Archives of Acoustics Vol. 51, No. 1, pp. 197–209 (2026), https://doi.org/10.24423/archacoust.2026.4316

Technical Note

Cube-Shaped Sound-Insulating Enclosures: Experimental Tests and Calculation Models

Krzysztof Kosała

Department of Mechanics and Vibroacoustics, Faculty of Mechanical Engineering and Robotics, AGH University of Krakow
Kraków, Poland

e-mail: kosala@agh.edu.pl

Received September 5, 2025; revised November 10, 2025; accepted December 10, 2025; available online January 15, 2026; version of record February 19, 2026; published issue March 27, 2026.

The research described in this article concerns sound-insulating enclosures used for sound sources imitating a noisy machine or device. It is a continuation of experimental tests and modelling studies previously conducted on a prototype test stand, in which the enclosure walls measured 0.7 m × 0.7 m. The main aim of the current research was to estimate the acoustic efficiency of the enclosures through experimental testing on a new stand with walls measuring 0.55 m × 0.55 m, conducted under conditions similar to those found in an industrial facility. Tests conducted on five wall types of varying thicknesses, made of materials such as steel, aluminium, and plexiglass, enabled the development of a calculation model for insertion loss (IL), which can be used based on the material data for the enclosure walls. The model was validated through further experimental tests covering four additional material variants, and a high correlation of the results was obtained. The influence of the calculation model used for the enclosure wall transmission loss on the IL result was also investigated. The results of the experimental tests and modelling studies were also compared with those obtained for a larger enclosure made of the same wall materials. The research described in the current article may have practical applications in the selection of walls of cube-shaped enclosures and in estimating their effectiveness in a cost-free manner, assuming that appropriate material data are used in the calculations.

Keywords: acoustical enclosures, insertion loss (IL), noise protection.

1. Introduction

Homogeneous sound-insulating baffles have a number of applications in noise protection solutions. They can be used individually as shields or wall elements of acoustic barriers, or as walls of sound-insulating enclosures, usually designed to completely or partially isolate excessively noisy machines or devices (Barron, 2003; Vér, Beranek, 2006; Pawelczyk, Wrona, 2022; Chiu, 2012). The basic parameter determining the effectiveness of an acoustic enclosure is the insertion loss (IL). This parameter can be determined experimentally, based on the difference in sound power levels between the unenclosed and enclosed sound sources, or by using computational models, such as statistical energy analysis (SEA) (Nieradka, Dobrucki, 2018; Kim *et al.*, 2014; Lei *et al.*, 2012; Ming, Pan, 2004), or numerical methods (Zhou *et al.*, 2011; Agahi *et al.*, 1999).

Selecting the proper material for an enclosure wall has a significant impact on its effectiveness; however, this is often dependent on economic and functional factors. The sound-insulating properties of a single baffle can be determined in a laboratory, but this approach cannot take into account all the factors that influence its final resistance to sound transmission, such as the mounting method being identical to the intended one in a specific enclosure, external dimensions of the panel other than those resulting from the dimensions of the laboratory measurement window, sealing method, etc. In order to estimate the sound insulation of a baffle with specific dimensions, calculation models (Kosała *et al.*, 2020a) can be used, with the accuracy dependent primarily on the material data used, which in reality may differ slightly from the commonly known and typical values given in the literature.

To test the effectiveness of material and construction solutions for baffles and enclosures, a prototype test stand was proposed within the framework of previous research, the main element being a heavy steel frame with the possibility of attaching baffles constituting the enclosure walls (Kosała *et al.*, 2020b). In addition to experimental tests of the effectiveness of an enclosed omnidirectional sound source imitating a noisy machine or device, a computational model has been developed to estimate the spectral characteristics of the IL of a sound-insulating enclosure without any cost (Kosała *et al.*, 2020c; Kosała, 2022; 2024). The experimental tests and modelling studies conducted so far concerned specific dimensions of the enclosure walls (0.7 m × 0.7 m), resulting from the developed test stand. As a continuation of this research, a smaller frame was developed, which enables the attachment of enclosure wall panels measuring 0.55 m × 0.55 m. The effectiveness of such cube-shaped sound-insulating enclosures, both experimental and model-based, is the subject of this article. A calculation model for the IL of enclosures made of five material sets of different thicknesses, such as steel, aluminium and plexiglass, was proposed and validated by further experimental studies on another four material sets. This model is a function of parameters related, among others, to the sound-insulating properties of the enclosure walls in question, such as the transmission loss (TL) and the single-number weighted sound insulation index of the wall R_w determined on its basis. These parameters were determined from the model proposed in previous studies, which is a combination of the models developed by Davy and Sharp, in specific frequency bands, respectively lower and higher, as described in (Kosała, 2019). The influence of TL, determined using the SoundFlow (Ahnert Feistel Media Group, 2011) and INSUL (Marshall Day Acoustics, 2017) programs, on the IL calculation results was also verified with respect to the experimental tests. The IL spectral characteristics obtained from the experimental tests of the enclosures were compared for four variants of wall materials, identical for the enclosure under study and the previously tested larger one with wall dimensions of 0.7 m × 0.7 m.

2. Measurement setup

The enclosure is based on a steel frame in the shape of a cube, allowing the installation of five walls measuring 0.55 m × 0.55 m with a thickness of 0.001 m to 0.09 m (Fig. 1). The frame is placed on a rubber plate, which ensures good tightness of the enclosure at the bottom. Each of the tested walls is tightened to the enclosure frame, through a rubber seal, using 0.01 m × 0.01 m square steel frames and a set of 11 holdfast mechanisms, except for the upper wall of the enclosure, which is attached using 12 mechanisms. The lower sides of the four walls mounted vertically to the frame are pressed against the frame using two holdfast mechanisms and a flat steel bar. This clamping solution allows the cubic shape of the enclosure cavity, measuring 0.55 m × 0.55 m × 0.55 m, to be maintained. Inside the enclosure, a loudspeaker is placed centrally on the floor (Fig. 1).

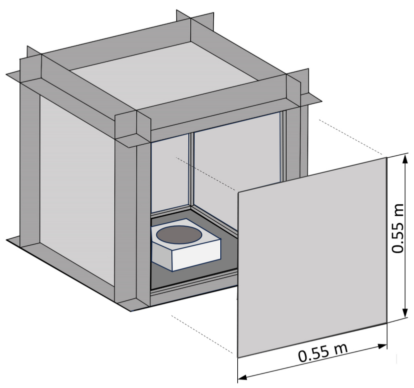

Fig. 1. Conceptual sketch of the enclosure in the form of a frame, loudspeaker, and five walls.

