A study on partial enclosures for noise control

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ABSTRACT

Partial enclosures are widely used to reduce noise radiation while keeping ventilation, lighting and easy access through openings at the same time. However, because of the complexity of the sound field around partial sound barriers, no method has been reported on how to design them to achieve the best noise reduction performance. This paper uses numerical simulations to study the properties and performance of partial enclosures. A case study on a simplified model consisting of several separate acoustic panels surrounding the sound source is carried out and some general rules are found which might be useful for optimizing the design. Experiments in the anechoic chamber are carried out to support simulation results.

1 INTRODUCTION

Sound absorption has been widely used to reduce noise radiation for a long time (Emms G, 2001 and Kang J, 2005). Theoretically, a completely closed enclosure achieves the best noise reduction performance; however, in many practical applications, the noise sources cannot be completely sealed and gaps must be kept for the purpose of access, maintenance, ventilation and/or heat dissipation. Partial enclosure is an option to reduce noise radiation in this case, but the openings in enclosures provide paths for noise transmission which deteriorates the passive noise reduction performance (Ver I, 2006). Such a partial enclosure needs to be designed to achieve a balance between noise reduction and ventilation.

Sound barriers are usually constructed to reduce direct sound within a local area because a shadow zone lies behind the barrier which is dominated by the diffracted sound and they have been applied in many applications such as along highways (Wang X, 2017 and Bowlby W, 1986). Previous study shows that applying sound absorption materials on the barriers will improve the performance of sound barriers while a large reflective surface deteriorates its performance (Watts G, 1999 and Pan J, 2004). Both theoretical and numerical simulation method such as Boundary Element Method (BEM) have been proposed to investigate the sound propagation and insertion loss of various kinds of noise barriers (Pierce A, 1974, Ishizuka T, 2004 and Wang X, 2015), but due to the complexity of the sound field, there is little work reported on partial enclosure design to achieve the best noise reduction performance.

In this paper, a numerical simulation method is used to study the performance of the partial enclosure first, and then experiments are carried out to prove its feasibility. It is found that partial enclosures can achieve satisfactory local noise reduction performance and they can even achieve global noise reduction with a proper design.

2 THEORY

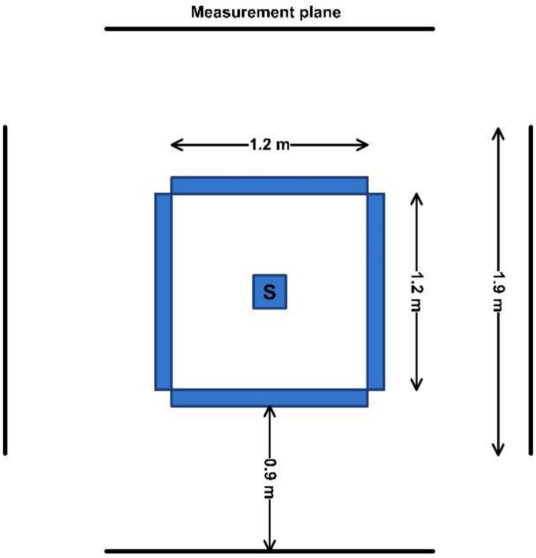
The insertion loss of partial enclosure can be estimated by (Ver I, 2006)

$$\mathrm{IL} = 10\log\left[1+\alpha\left(\frac{\Omega_{\mathrm{tot}}}{\Omega_{\mathrm{open}}}-1\right)\right]\ \mathrm{dB} \tag{1}$$

