Applied Acoustics 169 (2020) 107471

Contents lists available at ScienceDirect

Applied Acoustics

journal homepage: www.elsevier.com/locate/apacoust

Technical note

A note on using panel diffusers to improve sound field diffusivity in reverberation rooms below 100 Hz

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ARTICLE INFO

Article history:
Received 28 March 2020
Received in revised form 24 May 2020
Accepted 2 June 2020

Keywords:
Sound field diffusivity
Panel diffuser
Low frequency
Reverberation room

ABSTRACT

In many standard acoustic tests, the sound field in a reverberation room is required to be sufficiently diffuse. The standard deviation of sound pressure levels is commonly used to evaluate the spatial uniformity of a sound field; however, this paper shows that the standard deviation of squared sound pressures is a better indicator of sound field diffusivity at frequencies below 100 Hz where the sound field is very uneven. Fixed diffusers are recommended in ISO 354 to improve sound field diffusivity in reverberation rooms, and performing sound absorption measurements with an increasing number of diffusers is employed as a check on the diffusivity of the sound field above 500 Hz. This paper demonstrates that typical panel diffusers (as suggested in ISO 354) cannot increase the sound field diffusivity at low frequencies. While low frequency diffusivity can be improved with large panels at some frequencies, the diffusivity at other frequencies generally deteriorates. Experimental results in a reverberation room are presented to support the numerical simulation results and analyses.

1. Introduction

To obtain accurate results in acoustic tests such as the measurement of sound absorption coefficients, transmission loss and sound power using the reverberation room method, the sound field needs to be sufficiently diffuse [1–3]. An ideal diffuse sound field can be described as the superposition of plane waves with random phases and equal magnitudes with the propagation directions uniformly distributed over all angles of incidence [4]. Many different measures have been proposed to evaluate the diffusivity of a sound field, such as the spatial uniformity of pressure, uniformity of decay rate, and linearity of decay curves [4]. Two of the most commonly used descriptors are the spatial uniformity of the pressure field and the cross-correlation between pressures at neighboring positions [5].

Sizes and shapes of reverberation rooms have effects on the sound field diffusivity, and it has been found that oblique-shaped room with a reduced vertical dimension gives better field diffusivity and better measurement accuracy [6]. To further increase the diffusivity of the sound field in reverberation rooms, rotating vanes can be installed such that the motion of each vane causes fluctuations in sound pressure, Doppler effects, and the changes in the radiation impedance of a source. Overall, these contributions all improve the diffusivity of the sound field in the room [7].

Surface diffusers also affect the diffusivity of the sound field. It has been demonstrated that a semicircular diffuser alters the resonant frequencies and modal distributions of the sound field in a cavity [8]. The effect of panel diffusers on the spatial uniformity of the sound field has also been analyzed with the discovery that their orientations have an insignificant effect on the diffusion levels in a reverberation room [9].

Investigations on volume diffusers found that they can enhance the diffusivity in reverberation rooms [10,11]. D'Antonio et al. applied both spherical cap domes and suspended convex clouds spaced in a primitive root sequence to provide low frequency diffusion below 100 Hz [12]. In addition to the traditional surface and volume diffusers, Schroeder proposed phase grating diffusers, including Schroeder quadratic residue diffusers (QRDs), maximum length diffusers (MLDs) and primitive root diffusers (PRDs), and offered the possibility of producing optimum diffusion [13–15]. Recently, a sonic crystal diffuser consisting of two arrays of sonic crystals with slightly different periodicities was proposed and achieved high diffusivity at low frequencies [16].

The sound field is usually diffuse above the Schroeder frequency due to strong modal overlap [17], but achieving sound field diffusivity at low frequencies is often a challenge because there are only a few dominating modes in that frequency range. ISO 354 recommends panel diffuser implementations to achieve acceptable

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https://doi.org/10.1016/j.apacoust.2020.107471
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sound field diffusivity above 500 Hz in a reverberation room, and specifies sound absorption measurements to check that diffusivity [2]. The sound absorption coefficients of the test specimen are measured without diffusers, and then with increasing quantities of diffusers. The mean sound absorption coefficient is expected to approach a maximum and then remains constant with increasing numbers (total area) of diffusers.

This paper investigates the sound field diffusivity concentrating on the low frequency range. Measures describing the sound field diffusivity at low frequencies are investigated first, and then the effects of the number and size of panel diffusers on the sound field diffusivity in the low frequency range are discussed. Finally, the experimental results in a reverberation room are presented to support the numerical simulation results and analyses.

2. Measures to quantify sound field diffusivity

The standard deviation of the sound pressure levels (SPLs) is commonly used to evaluate the spatial uniformity of the sound field in reverberation rooms, i.e. [3]

$$s_L = \sqrt{\sum_{i=1}^{N_M} \frac{(L_i - L_M)^2}{N_M - 1}} \tag{1}$$

where L_i is the SPL at the ith receiver position, N_M is the number of receiver points, and L_M is the arithmetic mean value of the SPLs at all the receiver points.

