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Effects of sound radiation on Sabine absorption coefficient of modally-reactive panels

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Abstract: A modally-reactive panel in a room absorbs and radiates sound at the same time when it is excited acoustically by the enclosed sound field. The absorption of sound occurs when only part of the incident sound is reflected by the panel, but when the entire panel vibrates, the radiation of sound is also produced. In this paper, Statistical Energy Analysis (SEA) is used to establish a relationship between the acoustic-structural coupling of the panel with the sound field and the Sabine absorption coefficient of the panel. It is shown that the coefficient does not only consist of the sound absorption by the panel from the room but also, the sound radiation from the vibrating panel back into the room. Computational and experimental examples are presented for different acoustical properties of the panel and the sound field to illustrate the extent of influence of the sound radiation on the coefficient. The results provide a basic understanding of the conditions in both cases where the sound radiation has significant effects and negligible effects on the determination of the Sabine absorption coefficient of modally-reactive panels in rooms.

Keywords: Sabine absorption coefficient, Sound radiation, Acoustic-structural coupling, Modally-reactive panels, Statistical Energy Analysis

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1. INTRODUCTION

Sound absorption coefficient describes the ability of an absorptive surface to dissipate sound and it is defined as the ratio of absorbed to incident sound energy of the surface. The absorption coefficient of a locally-reactive surface in a room was described as a function of the incident angle of sound waves impinging on the surface and the normal acoustical impedance of the surface [1,2]. The behavior of the coefficient was then studied for different incident angles that correspond to individual acoustic modes of the enclosed sound field. When the sound field is diffuse, the absorption coefficient of the surface is known as statistical absorption coefficient (or random-incidence absorption coefficient), and it has also been derived analytically in terms of the acoustical impedance and investigated for different magnitudes and phases of the impedance [2,3]. In addition, a few numerical techniques were also developed for the estimation of the statistical absorption coefficient (e.g. [4]). However, when a surface is acoustically excited into vibration by the sound field in a room, it not only absorbs sound from the room but also, radiates sound back into the room. Since the radiation of sound from locally-reactive surfaces is insignificant, the sound radiation component of the surfaces has not been taken into account in these previous works. Hence, only the sound reflection and incidence components were considered in the analysis of the absorption coefficient.

In the area of architectural acoustics, modally-reactive surfaces are widely found on various structures in theatres, concert and lecture halls, and opera houses [5]. Examples of the structures are wooden stage floors, stage enclosures, flexible side walls, timber floors over some airspace, panel absorbers or reflectors, and diffusers. Unlike locally-reactive surfaces, modally-reactive surfaces can radiate sound efficiently depending on the structural modes that dominate the vibrational response of the surfaces [6,7]. There was also some evidence which showed that the sound radiation from modally-reactive surfaces significantly affects the reflected sound pressure in the far field of the surfaces [8,9]. However, it is unclear how the sound radiation affects the absorption coefficient of such surfaces and whether this influence is significant or not.

The sound absorption and radiation of a modally-reactive structure in a room are controlled by the acoustic-structural coupling between the enclosed sound field and the structure, and they are mutually dependent [10,11]. In other words, the sound reflection, incidence and radiation components of the structure are dependent on each other and cannot be separately described. So, in numerous experiments where the absorption coefficient of modally-reactive structures was measured [11–13], the sound radiation from the structures was inherited in the measured results. Also, it is known that the absorption coefficient is a parameter which depends on the conditions of the sound field in the room where the measurement is conducted [11,14,15]. This means that the sound absorption and radiation and thus, the absorption coefficient of the same absorptive surface, are functions of the enclosed sound field and they are different in different rooms. Therefore, in order to accurately predict the absorption coefficient of modally-reactive structures in the design of room acoustics, the development of a reliable technique that incorporates both absorption and radiation of the structures is essential. Subsequently, it is necessary to first understand the extent of influence of the sound radiation on the absorption coefficient of the structures under different conditions of the sound field and the structures.

In this paper, the Sabine absorption coefficient is used to describe the acoustical dissipativity of modally-reactive panels in rooms. In order to generalize the use of this coefficient to any enclosed sound field, it is not defined as the ratio of absorbed to incident sound energy but rather, only as a dissipation factor of inverse proportionality to the reverberation time of the sound field. Statistical Energy Analysis (SEA) is employed to establish a relationship between the Sabine absorption coefficient of a modally-reactive panel and the acoustic-structural coupling of the panel with the sound field in a room. The coupling is expressed explicitly in terms of the sound absorption and sound radiation of the panel. The effect of neglecting the radiation term on the determination of the Sabine absorption coefficient is investigated for different internal dampings of the panel and the sound field.

2. QUASI-TRANSIENT SOLUTION TO THE SOUND-FIELD RESPONSE BY SEA

Consider the coupling between a modally-reactive panel at the boundary of a room and a general sound field in the room. By using SEA, the quasi-transient energy-balance equations for the acoustic-structural coupled system can be obtained when the panel and the sound field are simultaneously driven by time-dependent input excitations, $\tilde{\Pi}_p(t)$ and $\tilde{\Pi}_a(t)$, respectively:

$$\frac{\partial \tilde{E}_p(t)}{\partial t} = \tilde{\Pi}_p(t) + \eta_{ap}\omega_0\tilde{E}_a(t) - (\eta_p + \eta_{pa})\omega_0\tilde{E}_p(t), \quad (1)$$

$$\frac{\partial \tilde{E}_a(t)}{\partial t} = \tilde{\Pi}_a(t) + \eta_{pa}\omega_0\tilde{E}_p(t) - (\eta_a + \eta_{ap})\omega_0\tilde{E}_a(t). \quad (2)$$

Figure 1 shows the two-subsystem model for the quasi-transient power flow between the panel and the sound field.