Figure 2 shows the enclosure frame with a loudspeaker (a) and the enclosure with steel walls (b). Acoustic tests were carried out in a room with a volume of 79 m^3. The signal, in the form of pink noise, generated using the Audacity program via an audio interface (EDIROL UA-5) and an amplifier (Bruel & Kjaer 2716-C), was fed to the sound source. A SVAN 945A sound level meter was used to measure sound pressure levels on a hemispherical measurement surface to determine the sound power levels of the source with and without the enclosure.

a) b)

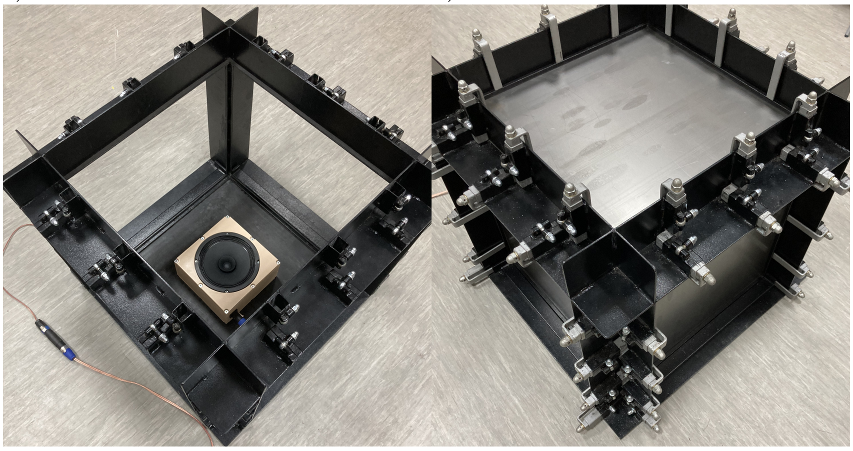

Fig. 2. View of the enclosure frame: a) without the walls, b) with steel walls installed.

For this purpose, the survey method was used in accordance with the ISO standard 3746 (International Organization for Standarization [ISO], 2010), taking into account the background noise correction factor K1 and the environmental correction factor K2. The surface sound pressure level was averaged from the results obtained from measurements carried out at four measurement points, located in accordance with the standard on a hemispherical measurement surface with a radius of 1.15 m.

The room acoustic absorption required for calculating the K2 correction factor was determined based on the reverberation time obtained from impulse responses recorded in the room. For this purpose, an omnidirectional sound source was used, consisting of six loudspeakers (BG 20/70W VISATON) assembled in a cubic casting, an amplifier (Bruel & Kjaer 2716-C), a BEHRINGHER ECM 8000 measuring microphone, and an EDIROL UA-5 audio interface, along with a DIRAC 6.0 Bruel & Kjaer system. The maximum length sequence (MLS) signal generated by the DIRAC program was used for the room excitation.

The performance of the acoustical enclosure in terms of IL was determined as the difference between the sound power levels radiated by the unenclosed and the enclosed source.

The IL of an acoustic enclosure can be determined more accurately under laboratory conditions in an anechoic room by determining sound power levels using the precision method (ISO, 2012), which was the subject of previous research (Kosała *et al.*, 2020c). In the current research, the use of the survey method was intended to reflect measurement conditions in which the nature of the sound field is similar to that occurring under industrial conditions, and it is considered a sufficient approach to validate the results obtained from the calculation model for the IL of sound-insulating enclosures proposed in this article.

An enclosure with a set of five identical walls was tested for different baffle types, with the material data shown in Table 1.

Table 1. Material data of the tested enclosure walls.

Parameter	Aluminium	Plexiglass	Steel
Density (p) [kg/m^3]	2800	1190	7850
Thickness (h) [m]	0.002 0.003	0.005 0.015	0.001
Young's modulus (E) [GPa]	70	3.5	207
Poisson's Ratio (ν) [–]	0.35	0.35	0.3
Loss factor (η) [–]	0.01	0.02	0.01

Assuming that the enclosure has perfectly rigid walls, the mode frequencies (eigenfrequencies) $f_{m,n,p}$ of the enclosure cavity can be determined according to the well-known formula (Hopkins, 2007):

$$f_{m,n,p} = \frac{c_0}{2}\sqrt{\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2 + \left(\frac{p}{L_z}\right)^2}, \tag{1}$$

where c_0 is the speed of sound in the air [m/s], m, n, and p are the particular mode numbers, and L_x, L_y, and L_z are dimensions of the enclosure cavity [m].

The eigenfrequencies calculated in this way for a simple, rectangular-shaped cavity are an approximation. In reality, the air cavity inside the enclosure is reduced by the presence of a loudspeaker, which has a case with dimensions of 0.24 m × 0.24 m × 0.115 m. Calculations of the eigenfrequencies for such a complex geometry of the air cavity inside the enclosure were possible using a numerical model and simulations performed in COMSOL Multiphysics 6.3.

Table 2 shows the values of the eigenfrequencies calculated for the enclosure cavity with and without a loudspeaker.

Table 2. Eigenfrequencies of the enclosure cavity with and without a loudspeaker.

Mode	Eigenfrequency [Hz]		
	Enclosure cavity without a loudspeaker		Enclosure cavity with a loudspeaker
	Equation (1)	COMSOL Multiphysics	COMSOL Multiphysics
1	312.1	312.0	297.3
2	312.1	312.0	297.3
3	312.1	312.0	319.5
4	441.4	441.3	427.9
5	441.4	441.3	427.9
6	441.4	441.3	436.1
7	540.6	540.4	534.3
8	624.2	624.1	594.6
9	624.2	624.1	601.2
10	624.2	624.1	643.8

The identical dimensions of the enclosure cavity without a loudspeaker ($L_x = L_y = L_z$) show that each of the axial modes (#1–3 and #8–10) as well as each of the tangential modes (#4–6) have the same values. For an enclosure cavity reduced by the presence of a loudspeaker, the values of the eigenfrequencies are more varied.