The concept of a diffuse sound field is based on the assumption that the acoustic energy density is uniformly distributed over the major portion of the interior space of the room [18], and the spatial uniformity of acoustic energy density is thus an important indicator of sound field diffusivity. The sound energy density is proportional to the squared sound pressure, for example, at the ith receiver point for plane wave radiation, it equals to $p_i^2/2\rho_0 c_0^2$, where ρ_0, the air density, and c_0, the speed of sound in air are both constants, and p_i is the sound pressure at this point. Therefore, we hypothesize that the standard deviation of squared sound pressures might be a better description of the spatial uniformity of the sound field, whereby:

$$\Delta p^2 = \sqrt{\sum_{i=1}^{N_M} \frac{(p_i^2 - p_M^2)^2}{N_M - 1}} \tag{2}$$

where p_M^2 is the mean of the squared sound pressures at all the receiver points. Instead of averaging SPLs at receiver points, Δp^2 uses the mean squared sound pressures to characterize the average sound field. To better compare with the commonly used standard deviation of SPLs, s_p is used in this manuscript, which is:

$$s_p = 10\log_{10}\left(\frac{p^2{}_M + \Delta p^2}{\bar{p}^2{}_M}\right) \tag{3}$$

where $\bar{p}^2{}_M$ is the average of the mean squared sound pressure p_M^2 over the whole frequency band of interest, so it is a dimensionless quantity.

Numerical simulations were performed on a model of a rectangular room to compare the standard deviations of SPLs and squared sound pressures. The room size is 6.0 m × 5.9 m × 6.5 m with the lowest resonant frequency of 26 Hz, and the acoustic impedance of all the boundaries of the room were set as (56472 – 68455i kg/m^{2}s), corresponding to a normal incidence sound absorption coefficient of approximately 0.013. The sound source is a point monopole near one corner of the room at (5.5, 5.4, 1.45) m with a strength of 10^{-4} m^3/s (the origin of the coordinate is at one corner of the room, as shown in Fig. 2). The commercial Finite Element Method (FEM) software COMSOL Multiphysics Version 5.0 was used to calculate the sound pressure in the room. Sound pressures at 2940 points evenly distributed over the entire volume of the room with an interval of 0.4 m were used to evaluate the diffusivity of the sound field.

The standard deviations of SPLs and squared sound pressures calculated with Eqs. (1) and (3), as well as the SPL at one receiver point are shown in Fig. 1 (a) and (b), respectively. Most peaks in the curve of the standard deviations of squared sound pressures in Fig. 1(b) correspond to the peaks of the SPL curve. This is reasonable because the sound fields at resonant frequencies are the least uniform; however, there are no such trends for the standard deviation of SPLs in Fig. 1(a).

The SPLs are averaged when calculating the s_L while the squared sound pressures are averaged when calculating the s_p. Averaging SPLs does not have the same physical meaning as averaging squared sound pressures. The later is more reasonable for describing sound field diffusivity because the squared sound pressure is proportional to the sound energy density. When the sound field is very uneven, averaging SPLs and squared sound pressures are very different, for example, the average of 70 dB and 40 dB is 55 dB when averaging the SPLs, while the value is 64 dB when they are averaged with squared sound pressure, so it is inappropriate to evaluate the spatial uniformity using the average of SPLs in this case. Because the sound field at low frequencies is dominated by a few modes and is usually uneven, the standard deviation of SPLs is not an effective indicator of sound diffusivity in this frequency range. However, at higher frequencies where the sound field is almost uniform in reverberation rooms, averaging SPLs and squared sound pressures give similar results, so s_L is still effective at high frequencies.

Fig. 2(a) shows the sound pressure distribution in the room at 64 Hz, which is close to a room resonant frequency. The standard deviation of squared sound pressures at 64 Hz is 10.2 dB, which is the highest below 100 Hz; however, the standard deviation of SPLs in Fig. 1(a) is only 6.2 dB at 64 Hz, which is even lower than that at some non-resonant frequencies such as 66 Hz. The sound pressure distribution at 66 Hz is shown in Fig. 2(b) for comparison and it is clearly more uniform than that in Fig. 2(a). These results suggest that the standard deviation of squared sound pressures is a more reasonable indicator of sound field uniformity when the sound field is apparently not uniform. Because the sound field is not uniform in the reverberation room at low frequencies, the standard deviation of squared sound pressures is used in the following sections for assessing the sound field diffusivity in reverberation rooms.

3. Effects of panel diffusers on sound field diffusivity

To investigate the effects of panel diffusers on sound field diffusivity, the model of the reverberation room at University of Technology Sydney (UTS) was used. The model is shown in Fig. 3(a), and the dimensions are as indicated in Fig. 3(b).

3.1. Number of panel diffusers

ISO 354 recommends using diffusers of different sizes, ranging from approximately 0.8 m^2 to 3 m^2 in area (for one side) to achieve an acceptable diffusivity [2], and the size of the panels used here is 1.5 m × 1.2 m with a thickness of 0.01 m. The panels are randomly placed in the reverberation room at the height between 3 m and 6 m. In the simulations, all the boundaries, including the floor, ceiling and walls of the reverberation room and the surfaces of the panels, are assumed to have the same impedance (56472 – 68455i kg/m^{2}s). There are N_M = 57 evaluation points