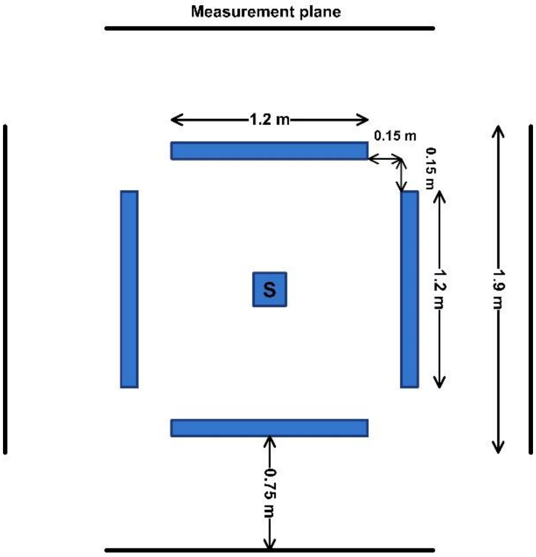
where Ω_{tot} is the solid angle of sound radiation of the enclosed source and $\Omega_{open} = S_{open}/r$ is the solid angle at which the enclosed source (located at a distance r) "sees" the opening of area S_{open}. It can be concluded from Eq. (1) that larger open area will lead to less insertion loss and deteriorate the performance of the partial enclosures, but it is beneficial to the ventilation. Therefore, partial enclosures have to be specially designed to achieve a balance between them. The equation is usually used for middle to high frequencies where the wavelength of the sound is much smaller than the geometry of the enclosure.

3 NUMERICAL SIMULATIONS AND EXPERIMENTS

A partial enclosure consisting of 4 panels around a sound source is investigated, as shown in Fig. 1. Each panel is of the same size 1.20 m × 0.90 m, 0.60 m away from the sound source. There are 55 points evenly distributed on a 1.90 m × 1.10 m plane, 0.75 m behind each of the 4 barriers to evaluate the insertion loss of the partial barrier. The interval between adjacent evaluation points is 0.19 m. The average sound pressure level difference at the 220 points in the free field and with the barrier is defined as the insertion loss. To investigate the influence of open area on insertion loss, two conditions are considered: 4 panels are connected to each other (closed) as shown in Fig. 1(a) and separate (open) as shown in Fig. 1(b). All the 4 panels are considered as rigid and the sound source is a point source at the height of 0.15 m.



(a)



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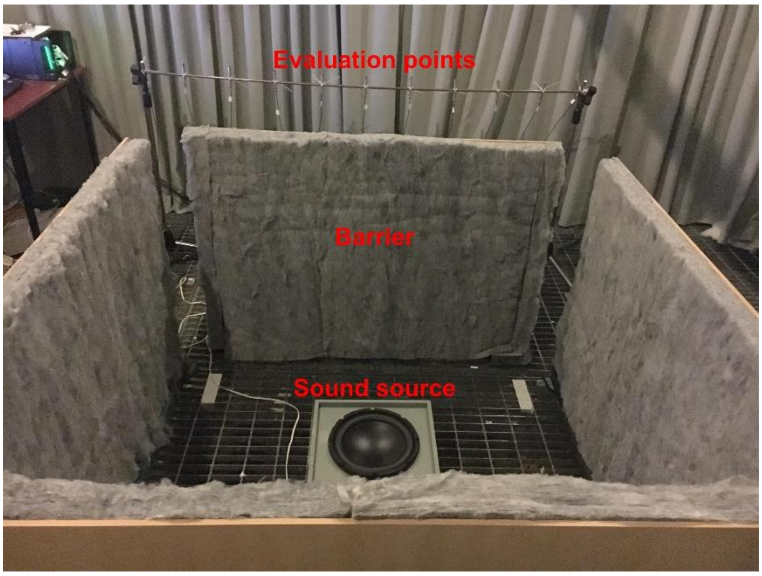
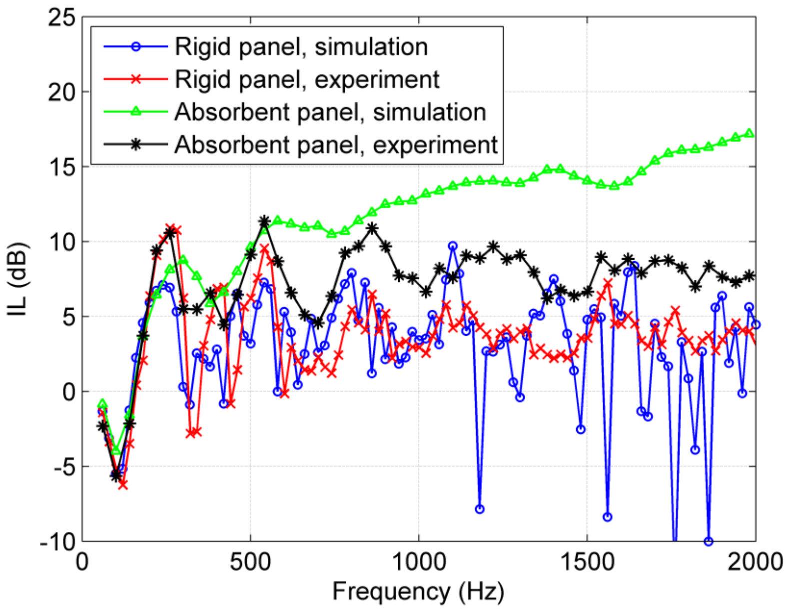


