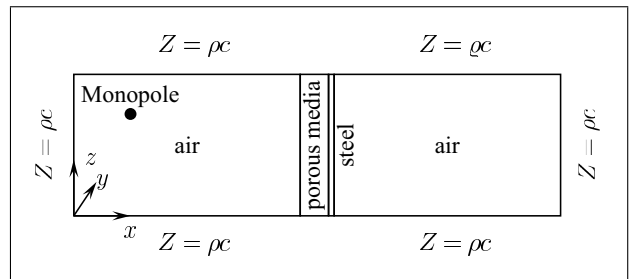


Figure 6. Test configuration used to simulate sound transmission through a lined steel plate.

mission, since this approach is more inhere with industrial requirements.

The basis of our prediction was the following simple equation, which relates the acoustic pressure p to the normal surface velocity v

$$v = \frac{1}{Z} p = Y p \tag{1}$$

via the normal surface impedance Z or the normal surface admittance Y of the material under investigation. Assuming that the acoustic pressure p on the side facing the excitation is practically unaffected by the presence of the porous layer, the particle velocity v can be said to depend on the normal surface admittance of the porous layer. Hence, the greater the value of Y the greater the velocity v will be. As we force continuity of the velocity at the air/plate interface, the plate must vibrate more strongly and radiates a higher sound level, thus causing higher sound transmission. The advantage of the above procedure is that the normal surface admittance Y can be easily determined by performing the impedance tube experiment [10, 11, 12, 13].

To test the hypothesis that a greater Y value will cause higher sound transmission with numerical examples, the sound transmission through a clamped steel plate measuring 400 mm × 250 mm × 2 mm and lined with four different materials was calculated. The set-up of the used configuration is shown in Figure 6. To speed up the numerical simulations, the plate used for the numerical experiments was slightly smaller than that used in section 2. A plate of the same size as that used in the experiments described in section 2 will be studied at the end of this section. The material properties of the porous materials used here are shown in Table I. The properties of the material named

