PREDICTION OF SOUND TRANSMISSION LOSS OF SINGLE LEAF WALLS DUE TO AN IMPROVEMENT OF THE SEA METHOD

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ABSTRACT

The SEA (statistical energy analysis) method can accurately predict the sound transmission loss of a finite wall. However, there is a large discrepancy between the calculated value and the measured value at lower frequencies, since the random incidence mass law deduced for the infinite panel is partly applied in this method.

In this paper, to improve this point, the radiation factor derived by investigating the directivity characteristic of transmitted waves through a finite wall was introduced into the coupling loss factor between source and receiving rooms. The experimental and theoretical values agreed well over a wide frequency range.

1. Introduction

It is a well known fact that the measured sound transmission losses of single leaf walls by use of the two-room method are generally greater than the calculated values by using the random incidence mass law, when the area of wall becomes smaller. The mass law of sound transmission through an infinite panel by A. London[1] is popular as the classical method.

On the other hand, M. J. Crocker and A. J. Price[2] introduced the Statistical Energy Analysis (SEA) Method to predict the sound insulation of the finite walls. The SEA method is characterized mainly in that the total sound transmission energy through a finite partition is the sum of the sound radiation energy on resonant mode and on nonresonant mode. Further, the transmission suite (source and receiving rooms) is also analyzed. The theoretical values agree well with the experimental ones in higher frequencies around the coincidence effect region. However, there is a large difference at low frequencies. Since these authors applied the 80° random incidence mass law of the infinite panel as the coupling loss factor (η_{13}) due to nonresonant transmission, this point remains to be improved.

There are some recent developments in the sound transmission theories for a finite wall. R. J. Donato[3] has deduced a correction factor at low frequencies by using a wavenumber approach. T. Kawai[4] has analyzed numerically the transmission mechanism of a normal incident plane wave through a circular plate. It was indicated that the plate vibrates in such a way that a piston motion given as the mass law and a flexural motion damped by the internal loss are superposed, further the later one is influenced by the edge conditions.

H. Sato[5] has studied the directivity characteristic of transmitted wave of a rectangular wall vibrating with an infinite baffle. It becomes non–directional in order the higher value of incident angle when the wavenumber of incident sound or the dimensions of wall diminishes from infinity. Therefore, the window whose area is small does not show characteristic of infinte wall in the lower frequency region. The above results are considered to correspond with the 80° random incidence mass law by Crocker and Price which agrees well with the measured value. The author ascertained that the calculated values by using the 80° random incidence mass law agreed with our own experimental values[6].

While, a later paper by A. Elmallawany[7] demonstrated to improve the coupling loss factor of nonresonant transmission at low frequencies using the correction factor deduced by Donato. Here the calculated values are in good agreement with the measured values at the low frequencies.

The author has attempted to compensate the SEA method for the finite size of the single leaf wall. The radiation factor deduced by Sato was introduced into the coupling loss factor of nonresonant transmission. The measured and theoretical values of sound transmission loss are made to reach good agreement over wide frequency range by using our method.

2. Improvement of the Coupling Loss Factor η_{13} for Nonresonant Transmission

Crocker and Price obtained the coupling loss factor η_{13} due to nonresonant mass law transmission between the source room and the receiving room as the following relationship[1,8] :

$$10 \log \eta_{13} = -\mathrm{TL_m} + 10 \log\left(\frac{Sc}{4V_1\omega}\right) \quad \cdots\cdots(1)$$

where S is the wall area; c is the speed of sound in air; V_1 is the volume of source room and ω is angular frequency. $\mathrm{TL_m}$ is the random incidence mass law TL value. This value is given by the TL of 80° random incidence mass law which is determined by comparing the theoretical value with the experimental one.

Using the correction factor according to Donato[3], Elmallawany obtained a least square equation and then improved the coupling loss factor as follows[7] ;

$$10 \log \eta_{13} = -\mathrm{TL_m'} + 10 \log\left(\frac{Sc}{4V_1\omega}\right) \quad \cdots\cdots(2)$$

$$\mathrm{TL_m'} = \mathrm{TL_m} + 5\left(\frac{ka}{2.3}\right)^{-0.72} \quad \cdots\cdots(3)$$

where $\mathrm{TL_m}$ is the transmission loss for a diffused sound field from the mass law, k is wave number, and a is half the smallest dimension of the panel based on the radius of the largest circle which can just fit into the rectangular area, respectively. Since the value in the parentheses of second term of Eq. (3) is limited within 1 to 6.5, this method can be applied to the relatively small walls or at the low frequency region.

