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Determination of sound insulation properties of homogeneous baffles using Finite Element Method

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Abstract The basic parameter of materials used in constructional solutions of anti-noise protection, is sound insulation, which can be determined in laboratory conditions and also using theoretical models. The use of numerical methods in the form of the Finite Element Method to calculate the mechanical impedance of a baffle and then the sound insulation of homogeneous baffles was presented in the article. A 1 mm thick steel plate with a square, rectangular and round shape was analyzed. The boundary conditions for simply supported and clamped plate were taken into account in the numerical calculations. The results of the calculations were compared to both the commonly used the mass law and to the experimental tests. These analyzes will be the starting point for analyzes of multi-layer baffles, for which it is no longer possible to apply the mass law.

Keywords: sound insulation, sound transmission loss, Finite Element Method, homogeneous baffles.

1. Introduction

The basic parameter of a baffle, important in the construction of noise protection such as a screen or sound-absorbing and insulating enclosures, is sound insulation [1-4]. The effectiveness of a single or multi-layer baffle is determined by measurement in laboratory conditions [2], by determining the spectral characteristics of airborne sound insulation *R*. The sound insulation *R* can also be determined using theoretical calculation models [5-9].

This article presents an attempt to use the Finite Element Method to determine the mechanical impedance of the baffle, which was then used to calculate the sound insulation of homogeneous baffles, which were steel plates with a thickness of 1 mm, of various shapes and external dimensions. The concept of mechanical impedance, used in the article, refers to a baffle, in contrast to the also used acoustic impedance, which applies to air. The boundary conditions for simply supported and clamped plate were taken into account in the numerical calculations. The obtained results of the sound insulation *R* were compared to both the calculations from the mass law model and to the experimental tests. The approach shown for a single baffle, made of homogeneous materials, e.g., steel, glass, plastics - the calculation of sound insulation on the basis of mechanical impedance, will be used in further research for multi-layer baffles.

2. Measurement methods and material specimens

The dimensions of the tested steel plates were appropriate to the conditions of a given laboratory environment. In the impedance tube, a sample in the form of a steel disc with a diameter of ϕ = 34.9 mm was tested, and in the laboratory for tests of sound insulation in interconnected reverberation chambers, plates with dimensions of 1 × 2 m and 0.7 × 0.7 m were tested.

The test stand enabling the measurement of the normal incidence sound transmission loss included a Mecanum Inc. impedance tube with a loudspeaker, an SMSL SA-36A Pro amplifier, type 378A14 PCB measuring microphones, a Siemens LMS SCADAS Mobile analyzer and a computer with Simcenter Testlab software. The internal diameter of the tube was 34.9 mm, which enabled measurements in two frequency ranges: 50 Hz - 2400 Hz and 119 – 5700 Hz while maintaining the distance between the microphone holders, which was 65 and 29 mm, respectively.

In the case of tests of the sound insulation *R*, measurements are performed in the conditions of a diffuse sound field. Depending on the specifics of a given research laboratory, acoustic measurements are usually carried out for a specific size of the baffle, resulting from the size of the measurement window. In the Department of Mechanics and Vibroacoustics of the AGH, as part of many years of research [4], a blanking barrier was developed, installed in the measuring window intended for testing samples with dimensions of

1 × 2 m, which allows the testing of smaller samples, amounting to 0.7 × 0.7 m. In such a case, the tested plate rests on one side on a flange with a width of 10 mm, lined with a rubber gasket, and on the other side is pressed using a pressure system consisting of an aluminum frame with a width of 10 mm, lined with a rubber gasket and a set of 16 screws.

A sample with dimensions of 1 × 2 m was attached in the traditional way, in the same way as the sound insulation tests of baffles in a laboratory are carried out. On one side, the plate rested against a pressure frame, and on the other, it was sealed with a permanently plastic mastic.

Normal incidence sound transmission loss tests in an impedance tube are usually performed with samples made of soft materials such as rock wool or polyurethane foam placed in the sample container. In the tested case, where it is a hard material, a thin steel disc with a diameter corresponding to the inner diameter of the tube was held in a vertical position in the sample container by the use of an adhesive paper tape. An attempt to use such a method of fixing the sample has brought a satisfactory result.

Table 1 shows the material data which were then used for numerical calculations.

Table 1. Material data of steel plate.

