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Significance of resonant sound transmission in finite single partitions

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Abstract

The resonant sound transmission has been regarded to be negligible in finite single partitions compared to the non-resonant transmission. In this study, the sound transmission coefficient is reconsidered by using the general modal expansion method in order to estimate the contribution of resonant components to the whole sound transmission at frequency bands below the coincidence frequency. By investigating the band-average difference between the total and non-resonant transmission losses, the validity of the aforementioned assumption is discussed. In the analysis, the participation factor comprised of size, thickness, and loss factor is employed. Based on this factor, the valid range of neglecting the resonant sound transmission is proposed, in which the inclusion of the resonant component makes difference being less than 1 dB.

1. Introduction

The transmission of sound through plate-like partitions has been studied by a number of researchers. However, most of works has dealt with infinite partitions, while the deviations of measured transmission loss data from predicted results could be observed even in the case of single leaf partitions. The major causes of discrepancies between theory and measurement are the finiteness and mounting (or boundary) conditions, and the non-diffuseness of test room. In relation to the non-diffuseness, the concept of limit angle has been introduced [1] and the predicted result in general agrees well with experimental one. However, as many researchers have pointed out, the limit angle is usually chosen arbitrarily and, in the case of multiple panels, the

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predicted results are quite sensitive to the change of the limit angle value. In this regard, Kang et al. [2] introduced the directional weighting function for incidence energy, which made the prediction very accurate compared to the field incidence calculations.

For finite panels, Sewell [3] obtained a complete solution using the classical modal expansion method that appreciably improves the prediction accuracy relative to the mass law. However, due to the fact that the forced transmission is considered only, the formula always overestimates the transmission loss. This feature was fully discussed by other authors [4,5] in connection with various experimental results. In regard to the same problem, Leppington et al. [6] presented an improved estimate below coincidence and overcame the frequency limitations in predicting the forced transmission. They also quantified the resonant transmission, from which the total transmission coefficient agreed better with the measured results than the previous methods at frequencies just below the coincidence frequency and at lower frequencies.

The radiation impedance of partition is one of the important factors in sound transmission and its evaluation is essential in the prediction. Area independent radiation impedance of an infinite panel is one of the inherent causes of discrepancy with experiments, because real partitions are definitely finite anyway. When the finite partitions are concerned, there occurs a sudden rise in radiation efficiency curve due to the fact that the edge modes are dominant at frequencies just below the coincidence frequency and the relative contribution of resonant transmission also increases. Thus, the sound transmission loss predicted either by infinite theory or non-resonant transmission generally overestimates in this frequency region.

A majority of researchers have assumed a negligible contribution from the resonant transmission, based on the fact that the radiation efficiencies of resonant modes are relatively small in comparison with the internal loss factors [3,5,7]. This might be true for most of cases, although some researchers have shown that the role of resonant transmission should not be neglected in some cases [6,8,9]. However, this is mainly due to the fact that there were few quantitative investigations on the relative contribution of the resonant transmission component. Takahashi [9] calculated the relative contribution of resonant components in sound transmission through a single partition, which was defined as the ratio of the resonant sound transmission coefficient to the non-resonant one. It was noted that, at frequencies well below the coincidence frequency, even a single resonance may affect the total transmission, in which the extent of influence depends strongly on the panel damping. However, because the relative contribution also depends on other parameters such as the thickness and the size, a contribution analysis including the effect of those factors is required and the validity of the assumption of a negligible contribution from the resonant transmission should be investigated in connection with the structural parameters, from which the effect on sound transmission loss should be investigated quantitatively.

In this study, the sound transmission of a rectangular panel in an infinite rigid baffle is revisited by utilizing the general modal expansion method and the relative contribution of resonant transmission is investigated at frequency bands below the coincidence frequency. In the analysis, the transmission coefficients are averaged over a frequency band and its approximate expressions at the center frequency are explicitly presented for further numerical calculations. From the relative contribution of resonant transmission, the validity of neglect of resonant transmission components is investigated. Additionally, the validity of determining the sample size together with the thickness and damping factor is also discussed, referring to the general guidelines in the usual sound insulation tests.