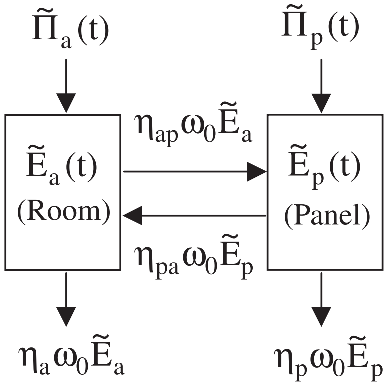

Fig. 1 Two-subsystem model for the quasi-transient power flow between the panel and the sound field.

$\tilde{E}_p$ and $\tilde{E}_a$ are time-dependent energies of the panel and the sound field, ω_0 is the centre frequency of the excitation band and t is time. η_p and η_a denote the damping loss factors of the uncoupled panel and the uncoupled sound field, η_{pa} denotes the coupling loss factor from the panel to the sound field and η_{ap} denotes the coupling loss factor from the sound field to the panel. η_p represents the combined loss due to internal structural damping and other dissipations at the joints of the panel, as well as sound radiation from the panel surface opposite to the sound field. η_a represents the total dissipation of sound by air, other parts of the boundary of the room and any objects inside the room. η_{pa} is directly proportional to the radiation efficiency of the panel [11,16], so it accounts for the sound radiation and describes the radiativity of the panel. η_{ap} accounts for the sound absorption and describes the absorptivity of the panel.

If $\tilde{\Pi}_p(t) = \Pi_p$ and $\tilde{\Pi}_a(t) = \Pi_a$ are steady-state input excitations that operate for sufficiently long during $t < 0$, then the coupled system would reach a steady state before $t = 0$. When the system is in the steady state at $t = 0$, the excitations are turned off [i.e., $\tilde{\Pi}_p(t) = \tilde{\Pi}_a(t) = 0$ for $t > 0$]. By using this initial condition and applying Laplace transformation to Eqs. (1) and (2), the decay of the sound field is obtained as [11]

$$\tilde{E}_a(t) = E_a[(d_1 - s_1)e^{-s_1 t} - (d_1 - s_2)e^{-s_2 t}]/(s_2 - s_1), \quad (3)$$

$$s_1 = 0.5\omega_0\left[\eta_a + \eta_{ap} + \eta_p + \eta_{pa} - \sqrt{(\eta_a + \eta_{ap} - \eta_p - \eta_{pa})^2 + 4\eta_{ap}\eta_{pa}}\right], \quad (4)$$

$$s_2 = 0.5\omega_0\left[\eta_a + \eta_{ap} + \eta_p + \eta_{pa} + \sqrt{(\eta_a + \eta_{ap} - \eta_p - \eta_{pa})^2 + 4\eta_{ap}\eta_{pa}}\right], \quad (5)$$

$$d_1 = \omega_0(\eta_p + \eta_{pa} + \eta_{pa}E_p/E_a). \quad (6)$$

In the above, E_p and E_a are initial energies of the panel and the sound field (i.e., at $t = 0$).

3. SABINE ABSORPTION COEFFICIENT OF THE PANEL

The mean Sabine absorption coefficient which is associated with all acoustical dissipations in a general sound field in a room can be expressed in terms of loss factor as [17]

$$\bar{\alpha}_{\mathrm{Sab}} = \left(A_p \alpha_{\mathrm{Sab}} + \sum_{i=1}^{M-1} A_i \alpha_{\mathrm{Sab},i} \right) / A_T + \alpha_{\mathrm{air}} = 4V_0 \eta_T \omega_0 / A_T c_0, \tag{7}$$

where there are M absorptive surfaces in the room including the panel. η_T is the total loss factor that corresponds to all the dissipations, V_0 is the volume of the room, A_T is the total area of the surfaces, c_0 is the speed of sound in air, A_p and α_{Sab} are the surface area and Sabine absorption coefficient of the panel, A_i and $\alpha_{\mathrm{Sab},i}$ are the area and Sabine absorption coefficient of the ith surface, and α_{air} corresponds to the air absorption.

From the above, if the panel is not present, then the two decay rates of the sound field, s_1 and s_2, only depend on η_a and are not affected by η_p, η_{pa} and η_{ap}. By setting $\alpha_{\mathrm{Sab}} = 0$ and $\eta_T = \eta_a$ in Eq. (7), it can be shown that

$$\sum_{i=1}^{M-1} A_i \alpha_{\mathrm{Sab},i} + A_T \alpha_{\mathrm{air}} = 4V_0 \eta_a \omega_0 / c_0.$$

Thus, when the sound field is coupled to the panel, Eq. (7) can be rewritten as

$$\alpha_{\mathrm{Sab}} = 4V_0 \omega_0 (\eta_T - \eta_a) / A_p c_0. \tag{8}$$

As can be seen in Eq. (3), the overall decay rate of the coupled sound field and thus, α_{Sab}, cannot be obtained analytically because the overall decay is non-exponential. In order to take into account the full quasi-transient solution to the sound-field response, approximate Sabine absorption coefficients of the panel (α_1 and α_2) which correspond, respectively, to $e^{-s_1 t}$ and $e^{-s_2 t}$ in Eq. (3) are determined first. Each approximate coefficient is then weighted with respect to the amplitude and decay of the associated exponential term given in Eq. (3) (i.e., α_1 with respect to $|d_1 - s_1|$ and e^{-s_1}, and α_2 with respect to $|d_1 - s_2|$ and e^{-s_2}). Subsequently, the value of α_{Sab} is defined to be the arithmetic average of both weightings. So, if the decay rate of the coupled sound field is obtained from s_1 only (i.e., $s_1 = \eta_T \omega_0$), $\alpha_1 = 4V_0(s_1 - \eta_a \omega_0)/A_p c_0$ from Eq. (8). If the decay rate is obtained from s_2 only (i.e., $s_2 = \eta_T \omega_0$), $\alpha_2 = 4V_0(s_2 - \eta_a \omega_0)/A_p c_0$ from Eq. (8). Then, α_{Sab} is determined from the weighted average of α_1 and α_2:

$$\alpha_{\mathrm{Sab}} = \frac{|d_1 - s_1| e^{-s_1} \alpha_1 + |d_1 - s_2| e^{-s_2} \alpha_2}{|d_1 - s_1| e^{-s_1} + |d_1 - s_2| e^{-s_2}}. \tag{9}$$

α_{Sab} can be evaluated by Eq. (9) when η_p, η_a, η_{pa} and η_{ap} are known.