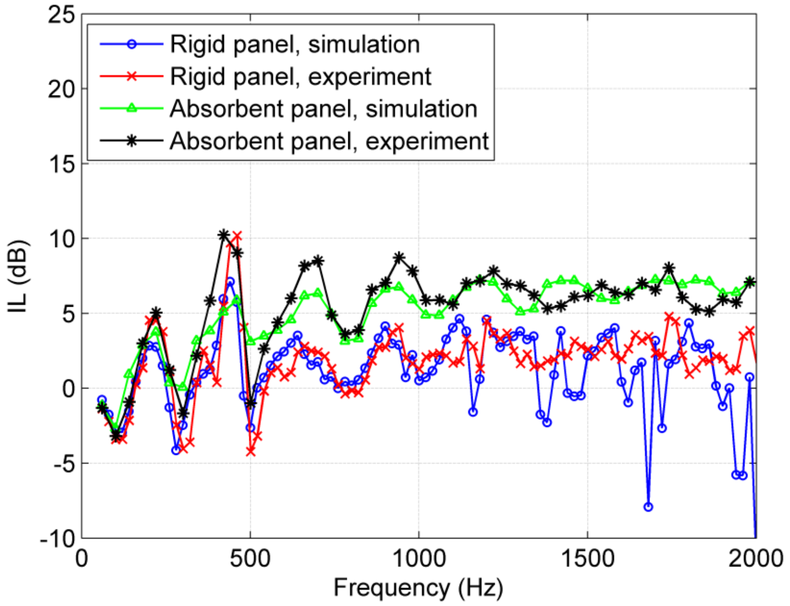
(c)

Figure 1: A partial barrier consisting of four panels, (a) closed setup, (b) open setup, (c) experimental setup in the anechoic chamber.

The average insertion loss at the 220 evaluation points is obtained by numerical simulations using a commercial BEM software Sysnoise 5.6 and the results are shown in Fig. 2(a) and 2(b). Experiments on the same setup as that in the simulations are carried out in an anechoic chamber to check the simulation results and the measured insertion loss is also plotted in Fig. 2(a) and 2(b) for comparison. It can be seen that the average insertion loss is about 3 dB above 800 Hz. The numerical simulation and experimental insertion loss agree reasonably well in low frequency range (below 1200 Hz). The reasons why there is a bigger difference at high frequencies might be: (1) The loudspeaker applied in experiments cannot be treated as a point source at high frequencies because of the directivity; (2) The size of the loudspeaker box is about 30 cm × 30 cm × 20 cm, and sound scattering from it plays more important role at high frequencies than at low frequencies.



(a)



(b)

Figure 2: The comparison between simulation and experimental ILs, (a) closed setup, (b) open setup.

To increase the insertion loss of the partial barrier, sound absorption material is attached on the inner side of the partial barriers. The thickness of the sound absorption material is 5 mm and its sound absorption coefficient is measured and shown in Fig. 3. The corresponding acoustic impedance applied on the surfaces in the numerical simulations is predicted by referring to (Attenborough K, 1992). The sound absorption coefficient calculated by the impedance is also shown in Fig. 3, which agrees well with the measured results. The numerical simulation and experimental insertion loss with sound absorption materials are shown in Fig. 2(a) and 2(b) as well. It can be seen that the insertion loss is increased to more than 5 dB above 800 Hz.