2. Background theory

2.1. Sound transmission loss

Consider a simply supported rectangular panel ($0<x<l_x$, $0<y<l_y$) in an infinite rigid baffle as shown in Fig. 1. The rectangular co-ordinates are chosen setting z as normal to the panel: $\mathbf{r}=(x,y,z)$ represents the position vector of a field point and $\mathbf{x}=(x,y,0)$ is the position vector of a surface point on the panel. The normal velocity, $v(\mathbf{x})$, vibrating with a time harmonic frequency ω satisfies

$$(D\nabla^4-\rho_s h\omega^2)\cdot v(\mathbf{x})=\mathrm{j}\omega[p_1(\mathbf{r})-p_2(\mathbf{r})]_{z=0}, \tag{1}$$

where ρ_s and h are mass density and thickness of partition, D means the flexural rigidity given by $Eh^3/12(1-v^2)$, and E and v being Young's modulus and the Poisson ratio, respectively. The common time factor $\exp(\mathrm{j}\omega t)$ will be omitted hereafter for simplicity. In Eq. (1), p_1 and p_2 are sound pressure in incident and transmitted field, respectively, which satisfy the wave equation and can be expressed in the integral equation form as

$$p_1(\mathbf{r})=2p_i-2\int_S g(\mathbf{r}|\mathbf{r}_0)\cdot\left.\frac{\partial p_1(\mathbf{r}_0)}{\partial z}\right|_{z=0}\mathrm{d}\mathbf{r}_0, \tag{2a}$$

$$p_2(\mathbf{r})=2\int_S g(\mathbf{r}|\mathbf{r}_0)\cdot\left.\frac{\partial p_2(\mathbf{r}_0)}{\partial z}\right|_{z=0}\mathrm{d}\mathbf{r}_0, \tag{2b}$$

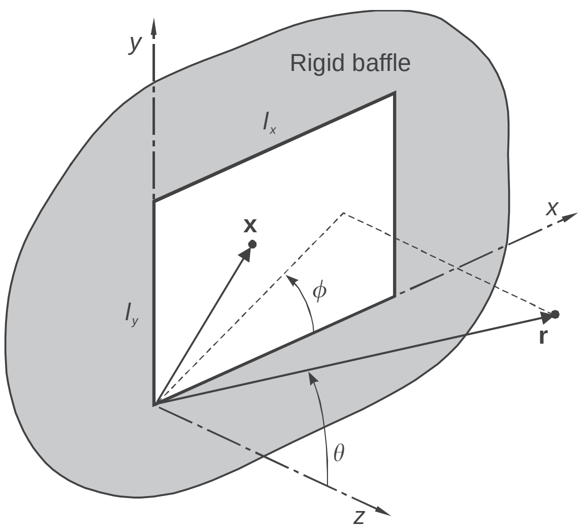

Fig. 1. Geometrical definition of co-ordinates and dimensions of the rectangular partition in an infinite rigid baffle.

where $p_i(=\exp[-\mathrm{j}(k_x x + k_y y + k_z z)])$ denotes the plane incident pressure, S is panel surface area, k the wave number, and $g(\mathbf{r}|\mathbf{r}_0)$ the Green function.

In order to solve Eq. (1) by using the general modal expansion method, the modal function of normal velocity, ψ_{mn} is adopted, which satisfy the following orthonormality:

$$\int_S \psi_{mn}(\mathbf{x})\psi_{qr}(\mathbf{x})\,\mathrm{d}\mathbf{x} = \delta_{mq}\delta_{nr}. \tag{3}$$

Here, δ_{mq} and δ_{nr} are the Kronecker delta and m, n, q, r are mode indices. All variables in Eq. (1) can be expanded by the panel modes, ψ_{mn}, as

$$[Dk_{mn}^4 - \rho_s h\omega^2]\cdot v_{mn} - 2\mathrm{j}\omega\sum_{q,r} R_{mnqr}\cdot v_{qr} = 2\mathrm{j}\omega p_{i,mn}, \tag{4}$$

where k_{mn} is the eigenvalue of Eq. (1). By expanding the radiated sound pressure in Eq. (2), one can obtain the radiation impedance, R_{mnqr}, which is given by

$$R_{mnqr} = 2\mathrm{j}\omega\rho_0\int_S\int_{S'} g(\mathbf{x}|\mathbf{x}')\cdot\psi_{qr}(\mathbf{x}')\psi_{mn}(\mathbf{x})\,\mathrm{d}\mathbf{x}'\,\mathrm{d}\mathbf{x}. \tag{5}$$

Eq. (5) can be also expressed as $R_{mnqr} = \rho_0 c_0(\theta_{mnqr} + \mathrm{j}\chi_{mnqr})$, in which ρ_0 and c_0 mean the density and the speed of sound in air, respectively, and θ_{mnqr} is often referred as the radiation resistance. In Eq. (4), the normal velocity, $v(\mathbf{x})$, and the incident pressure, p_i, are expanded by using the modal functions as

$$v(\mathbf{x}) = \sum_{m,n} v_{mn}\psi_{mn}(\mathbf{x}), \quad p_i = \sum_{m,n} p_{i,mn}\psi_{mn}(\mathbf{x}). \tag{6a, b}$$