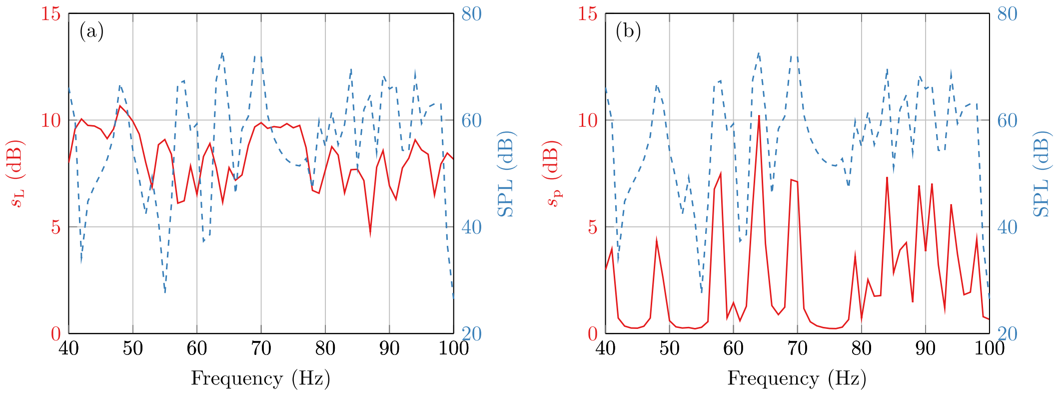

Fig. 1. The SPL at one receiver point (right y axis) and the standard deviations (left y axis) of: (a) SPLs and (b) squared sound pressures.

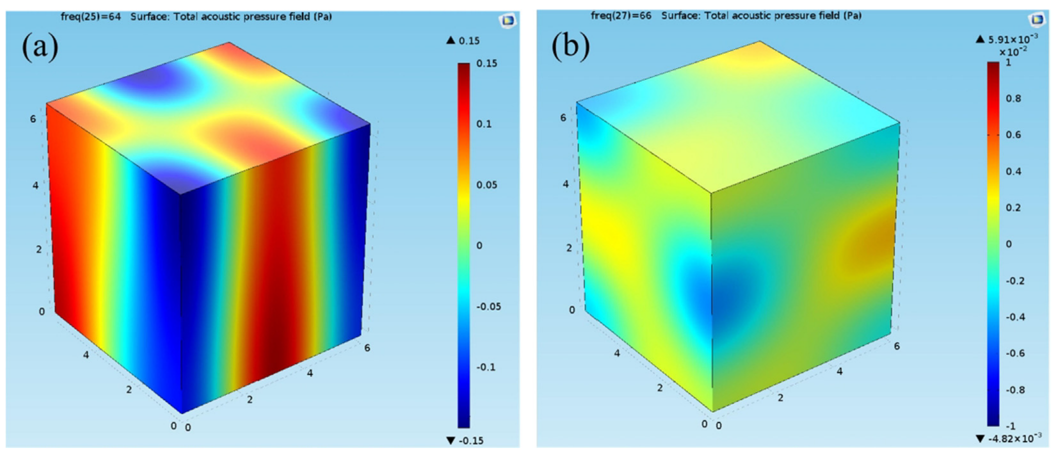

Fig. 2. The distribution of sound pressure at: (a) 64 Hz and (b) 66 Hz.

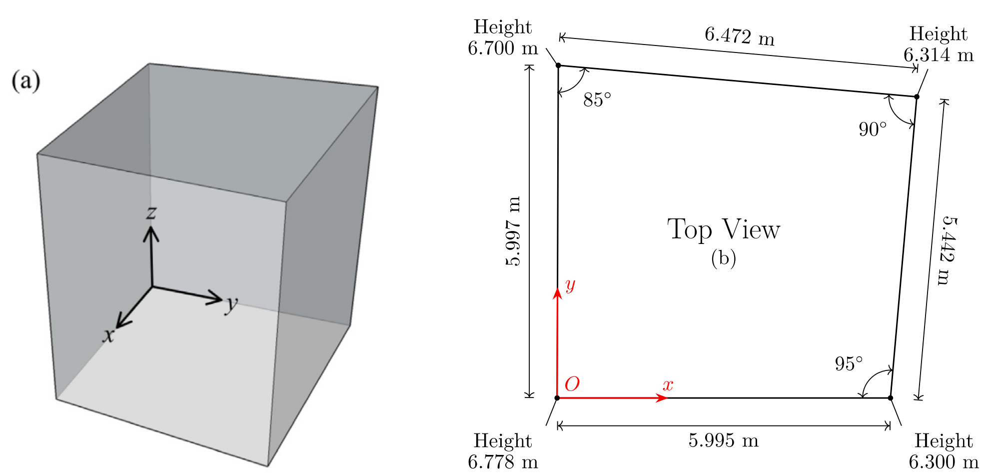

Fig. 3. (a) The model of the reverberation room, (b) the dimensions of the reverberation room.

and all of them are at least 2.0 m away from the sound source and 1.0 m away from the boundaries. The standard deviations of squared sound pressures in 1/12 octave bands with different numbers of panels are shown in Fig. 4.

Fig. 4(a) shows that there is no significant difference between the standard deviations with 3 or fewer panels in the low frequency range because the peak frequencies of all the curves are almost the same and the differences of standard deviations at all the frequencies are less than 2.6 dB. The standard deviations with more panels (up to 16, 25.3% of the total surface area of the reverberation room) are shown in Fig. 4(b). It is clear that the standard deviations decrease at some frequencies while increase at some other frequencies. This suggests that adding more panels is ineffective in increasing the diffusivity in the whole frequency range below 100 Hz.

