4pNSb2. Noise reduction from large machineries by using sound enclosures

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A sound enclosure is an effective measure to reduce the noise emitting from the large noise sources such as diesel engines and gas turbines. In this study, insertion loss prediction of the large enclosure is presented. Inside the enclosure, diffuse sound field is assumed, and there exist no air leakages. Insertion loss is predicted by using SEA (Statistical Energy Analysis). From the energy equilibrium equations, sound pressure inside the enclosure is derived in terms of the acoustic power from the machinery. Insertion loss is defined as the ratio between acoustic power inside and transmitted power outside the enclosure. It is shown that sound radiation from the panel vibration can be neglected compared to that transmitted through panel. Insertion loss predictions are compared to the measurements. The enclosure size is 6.4 m x 2.65 m x 4.8 m (L x W x H) and 4.5 m x 2.5 m x 2.0 m, where panel consists of 1.5 mm steel plate and 70 mm mineral wool. The comparisons show good agreements. It is concluded that to increase the insertion loss, panel must have a large sound transmission loss and sound absorption coefficient inside the enclosure must be high.

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INTRODUCTION

Acoustic enclosures are very effective noise control measures for reducing noise emitting from the sources like diesel engines and air compressors. Performance of the acoustic enclosures is described by the insertion loss (IL) defined as the difference between acoustic powers with and without the enclosure. In research vessels and naval ships where low noise is of prime concern, large acoustic enclosures are frequently used to reduce high noise from the diesel engines, in which wavelength is small compared to the dimensions of the enclosure. In Refs. 1-3, formulas to predict insertion loss are described depending on the size of the enclosure. Insertion loss of the large acoustic enclosures are mainly determined by the sound transmission loss of the enclosure panel and averaged sound absorption coefficient inside the enclosure.\textsuperscript{1,3} However, insertion loss can be degraded by leakages, poor treatment of the openings, and flanking transmission of structure-borne noise. Reference 3 noted that typical panel of the sound 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.

In this paper, we study the insertion loss of the large acoustic enclosures for diesel engines used in naval ships. The dimension of the enclosures ranges from several meters to more than ten meters. We use the SEA (Statistical Energy Analysis) to derive insertion loss based on the assumption that sound field inside the enclosure is diffuse. We compare the predicted insertion loss to the measurements for two cases of the large acoustic enclosures.

I. INSERTION LOSS PREDICTION BY SEA

The insertion loss (IL) of the sound enclosure is defined as\textsuperscript{1}

\[
IL = 10 \log\frac{W_0}{W_{Encl}}
\]  

(1)

where \(W_0\) is the acoustic power from the unenclosed source and \(W_{Encl}\) from the enclosed source.

![Figure 1. SEA modeling of the sound enclosure and surrounding space.](image-url)

We assume that sound field inside the enclosure is diffuse and employ the SEA to build energy balance equations between the sound enclosure and surrounding space. As shown Fig. 1, the model consists of three elements: sound field inside the enclosure (element 1), enclosure 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:
\[ E_1 = \langle p_i^2 \rangle > \frac{V}{\rho c^2}, \]  
\[ E_2 = \langle v^2 \rangle > M_p, \]  
\[ E_3 = \langle p_{outside}^2 \rangle > \frac{V_{outside}}{\rho c^2}, \]

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

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

\[ (\eta_1 + \eta_{12} + \eta_{13}) E_1 - \eta_1 E_2 - \eta_{13} E_3 = W / \omega, \]  
\[ -\eta_{12} E_1 + (\eta_2 + \eta_{21} + \eta_{23}) E_2 - \eta_{13} E_3 = 0, \]  
\[ -\eta_{13} E_1 - \eta_{23} E_2 + (\eta_3 + \eta_{31} + \eta_{32}) E_3 = 0, \]

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 outside space surrounding the enclosure is sufficiently large so that there is no energy flow from element 3 to element 1 or 2. Therefore, we set

\[ \eta_1 E_3 = \eta_{12} E_3 = 0. \]  

The internal and coupling loss factors are given as

\[ \eta_1 = \frac{cA\alpha}{4\omega V}, \quad \eta_{13} = \frac{cA\tau}{4\omega V}, \quad \eta_{21} = \eta_{23} = \frac{\rho c^2\alpha A}{\omega M_p}, \]  

in which \( \alpha \) is the averaged sound absorption coefficient inside the enclosure, and \( \tau, A, \sigma \) are parameters associated with the panel: sound transmission coefficient, area, and radiation efficiency respectively. We used radiation efficiency formula from Ref. 4. The coupling loss factor \( \eta_{12} \) can be computed from the relation

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

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

\[ n_1(\omega) = \frac{\omega^2 V}{2\pi^2 c^3}, \quad n_2(\omega) = \frac{\sqrt{3} A}{2\pi c_\lambda h}, \]
where \( c_L \) is the longitudinal wave speed of the panel, and \( h \) thickness of the panel. From Eqs. (5) and (6), we can obtain the relation between input power \( W_i \) and energy density \( E_i \). Alternatively, we can express \(< p_i^2 >\) in terms of \( W_i \) as

\[
\frac{A}{4\rho c} < p_i^2 > [\alpha + \tau + \delta] = W_i,
\]

(12)

where

\[
\delta = \frac{D(\eta_1 + \eta_{23})}{\eta_2 + 2\eta_{21}},
\]

(13)

in which \( D \) is the constant associated with radiation efficiency

\[
D = \frac{4\sqrt{3} \pi \rho_c \sigma}{\omega^2 \rho_e c h}.
\]

(14)

In Eq. (14), \( \rho_p \) is the surface density (kg/m²) of the panel. For a typical large sound enclosure encompassing marine diesel engines, non-dimensional constants \( \eta_2 \), and \( D \) are much smaller than 1. However, \( \delta \) is comparable to \( \tau \).