Eq. (4) contains the inter-modal coupling terms (R_{mnqr} for $m\neq q$ or $n\neq r$). The leading effects of the inter-modal couplings are the shift of resonance frequencies and the increase in internal damping, while its influence on the vibrational amplitude is not severe. Because the band-averaged characteristics of vibration response and the resulting sound transmission are of interest in this work, the exact locations of resonance are not of concern. About the effects of inter-modal couplings, some previous research results reveal that they are not significant [9,10]. If the inter-modal coupling terms are neglected, the modal coefficient is reduced to

$$v_{mn} = \frac{1}{\rho_s h}\frac{2\mathrm{j}\omega p_{i,mn}}{[(\omega_{mn}^e)^2 - \omega^2] + \mathrm{j}\eta_{mn}^e\omega_{mn}^2}, \tag{7}$$

where ω_{mn}^e and η_{mn}^e indicate fluid loaded resonant frequency and effective loss factor, respectively, that are given by

$$(\omega_{mn}^e)^2 = \omega_{mn}^2 - 2\left(\frac{\rho_0 c_0}{\rho_s h}\right)\omega\chi_{mn}, \quad \eta_{mn}^e = \eta + 2\left(\frac{\rho_0 c_0}{\rho_s h}\right)\omega\frac{\theta_{mn}}{\omega_{mn}^2}. \tag{8a, b}$$

Here, ω_{mn} means the in-vacuo resonance frequency of the panel and θ_{mn} means the self-radiation resistance, which is often referred as the modal radiation efficiency. The radiated sound power can be obtained as follows:

$$\Pi_t = \tfrac{1}{2}\mathrm{Re}\left\{\int_S p_2\cdot v^*\,\mathrm{d}S\right\} = \tfrac{1}{2}\mathrm{Re}\left\{\sum_{m,n}\sum_{q,r} R_{mnqr}v_{qr}v_{mn}^*\right\}. \tag{9}$$

Regarding the effects of coupled terms in the case of sound radiation into a free space (R_{mnqr} for $m \neq q$ or $n \neq r$), Davies [10] found that the coupling effect in sound radiation could be neglected provided that the frequency spacing is larger than the effective width of resonances and the structural damping is small. Although the coupling effects could be strong for the non-resonant sound radiation of the first few modes, the coupling terms are not significant as concluded in the previous works [9,10]. Excluding the coupled terms, the radiated sound power can be reexpressed as

$$\Pi_t = \frac{1}{2}\sum_{m,n} \rho_0 c_0 \theta_{mn} \left|\frac{Y_{mn}}{\rho_s h}\right|^2 \cdot |p_{i,mn}|^2, \tag{10}$$

where $Y_{mn}/(\rho_s h)$ denotes the velocity admittance function and Y_{mn} is given by

$$Y_{mn} = 2\mathrm{j}\omega[\{(\omega^e_{mn})^2 - \omega^2\} + \mathrm{j}\eta^e_{mn}\omega^2_{mn}]^{-1}. \tag{11}$$

Because the average of $|p_{i,mn}|^2$ over all incidence angles is related to the modal radiation efficiency as

$$\frac{1}{\pi}\int_0^{2\pi}\int_0^{\pi/2} |p_{i,mn}|^2 \sin\theta \, \mathrm{d}\theta \, \mathrm{d}\phi = \frac{4\pi c_0^2}{\omega^2}\cdot \theta_{mn}, \tag{12}$$

one can derive, using the angular expression of incident power, Π_i, as $S\cos\theta/2\rho_0 c_0$, the random incidence sound transmission coefficient, τ, as follows:

$$\tau(\omega) = \frac{\pi c_0^2}{S}\left(\frac{2\rho_0 c_0}{\rho_s h\omega}\right)^2 \sum_{m,n} \theta^2_{mn}|Y_{mn}|^2. \tag{13}$$

By averaging Eq. (13) over a frequency band of $\omega_l < \omega < \omega_u$, the average sound transmission coefficient is obtained as

$$\bar{\tau} = \frac{1}{\Delta\omega}\frac{\pi c_0^2}{S}\int_{\omega_l}^{\omega_u}\left(\frac{2\rho_0 c_0}{\rho_s h\omega}\right)^2 \cdot \left(\sum_{m,n}\theta^2_{mn}|Y_{mn}|^2\right)\mathrm{d}\omega, \tag{14}$$

where $\Delta\omega$ denotes the frequency bandwidth. Here, the validity of the neglect of the inter-modal coupling effects can be easily done by detailed numerical calculations. The inter-modal coupling terms can be included in the mathematical model of Eqs. (5) and (9), which is similar to Davies' derivation procedure [10]. However, the differences in TL due to neglecting the inter-modal coupling terms are very small: for example, for a 3-mm-thick glass window of $0.56 \times 1.68\,\mathrm{m}^2$ in size, the difference in TL is less than 0.1 dB at the frequency bands of interest.

