A simple formula for insertion loss prediction of large acoustical enclosures using statistical energy analysis method

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ABSTRACT: Insertion loss prediction of large acoustical enclosures using Statistical Energy Analysis (SEA) method is presented. The SEA model consists of three elements: sound field inside the enclosure, vibration energy of the enclosure panel, and sound field outside the enclosure. It is assumed that the space surrounding the enclosure is sufficiently large so that there is no energy flow from the outside to the wall panel or to air cavity inside the enclosure. The comparison of the predicted insertion loss to the measured data for typical large acoustical enclosures shows good agreements. It is found that if the critical frequency of the wall panel falls above the frequency region of interest, insertion loss is dominated by the sound transmission loss of the wall panel and averaged sound absorption coefficient inside the enclosure. However, if the critical frequency of the wall panel falls into the frequency region of interest, acoustic power from the sound radiation by the wall panel must be added to the acoustic power from transmission through the panel.

KEY WORDS: Sound enclosure; Insertion loss; Statistical energy analysis (SEA); Sound radiation.

INTRODUCTION

Acoustical enclosures are very effective noise control measures for reducing high noise emitting from the sources like diesel engines, air compressors, etc. The performance of an acoustical enclosure is described by the Insertion Loss (IL) defined as the difference between acoustic powers with and without the enclosure. In a small acoustical enclosure at low frequencies, in which dimension of the enclosure is small compared with the wavelength, there exist no resonances inside the enclosure, and insertion loss is mainly determined by the stiffness of the wall panel (Vér, 2006). In a large acoustical enclosure, in which dimension of the enclosure is large compared with the wavelength, wall panels and interior air cavity exhibit a large number of resonances. In a large acoustical enclosure, sound field inside the enclosure and vibrations of the panel can be treated statistically using an averaging concept rather than deterministically using individual modal behaviors. In research vessels and naval ships where low noise is of prime concern, large enclosures are widely used to reduce high noise from diesel engines, in which size of enclosures often reaches more than 10 meters.

Insertion loss of the acoustical enclosures is determined from the coupled motion of the sound field and vibration of wall panels. Lee and Ng (1998) studied insertion loss of the small enclosures whose sizes are less than 0.5 m. They solved coupled motion of air cavity and wall panel using an acoustic velocity potential and the finite element method. Lyon (1963) computed noise reduction of a small enclosure by assuming that the critical frequency of the wall lies above the first acoustic resonance of the enclosure. Al-Bassyouni and Balachandran (2005) investigated sound transmission through a flexible panel into an en-
closure, in which a spherical wave was generated outside the enclosure, and the largest dimension of the enclosure is 60.96 cm. They proposed an active noise control scheme based on a structural-acoustics model, in which they used piezoelectric patches as actuators. Earlier works on active noise control in small enclosures are found in the papers by Sampath and Balachandran (1997; 1999) and Balachandran et al. (1996).

Vér (1973; 2006) and ISO (2000) described formulas to predict insertion loss for various sizes of enclosures, which is an extension of Lyon’s work (1963). For large acoustical enclosures, Vér (1973) assumed a diffuse reverberant sound field and derived insertion loss from consideration of the power balance among several contributions like power dissipation by wall absorption, power loss through sound radiation of the wall panel, power dissipation through viscous damping effects, sound transmission to the exterior though openings and gaps, etc. Vér (1973) showed that the insertion loss of large acoustical enclosures is dominated by the sound transmission loss of the wall panel and averaged sound absorption coefficient inside the enclosure, if other effects such as leaks, silencer openings, flanking structure-borne noise path, and viscous dissipation of the wall panel are well controlled and negligible.

In ISO (2000), practical information about design and assembling enclosure panels is described. For instance, typical panel of a large acoustical enclosure consists of 1.5 mm steel sheet metal, 50 mm mineral wool for absorbent lining, and perforated plate covering with 30% opening at minimum.