Quantity	Symbol	Unit	Value
Density	ρ	kg/m^3	7850
Thickness	h	mm	1
Young's modulus	E	GPa	207
Poisson's ratio	υ	-	0.3
Loss factor	η	-	0.01

3. Calculations of the sound insulation

The phenomenon of the transmission (penetration) of acoustic energy through the baffle is very complex. The acoustic wave penetrates the baffle mainly due to the vibrations of the baffle. The vast majority of theoretical models (including the ones described in this paper) assume that the acoustic wave hits the baffle and stimulates it to mechanical vibrations. These vibrations, in turn, are the source of acoustic waves on both sides of the baffle, i.e. the reflected wave (remaining on the source side) and the wave passing (transmitted) to the protected medium (outside the insulating enclosure). These models ignore the phenomenon of couplings between the gaseous medium (acoustic wave) and the solid medium (baffle), on both sides of the baffle.

The specific sound insulation R of a baffle is defined as the ratio of the total energy of the acoustic wave incident on the baffle (incident wave) E_1 to the total acoustic energy transmitted through the baffle (transmitted wave) E_2 and is expressed by the formula [10,11]:

$$R = 10\log_{10}\frac{E_1}{E_2}. \tag{1}$$

With the condition assuming that the baffle surface areas on the incident wave side and on the transmitted wave side are the same, the acoustic wave intensity I can be replaced in place of the acoustic energy. The surface evenness condition is always met, as it is difficult to imagine that any enclosure would be constructed in such a way that one of its walls had a different surface on the inside (from the machine side) and on the outside (protected area). In this case, the sound insulation is expressed by the formula [10,11]:

$$R = -10\log_{10}\tau, \tag{2}$$

where $\tau = \frac{I_2}{I_1}$ is the transmission coefficient, I_1 is the sound intensity of the wave incident on the baffle, I_2 is the sound intensity of the wave transmitted through the baffle.

Another assumption of each of the theoretical models is the assumption of a large distance between the source and the baffle, and thus the assumption that the acoustic wave is a plane wave. From this assumption, there is a relationship between the sound pressure and the particle velocity. The dependence for a plane wave is expressed as:

$$p = \rho_0 c v, \tag{3}$$

where p is the sound pressure, ρ_0 is the air density, c is the speed of sound and v is the velocity of the acoustic particle.

Using the relationship (3), the transmission coefficient τ from equation (2) can be expressed by the relationship:

$$\tau = \frac{I_2}{I_1} = \frac{p_2 v_2}{p_1 v_1} = \frac{p_2^2}{p_1^2}. \tag{4}$$

Assuming the periodic waveforms of each wave: incident p_1, transmitted p_2 and reflected p_3, their waveform can be written as:

$$p_1 = Ae^{i(\omega t - kx)},$$
$$p_2 = Be^{i(\omega t - kx)}, \tag{5}$$
$$p_3 = Ce^{i(\omega t + kx)}.$$

The second law of Newton's dynamics shows that the pressure difference on both sides of the baffle, acting on a unit of the baffle area, is equal to the product of the baffle's impedance Z and the average velocity v_p of the analyzed unit of the baffle area. With the adopted designations, the dynamic equation of motion can be written in the form:

$$\Delta p = (A + B - C) = Zv_p. \tag{6}$$

Due to the fact that the velocities of acoustic particles on both surfaces of the baffle are equal to the speeds of the baffle itself, the relationship between them can be described as:

$$v_p = v_1 - v_3 = v_2. \tag{7}$$

Using the equations (3), (6) and (7), after several transformations, the transmission coefficient is expressed by the relationship:

$$\tau_\theta = \frac{p_2^2}{p_1^2} = \left[1 + \frac{Zcos(\theta)}{2\rho_0 c}\right]^{-2}. \tag{8}$$

In this way, after several transformations, it was shown that the value of the sound insulation coefficient sought is a function of the mechanical impedance of the baffle.

3.1. The mass law

The mass law, often used in practice and widely described in the literature, results from the simplest baffle model, namely the rigid body model. In this case, the impedance of such a baffle is expressed by the relationship:

$$Z = i\omega\mu, \tag{9}$$

where μ is the unit mass of the baffle area (in the literature, the term surface mass is also used).

In the mass law it is assumed that $i\omega\mu/2\rho_0 c \ll 1$, which leads to the simplification of relation (8) to the form:

$$\tau_\theta = \left[\frac{2\rho_0 c}{\omega\mu\cos(\theta)}\right]^2, \tag{10}$$

where τ_θ is the transmission coefficient of the baffle for the incident acoustic wave at the angle θ in relation toe the normal to the baffle.