3. IL calculation model

Previous studies have shown that, when determining the effectiveness of a sound-insulating enclosure, good results, i.e., small discrepancies between the calculation and experimental results. can be obtained using the model described in (Kosała, 2022). This model determines the effectiveness of a sound source enclosed by sound-insulating walls with dimensions of 0.7 m × 0.7 m. For cube-shaped enclosures, built of sound-insulating walls with dimensions 0.55 m × 0.55 m, a similar approach to that presented in (Kosała, 2022) is proposed, however, it is defined by the relationship:

$$\mathrm{IL} = \begin{cases} 10\log\left[\cos(kd) - 0.37\rho h\omega\sin(kd)/\rho_0 c_0\right]^2 & \text{for} \quad f \le 1.9 f_{0,0,1}, \\ 10\log\dfrac{\bar{\alpha}_{\mathrm{rand}}}{10^{-0.1\mathrm{TL}} + e^{-0.31R_w}} & \text{for} \quad f > 1.9 f_{0,0,1}, \end{cases} \tag{2}$$

where k is the sound wavenumber, d is the distance of the top panel of the tested enclosure from the floor, ρ is the material density, h is the material thickness, ω is the angular frequency, ρ_0 is the air density, c_0 is the speed of sound in the air, $f_{0,0,1}$ is the first axial mode frequency of the enclosure cavity, $\bar{\alpha}_{\mathrm{rand}}$ is the random-incidence sound absorption coefficient of bare enclosure walls, TL is the sound transmission loss of the bare enclosure walls, R_w is the weighted sound reduction index of the enclosure wall, and f_c is the coincidence frequency of the enclosure wall (Kosała, 2022; Vér, Beranek, 2006).

A new calculation model is proposed for IL for the relevant frequency ranges, depending on the eigenfrequency $f_{0,0,1}$ enclosure cavity. For the low-frequency range, in which $f \leq 1.9 f_{0,0,1}$ of the enclosure cavity, a modified Oldham model (Oldham, Hillarby, 1991; Kosała, 2022) for a clamped boundary condition is proposed. For the high-frequency range, where $f > 1.9 f_{0,0,1}$ of the enclosure cavity, the proposed model is a function of $\bar{\alpha}_{\text{rand}}$, TL, and R_w.

The quantity 0.32 in Eq. (16) from (Kosała, 2022) was replaced by 0.37 in Eq. (2) for the low-frequency range. This value was obtained based on the results of new experimental tests of the IL spectral characteristics, carried out for five variants of the wall materials given in Table 1. For the frequency range of 100 Hz to 500 Hz (centre frequencies of the ⅓-octave bands), using the assumed value of 0.37, a relatively low root mean square error (RMSE) value of 3.9 dB was obtained, averaged across five variants, with a simultaneously high value of the Pearson correlation coefficient $r = 0.88$ between the results of the model calculations and the experimental tests. Similarly, new experimental tests were used for the higher frequency range, from 630 Hz to 5 kHz. Using the assumption of a value of 0.31 for R_w resulted in a relatively low five-variant average of RMSE = 2.73 dB, with a high Pearson correlation coefficient of $r = 0.85$.

4. Results

4.1. Calculation of sound-insulating properties of the enclosure walls

To determine the TL of a homogeneous baffle with external dimensions of 0.55 m × 0.55 m, we needed to determine the IL of a sound-insulating enclosure using the proposed Eq. (1), the material data of the baffle, shown in Table 1, and the Davy–Sharp model proposed in (Kosała, 2019) were used. Previous studies have shown that such a combination of the two models, Davy (Davy, 2009) and Sharp (Sharp, 1973), for the relevant frequency ranges, resulted in smaller discrepancies in TL calculations for single homogeneous baffles with dimensions of 1 m × 2 m, compared to laboratory tests, than when using commercial software and when the models were used separately (Kosała, 2019). The calculation results, in the form of TL spectral characteristics in ⅓-octave frequency bands for five materials with thicknesses ranging from 0.001 m to 0.015 m, are shown in Fig. 3.

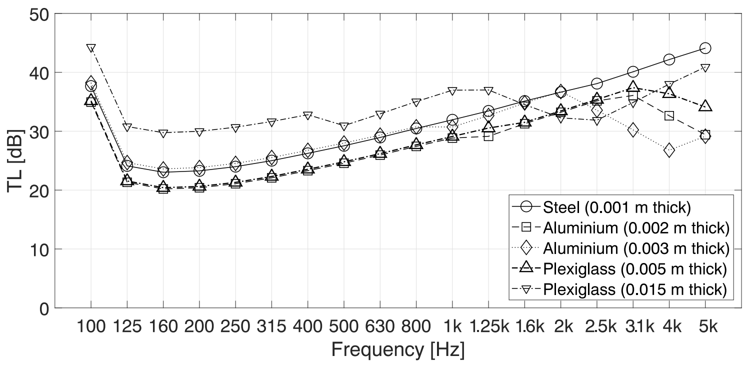

Fig. 3. TL of the tested enclosure walls obtained using the Davy–Sharp calculation model.

The curves for the aluminium baffle (0.003 m thick) and the plexiglass baffle (0.015 m thick) show a characteristic reduction in sound insulation in the higher frequency range, related to the occurrence of coincidence phenomena. For the remaining baffles, the coincidence frequency f_c is above the highest centre frequency of the ⅓-octave bands considered in this study, which is 5 kHz. The calculated f_c values for the tested enclosure walls are shown in Table 3.

In the low-frequency bands (Fig. 3), where the radiation efficiency of a finite-sized baffle is reduced, a specific increase in sound insulation can be observed. Based on the TL spectral characteristics, the weighted sound reduction index R_w was calculated for each type of enclosure wall tested. The values of this index, as well as the spectral adaptation indices C and C_{tr}, for the tested enclosure walls, are shown in Table 3. By comparing the R_w values (Table 3) for enclosure walls of dimensions 0.55 m × 0.55 m, made of steel (0.001 m thick) and

Table 3. Weighted sound reduction indices $R_w(C;C_{tr})$ and the coincidence frequencies f_c of the tested enclosure walls.

Enclosure wall material	$R_w(C;C_{tr})$ [dB]	f_c [Hz]
Steel (0.001 m thick)	32(−1;−3)	12 042
Aluminium (0.002 m thick)	29(−1;−3)	6072
Aluminium (0.003 m thick)	32(−1;−2)	4048
Plexiglass (0.005 m thick)	29(0;−2)	6843
Plexiglass (0.015 m thick)	34(0;0)	2281

aluminium (0.002 m thick) with the corresponding values of this parameter for baffles of dimensions 0.7 m × 0.7 m (Kosała, 2022), it can be seen that the smaller baffles have R_w values 1 dB higher than the larger baffles. This is influenced by the TL spectral characteristics, which in the low-frequency bands show slightly higher values for smaller baffles compared to larger ones. This also applies to baffles of larger dimensions (1 m × 2 m), for which the steel baffle (0.001 m thick) has R_w = 30 dB, and the aluminium baffle (0.002 m thick) has R_w = 27 dB (Kosała, 2019).

4.2. Calculation of IL for the enclosure using the proposed model

The calculation results obtained using the proposed IL Eq. (2) for the five tested variants of sound-insulating enclosures are shown in Fig. 4.