Sato has deduced the modified sound transmission loss $\mathrm{TL_S}$ of a finite wall for a random incidence sound field, baced on the theory of wave equation, as Eq. (4) and (5)[5].

$$\mathrm{TL_S} = \mathrm{TL_0} - 10 \log Q - 3 \quad \cdots\cdots(4)$$

$$Q \fallingdotseq \int_0^{2\pi} \lambda(ka) \sin\theta \, d\theta \quad \cdots\cdots(5)$$

where $\mathrm{TL_0}$ is the normal incidence mass law and Q represents spatial average of radiation factor $\lambda(ka)$ for a spatial direction of incident plane wave (incident–angle θ) in receiving room, and a is half a dimension of wall. The Q value of the square wall has been calculated numerically. In the case that the wall is not square, the derivation of Q is within ± 1 to 2 dB.

The author has tried to introduce this radiation factor into the coupling loss factor due to nonresonant transmission of the SEA method. To facilitate computer calculation, the values of Q calculated by Sato in Reference 5 are converted in the following procedure:

$$\Delta \mathrm{R} = -10 \log Q + 5 \quad \cdots\cdots(6)$$

and Eq. (7) represents the mathematical form.

$$\Delta R = A(ka)^B \qquad \cdots\cdots(7)$$

The two constants, A and B, can be determined by the least square method. The errors remain less than ±0.5dB. Finally, the coupling loss factor is obtained as follows:

$$10 \log \eta_{13} = -TL_S + 10 \log\left(\frac{Sc}{4V_1\omega}\right) \qquad \cdots\cdots(8)$$

$$TL_S = TL_0 + 9.2(ka)^{-0.51} - 8 \qquad \cdots\cdots(9)$$

where *ka* is limitied within 0.5 to 64.

3. Verification of the Improved SEA Method

The experimental work is done in the Kagoshima University transmission suite laboratory in conformity to the JIS Standards "Method for Laboratory Measurement of Sound Transmission Loss" (JIS A 1416). The volume of source and receiving rooms are 207m^3 and 102m^3, respectively. The panels were measured 2,468 by 2,377mm and divided vertically into two parts by a wooden stud (50× 100 mm section). The panel was nailed on to both the wooden frame and the stud. The distance of every nail is about 15cm. Table 1 shows the physical properties of the panels.

To calculate the sound transmission loss of the wall using the SEA theory[1,8], some parameters must be known beforehand; for example, the volume of the source and receiving rooms and the reverberation time of the receiving room when the object panel is mounted. Since the panel is divided by a rigid stud, the rectangular wall were calculated on the half area (1,234×2,377 mm^2) as a square wall having the same area. The internal loss factor η_{int} used in the prediction was 0.01. The sound speed in air was calculated by measuring the temperature of air in rooms. The panel edge conditions were intended to be fixed.

Figure 1 and 2 show the comparisons between the measured TL of single wall of aluminium panel