The two coupling loss factors, η_{pa} and η_{ap}, can be determined by solving the energy-balance equations of the coupled system at a steady state where $\partial \tilde{E}_p / \partial t = \partial \tilde{E}_a / \partial t = 0$. When the input excitation is given only to the sound field, $\tilde{\Pi}_p(t) = 0$ and thus, Eq. (1) yields the steady-state energy-balance equation of the panel:

$$0 = \eta_{ap} \omega_0 E_a^a - (\eta_p + \eta_{pa}) \omega_0 E_p^a. \tag{10}$$

If the situation is reversed with the input excitation given only to the panel, then $\tilde{\Pi}_a(t) = 0$. At the steady-state, Eq. (2) yields the corresponding energy-balance equation of the sound field:

$$0 = \eta_{pa} \omega_0 E_p^p - (\eta_a + \eta_{ap}) \omega_0 E_a^p. \tag{11}$$

In Eqs. (10) and (11), E_p^a and E_a^a are the steady-state energies of the coupled panel and the coupled sound field when only the sound field is excited. E_p^p and E_a^p are the steady-state energies when only the panel is excited. Figure 2

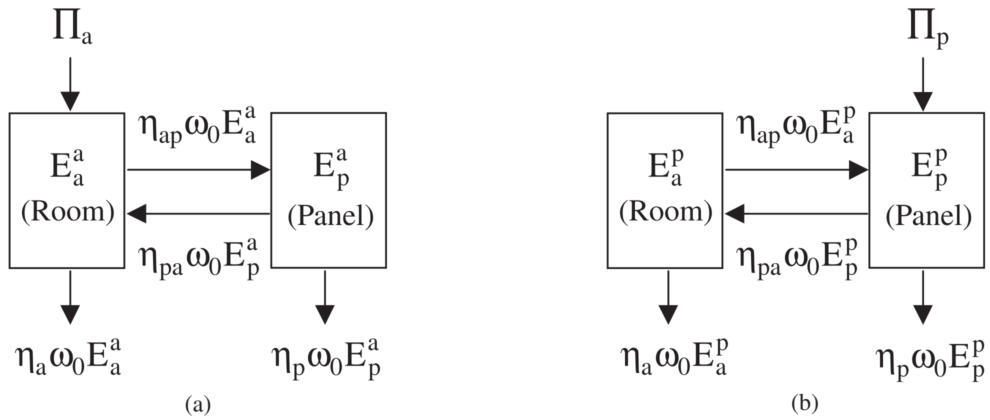

Fig. 2 Two-subsystem models for the steady-state power flow between the panel and the sound field. Input excitation is given only to (a) the sound field, and (b) the panel.

shows the two-subsystem models for the steady-state power flow between the panel and the sound field. Given the four steady-state energies, η_p and η_a, Eqs. (10) and (11) can be solved for η_{pa} and η_{ap}.

4. RESULTS AND DISCUSSION

In the application of SEA, the presumption that each subsystem in a coupled system must have a large number of resonant modes, has always been used without any questions of its necessity. Some new studies have been conducted recently to clarify the requirement in the number of these modes. In a system which has more than two subsystems, each subsystem does not need to have a large number of resonant modes if all indirect coupling paths are included in the analysis [18–20]. The analysis is called quasi-SEA when these paths are taken into account [21]. A large number of the modes are required only when the paths are not considered, and the large number ensure that the direct paths dominate the power flow over the indirect paths so that the errors due to the negligence of the latter can be ignored [19,20]. For instance, indirect paths do not exist in a system with only two subsystems. In this case, only the presence of a sufficient number of modal pairs is necessary where one of the subsystems needs to have only one resonant mode and the other subsystem has a few resonant modes [18,20]. Also, the concept of modal density is a major source of uncertainty because it constrains SEA to resonant modes only and the equipartition of energy among these modes. Thus, if this concept is not used, no modal-density terms are involved, and the coupling loss factors and energies of the subsystems consist of the combined influence by both resonant and non-resonant modes where the requirement in the number of resonant modes is less strict [22]. In the two-subsystem case for example, each subsystem requires at least two resonant modes [22]. The non-resonant response of a given subsystem in a coupled system can also be separately modelled within the framework of SEA as an extra subsystem with no resonant modes where the concept of modal density is not applicable [23]. Since this concept is not used in the determination of η_{pa} and η_{ap} as can be seen in Eqs. (10) and (11), the lowest frequency band of analysis in the following discussion is chosen such that the panel and the sound field, respectively, have at least two resonant modes.

Computational and experimental examples are employed to illustrate the effect of neglecting η_{pa} on the determination of α_{Sab} of a modally-reactive panel in a room. In the following computational examples, the physical model of the acoustic-structural coupled system consists of a rectangular-parallelepiped room and a simply-supported rectangular glass panel. The dimensions of the room are $(L_x, L_y, L_z) = (0.880, 1.725, 1.540)$ m and the panel forms a boundary of the room at $z = 1.540$ m. The properties of the panel are: $A_p = 0.880 \times 1.725\,\mathrm{m}^2$, thickness $= 6$ mm, material density $= 2{,}500\,\mathrm{kgm}^{-3}$ and longitudinal wave speed $= 5{,}200\,\mathrm{ms}^{-1}$. E_p^a, E_a^a, E_p^p and E_a^p are calculated by the well-established modal-coupling method [10,24]. A monopole is used as an input source to drive the sound field and it is located consecutively at twelve random positions in the room. E_p^a and E_a^a are obtained by averaging the energies of the panel and the sound field over all driving locations. A mechanical point force is used as an input source to drive the panel and it is located consecutively at twelve random positions on the panel surface. E_p^p and E_a^p are calculated as the average energies for all driving locations. Various combinations of values of η_p and η_a have been tested numerically and two of these are presented here. In the first, η_p is two times larger than η_a (i.e., $\eta_p = 0.001$, $\eta_a = 0.0005$) and in the second, η_p is five times larger than η_a (i.e., $\eta_p = 0.05$, $\eta_a = 0.01$). The values of the ratio of η_p to η_a in both combinations are typical for practical modally-reactive structures in normal rooms. They are selected to illustrate, respectively, the case where η_{pa} is significant and the case where η_{pa} is insignificant to the determination of α_{Sab}. The lower values of η_p and η_a represent a lightly damped panel and sound field in the uncoupled state, and the higher values represent a well-damped panel and sound field.