The SEA method (Lyon and DeJong, 1995) is known as one of the most powerful tools in investigating the acoustical and vibratory motion of the structures at mid and high frequencies. Ming and Pan (2004) investigated insertion loss of an acoustic enclosure using the SEA method. However, their SEA formulation was incomplete, since they didn’t include the full effect of structural-structural coupling between the component panels. Lei et al. (2011) improved Ming and Pan’s SEA formulation, and used complicated expression for the transmission coefficient of finite flexible plate. They considered insertion loss of a small aluminum enclosure (1.15 m × 1.0 m × 0.868 m, thickness is 2.5 mm), and obtained good agreement with the measurements below critical frequency. Sgard and Nelisse (2010) presented a hybrid statistical energy analysis in predicting insertion loss of L-shape enclosure. They used the method of image sources in computing the sound field inside the enclosure. Although SEA based prediction methods are helpful in investigating the acoustical performance of enclosures, the formulations are rather complicated to be used by engineers who have no background knowledge in SEA.

The aim of this paper is to present a simple formula for insertion loss prediction of large acoustical enclosures based on the SEA. We assume that the number of modes in the sound field and panel are sufficiently high enough to construct the SEA model, in which we can derive the power balance equations among subsystems systematically. We compare the predicted insertion loss to the measurements for two cases of large acoustical enclosures to confirm the accuracy of prediction.

**INSERTION LOSS PREDICTION USING THE SEA**

An example of large acoustical enclosures for marine diesel generators is shown in Fig. 1.

*Fig. 1 An example of large acoustical enclosures for marine diesel generators.*
The insertion loss of an acoustical enclosure is defined as

\[ IL = 10 \log \frac{W_0}{W_{\text{Encl}}} \]  \hspace{1cm} (1)

in which \( W_0 \) is the acoustic power from the unenclosed source and \( W_{\text{Encl}} \) from the enclosed source. We use the SEA method to model an enclosure and surrounding space as shown in Fig. 2.

![Fig. 2 SEA modeling of an acoustical enclosure and surrounding space.](image)

The SEA model in Fig. 2 consists of three elements: sound field inside the enclosure (element 1), vibration energy of the wall panel (element 2), and sound field (element 3) outside the enclosure. The energy densities of the elements are related to the physical parameters as follows:

\[ E_1 = \langle p_1^2 \rangle \frac{V}{\rho c^2}, \]  \hspace{1cm} (2)

\[ E_2 = \langle v^2 \rangle M_p, \]  \hspace{1cm} (3)

\[ E_3 = \langle p_2^2 \rangle \frac{V_{\text{outside}}}{\rho c^2}, \]  \hspace{1cm} (4)

in which \( \langle p_1^2 \rangle \) represents the sound pressure level inside the enclosure, \( V \) volume of the enclosure, \( \rho \) density of air, \( c \) speed of sound in air, \( \langle v^2 \rangle \) squared velocity of the enclosure panel, \( M_p \) mass of the panel, \( \langle p_2^2 \rangle \) sound pressure level outside the enclosure, and \( V_{\text{outside}} \) volume of the outside space.

When a sound source generates acoustic power \( W_0 \) at a frequency \( \omega \), governing equations for the SEA are given by

\[ (\eta_1 + \eta_{12} + \eta_{13})E_1 - \eta_{21}E_2 - \eta_{31}E_3 = \frac{W_0}{\omega}, \]  \hspace{1cm} (5)

\[ -\eta_{12}E_1 + (\eta_2 + \eta_{21} + \eta_{23})E_2 - \eta_{32}E_3 = 0, \]  \hspace{1cm} (6)

\[ -\eta_{13}E_1 - \eta_{23}E_2 + (\eta_3 + \eta_{31} + \eta_{32})E_3 = 0, \]  \hspace{1cm} (7)
in which \( \eta_i \) represents internal loss factor of the \( i_{th} \) element and \( \eta_{ij} \) coupling loss factor between \( i_{th} \) and \( j_{th} \) elements where \( i, j = 1, 2, 3 \).

We assume that the space surrounding the enclosure is sufficiently large so that there is no energy flow from element 3 to element 1 or 2.

\[
\eta_{31}E_3 = \eta_{32}E_3 = 0. \tag{8}
\]

The internal and coupling loss factors are given by

\[
\eta_i = \frac{cA\alpha}{4\omega V}, \quad \eta_{ij} = \frac{c\alpha}{4\omega V}, \quad \eta_{21} = \eta_{32} = \frac{\rho c\sigma}{\omega h\rho_p}, \tag{9}
\]

in which \( \alpha \) is the averaged sound absorption coefficient inside the enclosure, and \( \tau, A, \sigma, h, \rho_p \) are parameters associated with the panel: sound transmission coefficient, area, radiation efficiency, thickness, and density respectively. Although the wall panel of a large acoustic enclosure usually consists of a steel plate and mineral wool (density is 40–50 kg/m\(^3\)), we assume that vibration behavior of the steel plate is not affected by the soft mineral wool, for which case \( \eta_{2i} = \eta_{3j} \) holds.