It is found by comparing Figs. 2(a) and 2(b) that the insertion loss when the four panels are separate is less than that when they are connected to each other, but the difference is very little which means that the gap of about 0.2 m between adjacent panels can be kept without affecting the insertion loss significantly.

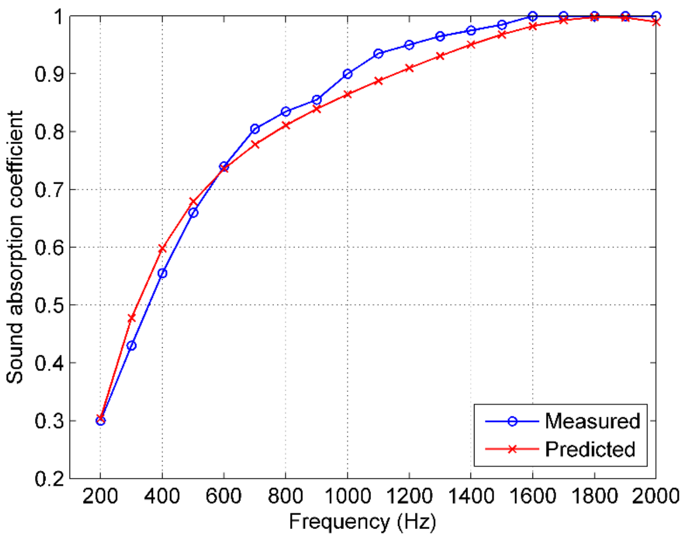


Figure 3: The measured and predicted sound absorption coefficient of the material used in experiments.

To further investigate the effect of the open area, a partially opened enclosure consisting of five panels as shown in Fig. 4 is investigated. The 5 separate panels are of size 0.432 m × 0.67 m, 0.432 m × 0.598 m and 0.67 m × 0.598 m. The sound source is at the centre of the enclosure.

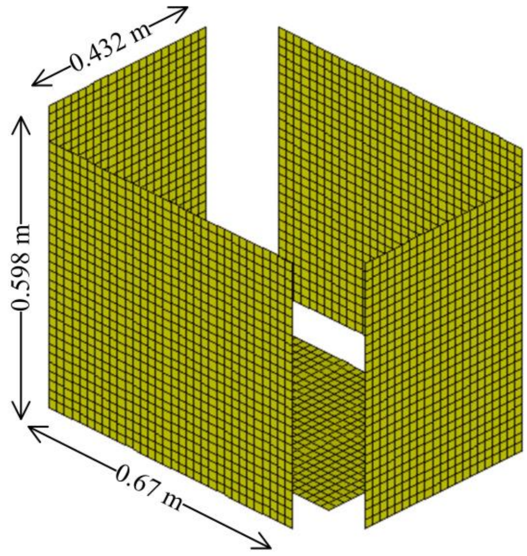


Figure 4: The model of the partially opened enclosure in Sysnoise.

Assume that the inner side of the five panels are sound absorbent, the average sound pressure level at 1106 evaluation points evenly distributed on a sphere of radius 5 m centered at the sound source is used to evaluate the passive noise reduction performance of the enclosure. The average sound pressure level at the evaluation points when the gap between two adjacent panels is 0.1 m, 0.2 m, 0.4 m and 0.8 m is shown in Fig. 5.