Figure 3 presents the values of η_{pa} and η_{ap} in 1/3-octave frequency bands for the two combinations of η_p and η_a. The corresponding values of Sabine absorption coefficient with and without including the sound radiation from the panel [i.e., omitting the terms with η_{pa} in Eqs. (1)–(6)]

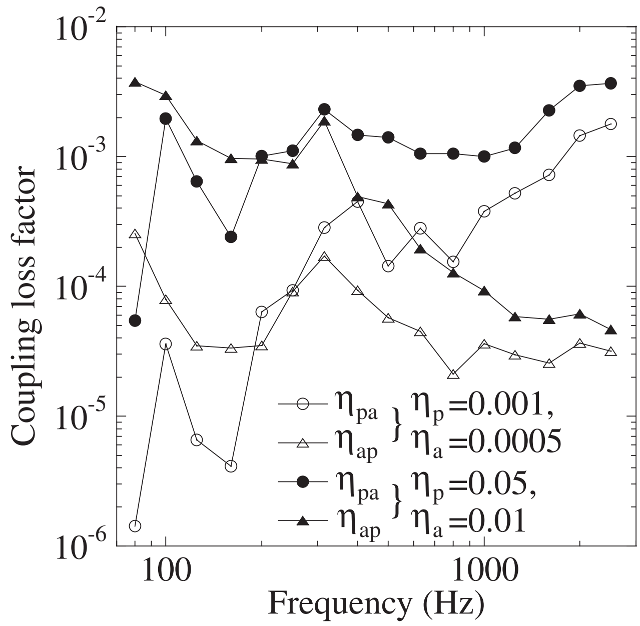

Fig. 3 Coupling loss factors for the coupling between the glass panel and the sound field in the room.

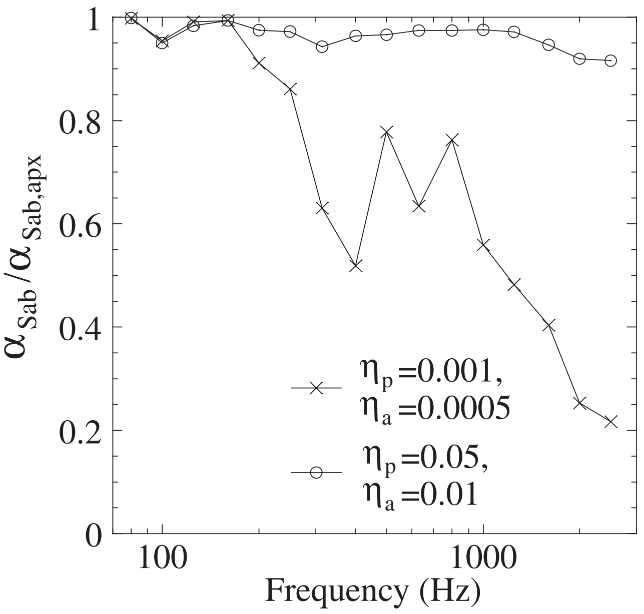

Fig. 4 Ratio of Sabine absorption coefficients of the glass panel in the room with and without including the sound radiation from the panel.

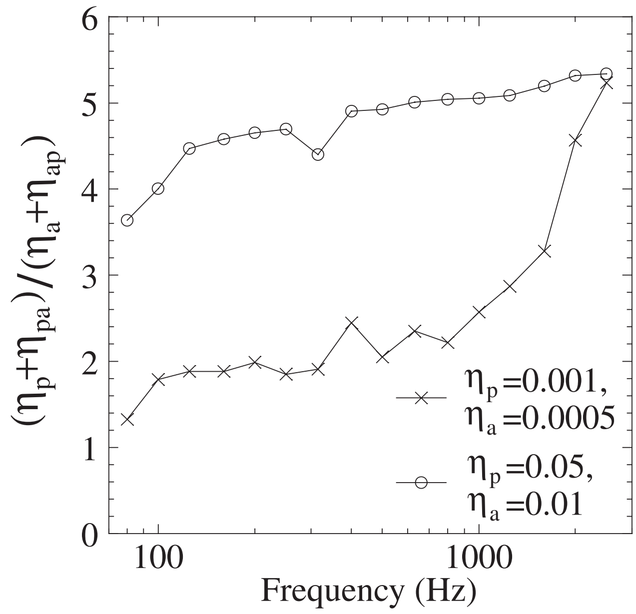

Fig. 5 Ratio of the total acoustical dissipativity of the panel to the total acoustical dissipativity of the sound field in the room.