We used the radiation efficiency formula from the paper by Crocker and Price (1969). The coupling loss factor \( \eta_{12} \) can be computed from the relation

\[
\eta_{12} = \eta_{21} \frac{n_2}{n_1}, \tag{10}
\]

in which \( n_1 \) and \( n_2 \) are modal densities of the element 1 and 2 given by

\[
n_1(\omega) = \frac{\omega^2 V}{2\pi^2 c^3}, \quad n_2(\omega) = \frac{\sqrt{3}A}{2\pi c_L h}. \tag{11}
\]

In Eq. (11), \( c_L \) is the longitudinal wave speed of the panel given by

\[
c_L = \sqrt{\frac{E}{ho_p(1-\nu^2)}}, \tag{12}
\]

in which \( E \) and \( \nu \) are Young’s modulus and Poisson’s ratio of the wall panel respectively. From Eqs. (5) and (6), we can obtain the relation between input power \( W_0 \) and energy density \( E_1 \). Alternatively, we can express \( < p_i^2 > \) in terms of \( W_0 \) as

\[
A < p_i^2 > \geq \frac{(\alpha + \tau + \delta)}{4\rho c} \eta_{21}E_3 = W_0, \tag{13}
\]

in which

\[
\delta = \frac{D(\eta_2 + \eta_{21})}{\eta_2 + 2\eta_{21}}, \tag{14}
\]

where \( D \) is the non-dimensional constant given by
\[ D = \frac{4\sqrt{3} \pi c^3 \sigma}{\omega^2 \rho c / h^2}. \]  

(15)

Note that expression of \( \delta \) is different from Vér’s papers (Vér, 1973; 2006). To simplify Eq. (13), we need to know how small or large \( \delta \) is compared to \( \alpha \) or \( \tau \). For typical large enclosures in marine applications, structural damping of the panel, \( \eta_2 \) in Eq. (14) is order of \( O(10^{-2}) \), while \( \eta_{21} \) is order of \( O(10^{-4} \sim 10^{-5}) \). Hence, \( \delta \) is roughly equal to \( D \), which is order of \( O(10^{-2} \sim 10^{-3}) \) and comparable to \( \tau \).

The acoustic power \( W_{\text{trans}} \) transmitted to the outside of the enclosure, which is equal to \( W_{\text{Enc}} \) in Eq. (1), consists of the power transmitted through the panel and the power generated by the panel radiation. After rearrangement of Eq. (7), the acoustic power \( W_{\text{trans}} \) is given by

\[
W_{\text{trans}} = \omega \eta_{13} E_1 + \omega \eta_{23} E_2 = \frac{A \tau < p^2 >}{4 \rho c} + \rho c \sigma A < v^2 >.
\]

(16)

Note that the power generated by the panel radiation, \( \omega \eta_{23} E_2 \), is absent in the previous works (Vér, 1973, 2006). Eq. (16) can be rewritten as

\[
W_{\text{trans}} = \frac{A < p^2 >}{4 \rho c} [\tau + \beta],
\]

(17)

in which

\[
\beta = \frac{D \eta_{21}}{\eta_2 + 2 \eta_{21}}.
\]

(18)

The insertion loss of the sound enclosure is given by

\[
IL = 10 \log \frac{W_0}{W_{\text{trans}}} = 10 \log \frac{\alpha + \tau + \delta}{\tau + \beta}.
\]

(19)

So far, we have assumed that the transfer of the acoustic power among elements occurs through one big panel whose area is equal to the interior surface area of the enclosure. In reality, the enclosure consists of 6 sides, of which panels might have different thickness and material properties. For instance, floor of the enclosure is usually much stiffer and heavier than wall panel. When construction of the panel for each side is not the same, governing equations for the SEA must include summation over different panels. If total number of panels is \( N \), Eqs. (5)-(8) become

\[
(\eta_1 + \sum_j \eta_{1j}^f + \sum_j \eta_{1j}^l) E_i - \sum_j \eta_{1j}^f E_2^j = W_0 / \omega,
\]