In this case, the specific sound insulation of the baffle is determined from the equation:

$$R_\theta = 20\log_{10}\frac{\omega\mu\cos(\theta)}{2\rho_0 c}. \tag{11}$$

It is the baffle sound insulation in the case of an incident wave at the angle θ (angle measured to the normal to the baffle). In the case of the sound insulation, it is always interesting what it is in the case of a diffuse field (waves are falling from all directions). In the case of the mass law, it is assumed that it is enough to determine the sound insulation value for θ=0° and subtract 5 dB from the result, that is:

$$R = 20\log_{10}\frac{\omega\mu}{2\rho_0 c} - 5. \tag{12}$$

There is also a relationship in the literature [10,11]:

$$R = 20\log_{10}(f\mu) - 47.5, \tag{13}$$

where f is the frequency of the acoustic wave (excitation frequency) and μ is the unit mass of the baffle area.

The equation (13) results directly from the equation (12) after accepting and substituting the value of the acoustic impedance of the air, i.e. $\rho_0 c = 415$ Ns/m³.

3.2. Finite Element Method (FEM)

In the work, the authors propose to use FEM to determine the mechanical impedance of the baffle. The analyzed model consists in loading the baffle (modeled as a bending plate) with a uniform surface load and determining the value of the average vibration velocity determined in all nodes of the mesh. In this paper, the authors limited their considerations to homogeneous plates and compared the results of the calculations with the commonly accepted (described above) law of mass. These analyzes will be the starting point for analyzes of multi-layer baffles, for which it is no longer possible to apply the mass law.

4. Verification of calculation models

The results of the sound insulation calculations using the Finite Element Method were related to the law of mass and experimental tests. For the boundary conditions of simply supported and clamped plate, the results were shown in Figs. 1–3 and 4–6, respectively.

When comparing the calculation results shown in Figs. 1–6, it should be taken into account that the results of the experimental tests concern the centre frequencies of 1/3 octave bands, while the sound insulation R using the Finite Element Method was calculated separately for each specific excitation frequency. This may result in some discrepancies.

From the results shown in Fig. 1 and 4, a certain convergence of the FEM method with the results from the impedance tube can be seen, while in the case of simply supported plate, these results are either underestimated (200 Hz <f <800 Hz) or slightly overestimated (f <200 Hz), but in both cases with small discrepancies (Fig. 1). However, for the clamped plate boundary conditions and frequencies below 1.6 kHz (Fig. 4), the results are overestimated with significant discrepancies. For the round sample, which has the smallest dimensions among the analyzed plates, the law of mass does not apply at frequencies below 1 kHz, both for the both case of boundary conditions. Large divergences in the obtained results of calculations of sound insulation R using FEM for f<800 Hz (Fig. 1 and 4) in relation to the mass law result from the fact that the mass law is used for excitations with frequencies higher than frequency of the first baffle resonance.

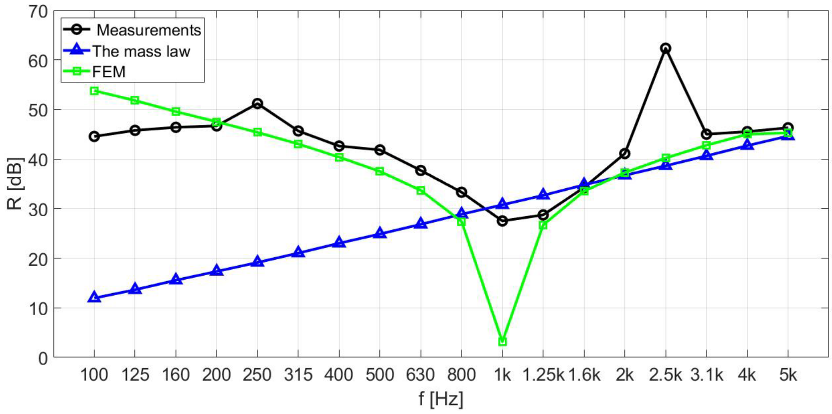

Figure 1. The sound insulation of a round steel plate with a thickness of 1 mm and a diameter of ø = 34.9 mm, determined from experimental tests and calculated using the mass law and the Finite Element Method for boundary conditions of a simply supported plate.