are calculated. The ratio of $\alpha_{\rm Sab}$ to $\alpha_{\rm Sab,apx}$ is shown in Fig. 4 where $\alpha_{\rm Sab,apx}$ denotes the approximated Sabine absorption coefficient when the radiation is excluded. The figure indicates that $\alpha_{\rm Sab} < \alpha_{\rm Sab,apx}$ in all bands, which implies that the sound radiation has an effect of reducing values of the Sabine absorption coefficient. As depicted, there are two different cases where the influence of the sound radiation on the coefficient is either significant or insignificant. When $\eta_{\rm p}$ is close to $\eta_{\rm a}$ (i.e., $\eta_{\rm p} = 0.001$, $\eta_{\rm a} = 0.0005$), $\alpha_{\rm Sab}$ is only about 65% of $\alpha_{\rm Sab,apx}$ or less in many bands. In this case, the negligence of the radiation incurs large errors in the estimation of the coefficient. When $\eta_{\rm p}$ is much larger than $\eta_{\rm a}$ (i.e., $\eta_{\rm p} = 0.05$, $\eta_{\rm a} = 0.01$), $\alpha_{\rm Sab}$ is within 95% of $\alpha_{\rm Sab,apx}$ in most of the bands, and the sound radiation only plays a minor role in the prediction of the Sabine absorption coefficient. Thus, it is not important whether the terms with $\eta_{\rm pa}$ are included or excluded in the determination of the coefficient. The observation is explained as follow.

In Eq. (9), $e^{-s_1 t}$ and $e^{-s_2 t}$ have a more significant weighting than the absolute value terms, and the Sabine absorption coefficient is then dependent on s_1 and s_2. Furthermore, Eqs. (4) and (5) for both the decay rates suggest that the value of $\eta_{\rm p} + \eta_{\rm pa}$ relative to that of $\eta_{\rm a} + \eta_{\rm ap}$ determines whether both $\eta_{\rm ap}$ and $\eta_{\rm pa}$ control the coefficient or only $\eta_{\rm ap}$ controls the coefficient. $\eta_{\rm p} + \eta_{\rm pa}$ and $\eta_{\rm a} + \eta_{\rm ap}$, respectively, represent the total acoustical dissipativity of the panel and of the sound field. In general, two conditions are usually encountered namely, $\eta_{\rm a} + \eta_{\rm ap} \ll \eta_{\rm p} + \eta_{\rm pa}$ and $\eta_{\rm a} + \eta_{\rm ap} \approx \eta_{\rm p} + \eta_{\rm pa}$. The condition of $\eta_{\rm a} + \eta_{\rm ap} \gg \eta_{\rm p} + \eta_{\rm pa}$ is rare and will not be considered here. It exists only when one uses a room which is acoustically dead and a metal panel which is undamped such that $\eta_{\rm a} \gg \eta_{\rm p}$. For the two usual conditions, s_1 and s_2 can be mathematically approximated as $(\eta_{\rm a} + \eta_{\rm ap} - C)\omega_0$ and $(\eta_{\rm p} + \eta_{\rm pa} + C)\omega_0$ where $C = \eta_{\rm ap}\eta_{\rm pa}/(\eta_{\rm p} + \eta_{\rm pa} - \eta_{\rm a} - \eta_{\rm ap})$. Thus, $s_1 \ll s_2$ and hence, $e^{-s_1 t} \gg e^{-s_2 t}$ for $t > 0$ where s_1 controls the decay of the sound field. Therefore, from the discussion in Sect. 3, $\alpha_{\rm Sab} \approx \alpha_1 \approx 4V_0\omega_0(\eta_{\rm ap} - C)/A_{\rm p}c_0$ where the dependence of $\alpha_{\rm Sab}$ on $\eta_{\rm pa}$ is only described in C.

Figure 5 indicates that when $\eta_{\rm p}$ is five times larger than $\eta_{\rm a}$ (i.e., $\eta_{\rm p} = 0.05$, $\eta_{\rm a} = 0.01$), $\eta_{\rm a} + \eta_{\rm ap}$ is much smaller than $\eta_{\rm p} + \eta_{\rm pa}$. In this case, $|C| \ll |\eta_{\rm ap}|$ and Fig. 6 shows that $|C/\eta_{\rm ap}| \ll 1$. Therefore, only $\eta_{\rm ap}$ determines the Sabine absorption coefficient because $\eta_{\rm pa}$ affects C which is of minor importance. Physically, the total dissipativity of the panel is much larger than that of the sound field such that the power flow from the panel to the sound field is negligibly small compared to the power flow from the sound field to the panel. This is evident in Fig. 7 where the ratio of radiated sound power to absorbed sound power (i.e., $\eta_{\rm pa}E_{\rm p}^{\rm a}/\eta_{\rm ap}E_{\rm a}^{\rm a}$) is significantly small in all the bands when $\eta_{\rm p} = 0.05$ and $\eta_{\rm a} = 0.01$. As a result, the sound radiation from the panel does not have much influence on the estimation of the Sabine absorption coefficient as seen in Fig. 4.

When $\eta_{\rm p}$ is close to $\eta_{\rm a}$ (i.e., $\eta_{\rm p} = 0.001$, $\eta_{\rm a} = 0.0005$), $\eta_{\rm a} + \eta_{\rm ap} \approx \eta_{\rm p} + \eta_{\rm pa}$ in most of the bands (see Fig. 5). In this case, the magnitude of C becomes large and is comparable to that of $\eta_{\rm ap}$ as illustrated in Fig. 6. Since C is directly proportional to $\eta_{\rm ap}$ and $\eta_{\rm pa}$ (see the expression for

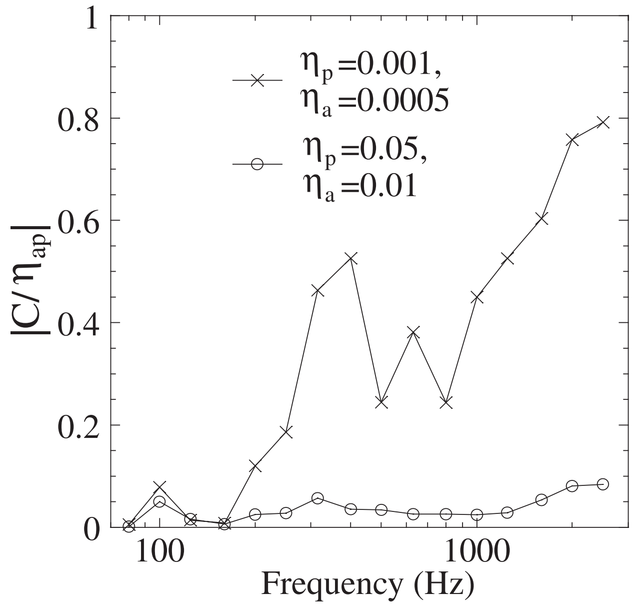

Fig. 6 Absolute ratio of C to the coupling loss factor from the sound field in the room to the glass panel.