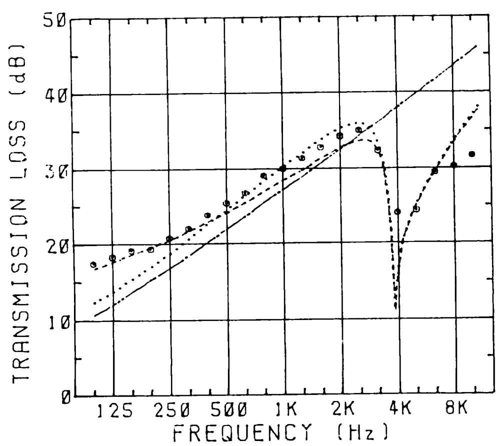
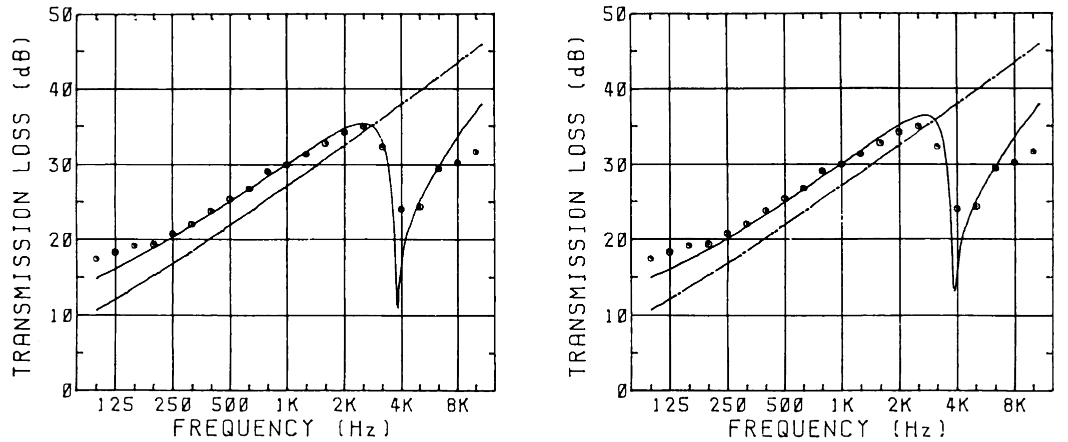


Figure 1. Theoretical and experimental transmission losses for 3.2mm aluminium panel.
○ ; experimental data,
— - — ; theoretical random incidence mass law,
······; theoretical according to Crocker and Price, Eq. (1),
- - - -; theoretical according to Elmallawany, Eq. (3),
η_{int}=0.01.

Table 1. Surface Density and Speed of Longitudinal Waves of Panels

Matearial	m kg/m²	c_L m/sec
3.2mm Aluminium	8.6	5150
1.2mm Aluminium	3.2	5150
1.5mm Steel	11.7	5050



(a) 1,234mm×2,377mm; Dimensions of panel in computing. (b) 2,468mm×2,377mm; Dimensions of panel in computing.

Figure 2. Theoretical and experimental transmission losses for 3.2mm aluminium panel.

○ ; experimental data,

—·— ; theoretical random incidence mass law,

——; theoretical according to our method, Eq. (9),

$\eta_{int}=0.01$.

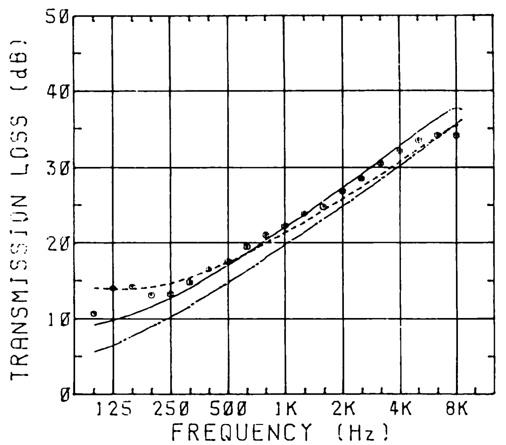


Figure 3. Theoretical and experimental transmission losses for 1.2mm aluminium panel.

○ ; experimental data,

—·— ; theoretical random incidence mass law,

----; theoretical according to Elmallawany, Eq. (3),

——; theoretical according to our method, Eq. (9),

$\eta_{int}=0.01$.

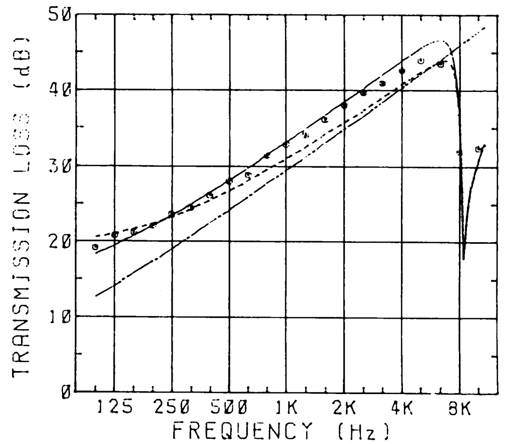


Figure 4. Theoretical and experimental transmission losses for 1.5mm steel panel.