3.2. Size of panel diffusers

Different panel sizes, specifically: 1.5 m $\times$ 1.2 m, 3.0 m $\times$ 2.4 m, and 4.5 m $\times$ 3.6 m were investigated and the thickness of all the

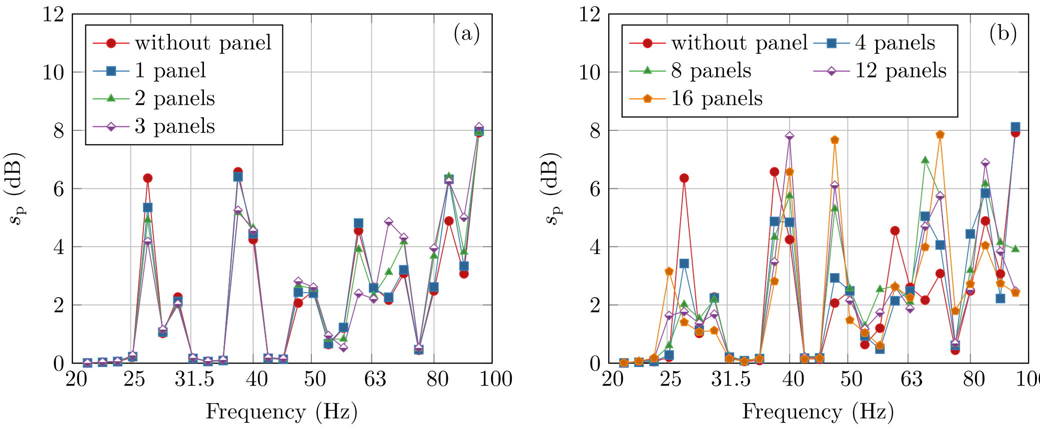

Fig. 4. Standard deviations of squared sound pressures in 1/12 octave bands with different numbers of 1.5 m × 1.2 m panels in the reverberation room: (a) 0, 1, 2, and 3 panels, (b) 0, 4, 8, 12 and 16 panels.

panels was again set as 0.01 m. The standard deviations of squared sound pressures in 1/12 octave bands are shown in Fig. 5. It can be observed that for a small panel, the standard deviation of squared sound pressures is almost the same as that without the panel because the panel is smaller than the wavelength of the frequency of interest thus has little effect on the modal distributions in the room.

As the panel becomes larger, the standard deviations of squared sound pressures at some frequencies decrease while those at some other frequencies increase. Although large panels can change the distribution of a sound field at the original resonant frequencies, thus increasing the spatial uniformity at these frequencies, they introduce new resonant frequencies at which the spatial uniformity decreases. Sufficient sound field diffusivity can be achieved if the modal density is sufficiently high. The numbers of modes in 1/3 octave bands with different numbers and sizes of panels are listed in Table 1.

The empirical requirement for sufficient sound field diffusivity is that there are more than 20 modes in each 1/3 octave band [6]. It is clear in Table 1 that the number of modes remains almost unchanged with more or larger panels below the 1/3 octave band with the center frequency 80 Hz, where the sound field is not sufficiently diffuse (fewer than 20 modes in each 1/3 octave band). Therefore, adding panel diffusers does not increase the modal density of the room, and does not increase the sound field diffusivity at low frequencies.

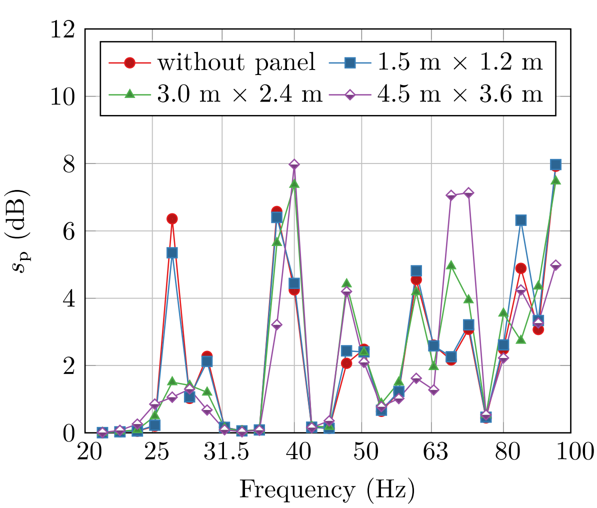

Fig. 5. The standard deviations of squared sound pressures in 1/12 octave bands with a panel of different sizes in the reverberation room model.

Another possible approach to improve sound field diffusivity is to distribute the resonant frequencies more uniformly [19]. However, according to Fig. 6, there is no significant difference between the standard deviations of squared sound pressures at resonant frequencies with and without the panel, so adding large panel diffusers is not an effective way to improve sound field diffusivity in the low frequency range.

The sound field diffusivity with more panels of larger sizes was also investigated. When the panel size is 4.5 m × 3.6 m, the standard deviations of squared sound pressures are shown in Fig. 7. The standard deviations tend to become smaller at some frequencies when more panels are introduced; however, those at some other frequencies increase at the same time because of the new resonant frequencies introduced. The total area of the 3 panels is about 42% of the total surface area of the room, which has exceeded 15–25% as suggested in ISO 354, but they still cannot increase the diffusivity at all the frequencies below 100 Hz. Besides, because the panels are very large (almost half the size of the reverberation room), they are not feasible to be used in practice. Therefore, adding panel diffusers does not work as an effective way to increase sound field diffusivity in the low frequency range.

It should be noted that although panel diffusers have almost no effect on the sound field diffusivity at low frequencies, they are effective for the sound field at high frequencies (see Figs. A1 and A2 in the Appendix). One way to guarantee sufficient sound field diffusivity in the low frequency range is to build a large reverberation room specially for low frequency sound, while using another smaller reverberation room for measurements at middle to high frequencies. Another alternative method is to design new diffusers based on metasurfaces [20,21], which are small compared to the wavelength but works effectively to increase the diffusivity at low frequencies. This is the research we are currently working on.