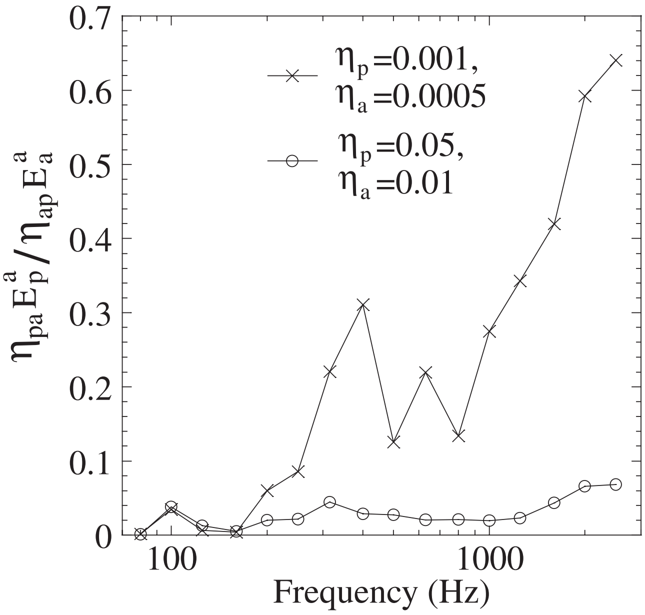

Fig. 7 Ratio of the radiated sound power to the absorbed sound power of the glass panel in the room.

C in the foregoing paragraph), both coupling loss factors are now in control of the Sabine absorption coefficient. This implies that physically, the total dissipativity of the panel is comparable to that of the sound field such that the power flow from the panel to the sound field is comparable to the power flow from the sound field to the panel. The phenomenon is shown in Fig. 7 where the values of $\eta_{pa}E_p^a$ are comparable to those of $\eta_{ap}E_a^a$ in many bands when $\eta_p = 0.001$ and $\eta_a = 0.0005$. Consequently, the sound radiation from the panel has a considerable influence on the prediction of the Sabine absorption coefficient (see Fig. 4).

In the experimental examples, the physical model of the acoustic-structural coupled system consists of a standard rectangular-parallelepiped reverberation room and six rectangular particle-board panels. The dimensions of the room are $(L_x, L_y, L_z) = (6.840, 5.565, 4.720)$ m, and the panels are identical and placed on the floor of the room. Each panel has these properties: $A_p = 1.5 \times 1.2\,\text{m}^2$, thickness = 10 mm, material density = 691 kgm^{-3} and longitudinal wave speed = 2,540 ms^{-1}. In order to avoid the contact between the panel surface and the floor, each panel is mounted along its edges to a wooden frame by screws where the frame is positioned below the panel. The thickness and height of the frame is 20 mm and 90 mm, respectively. As a result, an air gap of 90 mm in depth is formed between each panel and the floor, and the panels act as panel absorbers. 1/3-octave values of η_a for the sound field in the room in the absence of the panels, and those of η_p for the panels deduced from measurements in the room and an anechoic chamber, are provided [11]. Two different loudspeakers were used one after another as input sources to drive the sound field in the room, and a non-contacting electromagnetic driver was used as an input source to drive the panels. The cones of both loudspeakers are the same but one of the cones was attached onto a box and the other cone was left bare [11,25]. For convenience, the former is called "large speaker" and the latter is called "small speaker." The photos of the loudspeakers and their details are, respectively, available in Refs. [11] and [25]. There are two purposes of using two different loudspeakers. The first is to provide a simple means of changing η_a so that the coupling between the sound field and the panels could be varied. The second is to provide a way to confirm that all measurements were correct where the measured values of η_p must be the same for both loudspeakers. The data for η_a and η_p is presented in Fig. 8, and it is obvious that the use of the loudspeaker box significantly changes η_a in the bands below 250 Hz, but does not affect η_p that depends solely on uncoupled modal properties of the panels. The box acts as a small air cavity which couples with the sound field through the loudspeaker diaphragm that vibrates when the sound field decays, and provides sound absorption in the low frequency range. 1/3-octave values of E_p^a/E_a^a and E_a^p/E_p^p for the coupling between the sound field and one panel are also available [11]. By substituting the values of η_a, η_p and the energy ratios into Eqs. (10) and (11), η_{pa} and η_{ap} can be evaluated, and the data is shown in Fig. 9.

When the sound field is coupled to six panels, values of η_{ap}, E_p^a, E_p^p and A_p are increased by six times in the calculation of the sound-field decay rates, the amplitude of the exponential terms, α_1 and α_2. $C^\#$ is also evaluated by the expression for C where # denotes a multiple of six to η_{ap}. Figure 10 shows that α_{Sab} is always less than $\alpha_{Sab,apx}$

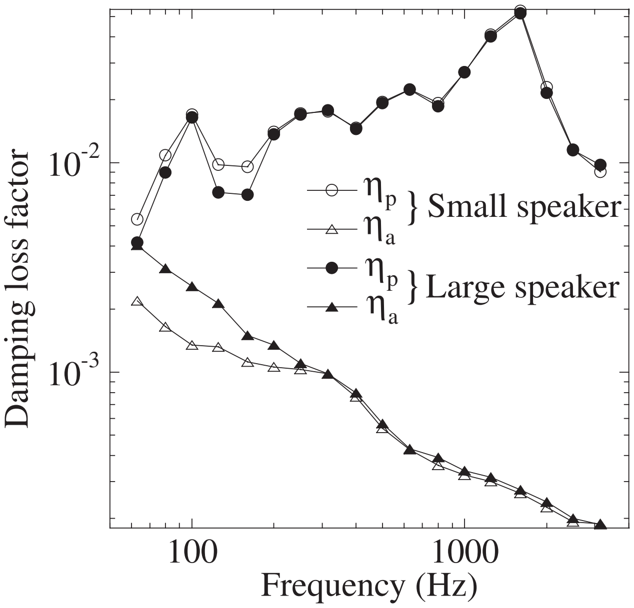

Fig. 8 Damping loss factors of the sound field in the reverberation room and the particle-board panels.