○ ; experimental data,

—·— ; theoretical random incidence mass law,

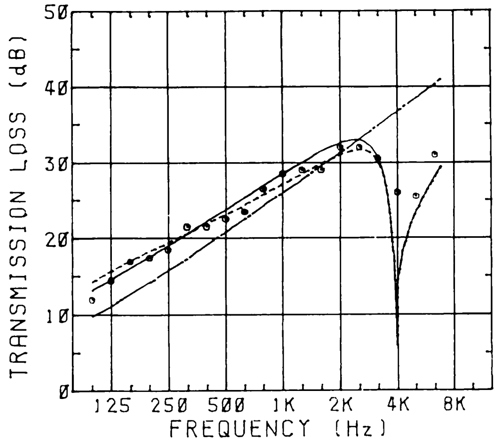
----; theoretical according to Elmallawany, Eq. (3),

——; theoretical according to our method, Eq. (9),

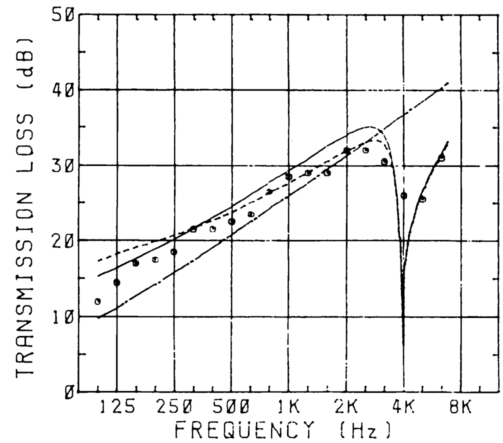
$\eta_{int}=0.01$.

(3.2mm thick) and the calculated one from each coupling loss factor described above. To facilitate the comparisons, the random incidence mass law based on the infinite aluminium panel is drawn in each figure as a dashed—dotted line.

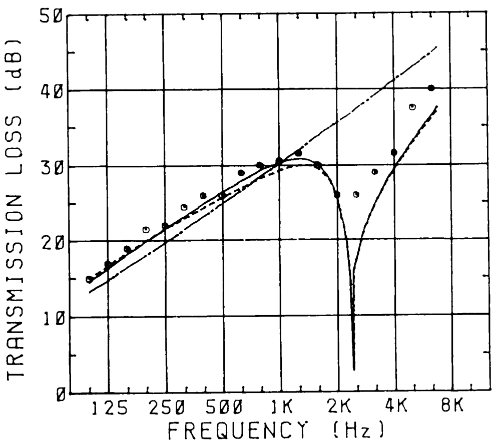
The calculated value by Eq. (1) according to Crocker and Price and the our own experimental



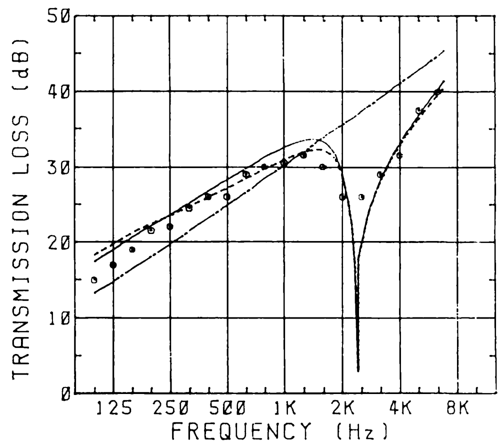
(a)– 1 3mm glass ($\eta_{int}=0.01$)



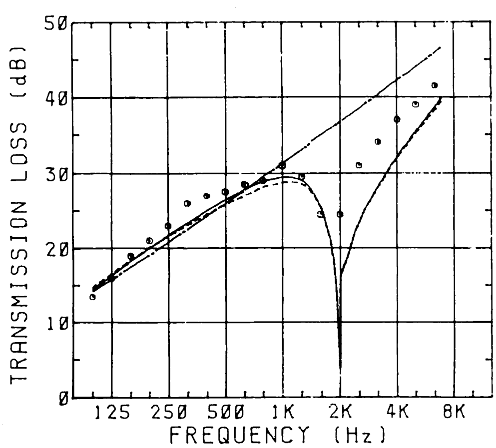
(a)– 2 3mm glass ($\eta_{int}=0.03$)