3.3. Sound absorption coefficients

This section investigates the relationship between the sound field diffusivity and the variation trends of the sound absorption coefficient as the number (total area) of the panel diffusers increases. Numerical simulations on sound absorption coefficients at low frequencies were carried out with COMSOL. The inverse Fast Fourier Transform (IFFT) was applied on the sound pressure in the frequency domain to calculate the impulse response from the source to the receiver position. The impulse response is then filtered in 1/12 octave bands, and the energy decay curve can be

Table 1
Number of modes in 1/3 octave bands.

Panel size	Center frequency (Hz)	25	31.5	40	50	63	80	100
	without panel	2	2	3	3	9	14	23
1.5 m × 1.2 m	1 panel	2	2	3	3	9	14	23
	2 panels	2	1	3	3	8	14	23
	3 panels	2	1	3	3	9	14	23
4.5 m × 3.6 m	1 panel	2	1	3	3	10	14	25
	2 panels	2	1	4	3	10	13	26

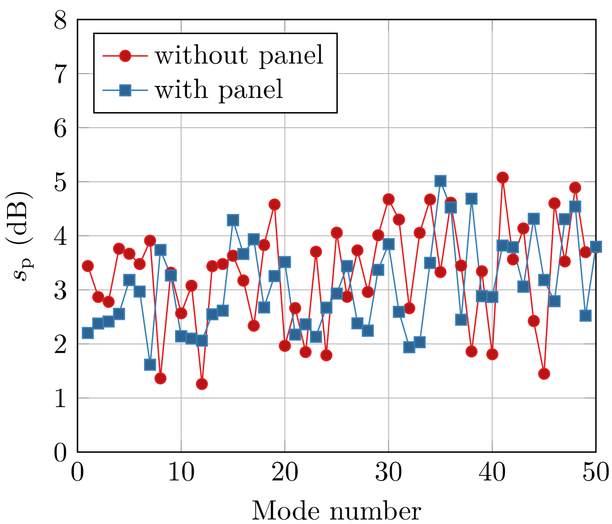

Fig. 6. Standard deviations of squared sound pressures at the first 50 resonant frequencies with and without a 4.5 × 3.6 m panel.

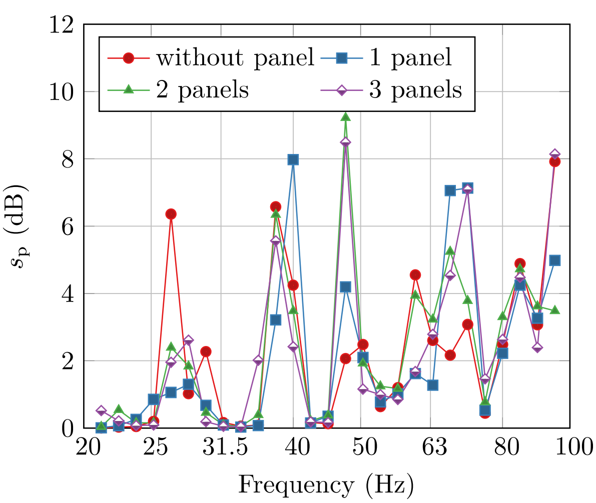

Fig. 7. Standard deviations of squared sound pressures in 1/12 octave bands with different numbers of 4.5 m × 3.6 m panels.

obtained with its backward integration [2]. The reverberation time can be calculated according to the energy decay curve. Assuming that the reverberation times with and without the specimen are T_2 and T_1, respectively, according to Sabine's reverberation formula [17], the sound absorption coefficient of the specimen is then:

$$\alpha = 0.161\frac{V}{S_0}\left(\frac{1}{T_2} - \frac{1}{T_1}\right) \quad (4)$$

where V is the volume of the reverberation room, and S_0 is the area of the specimen.

To perform the sound absorption calculation in numerical simulations, it was assumed that a test specimen of 3.2 m × 3.2 m × 0.05 m was placed on the floor (in the middle) of the reverberation room. The impedance of the specimen was set as (156 + 917i kg/m^{2}s), corresponding to a normal incidence sound absorption coefficient of 0.22, which is a common value for normal porous sound absorption material in the low frequency range. All the panel diffusers have a sound absorption coefficient of approximately 0.013, which is the same as that of the floor, ceiling and side walls of the reverberation room. When there are different numbers of panels randomly placed in the reverberation room, the sound absorption coefficient in 1/12 octave bands calculated with Eq. (4) are shown in Fig. 8. In Fig. 8, the sound absorption coefficients tend to remain unchanged as 8 or more panels are introduced, which is an indicator of sufficient diffusivity as suggested in ISO 354; however, the standard deviations of squared sound pressures are still very large with these panels (see Fig. 4(b) for details). For example, the standard deviation in the 1/12 octave band with center frequency 47.6 Hz when there are 16 panels is larger than 7 dB, which indicates that the sound field in this frequency range is very uneven. This investigation shows that sound absorption measurement is not an effective way to check the diffusivity at frequencies below 100 Hz. However, the procedure suggested in ISO 354 to check the sound field diffusivity is, as expected, quite valid in the middle and high frequency ranges (see Fig. A3 in the Appendix).