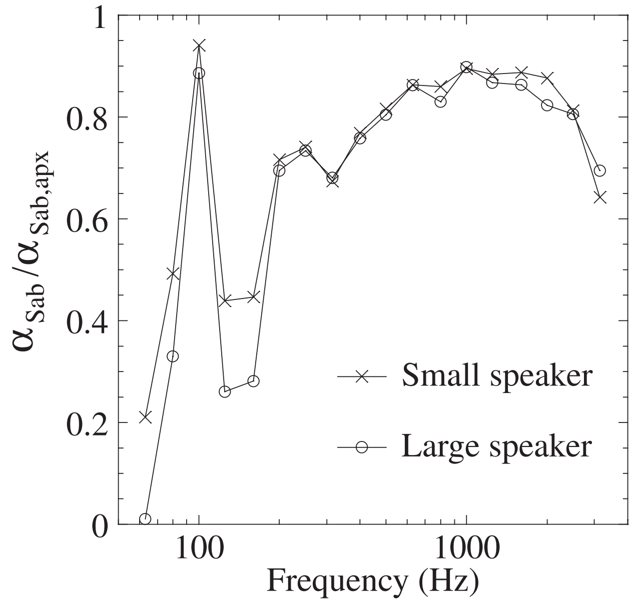

Fig. 10 Ratio of Sabine absorption coefficients of the particle-board panels in the reverberation room with and without including the sound radiation from the panels.

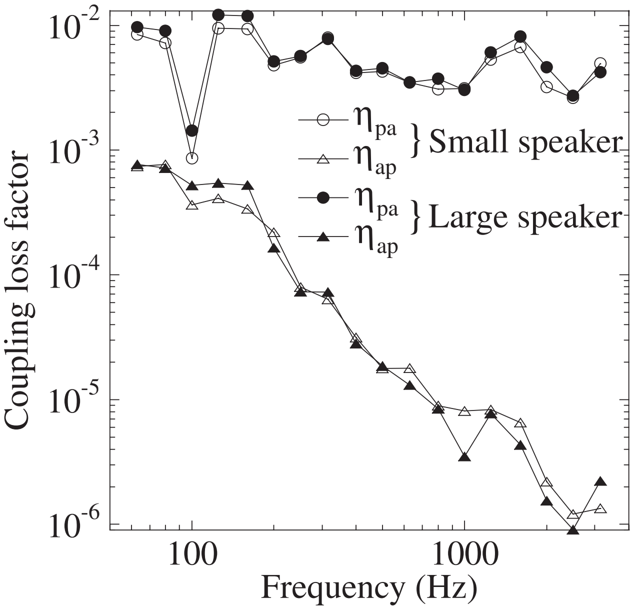

Fig. 9 Coupling loss factors for the coupling between one particle-board panel and the sound field in the reverberation room.

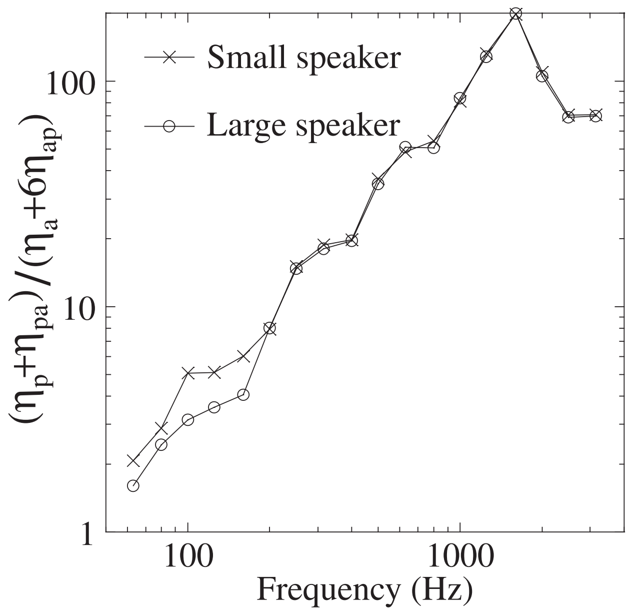

Fig. 11 Ratio of the total acoustical dissipativity of one particle-board panel to the total acoustical dissipativity of the sound field in the reverberation room.

which is similar to the computational results in Fig. 4, and suggests that the sound radiation affects the Sabine absorption coefficient in such a way that values of the coefficient are decreased. In most of the bands below 400 Hz, the radiation has a significant influence on the coefficient where α_{Sab} is about 70% of $\alpha_{\mathrm{Sab,apx}}$ or less for both loudspeakers. Figure 8 indicates that values of η_{a} and η_{p} in these bands are close to each other and thus, values of $\eta_{\mathrm{a}} + 6\eta_{\mathrm{ap}}$ are of the same order of magnitude as those of $\eta_{\mathrm{p}} + \eta_{\mathrm{pa}}$ in the bands (see Fig. 11). In this case, the magnitude of $C^{\#}$ is comparable to that of $6\eta_{\mathrm{ap}}$ as illustrated in Fig. 12. Since $C^{\#}$ depends significantly on both η_{ap} and η_{pa}, these coupling loss factors are in control of the Sabine absorption coefficient. As a result, values of $\eta_{\mathrm{pa}}E_{\mathrm{p}}^{\mathrm{a}}$ are comparable to those of $\eta_{\mathrm{ap}}E_{\mathrm{a}}^{\mathrm{a}}$ and hence, the power flows to and from each panel are comparable in the bands (see Fig. 13). Therefore, the negligence of the sound radiation