2.2. Classification of transmission components

It is well known that the sound transmission through a partition can be decomposed into two components: resonant and non-resonant transmission. Sound transmission by the modes, of which the resonance frequencies are within the frequency band of interest, is called the resonant transmission, whereas the non-resonant transmission is owing to the modes resonant outside the interested band [8,9]. The resonant sound transmission is caused by the interference between the multiply reflected waves from boundaries, travelling with free wave speed, and the bounded partition modes. The non-resonant sound transmission is associated with the response of the partition, in which the waves are forced to propagate with the trace wave speed of the incident wave. Consequently, the former is prone to be controlled by the damping, but the latter is not. Whether a mode is resonant or not should be determined in relation to the locations of resonance as well as the effective bandwidth of the mode. That is, if the bandwidth of a mode overlaps the frequency band under consideration, the mode should be regarded as being resonant. The two components can be classified by dividing the summation in Eq. (14) into two parts. Fig. 2 shows the velocity admittance functions and radiation efficiencies, which are principal factors in sound transmission, of several modes of a 2-mm-thick steel panel of $1.5 \times 1.5\,\mathrm{m}^2$ that are resonant in the 1/3-octave band centered at 500 Hz. Characteristics of non-resonant modes in the same band are depicted in Fig. 3. As shown in Fig. 2, resonant modes have low values of radiation efficiency. However, the values of velocity admittance are higher than those of non-resonant modes, although they depend on the structural damping.

2.2.1. Resonant transmission

Direct calculation of Eq. (14) enables the quantification of the relative contribution from the two transmission components. However, it is actually a laborious and time-consuming job, especially in the case of high modal density, and thus the approximate expressions would be more useful in estimating such contribution. For resonant components, the fact that the radiation efficiency is slowly varying with frequency can be exploited: Eq. (14) can be approximately rewritten as

$$\bar{\tau}_r \cong \frac{1}{\Delta\omega}\frac{\pi c_0^2}{S}\sum_{m,n}\left(\frac{2\rho_0 c_0}{\rho_s h \omega_c}\right)^2 \theta_{mn,c}^2 \int_{\omega_l}^{\omega_u} |Y_{mn}|^2\,\mathrm{d}\omega, \tag{15}$$

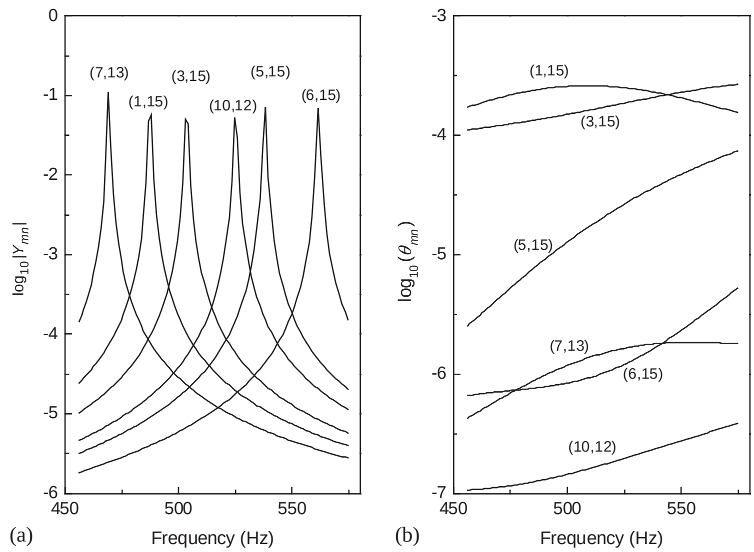

Fig. 2. Velocity admittance functions and radiation efficiencies for several resonant modes at 500 Hz band. Bracketed number indicates the mode index. (a) Admittance function, Y_{mn} and (b) modal radiation efficiency, θ_{mn}.