4. Experiments

The experiments were carried out in the reverberation room at UTS, as shown in Fig. 9. Perspex panels of 1.5 m × 1.2 m × 0.01 m were used in the experiments, and the spatial uniformity without

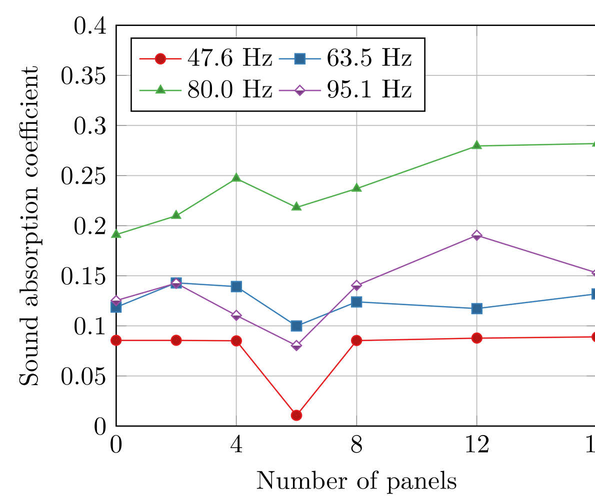

Fig. 8. Sound absorption coefficients calculated with different numbers of 1.5 m × 1.2 m panels.

Fig. 9. Photos of the experimental setup in the reverberation room at UTS: (a) without panels, (b) with panels hanging on the ceiling, (c) with a large panel on the ground, (d) the omni-directional sound source.

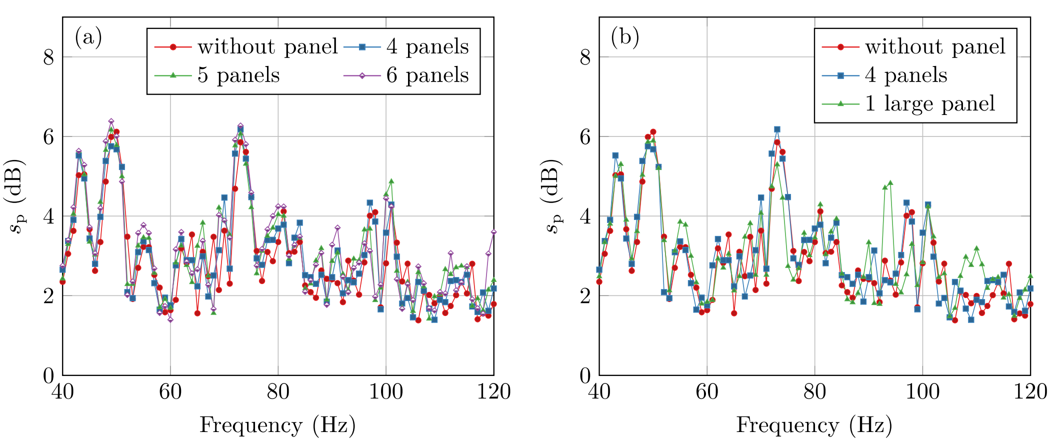

Fig. 10. Standard deviations of squared sound pressures in 1/12 octave bands without panels and with: (a) different numbers of panels, and (b) 4 panels and 1 large panel.

and with 4, 5, and 6 panels, and a large panel consisting of 2 perspex panels glued together were investigated. The corresponding photos are shown in Fig. 9(a)–(c). The sound source was a Brüel & Kjær Type 4292 omni-directional sound source and it was located near one of the corners of the reverberation room at the height of 1.45 m, as shown in Fig. 9(d). The SPLs at 60 points in a line with a spacing of 5 cm were measured simultaneously with Brüel & Kjær Type 4957 microphones. The microphone array was put on a frame at the height of 1.2 m, and it was moved 10 cm forward at a time to cover an area of 2.95 m $\times$ 4.0 m and there were a total of 60 $\times$ 41 = 2461 measurement points. The signal to noise ratio is more than 20 dB in the whole frequency band of interest. The sound pressures at low frequencies were compensated during data processing to ensure that the frequency response of the loudspeaker is almost flat over the whole frequency range of interest.

The standard deviations of squared sound pressures between 40 Hz and 120 Hz with different numbers of perspex panels are shown in Fig. 10 (a). It can be observed that the standard deviations almost remain unchanged with different numbers of panels, which indicates that small panels have little effect on the diffusivity of the sound field. The standard deviations of squared sound pressures without and with 4 panels as well as with the large panel are shown in Fig. 10(b). Compared with the cases without or with 4 panels, the standard deviations at some frequencies decrease after introducing the large panel, while those at some other frequencies increase. Therefore, adding panels is not an effective way to increase the sound field diffusivity in the low frequency range, which is consistent with the conclusions in numerical simulations.

5. Conclusions

The sound field diffusivity in a reverberation room in the low frequency range is discussed in this paper. The standard deviation of squared sound pressures was found to be a better indicator of sound field diffusivity than that of SPLs in the low frequency range. The effect of the size and number of the panel diffusers on the sound field diffusivity in reverberation rooms below 100 Hz was investigated with COMSOL. It was found that small panels have little effect on sound field diffusivity at low frequencies. Although large panels can improve the sound field diffusivity at some low frequencies, they deteriorate the sound field diffusivity at some other frequencies. Therefore, adding panel diffusers is not an effective way to improve sound field diffusivity below 100 Hz because it does not increase the modal density or make the modal distribution more uniform. It was also found that performing sound absorption measurements with increasing number (total area) of diffusers cannot be used as an effective way to check the sound diffusivity at low frequencies. Experiments in a reverberation room demonstrate the validity of the numerical simulation results and analyses. Future work can be exploring effective ways to improve the sound field diffusivity at low frequencies by using acoustic metamaterials.