(20)

\[- \eta_{1j}^l E_i + (\eta_{1j}^l + \eta_{1j}^f + \eta_{1j}^l) E_2^j = 0, \quad j = 1, \ldots, N, \]

(21)

\[
\sum_j \eta_{1j}^l E_i + \sum_j \eta_{1j}^f E_2^j = W_{\text{trans}} / \omega.
\]

(22)

After eliminating \( E_2^j \) from Eq. (21), and rearranging Eqs. (20) and (22), we can derive \( IL \) as
\[ IL = 10 \log \left[ \eta_1 + \sum_{j=1}^{N} \eta_1^j \left( \frac{\eta_2^j + \eta_2^{21}}{\eta_2^j + 2\eta_2^{21}} \right) \right] / \left( \sum_{j=1}^{N} \frac{\eta_2^j \eta_2^{21}}{\eta_2^j + 2\eta_2^{21}} \right) \]  \tag{23}

We introduce averaged parameters defined as

\[ \tau_{\text{avg}} = \sum_{j=1}^{N} \frac{j}{A_j} \left( \frac{A_j}{A} \right), \delta_{\text{avg}} = \sum_{j=1}^{N} J \left( \frac{A_j}{A} \right), \beta_{\text{avg}} = \sum_{j=1}^{N} B \left( \frac{A_j}{A} \right), \]

in which \( \tau_j \) and \( A_j \) are sound transmission coefficient and area of the \( j \)th panel, and \( \delta_j \) and \( \beta_j \) are coefficients associated with the material properties of the \( j \)th panel. Using the averaged parameters, we can rewrite summation expressions as

\[ \sum_{j} \eta_1^j = \sum_{j} \frac{cA_j \tau_j}{4\omega V} = \frac{cA}{4\omega V} \tau_{\text{avg}}, \]  \tag{24}

\[ \sum_{j} \eta_2^j \left( \frac{\eta_2^j + \eta_2^{21}}{\eta_2^j + 2\eta_2^{21}} \right) = \frac{cA}{4\omega V} \sum_{j} D_j \frac{A_j \left( \eta_2^j + \eta_2^{21} \right)}{A(\eta_2^j + 2\eta_2^{21})} = \frac{cA}{4\omega V} \sum_{j} J \left( \frac{A_j}{A} \right) = \frac{cA}{4\omega V} \delta_{\text{avg}}, \]  \tag{25}

\[ \sum_{j} \frac{\eta_2^j \eta_2^{21}}{\eta_2^j + 2\eta_2^{21}} = \frac{cA}{4\omega V} \sum_{j} D_j \frac{A_j \eta_2^{21}}{A(\eta_2^j + 2\eta_2^{21})} = \frac{cA}{4\omega V} \sum_{j} B \left( \frac{A_j}{A} \right) = \frac{cA}{4\omega V} \beta_{\text{avg}}. \]  \tag{26}

The insertion loss in Eq. (23) becomes

\[ IL = 10 \log \frac{\alpha + \tau_{\text{avg}} + \delta_{\text{avg}}}{\tau_{\text{avg}} + \beta_{\text{avg}}}. \]  \tag{27}

When all panels consist of the same construction, they have the same sound transmission coefficient. However, \( \delta_j \) and \( \beta_j \) are dependent on the panel size, since \( \delta_j \) and \( \beta_j \) contain radiation efficiency that is a function of the perimeter and area of the panel (Crocker and Price, 1969). For typical large enclosures, it is satisfied that \( \alpha \gg \tau_{\text{avg}} \) and \( \alpha \gg \delta_{\text{avg}} \), which will be confirmed in the numerical examples. Hence, Eq. (27) is simplified as

\[ IL \approx 10 \log \alpha / (\tau_{\text{avg}} + \beta_{\text{avg}}). \]  \tag{28}

Moreover, except near or above the critical frequency of the panel, contribution from the sound radiation is much smaller than the one from transmission through the panel, which means \( \tau_{\text{avg}} \gg \beta_{\text{avg}} \). Insertion loss in Eq. (28) can be further simplified as

\[ IL \approx 10 \log \left( \alpha / \tau_{\text{avg}} \right) = 10 \log (\alpha) + TL, \]  \tag{29}

in which \( TL \) is the sound transmission loss defined as

\[ TL = -10\log(\tau_{\text{avg}}). \]  \tag{30}

Eq. (29) is the same as found in the references (Vér, 1973; 2006).
MEASUREMENT OF THE INSERTION LOSS