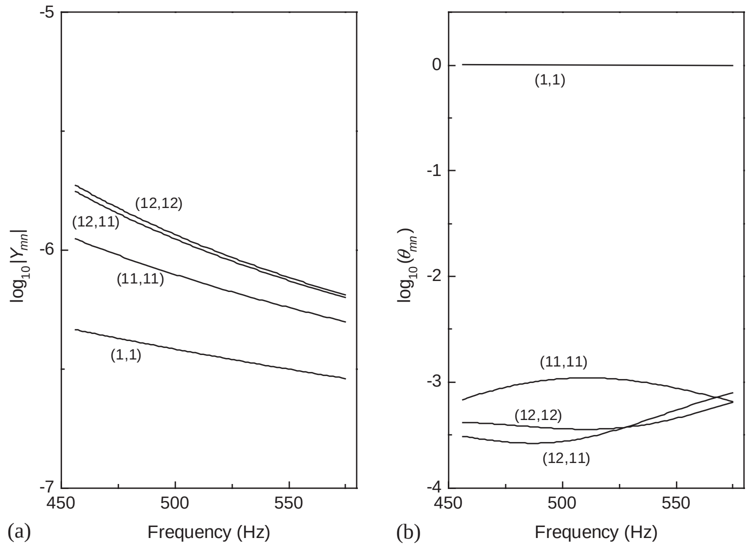

Fig. 3. Velocity admittance functions and radiation efficiencies for several non-resonant modes at 500 Hz band. Bracketed number indicates the mode index. (a) Admittance function, Y_{mn} and (b) modal radiation efficiency, θ_{mn}.

where subscript 'c' indicates the value at the center frequency of a band and the summation is done only for the resonant modes. Actually, there occurs error due to the assumption of the slowly varying radiation efficiency, however, it can be easily shown that the errors decreases as the mode count increases and an acceptable range can be defined by restricting the frequency region. That is to say, the validity of Eq. (15) is related with the number of modes in a band considered. The reasonable approximation of the integration is $2\pi/\eta^e_{mn}\omega_{mn}$ provided that there are at least several resonant modes within the frequency band of interest [11]. The resonant sound transmission coefficient can be further approximated as follows:

$$\bar{\tau}_r \cong \frac{1}{\Delta\omega}\frac{\pi c_0^2}{S}\left(\frac{2\rho_0 c_0}{\rho_s h\omega_c}\right)^2 \sum_{m,n}\frac{2\pi\theta^2_{mn,c}}{\eta^e_{mn}\omega_{mn}} \cong \frac{1}{\Delta\omega}\frac{\pi c_0^2}{S}\left(\frac{2\rho_0 c_0}{\rho_s h\omega_c}\right)^2 \frac{2\pi}{\eta^e_c\omega_c}\sum_{m,n}\theta^2_{mn,c}. \tag{16}$$

When at least several modes exist in the band, the summation can be replaced by integration and can be approximated as

$$\sum_{m,n}\theta^2_{mn,c} \cong \Delta N\overline{\theta^2_c}, \tag{17}$$

where ΔN means the mode count of that band and $\overline{\theta^2_c}$, the average of the square of modal radiation efficiency, approximated by the integration, is given by [6]

$$\overline{\theta^2_c} = \frac{l_x^2 + l_y^2}{\pi(kS)^2}\frac{\alpha^2}{(1-\alpha)^3}\left[(5-4\alpha)\sin^{-1}(\alpha^{1/2}) + (3-2\alpha)\sqrt{\alpha(1-\alpha)}\right], \tag{18}$$

where α is ω_c/ω_{co} with ω_{co} being the coincidence frequency. Then, Eq. (16) becomes

$$\bar{\tau}_r \cong \frac{2\pi^2 c_0^2}{S}\left(\frac{2\rho_0 c_0}{\rho_s h\omega_c}\right)^2 \frac{n(\omega)\overline{\theta^2_c}}{\eta^e_c\omega_c}, \tag{19}$$

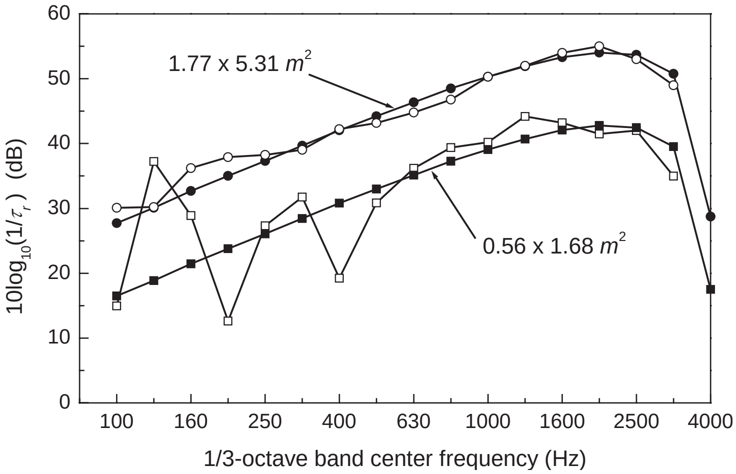

Fig. 4. A comparison between calculated values by Eqs. (14) and (20) for a 3-mm-thick single glass pane of two different sizes: solid symbol, Eq. (20); hollow symbol, Eq. (14).