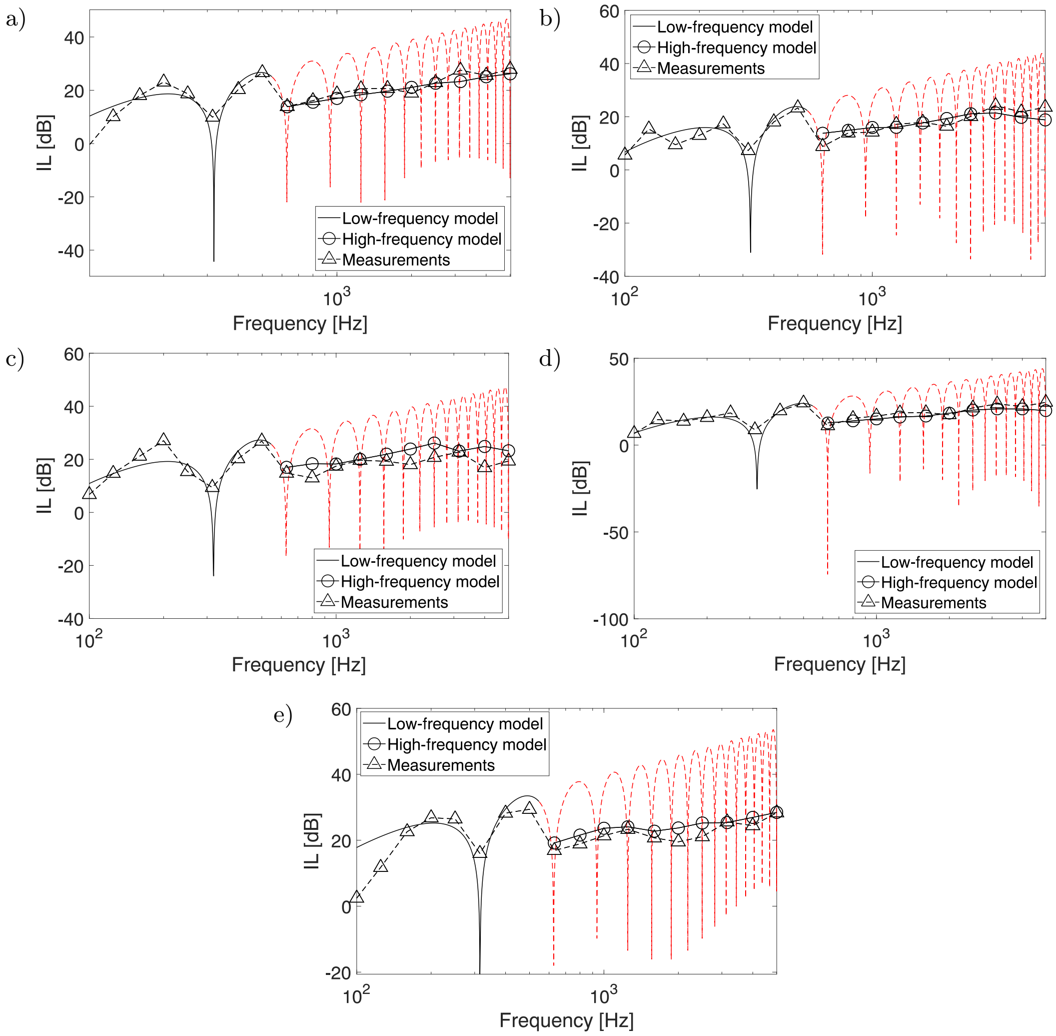

Fig. 4. IL of enclosures with walls made of different materials: a) steel (0.001 m thick), b) aluminium (0.002 m thick), c) aluminium (0.003 m thick), d) plexiglass (0.005 m thick), e) plexiglass (0.015 m thick), obtained using the low-frequency and high-frequency models, compared with experimental measurements.

After recalculating the IL values for the ⅓-octave bands in the low-frequency range, it was possible to compare the results of calculations and experimental tests using the Pearson correlation coefficient r and the RMSE. Close agreement between the calculated and measured results was obtained in both the lower and higher frequency ranges, as shown in Table 4. For the lower frequency range (centre frequencies of the ⅓-octave bands below 600 Hz), the Pearson correlation coefficients for all types of enclosure walls are high ($r \geq 0.8$). For the frequency range higher than 600 Hz, the values of $r \geq 0.86$, except for the 0.003 m thick aluminium baffle, for which $r = 0.61$. Considering the full frequency range, the Pearson coefficient remains high ($r \geq 0.73$).

Table 4. Pearson correlation coefficients r and RMSEs for predicted IL compared with experimental tests for the five enclosure wall materials.

Enclosure wall material			Steel	Aluminium	Aluminium	Plexiglass	Plexiglass
Thickness (h) [m]			0.001	0.002	0.003	0.005	0.015
r	Frequency range	low	0.88	0.87	0.85	0.95	0.80
		high	0.94	0.86	0.61	0.92	0.91
		full	0.9	0.87	0.73	0.94	0.81
RMSE [dB]	Frequency range	low	4.5	2.8	3.7	2.0	6.6
		high	1.9	2.7	4.2	2.3	2.5
		full	3.3	2.7	4.0	2.2	4.8

The RMSE values for the low-frequency range are from 2 dB (0.005 m thick plexiglass baffle) to 6.6 dB (0.015 m thick plexiglass wall). For the higher frequency range, the RMSE ranges from 1.9 dB (0.001 m thick steel baffle) to 4.2 dB (0.003 m thick aluminium baffle). Over the entire frequency range, the RMSE is between 2.2 dB and 4.8 dB.

In Fig. 4, the red dashed line indicates the further course of IL, after applying the modified Oldham formula (line 1 of Eq. (2)), but for $f > 1.9 f_{0,0,1}$. The nulls on the graphs correspond to the axial resonant frequencies of the enclosure cavity $f_{0,0,1}$, $f_{0,0,2}$, etc. Table 5 shows the eigenfrequencies of the enclosure cavity determined by this method and those calculated using Eq. (1).

Table 5. Eigenfrequencies of the enclosure cavity calculated from Eq. (1) and Eq. (2).