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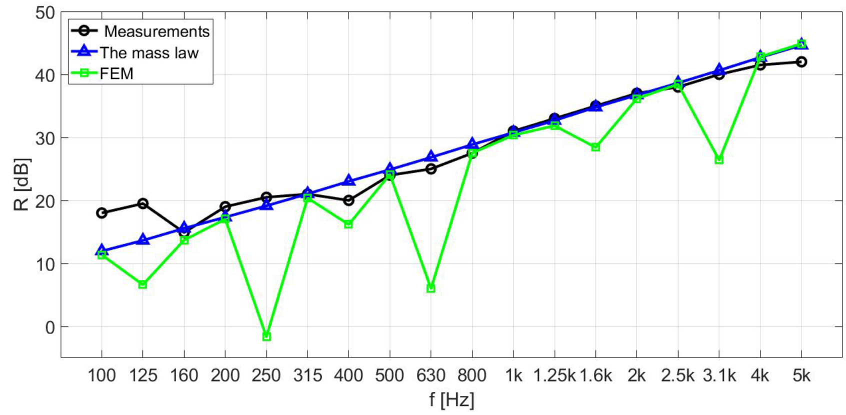

Figure 2. The sound insulation of a square steel plate 1 mm thick and a side length of 0.7 m, determined from experimental tests [2] and calculated using the mass law and the Finite Element Method for boundary conditions of a simply supported plate.

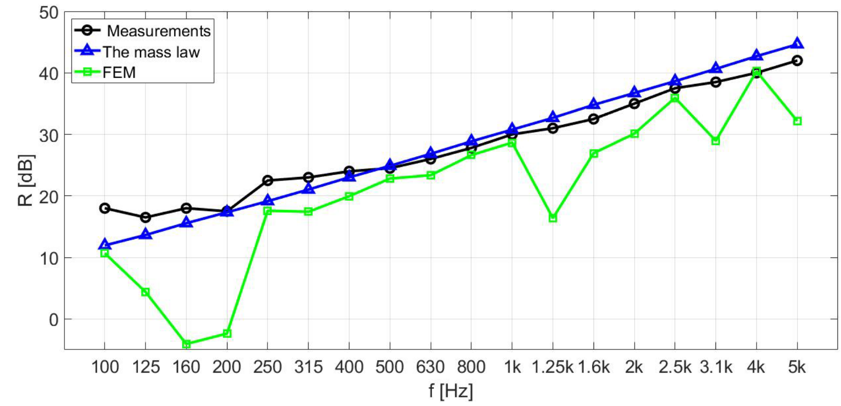

Figure 3. The sound insulation of a rectangular steel plate 1 mm thick and sides 1 × 2 m long, determined from experimental tests [2] and calculated using the mass law and the Finite Element Method for boundary conditions of a simply supported plate.

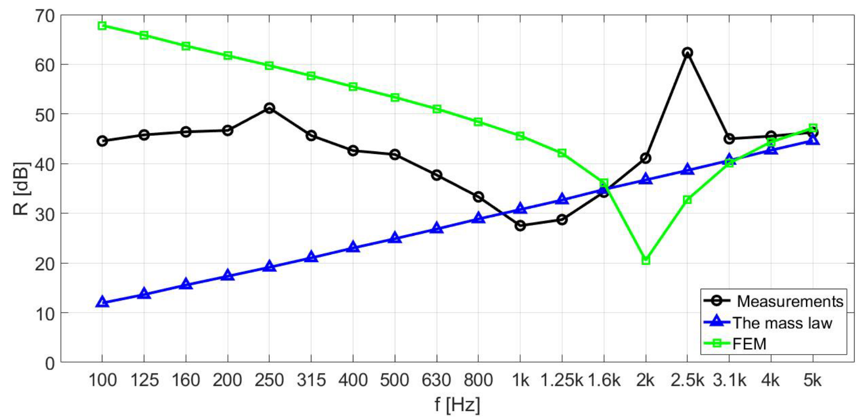

Figure 4. The sound insulation of a round steel plate with a thickness of 1 mm and a diameter of ø = 34.9 mm, determined from experimental tests and calculated using the mass law and the Finite Element Method for boundary conditions of a clamped plate.

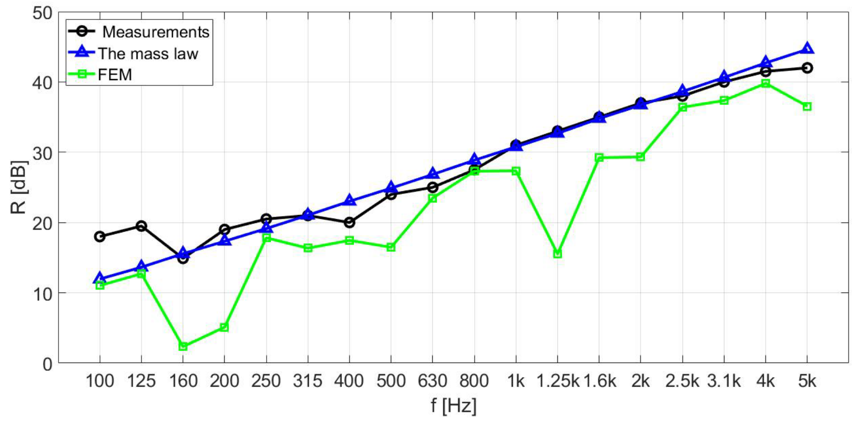

Figure 5. The sound insulation of a square steel plate 1 mm thick and a side length of 0.7 m, determined from experimental tests [2] and calculated using the mass law and the Finite Element Method for boundary conditions of a clamped plate.