where $n(\omega)$ means the modal density of partition, which is given by, for the case of a rectangular panel, $\sqrt{3}S/2\pi hc_L$ with c_L being the velocity of longitudinal wave. Finally, the resonant transmission coefficient can be approximately expressed as

$$\bar{\tau}_r \cong \frac{\pi}{2}\left(\frac{2\rho_0 c_0}{\rho_s h\omega_c}\right)^2\left(\frac{\omega_{co}}{\omega_c}\right)\left(\frac{\overline{\theta_c^2}}{\eta_c^e}\right), \tag{20}$$

which has similar form with previous results obtained from energy considerations [12,13]. Regarding the rationality of Eq. (20), Fig. 4 shows a typical example of differences between calculated values by Eqs. (14) and (20) for a 3-mm-thick single glass pane of two different sizes. It can be seen that the errors decreases as the area increases, that is, as the mode count increases. Therefore, provided that there are at least several resonant modes within the frequency band of interest, Eq. (20) can be regarded as a reasonable approximation for the resonant components in Eq. (14).

2.2.2. Non-resonant transmission

Regarding the non-resonant sound transmission, the well-known Sewell's work [3] is adopted in this work, because the derivation procedures are basically identical with performing the integration in Eq. (14) for non-resonant modes. Here, the Sewell's formula [3] is used with slight modification of excluding some terms that contributes very small amount as follows:

$$\bar{\tau}_n \cong \left(\frac{2\rho_0 c_0}{\rho_s h\omega_c}\right)^2\left(1-\frac{\omega_c^2}{\omega_{co}^2}\right)^{-2}\ln[k_c\sqrt{S}]. \tag{21}$$

Although a modification to Sewell's formula is available [6], which improves the estimates of TL at frequencies above $0.5\omega_{co}$, the results at frequencies below $0.5\omega_{co}$ are almost same. Reflecting the fact that the effects of panel size and the role of resonant transmission in the low frequency range are more prominent in general, Sewell's formula is sufficient for the present study. Here, it should be noticed that Eq. (21) is generally applicable to frequencies below $0.5\omega_{co}$.

2.3. Relative contributions

From Eqs. (20) and (21), the ratio between resonant and non-resonant transmission is given by

$$\frac{\bar{\tau}_r}{\bar{\tau}_n} = \frac{\pi}{2}\left(\frac{\omega_{co}}{\omega_c}\right)\left(1 - \frac{\omega_c^2}{\omega_{co}^2}\right)^{-2} \frac{1}{\ln[k_c\sqrt{S}]}\left(\frac{\bar{\theta}^2}{\eta_c^e}\right). \quad (22)$$

Fig. 5 shows a comparison of the calculated contributions of resonant and non-resonant transmission by using Eq. (14) and the approximated expression given by Eq. (22), for various steel panels of different size having the same thickness. Here, one can observe that the lowest frequency bands having the mode count more than 5 are 200, 100, and 50 Hz band for the panel of 0.8×0.8, 1.2×1.2, and $1.6 \times 1.6\,\text{m}^2$, respectively. It should be reminded that the condition of mode count more than 5 is the necessary condition for validating the approximations in Eq. (16). Because, at very low frequency bands, the existence of even a single resonant mode can affect the total transmission significantly [9], Eq. (22) just yields an averaged trend only, especially for the panel of $0.8 \times 0.8\,\text{m}^2$. Here, it should be noticed that the partition has been assumed as simply supported at edges. Since most of partitions are mounted firmly, the clamped end is probably more realistic, while the clamped edge makes the mathematical analysis to be complicated. The edge condition has been known to change the radiation characteristics, which can be represented by the radiation efficiency, as well as the vibration response. From detailed analysis, some researchers have shown that the radiation efficiency of a clamped partition is twice that of a simply supported partition at frequencies below the coincidence [14,15]. From the facts that the non-resonant transmission is less sensitive to edge conditions than resonant component and that resonant transmission is proportional to the square of the radiation efficiency as can be seen in Eq. (20), the relative contribution of resonant transmission should be lager than depicted values in Fig. 5. However, the actual edge conditions are in between clamped and simply supported