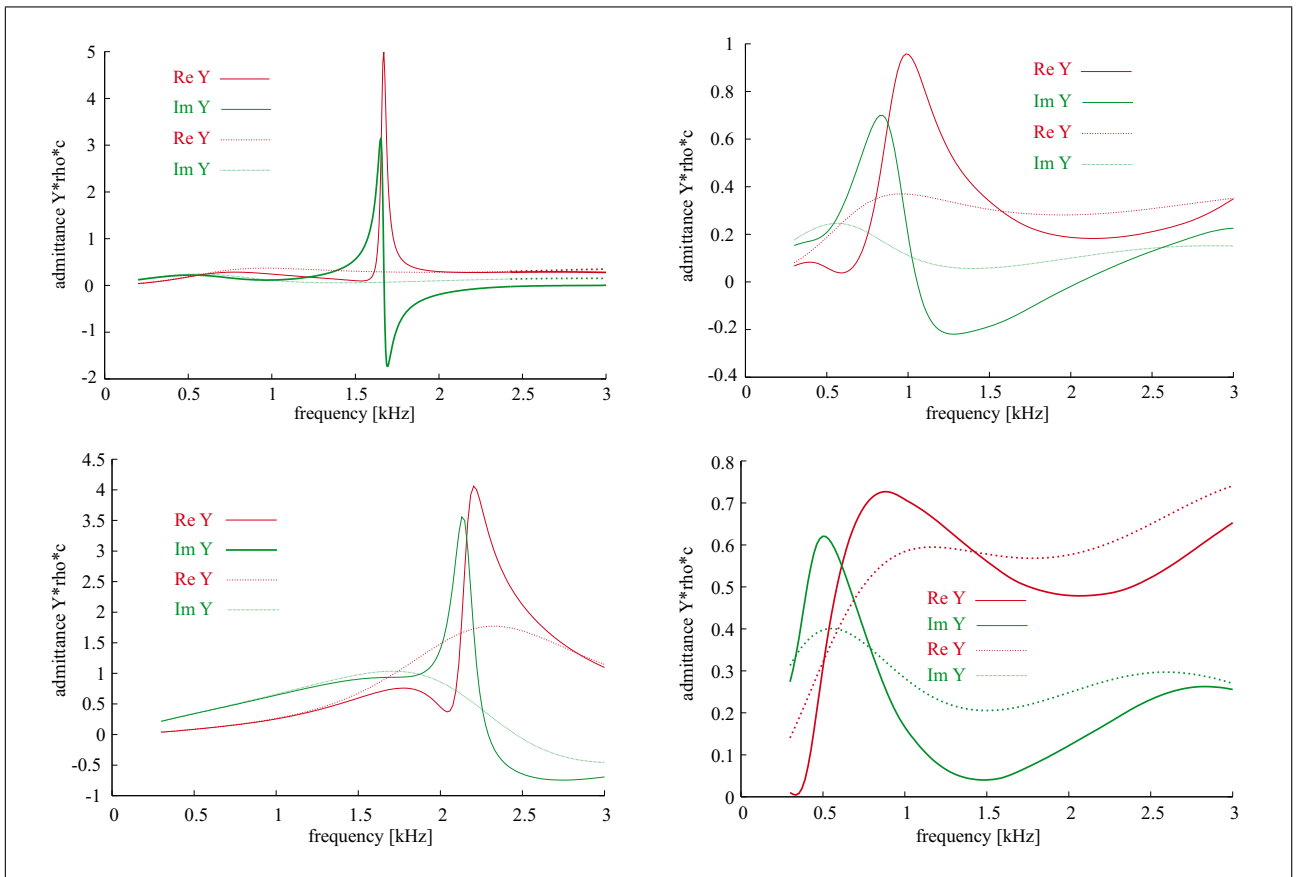


Figure 7. Surface impedance at normal incidence in the case of a 30 mm layer of foam A (upper left), foam B (upper right), melamine foam (lower left) and felt (lower right) calculated using the complete Biot model (solid line) and the rigid frame model (dashed line).

foam A were based on [14, Tab. 6.2, page 141]. The material named foam B was generated from the foam A parameters by modifying its Young’s modulus. The Biot parameters of melamine were taken from [6]. Using these parameters, the surface admittance Y was evaluated numerically (see [14] for details) to simulate the results of an impedance tube experiment. The values calculated for Y are given in Figure 7. In addition to the values obtained using the complete Biot theory (solid line) the values obtained assuming a rigid frame (dashed line) are given in Figure 7. Considering the frame of the porous material to be motionless is often referred to as the equivalent fluid model.

Applying the above mentioned hypothesis gives for each material a frequency range where higher sound transmission can be expected (see Table II). To determine a frequency range of higher sound transmission for a specific lining, frequency ranges where the measured value of the surface admittance Y shows peaks or the Z value tends to zero must be detected. These frequency ranges could be clearly identified for foam A at around 1.6 kHz and for the melamine layer at around 2.1 kHz. They were less clearly expressed for foam B and nearly indistinguishable for the felt layer.

The test configuration shown in Figure 6 was used to verify if higher sound transmission will occur in the frequency ranges given in Table II. The finite element

Table II. Frequency ranges where a higher sound transmission is expected.

Material	Frequency range
foam A	1500–1800 Hz
foam B	800–1500 Hz
melamine	2000–2500 Hz
felt	500–1000 Hz

method was used to simulate the transmission of sound through the lined steel plate. The clamped plate measuring 400 mm×250 mm×2 mm was discretized using 30×20×1 quadratic volume elements. The porous layer was also discretized using 30×20×1 quadratic volume elements. The Biot model in its $\{u, p\}$ formulation [15] was used to model the acoustical behavior of the porous layer. The two air layers were meshed with 30×20×10 elements with linear basis functions in y - and z -direction and quadratic basis functions in x -direction. The characteristic impedance of air was used as boundary condition for the air layers. Simulations performed with a finer mesh showed that the finite element model of the clamped plate is valid up to at least 3 kHz. A monopole source was used to simulate the acoustical source. Again, the porous materials were modeled using the complete Biot theory [16] and the model of a porous material having a rigid frame. The assumption of