Axial mode number	m	n	p	Eigenfrequency [Hz] $f_{m,n,p}$	
				Equation (1)	Modified Oldham model (Eq. (2))
1	0	0	1	312	319
2	0	0	2	624	628
3	0	0	3	936	939
4	0	0	4	1248	1250
5	0	0	5	1560	1562
6	0	0	6	1873	1874
7	0	0	7	2185	2186
8	0	0	8	2497	2498
9	0	0	9	2809	2810
10	0	0	10	3121	3122
11	0	0	11	3433	3434
12	0	0	12	3745	3746
13	0	0	13	4057	4058
14	0	0	14	4369	4370
15	0	0	15	4681	4682

The eigenfrequency values of the enclosure cavity obtained by the two methods differ slightly, as shown in Fig. 5. The greatest differences are for the (0,0,1) mode; however, they decrease with increasing frequency.

It should be taken into account that the calculations using Eq. (1) are approximate, since the walls of the tested enclosure cannot be perfectly rigid when excited by a sound source.

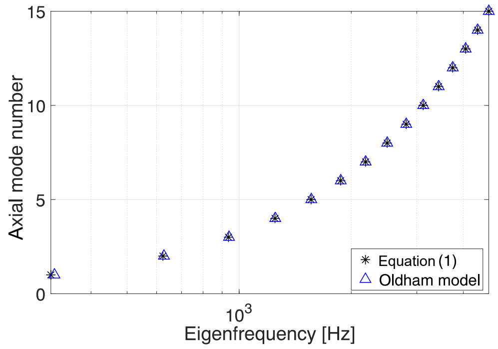

Fig. 5. Comparison of eigenfrequencies of the enclosure cavity obtained from Eq. (1) and Eq. (2).

5. Model validation and the influence of the wall's TL on the enclosure's IL

The proposed model for the sound insulation effectiveness of enclosures was tested in preliminary studies for five variants, each comprising five identical walls made of different materials. Based on the material data and the calculation of the enclosure wall's TL using the Davy–Sharp model, the IL of these enclosures was determined. In the next stage of the study, the model was validated using four additional sets of five walls made of the following materials: 1.5 mm and 2 mm thick steel, 10 mm thick plexiglass, and 1 mm thick aluminium. At this stage, the influence of other models, such as SoundFlow and INSUL, on the final results of the enclosure's IL calculations was examined. First, the TL spectral characteristics were determined and the corresponding R_w indices were calculated, as shown in Fig. 6.

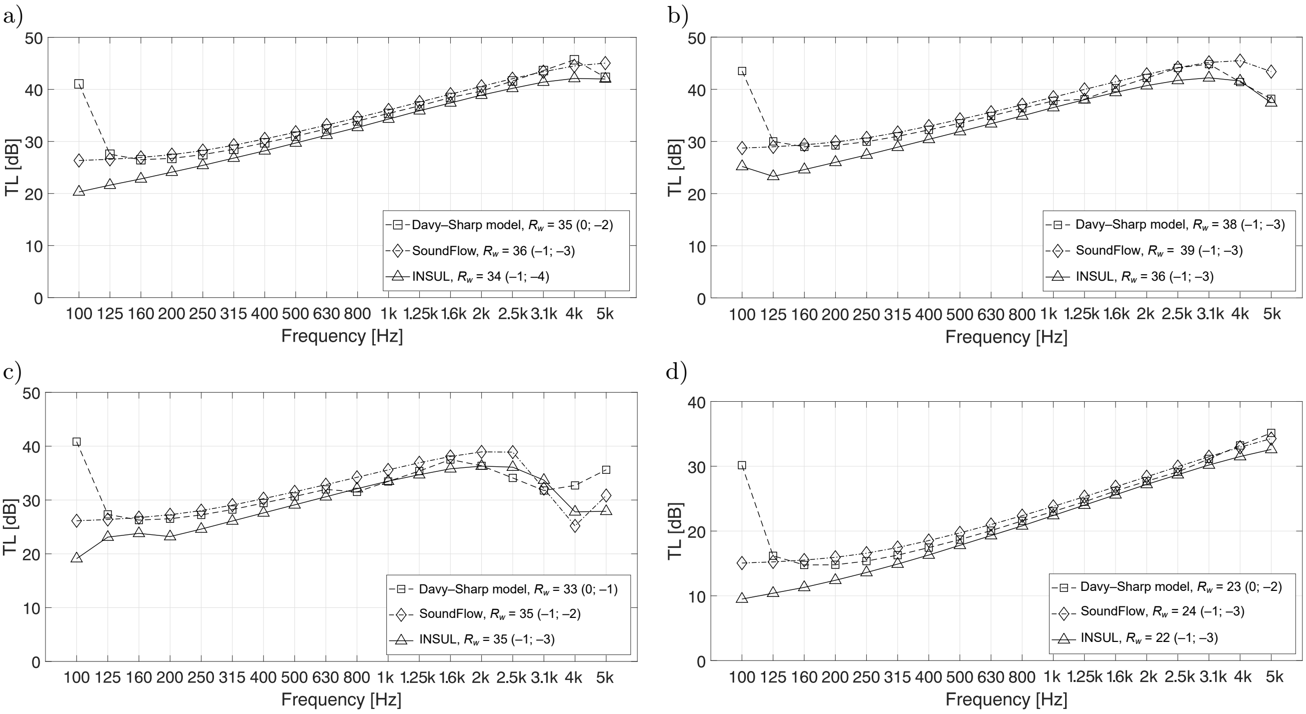

Fig. 6. TL calculated using the Davy–Sharp model, AFMG SoundFlow, and INSUL for enclosure walls made of: a) steel 0.0015 m thick, b) steel 0.002 m thick, c) plexiglass 0.01 m thick, and d) aluminium 0.001 m thick.

For the calculations in the AFMG SoundFlow and INSUL programs, the baffle dimensions were assumed to be 0.6 m × 0.6 m, because it was not possible to take into account the actual wall dimensions of 0.55 m × 0.55 m. Figure 6 shows discrepancies in the TL calculations, especially in the lower frequency bands. It should be noted that, although the IL calculations take into account the TL of the wall starting from the centre frequency of the ⅓-octave band at 630 Hz, the discrepancies are relatively small in the higher frequency range. The weighted sound reduction index R_w, on which IL also depends, is calculated for the ⅓-octave bands from 100 Hz to 3.1 kHz. The discrepancies between the calculated R_w values range from 2 dB to 3 dB. In the higher frequency bands, a noticeable reduction in sound insulation due to coincidence is observed for a 0.01 m thick plexiglass baffle. Since the results are presented in ⅓-octave frequency bands, the coincidence frequency value f_c = 3421 Hz, cannot be precisely identified in the graphs. For the AFMG SoundFlow and INSUL models, f_c = 4 kHz, and for the Davy–Sharp model, f_c = 3.1 kHz.

The results of the enclosure IL calculations for the four tested new wall material sets, including the higher frequency ranges of the three models for which TL was calculated, are shown in Fig. 7.