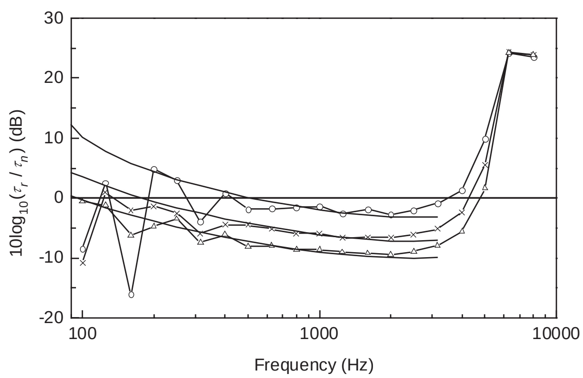

Fig. 5. Variation of the relative contribution factor, $10\log_{10}(\tau_r/\tau_n)$, calculated by Eq. (14) with the change of the size of finite steel partition ($h = 2\,\text{mm}$, $\eta = 0.002$): —○—, $0.8 \times 0.8\,\text{m}^2$; —×—, $1.2 \times 1.2\,\text{m}^2$; —△—, $1.6 \times 1.6\,\text{m}^2$; ——, approximated values by Eq. (22) for each size.

boundary conditions, and the increase of radiation efficiency is hard to be quantified accurately. Due to these features, the edge is assumed to be simply supported in this study, which can offer a basic guide.

3. Results

In order to investigate the effects of physical parameters such as size, thickness, and damping, the following non-dimensional factor is employed in this study:

$$\phi = \frac{S\eta^e}{h^2}. \tag{23}$$

This parameter was presented originally by Pope [8], which was called the ''participation factor''. The physical basis for choosing such a parameter was on the concept of both wave numbers and frequency matching: however, further detailed physical interpretation can be given from the consideration of sound transmission mechanism. Eq. (23) can be expressed as

$$\phi = \frac{\eta^e}{(h/l_x)(h/l_y)}. \tag{24}$$

Here, reflecting the fact that the added mass by the fluid loading is proportional to $\rho_0 l$ [16], the ratio, h/l in the denominator can be said to represent a ratio of the mass inertia of partition to the fluid loading. Thus, the participation factor in Eq. (23) is an important, representative parameter that relates the three basic energy transmission mechanisms: mass inertia, fluid loading, and energy loss. Fig. 6 depicts the variation of the ratio of resonant and non-resonant transmission coefficient by varying the participation factor in Eq. (23), i.e., changing the involved parameters, for a glass window ($\rho_s = 2400\,\mathrm{kg/m^3}$, $E = 7.0 \times 10^{10}\,\mathrm{Pa}$) and a steel panel ($\rho_s = 7700\,\mathrm{kg/m^3}$, $E = 19.5 \times 10^{10}\,\mathrm{Pa}$). In these figures, results for a glass and a steel panel are slightly different, because the relative importance of each sound transmission mechanism is also influenced from the other physical parameters of partitions; nevertheless, the general trend is almost same. One can find that the relative contribution of resonant transmission to the overall value is inversely proportional to the logarithm of the participation factor. It should be mentioned that there exists an uncertainty in determining the loss factor and, thus, one can only assume the values appropriately at prediction stage, if measured values were not available. It is true that the second term in Eq. 8(b) is sufficiently small and η^e can be approximated by the internal loss factor, but it would be more reasonable and practical to assume the total loss factor including edge losses for η^e.

In Fig. 7, the difference between the total transmission loss ($TL_t = -10\log_{10}(\tau_n + \tau_r)$) and the non-resonant transmission loss ($TL_n = -10\log_{10}(\tau_n)$) is illustrated as a function of the participation factor. Although the participation factor in Eq. (23) does not contain any frequency parameter, the difference in TL values changes along the frequency band when ϕ is applied to the formulas in Eqs. (14) and (21). Curves in Fig. 7 are displayed with a range: the minimum and maximum differences depicted by dashed lines correspond to the values for the lowest and highest frequency bands of interest, respectively, and the solid line depicts the frequency-averaged values. It should be noticed here that the results are applicable only to frequency bands below $0.5\omega_{co}$ and to frequency bands in which five or more resonant modes exist, which is due to the assumptions