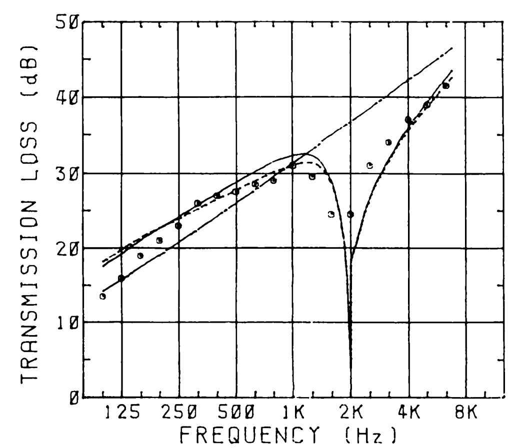
(b)– 1 5mm glass ($\eta_{int}=0.01$)



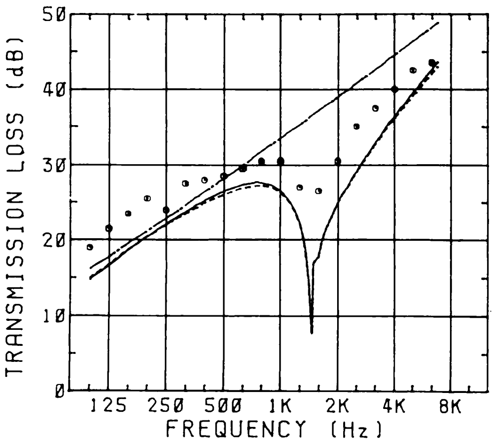
(b)– 2 5mm glass ($\eta_{int}=0.03$)



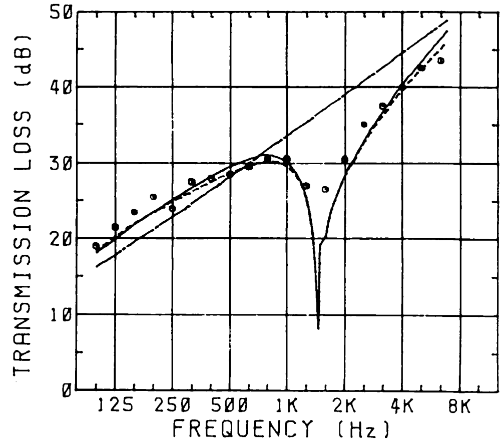
(c)– 1 6mm glass ($\eta_{int}=0.01$)



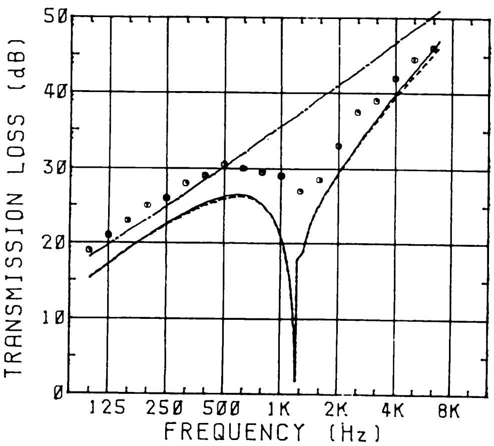
(c)– 2 6mm glass ($\eta_{int}=0.03$)



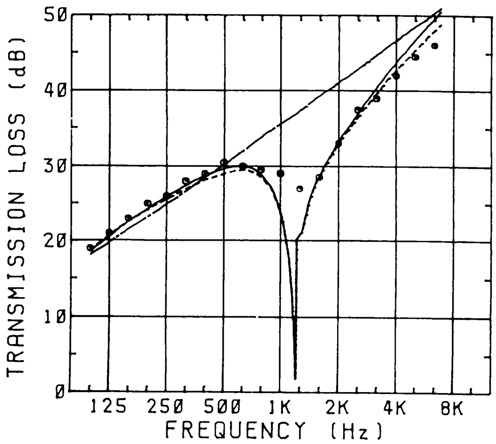
(d)— 1 8mm glass ($\eta_{int}=0.01$)



(d)— 2 8mm glass ($\eta_{int}=0.03$)



(e)— 1 10mm glass ($\eta_{int}=0.01$)



(e)— 2 10mm glass ($\eta_{int}=0.03$)

Figure 5. Theoretical and experimental transmission losses for glasses.