The first example is an enclosure for diesel engines in naval ships. The enclosure dimension is 6.4 m × 2.65 m × 4.8 m (L × W × H), and construction of the panel is as follows: 1.5 mm steel plate + damping layer + 70 mm mineral wool + perforated metal sheet. The enclosure was installed in a large factory and we measured the insertion loss by using a speaker as the sound source. In-situ measurement procedure of the insertion loss for sound enclosure is described in ISO (1995) which requires measurement of the sound power with and without the enclosure. Since sound power measurement is a standard procedure which needs to be done in accordance with ISO (2010) or ISO (2010), we skipped details of the measurement here. In addition, we measured TL of the panel and absorption coefficient inside the enclosure as shown in Table 1. The high absorption value at 125 Hz was probably due to the modes inside the enclosure. Although the composite panel has basically sound absorbing surface, not all the interior surface is covered by the panel. The bottom is hard floor, and many accessory parts and equipments are attached to the wall and ceiling, which may explain low absorption coefficient at high frequencies.

If no TL data is available, we can predict TL of the panel by assuming a mass-law (Vér, 2006). However, TL and sound absorption coefficient are best obtained by measurement, not by simulation.

To check if the number of modes is large enough to apply the SEA, we computed the number of modes at 125 Hz by multiplying bandwidth to modal densities in Eq. (11). The number of modes of the sound field and the plate of the smallest area (2.65 m × 4.8 m) at 125 Hz in one-octave band is 36 and 239 respectively, which is sufficiently high for applying the SEA.

The structural damping of a thin steel plate is in general order of O(10^{-3} \sim 10^{-4}) . However, such small value of damping can be obtained only for a single plate hanging in the air by strings. In our example, the panel consists of a steel plate, damping layer, and mineral wool, for which we have assumed \( \eta_2 = 0.05 \) as a typical value. As long as frequency range of the interest is below the critical frequency of the wall panel, structural damping of order of O(10^{-2} \sim 10^{-3}) does not affect the approximate formula, \( IL = 10\log(\alpha) + TL \) as shown below. However, beyond the critical frequency, damping can affect the insertion loss significantly. We assumed that vibration behavior of the composite panel is dominated by the steel plate so that the radiation efficiency of the panel is the same as that of 1.5 mm steel plate.

In Table 1, we showed TL, \( \tau(=\tau_{avg}) \), \( \alpha \), \( \delta_{avg} \), and \( \beta_{avg} \), from which it is confirmed that \( \alpha >> \tau \), \( \alpha >> \delta_{avg} \), and \( \tau >> \beta_{avg} \), where the only exception is \( \beta_{avg}/\tau = 0.41 \) at 8,000 Hz. The sound transmission loss of a plate shows a dip at the critical frequency given by

\[
f_c = \frac{c^2}{\pi h} \sqrt{\frac{3\rho_c(1-v^2)}{E}}.
\]

The critical frequency of the 1.5 mm steel plate is 7,800 Hz. Since internal loss factor of the panel \( \eta_2 \) is much larger than the coupling loss factor \( \eta_{21} \), it can be shown from Eqs. (15) and (18) that \( \beta \propto \sigma^2 \). As frequency increases, radiation efficiency \( \sigma \) increases monotonically. At the critical frequency, radiation efficiency reaches the highest value, which explains why \( \beta_{avg} \) becomes comparable to \( \tau \) at 8,000 Hz in Table 1. We compared predicted insertion loss by Eq. (29) to measurement in Fig. 3, which shows good agreement. At 8,000 Hz, insertion loss by Eq. (28) is 32.9 dB, while insertion loss by Eq. (29) is 34.3 dB. Therefore, it is recommended to use Eq. (28) if the critical frequency of the wall panel falls into the frequency region of interest.

Table 1 TL and sound absorption coefficient for the 1st sound enclosure example.