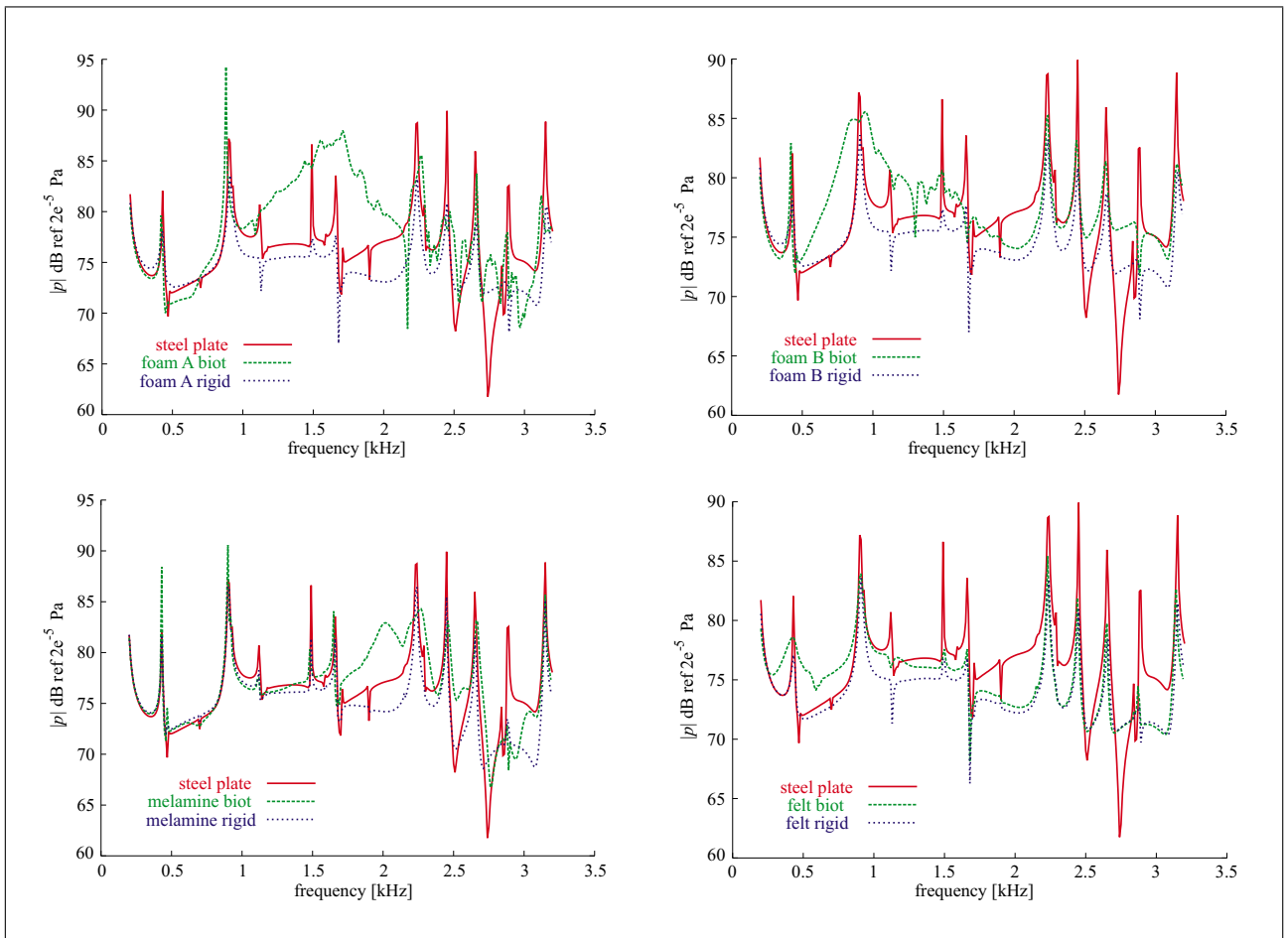


Figure 8. Sound power transmitted through a steel plate lined with a 30 mm layer of foam A (upper left), foam B (upper right), melamine foam (lower left) and felt (lower right) calculated using the complete Biot model (labeled 'biot') and the rigid frame model (labeled 'rigid').

a motionless frame reduces the number of unknown quantities at each field point from four, the sound pressure and three displacements, for the complete Biot model to only one, the sound pressure, when assuming a rigid frame. This model, therefore requires much fewer computational resources and is thus often used rather than the complete Biot model.