○ ; experimental data,

—-— ; theoretical random incidence mass law,

----; theoretical according to Elmallawany, Eq. (3),

——; theoretical according to our method, Eq. (9),

value were in good agreement at high frequencies above 250Hz (Figure 1). However, the discrepancy becomes greater at lower frequency below 250Hz. On the other hand, the computed value using Eq. (2) and (3) from Elmallawany agreed very well with the measured one at low frequencies below 315Hz. But the deviation in the range of 400 to 2.5kHz as wellknown as the mass control region cannot be neglected. The above results can be considered to be caused by the applicable range of the correction factor according to Eq. (3). This range is limited below 500Hz for the measured wall.

The value calculated by using Eq. (8) and (9), due to the present improved coupling loss factor, predicts the transmission loss with high accuracy as shown in Figure 2— (a). Comparing the results shown in Figure 2— (b), in which the dimensions of wall are 2,468mm and 2,377mm in computing, we found that our method can also estimate the effect of a stud. The applicable range of frequency using Eq. (9) is lower than 4kHz for the measured wall.

Figure 3 and 4 show the results for 1.2mm thick aluminium panel and 1.5mm thick steel panel, in order. The solid and broken lines show the calculated values by using our method and Elmallawany's method, respectively. The applicable range of each method is equal to the 3.2mm thick aluminium panel.

As shown in Figures 1–4, the calculated values by using Eq. (8) and (9), our method, agree accurately with the measured values except a few low frequencies below 250Hz. However, it has to be mentioned that the Elmallawany's method agrees much better below 250Hz.

Kuga[9] has carried out the measurement for many materials. As the system of measurement was described in detail in Reference 9, our method was applied to the five kinds of glasses which the edges has been clamped by the wood frame with the soft rubber 5mm thick on the both sides. The effective dimensions for sound transmission are 1,470mm and 860mm except the area of wood frame.

In computing based on the SEA method, the volumes of both source and receiving rooms were the same value (125m^3), and the reverberation time of receiving room took the value in Table 1 of Reference 9, when the standardized test concrete panel (using mortared concrete block) was mounted, for every glasses commonly. The sound velocity in air took the value at normal temperature and the edge condition was assumed to be fixed. The two values of the internal loss factor of glasses in calculating employed 0.01 and 0.03.

In Figure 5, the comparisons of the theoretical values and the experimental values are shown. The three kinds of lines for the calculated values indicated the random incidence mass law (—•—), Elmallawany's method (----, applicable range≦800Hz), and our method (—— , applicable range≦6kHz).

It is found that the calculated values are varied greatly by the internal loss factor used. It is mentioned that the theoretical value using Elmallawany's method is nearly equal to the value using our method at low frequencies < 800Hz. Above the critical coincidence frequency, the theoretical value using the internal loss factor $\eta_{int} = 0.03$ is in good agreement with the TL of each glass. Below the critical coincidence frequency, while the computed values of the 3mm thick glass using $\eta_{int} = 0.01$ agree well, but the used η_{int} value increased in proportion to the thickness of glass, finally $\eta_{int} = 0.03$ can predict well the TL of the 10mm thick glass. It can be considered that the increasing of the surface density causes to lose the more energy at the edge by a relation between the surface density and the edge condition.

4. Conclusions

It has been shown that the improved SEA method using the present coupling loss factor can predict the sound transmission loss with high accuracy and can estimate also the effect of a stud. However, an attention has to be paid to the applicable range of frequency and to the value of the internal loss factor η_{int}. The deviation between computed and measured values was significantly influenced by the used η_{int} value, and the deviation below the critical coincidence frequency f_c is more sensitive than that above f_c. It can be presumed that the η_{int} value changes due to both the surface density and the edge condition.

Therefore, in order to obtain the extended coupling loss factor applicable for the wide frequency range, it will be necessary to study the frequency characteristic of the internal loss factor.

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