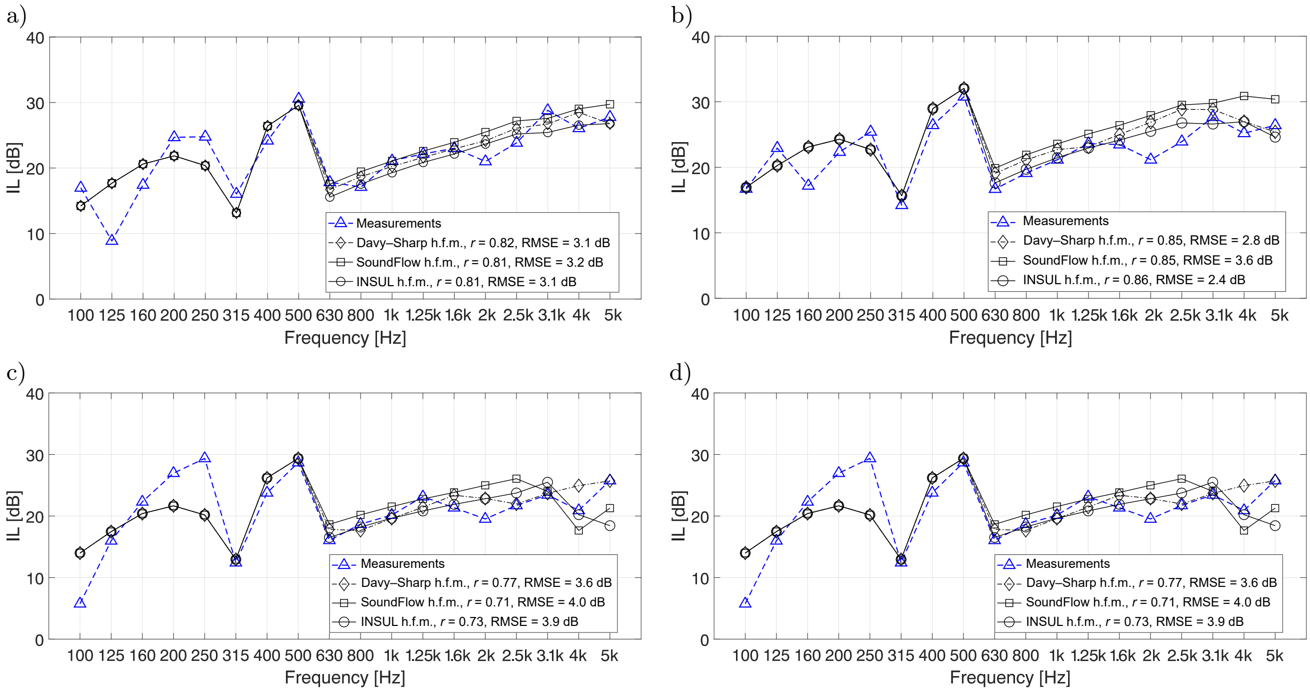

Fig. 7. IL calculated using the proposed calculation model: modified Oldham for the low-frequency range and Davy–Sharp, AFMG SoundFlow, and INSUL for the high-frequency range, for enclosure walls made of: a) steel 0.0015 m thick, b) steel 0.002 m thick, c) plexiglass 0.01 m thick, and d) aluminium 0.001 m thick.

The smallest RMSE discrepancies occurred for the 0.002 m thick steel baffle (RMSE_{avr} = 2.9 dB), and the largest for the 0.001 m thick aluminium baffle (RMSE_{avr} = 4.2 dB). The average Pearson correlation coefficients r for all four baffles calculated using the individual models are: 0.84 for Davy–Sharp, 0.8 for AFMG SoundFlow, and 0.8 for Insul. The average RMSEs are: 3.40 dB for Davy–Sharp, 3.73 dB for AFMG SoundFlow, and 3.43 dB for INSUL. These results indicate that the Davy–Sharp model showed both the highest correlation of IL characteristics with the experimental results among the three tested models and the lowest average RMSE.

6. Effectiveness of sound-insulating enclosures of different sizes made from the same wall materials

Experimental tests and modelling studies of the effectiveness of small enclosures (wall dimensions of 0.55 m × 0.55 m) described in this article, and large enclosures (wall dimensions of 0.7 m × 0.7 m) described in (Kosała, 2022), carried out on sets of identical wall materials, such as steel 0.001 m thick, aluminium 0.002 m thick, and plexiglas 0.005 m and 0.015 m thick, enabled a comparison of their sound-insulting properties. Figure 8 shows the results of the experimental tests, and Fig. 9 presents the results of the modelling studies.

The IL spectral characteristics (Fig. 8) show that IL is higher for smaller enclosures, especially in the mid- and high-frequency bands. For smaller enclosures in these ranges, greater variations in IL (5 dB to 10 dB) are observed for the tested materials, compared to larger enclosures (approximately 3 dB to 5 dB). In larger enclosures, the influence of the enclosure cavity eigenfrequencies on the IL spectral characteristic is observed up to 315 Hz,

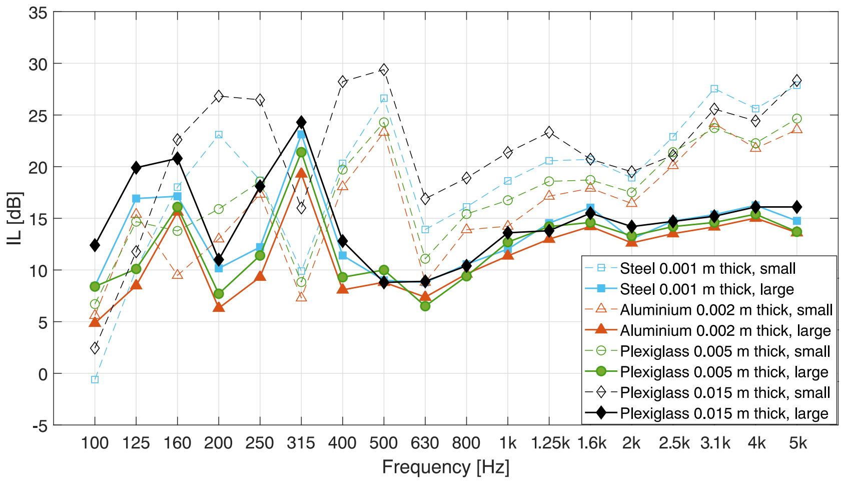

Fig. 8. IL obtained from experimental tests of small and large enclosures made of wall materials: steel (0.001 m thick), aluminium (0.002 m thick), and plexiglass (0.005 m and 0.015 m thick).