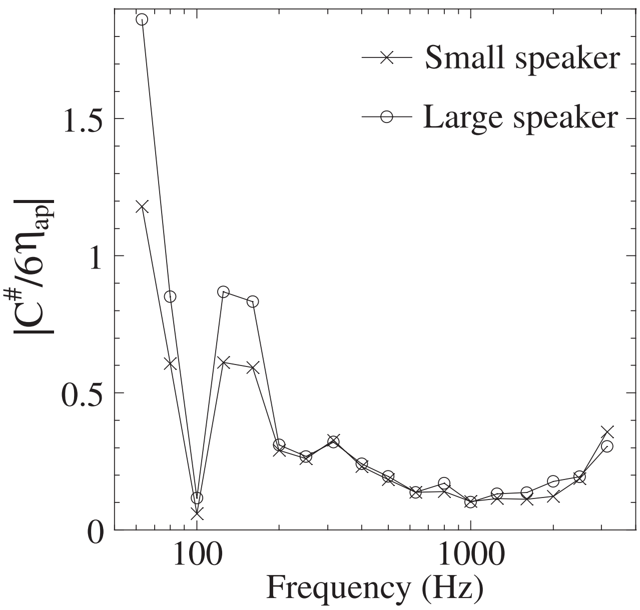

Fig. 12 Absolute ratio of $C^{\#}$ to the total coupling loss factor from the sound field in the reverberation room to the particle-board panels.

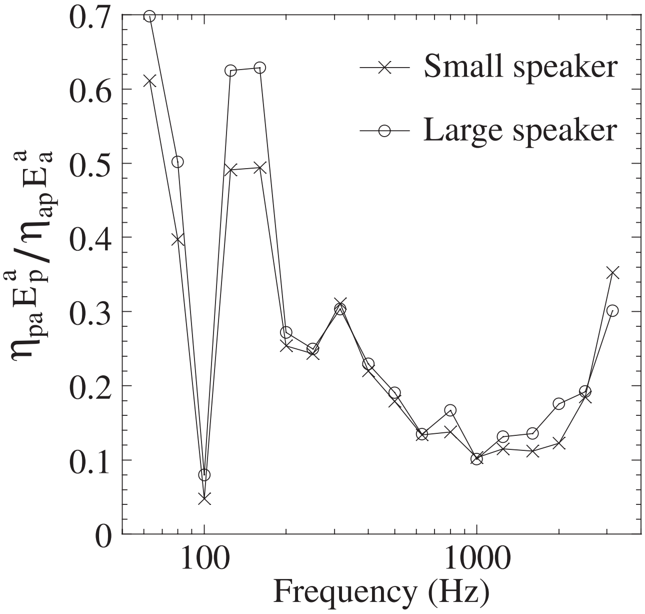

Fig. 13 Ratio of the radiated sound power to the absorbed sound power of one particle-board panel in the reverberation room.

causes large errors in the prediction of the Sabine absorption coefficient as depicted in Fig. 10.

Due to the decreasing trends of η_a and η_{ap} with frequency (see Figs. 8 and 9), these loss factors, respectively, become much smaller than η_p and η_{pa} in the 400-Hz band and above. So, $\eta_a + 6\eta_{ap} \ll \eta_p + \eta_{pa}$ in these bands (see Fig. 11) and thus, $|C^{\#}| \ll |6\eta_{ap}|$ as shown in Fig. 12. This means that $C^{\#}$ and thus, η_{pa}, are of minor importance and only η_{ap} determines the Sabine absorption coefficient. It is obvious from Fig. 13 that for each panel, the radiated power (i.e., $\eta_{pa}E_p^a$) is less than 20% of the absorbed power (i.e., $\eta_{ap}E_a^a$) in most of the bands above 400 Hz. The results suggest that the effect of the sound radiation on the value of the Sabine absorption coefficient is not significant and this is illustrated in Fig. 10.

5. CONCLUSIONS

In this paper, SEA is used to establish a relationship between the Sabine absorption coefficient of a modally-reactive panel in a room and the sound absorption and radiation of the panel. The absorptivity and radiativity of the panel are, respectively, described by η_{ap} and η_{pa}. The conditions under which the sound radiation from the panel is either significant or insignificant to the determination of the Sabine absorption coefficient, are investigated. The value of $\eta_p + \eta_{pa}$ relative to $\eta_a + \eta_{ap}$ is used as a general indicator of whether the radiation is significant or not. $\eta_p + \eta_{pa}$ describes the total acoustical dissipativity of the panel and $\eta_a + \eta_{ap}$ describes the total acoustical dissipativity of the sound field. When η_p is much larger than η_a, $\eta_a + \eta_{ap} \ll \eta_p + \eta_{pa}$ and the panel has a greater ability to dissipate energy compared to the sound field. Correspondingly, the power flow from the panel to the sound field is negligibly small compared to the power flow from the sound field to the panel. In this case, η_{ap} is shown to determine the Sabine absorption coefficient and η_{pa} only has a minor effect on the coefficient. In other words, it is not important whether or not the sound radiation is taken into account in the estimation of the coefficient. However, when η_p is close to η_a, $\eta_a + \eta_{ap} \approx \eta_p + \eta_{pa}$ and the dissipativity of the sound field is comparable to that of the panel. In this case, the power flows to and from the panel are comparable where both η_{ap} and η_{pa} are shown to be in control of the Sabine absorption coefficient. Therefore, since both the sound absorption and radiation of the panel are considerable, the latter must be accounted for in the prediction of the coefficient. The results in this paper provide a basic understanding of effects of the sound radiation from modally-reactive panels in rooms on the determination of the Sabine absorption coefficient of the panels.

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