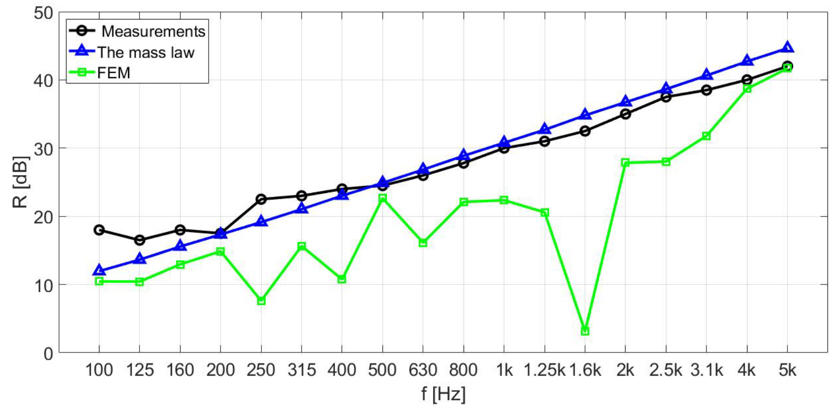

Figure 6. The sound insulation of a rectangular steel plate 1 mm thick and sides 1 × 2 m long, determined from experimental tests [2] and calculated using the mass law and the Finite Element Method for boundary conditions of a clamped plate.

Figures 2, 3, 5 and 6 show that the well-known model of the law of mass for the case of the tested steel plate with a square (0.7 × 0.7 m) and rectangular (1 × 2 m) shape satisfactorily reflects the results of experimental tests carried out in a set of coupled reverberation chambers. The smallest of the four analyzed variants of discrepancy with the use of the Finite Element Method in relation to the experimental tests carried out in reverberation chambers apply to a 0.7 × 0.7 m square plate for the boundary conditions of a simply supported plate. The results concerning the slabs with the largest dimensions, amounting to 1 × 2 m, showed the greatest discrepancies in relation to the experimental tests, both for the conditions of a simply supported (Fig. 3) and clamped plate (Fig. 6). The sound insulation curves R obtained using FEM, shown in Fig. 1-3 and Fig. 4-6, are mainly related to the external dimensions of the analyzed baffles. For baffles with the smallest dimensions, the insulation curves have a characteristic decrease to a certain frequency, which is 1 kHz (Fig. 1) or 2 kHz (Fig. 2). For the other tested baffles (plates with larger dimensions than a round plate), a trend of increasing insulation with increasing frequency can be observed. Despite these discrepancies in the results, the proposed FEM method can be used to search for sound insulation of multi-layer baffles.

5. Conclusions

Calculations of the sound insulation using the Finite Element Method for three steel plates, differing in shape and dimensions, showed the smallest discrepancies compared to the experimental tests for plates of smaller dimensions. Despite two different acoustic conditions, resulting from the random (FEM) or perpendicular (impedance tube) incident of the wave on the sample, there is quite good convergence in the obtained results for a simply supported plate, which applies to a steel disc with a diameter of 34.9 mm. An experiment on the possibility of using an impedance tube, in which the sound insulation determined by the normal incidence sound transmission loss coefficient is tested for soft materials, for a steel disc mounted in a tube, it was successfully completed. Comparable results were obtained with the Finite Element Method.

When it comes to samples with larger dimensions, the presented approach using the Finite Element Method is so far burdened with significant discrepancies in relation to the results from the tests carried out in the reverberation chamber set. The discrepancies could be influenced, among others, by various methods of fixing the samples in the measuring window of the laboratory. Not without significance in comparing the results is the fact of calculating the sound insulation using FEM for specific excitation frequencies and, in the case of the method used in the laboratory test of sound insulation, the centre frequencies of 1/3 octave bands. The approach of using mechanical impedance and numerical methods for a single material, in this case - a hard plate, presented by the authors of the paper, promises the possibility of modeling the sound insulation of baffles consisting of multiple layers.

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Additional information

The author(s) declare: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

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