CRediT authorship contribution statement

Shuping Wang: Methodology, Software, Experiment, Writing - original draft. **Jiaxin Zhong:** Experiment, Writing - review & editing. **Xiaojun Qiu:** Supervision, Writing - review & editing. **Ian Burnett:** Writing - review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

This research is supported under the Australian Research Council's Linkage Project (No. LP160100616).

Appendix

The numerical simulations at middle to high frequencies were carried out using ODEON, a commercial room acoustics software. The models of the reverberation room with 1, 2, 3, and 4 panels of 1.5 m × 1.2 m × 0.01 m randomly distributed in it are shown in Fig. A1. The sound absorption coefficients of all the surfaces (including those of the panels) were set as 0.013. The sound source

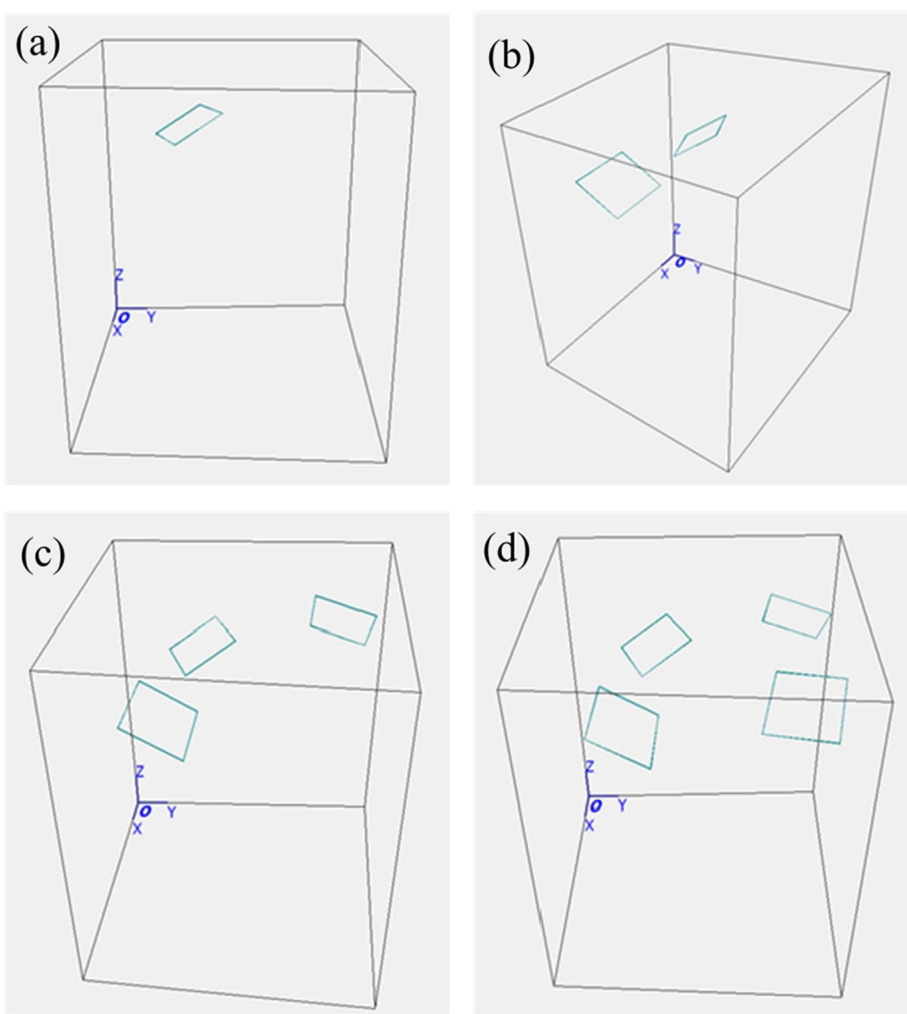

Fig. A1. Room models created in ODEON with: (a) 1 panel, (b) 2 panels, (c) 3 panels, and (d) 4 panels.

was at (5.5, 5.0, 1.45) m and sound pressures at N_M = 22 points evenly distributed at the height of 1.2 m above the floor with an interval of 1.0 m were used to evaluate the spatial uniformity of the sound field.

The standard deviations of squared sound pressures in octave bands above 500 Hz are shown in Fig. A2. It can be seen that implementing more panels helps to decrease the standard deviations of squared sound pressures first and they remain almost constant once there are sufficient panels. Another observation is that the improvement of spatial uniformity by implementing more panels becomes less apparent as the frequency increases. The reason is that the sound field is already sufficiently diffuse at high frequencies (for example, 8 kHz), and there is no need to introduce panel diffusers to improve the spatial uniformity in this frequency range.

Fig. A3 shows the sound absorption coefficients calculated with Eq. (4) at middle to high frequencies with different numbers of panel diffusers. It was assumed that a 3.2 m × 3.2 m × 0.05 m test specimen with the sound absorption coefficient 0.95 was placed on the floor (in the middle). It can be seen from Fig. A3 that the calculated sound absorption coefficients generally increase with the number of panels and remain almost constant when there are a sufficient number of panels, which is consistent with the guidelines in ISO 354, and the sound field is almost diffuse in this frequency range (with s_p less than 0.3 dB, see Fig. A2), therefore, the procedure suggested in ISO 354 to check the sound field diffusivity is valid in the middle and high frequency ranges.