<table>
<thead>
<tr>
<th>Hz</th>
<th>( \alpha )</th>
<th>TL (dB)</th>
<th>( \tau )</th>
<th>( \delta_{avg} )</th>
<th>( \beta_{avg} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.36</td>
<td>15.4</td>
<td>2.9E-2</td>
<td>1.0E-2</td>
<td>5.8E-6</td>
</tr>
<tr>
<td>250</td>
<td>0.39</td>
<td>29.2</td>
<td>1.2E-3</td>
<td>3.5E-3</td>
<td>1.3E-6</td>
</tr>
<tr>
<td>500</td>
<td>0.36</td>
<td>33.6</td>
<td>4.4E-4</td>
<td>1.3E-3</td>
<td>3.4E-7</td>
</tr>
<tr>
<td>1,000</td>
<td>0.43</td>
<td>33.5</td>
<td>4.5E-4</td>
<td>4.7E-4</td>
<td>9.7E-8</td>
</tr>
<tr>
<td>2,000</td>
<td>0.42</td>
<td>37.4</td>
<td>1.8E-4</td>
<td>2.0E-4</td>
<td>3.5E-8</td>
</tr>
<tr>
<td>4,000</td>
<td>0.42</td>
<td>36.1</td>
<td>2.5E-4</td>
<td>1.2E-4</td>
<td>2.4E-8</td>
</tr>
<tr>
<td>8,000</td>
<td>0.38</td>
<td>38.5</td>
<td>1.4E-4</td>
<td>4.2E-3</td>
<td>5.7E-5</td>
</tr>
</tbody>
</table>
The 2nd example is from the paper by Vér (2006), in which the enclosure dimension is $4.5 \times 2.5 \times 2 \text{ m}$ and panel is constructed of $1.5 \text{ mm}$ steel plate and $70 \text{ mm}$ mineral wool for interior lining. Measurements of $TL$ and sound absorption coefficient were reported in the reference cited therein. In the paper by Vér (2006), it was assumed that there exist leaks accounting for $0.01\%$ of the panel area. If leakage exists, sound transmission coefficient in Eq. (30) should be modified as

$$\tau_{\text{leak}} = \tau_{\text{avg}} + \varepsilon,$$  \hspace{1cm} (32)

in which $\varepsilon$ denotes the ratio of the leakage area to the panel area and it is assumed that $\varepsilon \ll 1$.

In Fig. 4, we compared two predictions (no leaks and $0.01\%$ leaks) to measurement, in which Eq. (29) are used. Vér (2006) concluded that $0.01 \%$ leaks yielded a prediction that matches reasonably well the measured data. However, prediction with assumption of no leaks results in a significant overestimation. In the 1st example, we did not assume any leaks, since the effect of leaks was already included in $TL$ that was in-situ measured. For the 2nd example, we confirmed that $\alpha \gg \tau_{\text{leak}}$, $\alpha \gg \delta_{\text{avg}}$, and $\tau_{\text{leak}} \gg \beta_{\text{avg}}$ except that $\beta_{\text{avg}} / \tau_{\text{leak}} = 0.51$ at $8,000 \text{ Hz}$. 

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**Fig. 3** Comparison of the predicted and measured insertion loss of the sound enclosure for a marine diesel engine. The dimension is $6.4 \times 2.65 \times 4.8 \text{ m} (L \times W \times H)$.  

**Fig. 4** Comparison of the predicted and measured insertion loss of the sound enclosure. Measurements of $IL$, $TL$ and sound absorption coefficient were from Vér (2006). The dimension is $4.5 \times 2.5 \times 2 \text{ m}$ and $0.01\%$ leaks was assumed.
CONCLUSIONS

To obtain high insertion loss of the large acoustical enclosures, it is important that sound transmission loss of the enclosure panel and averaged sound absorption coefficient inside the enclosure must be as high as possible. Since leakage can severely degrade the sound transmission loss, extreme care is needed to seal all leaks. In addition, damping of the panel must be large in order to minimize the contributions from sound radiation. We derived a simple formula for insertion loss of large acoustic enclosures using the SEA. It was found if the critical frequency of the wall panel falls above the frequency region of interest, insertion loss is dominated by the sound transmission loss of the wall panel and averaged sound absorption coefficient inside the enclosure, which was shown in the previous works. However, if the critical frequency of the wall panel falls into the frequency region of interest, it is recommended to use the formula developed in this paper.

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