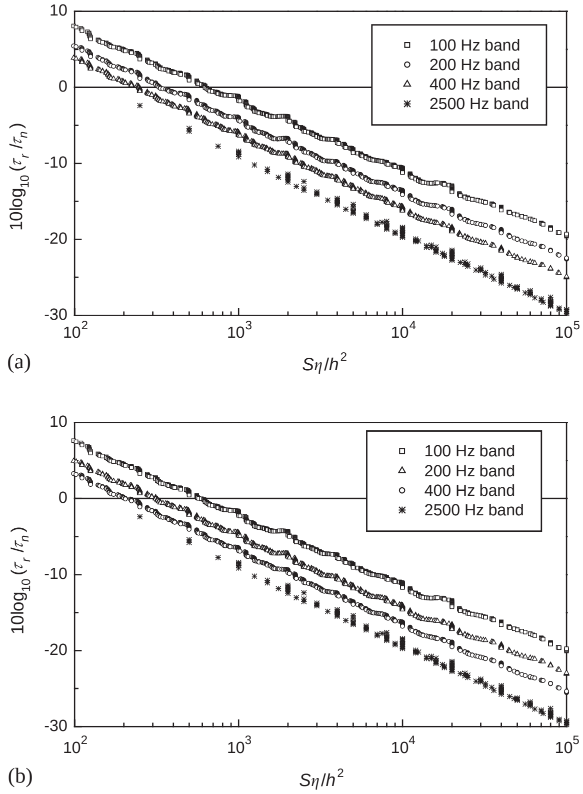

Fig. 6. Variation of the relative contribution factor, $10\log_{10}(\tau_r/\tau_n)$, with the change of participation factor for several frequency bands: (a) glass window and (b) steel panel.

made in deriving Eq. (22). From this figure, an appropriate range of parameters can be selected, which guarantees the validity in neglecting the resonant sound transmission in evaluating the total transmission loss. For example, when the averaged difference in TL, i.e., the prediction error, is to be smaller than 1 dB, the participation factor should be larger than about 3×10^3 on the basis of minimum value of TL difference in Fig. 7: This implies that the damping factor should be larger than 1.4×10^{-2} for a 3-mm-thick glass of $1.5 \times 1.25\,\mathrm{m}^2$ in size. For another example, a 9-mm-thick glass of the same size but with $\eta^e = 1 \times 10^{-2}$, the transmission loss predicted only from the non-resonant transmission would be lager than the total transmission loss by 4 dB.

In the guideline of the ISO standard for testing the sound insulation through a partition [17], a specimen area of $10\,\mathrm{m}^2$ or larger one is recommended. If a specimen of 1-mm-thick steel panel is considered, the participation factor is 2×10^4 for $\eta = 2 \times 10^{-3}$, and then the neglect of resonant sound transmission is definitely valid. However, if the size is $1.5 \times 1.25\,\mathrm{m}^2$, which is the recommended dimension for glass windows, the resonant transmission should be included depending on the thickness and damping factor. Here, it should be recalled again that the damping factor in the actual test situation is generally higher than the internal loss factor. Therefore, it should be said that the current guideline of correlating the measured and predicted sound transmission loss is not absolutely acceptable and there is a good possibility of discrepancy

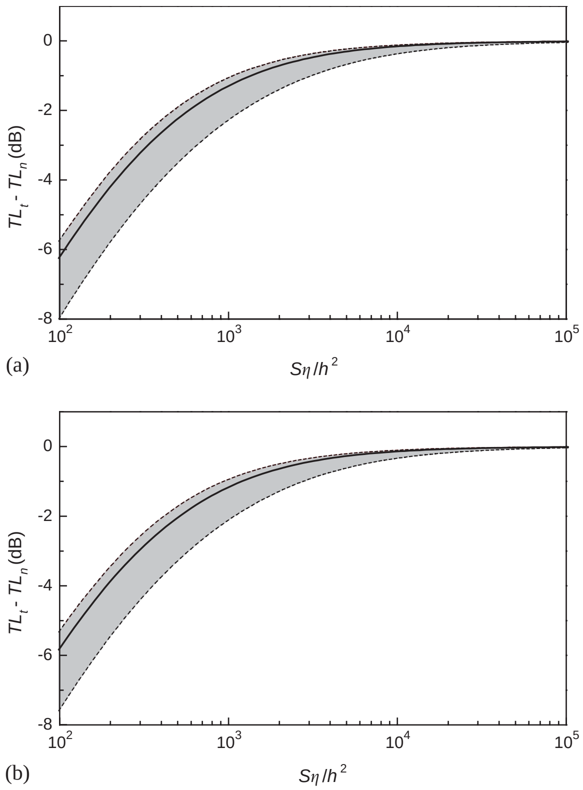

Fig. 7. Difference between total TL and non-resonant TL: ——, average value; - - - - -, minimum and maximum values. (a) Glass window and (b) steel panel.

between predicted and measured TL value without carefully checking the physical condition of test specimen.

4. Conclusions

In this study, the resonant sound transmission coefficient of a finite rectangular panel in an infinite rigid baffle is revisited and the relative importance of resonant transmission in the total transmission is investigated at frequency bands below the coincidence frequency. By investigating the average difference between the total transmission loss and the non-resonant transmission loss, the validity of neglecting the resonant transmission components in the prediction of TL is discussed: the analysis is based on the non-dimensional participation factor, which is composed of size, thickness, and damping factor of the specimen. As a result, in order to guarantee that the resonant transmission is negligible, the participation factor should be larger than 3×10^3 that yields the discrepancy between measured and predicted TL value less than 1 dB.

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