The acoustic power transmitted to outside the sound enclosure consists of the power transmitted through the panel and the power generated by the panel radiation

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

(15)

Eq. (15) can be rewritten as

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

(16)

in which

\[
\beta = \frac{D \eta_{23}}{\eta_2 + 2\eta_{21}}.
\]

(17)

The insertion loss of the sound enclosure is defined as the power ratio given by

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

(18)

For a typical large sound enclosure, it is satisfied that \( \alpha >> \tau \) and \( \alpha >> \delta \). Moreover, contribution from the sound radiation is much smaller than the one from transmission through the panel, which means \( \tau >> \beta \). Hence, insertion loss in Eq. (18) becomes

\[
IL \approx 10 \log (\alpha / \tau) = 10 \log (\alpha) + TL,
\]

(19)

where \( TL \) is the sound transmission loss of the panel defined as
II. MEASUREMENT OF THE INSERTION LOSS

The first example is the sound enclosure for diesel engines in naval ships. The enclosure dimension is 6.4 m x 2.65 m x 4.8 m (L x W x H), and panel consists of 1.5 mm steel plate and 70 mm mineral wool. 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 11546-2.\(^2\) In addition, we measured \(TL\) of the panel and absorption coefficient inside the enclosure. We assumed that internal loss factor of the panel is \(n_i = 0.05\). In Table 1, we showed \(TL\), \(\tau\), \(\alpha\), \(\delta\), and \(\beta\) from which it is confirmed that \(\alpha \gg \tau\), \(\alpha \gg \delta\), and \(\tau \gg \beta\), where the only exception is \(\beta / \tau = 0.41\) at 8000 Hz. We compared predicted insertion loss by Eq. (19) to measurement in Fig. 2, which shows good agreement.

<table>
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<tr>
<th>Hz</th>
<th>(TL) (dB)</th>
<th>(\tau)</th>
<th>(\alpha)</th>
<th>(\delta)</th>
<th>(\beta)</th>
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<tr>
<td>125</td>
<td>15.4</td>
<td>2.9E-2</td>
<td>0.36</td>
<td>1.9E-2</td>
<td>1.8E-5</td>
</tr>
<tr>
<td>250</td>
<td>29.2</td>
<td>1.2E-3</td>
<td>0.39</td>
<td>6.1E-3</td>
<td>3.8E-6</td>
</tr>
<tr>
<td>500</td>
<td>33.6</td>
<td>4.6E-4</td>
<td>0.36</td>
<td>2.1E-3</td>
<td>9.3E-7</td>
</tr>
<tr>
<td>1000</td>
<td>33.5</td>
<td>4.5E-4</td>
<td>0.43</td>
<td>8.0E-4</td>
<td>2.6E-7</td>
</tr>
<tr>
<td>2000</td>
<td>37.4</td>
<td>1.8E-4</td>
<td>0.42</td>
<td>3.4E-4</td>
<td>9.2E-8</td>
</tr>
<tr>
<td>4000</td>
<td>36.1</td>
<td>2.5E-4</td>
<td>0.42</td>
<td>2.0E-4</td>
<td>6.4E-8</td>
</tr>
<tr>
<td>8000</td>
<td>38.5</td>
<td>1.4E-4</td>
<td>0.38</td>
<td>4.2E-3</td>
<td>5.8E-5</td>
</tr>
</tbody>
</table>

FIGURE 2. Comparison of the predicted and measured insertion loss of the sound enclosure for a marine diesel engine. The dimension is 6.4 m x 2.65 m x 4.8 m (L x W x H).

The 2\(^{nd}\) example is from Ref. 1, in which the enclosure dimension is 4.5 m x 2.5 m x 2 m and panel is constructed of 1.5 mm steel plate and 70 mm mineral wool for interior lining. Measurements of \(TL\) and sound absorption coefficient were reported in the reference cited therein. In Ref. 1, it was assumed that leaks account for 0.01 % of the panel area. If leakage exists, sound transmission loss in Eq. (20) should be modified as

\[ TL = -10 \log (\tau). \]
\[ TL = -10 \log(\tau + \epsilon), \]  

(21)

in which \( \epsilon \) denotes ratio of the leakage area to the total panel area.

In Fig. 3, we compared two predictions (no leaks and 0.01\% leaks) to measurement. In Ref. 1, it was concluded that 0.01\% leaks yielded a prediction that matches reasonably well the measured data. However, prediction with assumption of no leaks results in 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.

![Graph showing comparison of predicted and measured insertion loss of the sound enclosure.](image)

**FIGURE 3.** Comparison of the predicted and measured insertion loss of the sound enclosure. Measurements of \( IL, TL \) and sound absorption coefficient were from Ref. 1. The dimension is 4.5 m x 2.5 m x 2 m and 0.1 \% leaks was assumed.

## III. CONCLUDING REMARKS

To obtain high insertion loss of the large acoustic enclosures, it is important that sound transmission loss of the enclosure panel and averaged sound absorption coefficient inside the enclosure must be high. Since leakage can severely degrade the sound transmission loss, extreme care is needed to seal all the leaks. In addition, damping of the panel must be large. It was found that in determination of the transmitted acoustic power outside the enclosure, sound radiation from the panel is negligible compared to the one from transmission through the panel.

## REFERENCES