The panel diffusers such as the perspex ones used in the experiments will vibrate in practical applications and there exist sound transmissions through them. Fig. A4 shows the results when the vibrations and sound transmissions of 16 panels of size 1.5 m × 1.2 m × 0.01 m with free boundary conditions are taken into consideration. In the simulations, the Young's modulus of the panels is set as 3 × 10^9 Pa, the Poisson's ratio is 0.39 and the density is 1190 kg/m^3, which are close to those of a typical perspex panel as used in the experiments. Fig. A4(a) shows the displacements on the surface of the 16 panels at 80 Hz and Fig. A4(b) shows the standard deviations of squared sound pressures without and with the vibrations taken into consideration. It is clear that the standard deviations change at some frequencies but overall there is no significant difference between them, so the vibrations and sound transmissions through the panel diffusers have an insignificant effect on the sound diffusivity level in the reverberation room at low frequencies.

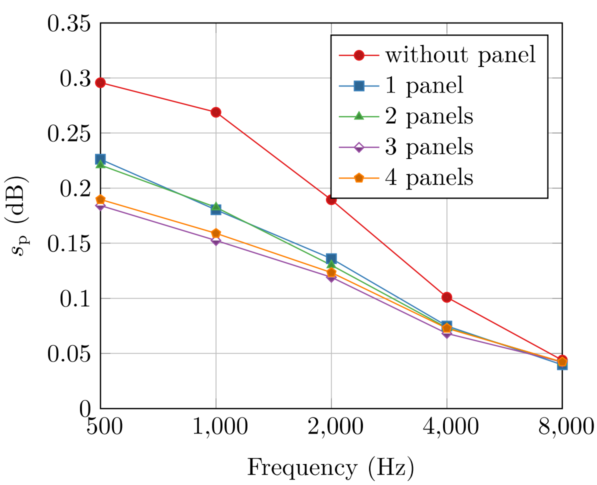

Fig. A2. Standard deviations of squared sound pressures in octave bands at middle to high frequencies with different numbers of 1.5 m × 1.2 m panels.

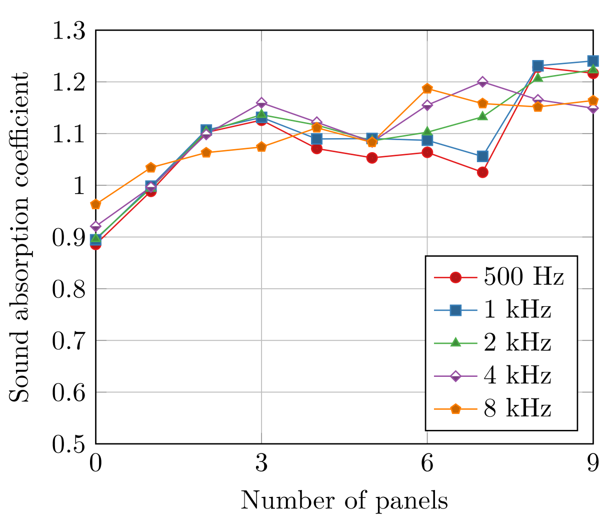

Fig. A3. Sound absorption coefficients calculated with different numbers of 1.5 m × 1.2 m panels at middle to high frequencies.

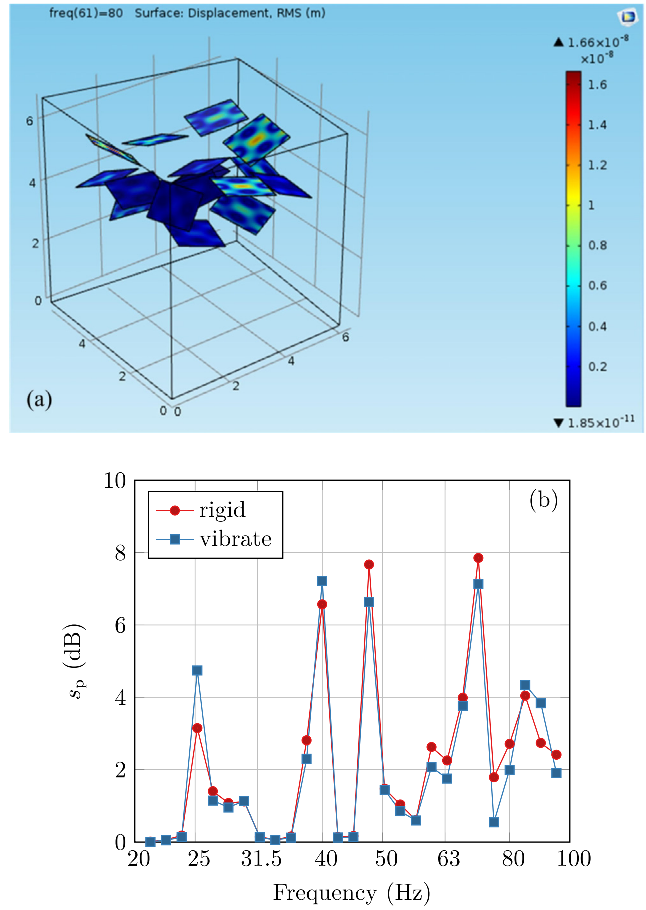

Fig. A4. (a) The displacements on the surfaces of the 16 panels at 80 Hz, (b) the standard deviations of squared sound pressures when the panels are rigid and when the vibration and sound transmission are taken into consideration.

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