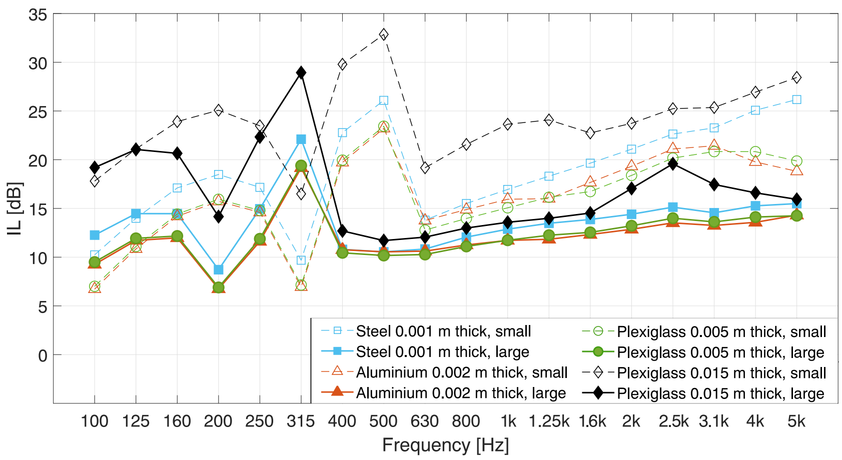

Fig. 9. IL obtained using calculation models for small and large enclosures made of wall materials: steel (0.001 m thick), aluminium (0.002 m thick), and plexiglass (0.005 mm and 0.015 m thick).

corresponding to 1.5 f_c of the enclosure wall, while for smaller enclosures this influence is more dominant, up to 500 Hz, corresponding to 1.9 f_c of the enclosure wall. Among the tested baffles, the most effective is the 0.015 m thick plexiglass wall; however, for smaller enclosures, the influence of coincidence, which is evident in the TL spectral characteristic of these walls, is more noticeable in the IL curves, whereas for larger enclosures, the IL characteristic in the 2 kHz region is flatter (f_c = 2281 Hz). The IL curves obtained from the calculation models (Fig. 9) closely match the corresponding curves obtained from the experimental tests (Fig. 8).

Table 6 summarizes the Pearson correlation coefficients r and the RMSEs for the IL calculation models compared to the experimental tests for enclosures with wall dimensions of 0.55 m × 0.55 m and 0.7 m × 0.7 m.

Table 6. Pearson correlation coefficients and RMSEs for predicted IL in comparison to experimental tests for the same enclosure wall materials, and enclosure wall dimensions of 0.7 m × 0.7 m and 0.55 m × 0.55 m.

Enclosure wall material	Thickness h [m]	Enclosure wall dimensions [m]			
		0.7 × 0.7		0.55 × 0.55	
		r	RMSE [dB]	r	RMSE [dB]
Steel	0.001	0.87	1.80	0.90	3.33
Aluminium	0.002	0.86	2.12	0.87	2.74
Plexiglass	0.005	0.89	1.75	0.94	2.17
Plexiglass	0.015	0.87	2.98	0.81	4.80
Average		0.87	2.16	0.88	3.26

The two models for smaller and larger enclosures, proposed in (Kosała, 2022), showed a high correlation coefficient between the IL spectral characteristics calculated using the models and those obtained from experimental tests. The average value of these coefficients is 0.9. The IL estimation using the calculation models showed approximately 1 dB lower average RMSE values for larger enclosures compared to smaller ones. In both cases, the average RMSE values are not high, and range from about 2 dB to 3 dB, except for the small enclosure made of 15 mm plexiglass, for which the prediction error is close to 5 dB.

7. Conclusions

As a result of experimental tests aimed at determining the effectiveness of an enclosed sound source imitating a noisy machine or device, IL characteristics were determined, first for five different wall material variants, and in the second stage of the study, for four additional variants.

In total, nine wall material variants were tested: steel with thicknesses of 0.001 m, 0.0015 m, and 0.002 m, aluminium with thicknesses of 0.001 m, 0.002 m, and 0.003 m, and plexiglass with thicknesses of 0.005 m, 0.01 m, and 0.015 m. The determined IL curves are very similar, and it can be roughly concluded that, in the medium- and high-frequency ranges, they are shifted towards higher IL values, corresponding to the single-number weighted sound insulation index R_w of the given wall. The poorest performance was observed in enclosures made of 0.001 m thick aluminium walls (R_w = 22 dB), while the best performance was achieved by enclosures made of 0.002 m thick steel walls (R_w = 38 dB). The IL curves, up to a certain centre frequency of the ⅓-octave bands, equal to 500 Hz, i.e., for $f \leq 1.9 f_{0,0,1}$, exhibit characteristic shapes, corresponding to the enclosure air cavity resonances. Above this frequency, the IL values are less differentiated across subsequent centre frequencies.

The proposed calculation model for the IL of sound-insulating enclosures was developed based on experimental tests conducted in the first stage, covering five variants of material walls. In a subsequent stage, the model was validated using four additional variants, taking into account the influence of the calculation model for the wall's TL on the final IL results. The studies showed that the model proposed in earlier studies, a combination of the Davy and Sharp models, yielded better agreement with experimental results compared to the models implemented in SoundFlow and INSUL software. The Davy–Sharp model showed both the highest correlation of IL characteristics with the experimental results of the three models tested and the lowest average RMSE. However, it should be noted that only the Davy–Sharp model could take into account the actual dimensions of the enclosure wall of 0.55 m × 0.55 m. Limitations in input data in the other software required the use of wall dimensions of 0.6 m × 0.6 m, i.e., only close to the actual ones, which could also have a certain impact on the accuracy of calculations using these models. A comparison of the effectiveness of enclosures made of the same materials, with walls measuring 0.55 m × 0.55 m, was made against larger enclosures with walls measuring 0.7 m × 0.7 m, analysed in previous studies. Based on the comparison of the IL spectral characteristics, it was found that IL is higher for smaller enclosures, especially in the mid- and high-frequency bands. Based on IL calculations using models, the average RMSE values were approximately 1 dB lower for larger enclosures compared to smaller ones. In both cases, the average RMSE values were not high, ranging from about 2 dB to 3 dB, except for the small enclosure made of 15 mm plexiglass, where the prediction error was close to 5 dB. Both the models for smaller and larger enclosures, proposed in (Kosała, 2022), showed high correlation coefficient between calculated and experimental IL spectral characteristics, with an average r = 0.9. The proposed calculation models can be helpful in cost-free estimation of the acoustic efficiency of cube-shaped sound-insulating enclosures with the studied wall dimensions, provided that appropriate material data needed for these calculations are available.