Using the numerical model described above, the transmitted sound power was evaluated 40 mm from the center of the plate. The results obtained using the four different materials (see Table I) are shown in Figure 8. The sound power transmitted through the steel plate lined with each of the materials—determined using the two different material models—is compared in this figure with the sound power transmitted through the untreated steel plate. Numerical results show that the higher sound transmission is observed very close to the frequency ranges given in Table II only when the complete Biot theory was used. The effect of higher transmission of sound, caused by resonances of the frame cannot be reproduced with the simpler model as it assumes a motionless frame. Hence, reliable numerical results in frequency ranges close to frame resonances can be obtained only by using the complete Biot model, which is much more costly from the numeri-

cal point of view. But in frequency ranges where no such resonances occur, the less resource demanding rigid frame model yields reliable results.

The above presented method of predicting the frequency ranges in which higher sound transmission occurs was based only on the properties of the porous material and not on the size of the vibrating structure. Therefore, a prediction must be also possible for the experiment described in section 2. The predicted frequency range of the melamine foam used for the actual measurements ranged from 2.0 to 2.5 kHz (see Table II). The measurements (see Figure 3 and Figure 4) gave higher sound transmission in a slightly higher frequency range. Upon simulating this experiment, in the case of a plate 890 mm×755 mm in size and taking the same set-up as that depicted in Figure 6, higher sound transmission was obtained in the predicted frequency range, which again is slightly lower than the measured one (see Figure 9 and Figure 10). These differences may be attributable to uncertainties of the material properties, especially the stiffness of the frame, which strongly influences the frequency range. This effect becomes obvious if one compares the results obtained on foam A and foam B. The only difference between these two materials was in their mechanical properties. The less

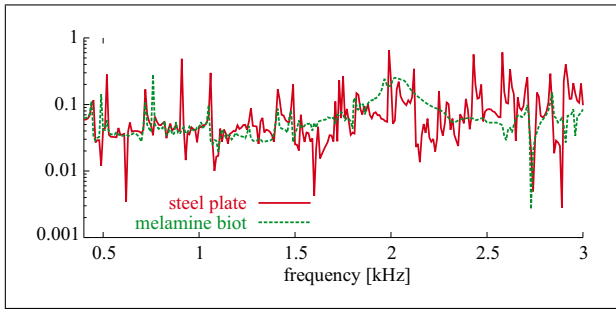


Figure 9. Sound power transmitted through a steel plate as compared to a steel plate lined with a 30 mm melamine layer. Results of the simulation of the measurements carried out at LMA. Frequency range: 400–3000 Hz.

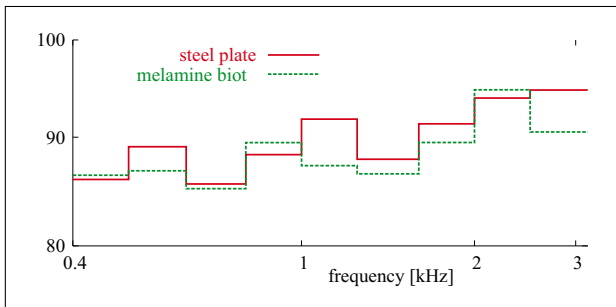


Figure 10. Sound power transmitted through a steel plate as compared to a steel plate lined with a 30 mm melamine layer. Results of the simulation of the measurements carried out at LMA. Frequency range: 400–3000 Hz (third octaves).

stiff frame of foam B resulted in a shift of the frequency range of higher sound transmission toward lower frequencies.

4. Conclusions

Previous measurements carried out at LMA have shown that adding a porous layer onto a steel plate can increase the transmitted sound power in a specific frequency band. The Biot theory for porous materials shows that this effect occurs around frequencies where the quarter wavelength of the compressional wave in the porous media is equal to the thickness of the porous layer. Based on the presented data it was shown that the frequency ranges at which higher sound transmission occur, can be predicted from the value of the normal surface admittance of the porous layer, which can be determined by performing impedance tube experiments.

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