Testing the acoustic efficiency of sound-insulating enclosures is a significant challenge, as their effectiveness must be evaluated differently compared to that of sound-absorbing or insulating enclosures. In the latter, the impact of resonances within the enclosure, which sometimes counteracts their effectiveness, is mitigated by sound-absorbing material placed near the sound source. In contrast, sound-insulating enclosures rely on reflective walls, and their performance is strongly influenced by cavity resonances. Despite these challenges, sound-insulating enclosures have many practical applications. These are particularly relevant in cases where smooth, sound-reflecting transparent walls, such as plastic or glass, are required to monitor the operation of the enclosed device. Another area of application involves situations where, due to the risk of bacterial growth in porous and fibrous materials, sound-absorbing and insulating enclosures are inappropriate.

The obtained research results may have practical applications for selecting wall materials for cube-shaped sound-insulating enclosures.

Fundings

This research was funded by a research subvention supported by the Polish Ministry of Education and Science, grant number 16.16.130.942.

Conflict of interest

The author declares that he has no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Author's contribution

All author reviewed and approved the final manuscript.

Data availability

The data that support the findings of this study are available from the corresponding author upon reasonable request.

References

1. Agahi P., Singh U.P., Hetherington J.O. (1999), Numerical prediction of the insertion loss for small rectangular enclosures, *Noise Control Engineering Journal*, **47**(6): 201–208, https://doi.org/10.3397/1.599316.
2. Ahnert Feistel Media Group (2011), AFMG SoundFlow User's Guide.
3. Barron R.F. (2003), *Industrial Noise Control and Acoustics*, Marcel Dekker, New York.
4. Chiu M.-C. (2012), Shape optimization of two-layer acoustical hoods using an artificial immune method, *Archives of Acoustics*, **37**(2): 181–188, https://doi.org/10.2478/v10168-012-0024-5.
5. Davy J.L. (2009), Predicting the sound insulation of single leaf walls: Extension of Cremer's model, *The Journal of the Acoustical Society of America*, **126**(4): 1871–1877, https://doi.org/10.1121/1.3206582.
6. Hopkins C. (2007), *Sound Insulation*, Elsevier, Oxford.
7. International Organization for Standardization (2010), *Acoustics – Determination of sound power levels and sound energy levels of noise sources using sound pressure – Survey method using an enveloping measurement surface over a reflecting plane* (ISO Standard No. 3746:2010), https://www.iso.org/standard/52056.html.
8. International Organization for Standardization (2012), *Acoustics – Determination of sound power levels and sound energy levels of noise sources using sound pressure – Precision methods for anechoic rooms and hemi-anechoic rooms* (ISO Standard No. 3745:2012), https://www.iso.org/standard/45362.html.
9. Kim H.-S., Kim J.-S., Lee S.-H., Seo Y.-H. (2014), A simple formula for insertion loss prediction of large acoustical enclosures using statistical energy analysis method, *International Journal of Naval Architecture and Ocean Engineering*, **6**(4): 894–903, https://doi.org/10.2478/IJNAOE-2013-0220.
10. Kosała K. (2019), Calculation models for analysing the sound insulating properties of homogeneous single baffles used in vibroacoustic protection, *Applied Acoustics*, **146**: 108–117, https://doi.org/10.1016/j.apacoust.2018.11.012.
11. Kosała K. (2022), Experimental tests and prediction of insertion loss for cubical sound insulating enclosures with single homogeneous walls, *Applied Acoustics*, **197**: 108956, https://doi.org/10.1016/j.apacoust.2022.108956.
12. Kosała K. (2024), Sound insulation performance of cube-shaped enclosures, *Vibrations in Physical Systems*, **35**(2): 2024212, https://doi.org/10.21008/j.0860-6897.2024.2.12.

13. Kosała K., Majkut L., Mleczko D., Olszewski R., Wszołek T. (2020a), Accuracy of prediction methods for sound insulation of homogeneous single baffles, *Vibration in Physical Systems*, **31**(2): 2020210, https://doi.org/10.21008/j.0860-6897.2020.2.10.

14. Kosała K., Majkut L., Olszewski R. (2020b), Experimental study and prediction of insertion loss of acoustical enclosures, *Vibrations in Physical Systems*, **31**(2): 2020209, https://doi.org/10.21008/j.0860-6897.2020.2.09.

15. Kosała K., Majkut L., Olszewski R., Flach A. (2020c), Laboratory tests of the prototype stand to determine the acoustic properties of materials used in noise protection, [in:] *21st Century Technologies – Current Problems and New Challenges* [in Polish: *Technologie XXI Wieku – Aktualne Problemy i Nowe Wyzwania*], Vol. 1, pp. 7–20, Wydawnictwo Naukowe TYGIEL, Lublin.

16. Lei Y., Pan J., Sheng M.P. (2012), Investigation of structural response and noise reduction of an acoustical enclosure using SEA method, *Applied Acoustics*, **73**(4): 348–355, https://doi.org/10.1016/j.apacoust.2011.10.008.

17. Marshall Day Acoustics (2017), Sound Insulation Prediction Program, INSUL, User's Manual.

18. Ming R., Pan J. (2004), Insertion loss of an acoustic enclosure, *The Journal of the Acoustical Society of America*, **116**(6): 3453–34599, https://doi.org/10.1121/1.1819377.

19. Nieradka P., Dobrucki A. (2018), Insertion loss of enclosures with lined slits, [in:] *Proceedings of the 11th European Congress and Exposition on Noise Control Engineering Euronoise Conference*, pp. 893–898.

20. Oldham D.J., Hillarby S.N. (1991), The acoustical performance of small close fitting enclosure, part 1: Theoretical models, *The Journal of Sound and Vibrations*, **150**(2): 261–281, https://doi.org/10.1016/0022-460X(91)90620-Y.

21. Pawelczyk M., Wrona S. (2022), *Noise-Controlling Casings*, CRC Press, Boca Raton.

22. Sharp B.H. (1973), *A study of techniques to increase the sound insulation of building elements*, Technical Report, US Department of Commerce, National Technical Information Service, Washington.

23. Vér I.L., Beranek L.L. (2006), *Noise and Vibration Control Engineering – Principles and Applications*, 2nd ed., John Wiley & Sons, Hoboken, New Jersey.

24. Zhou L., Carter A.E., Herrin D.W., Shi J., Copley D.C. (2011), Airborne path attenuation of partial enclosures: Simulation and sensitivity study, *Applied Acoustics*, **72**(6): 380–386, https://doi.org/10.1016/j.apacoust.2010.12.012.