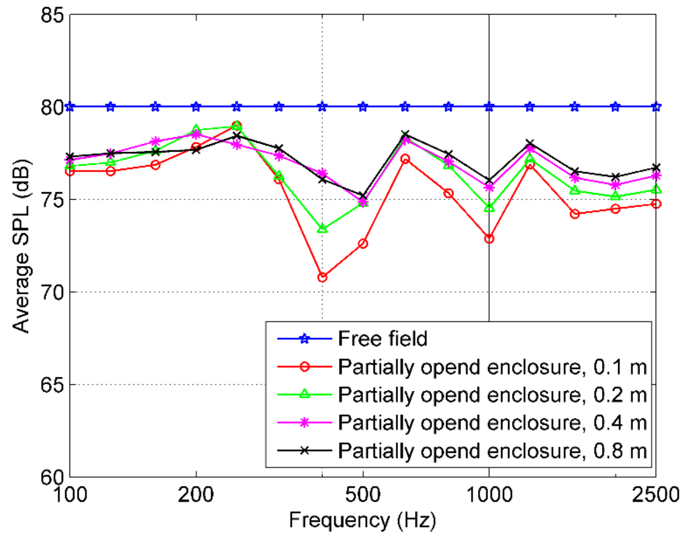


Figure 5: The average sound pressure level at evaluation points.

The difference of the average sound pressure level (SPL) at all the evaluation points in the free field and when the partially opened enclosure exists can be regarded as the sound power reduction. It is clear in Fig. 5 that the sound power reduction is the highest when the gap is 0.1 m and it decreases as the gap between panels becomes larger. It can also be seen from Fig. 5 that there are some peaks and valleys of the noise reduction which is related to the positions and sizes of the panels. When the gap is 0.1 m, the noise reductions are the highest at 400 Hz and 1000 Hz, which are 9.2 dB and 7.1 dB, and the least at 250 Hz and 630 Hz, which are 1.0 dB and 2.8 dB. Figure 6 and Fig. 7 shows the distribution of sound pressure level at plane z = 0.598 m (the top opening of the enclosure) and y = 0.335 m at these 4 frequencies with and without the partial enclosure. The dashed lines in Figs. 6(a)-6(d) and Figs. 7(a)-7(d) represent the positions of the side walls. It can be seen from the results that the top opening is the main path for sound to transmit from inside the partial enclosure to the outside, and the gaps between side walls have different effects on sound propagation at different frequencies.

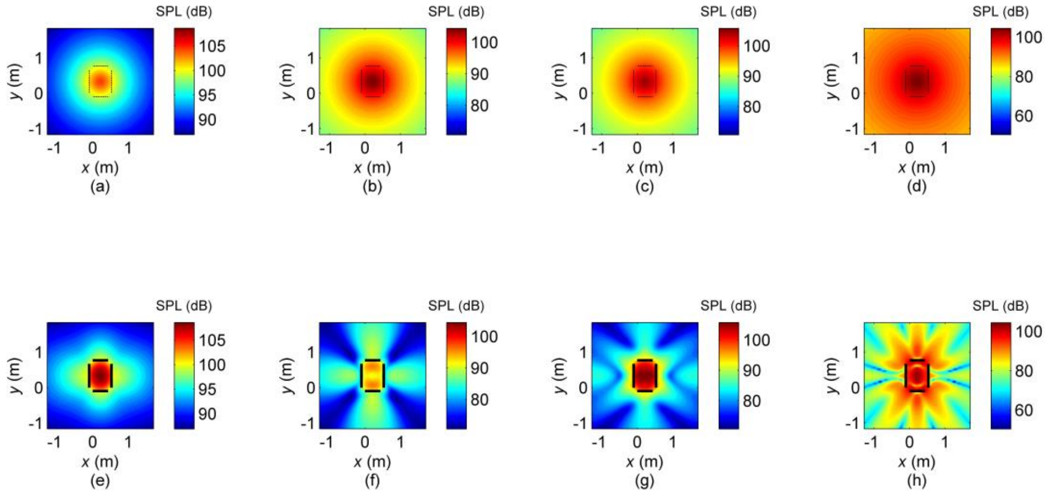


Figure 6: The sound pressure level at plane z = 0.598 m, (a) 250 Hz, in the free field, (b) 500 Hz, in the free field, (c) 630 Hz, in the free field, (d) 1000 Hz, in the free field, (e) 250 Hz, with the enclosure, (f) 500 Hz, with the enclosure, (g) 630 Hz, with the enclosure, (h) 1000 Hz, with the enclosure.

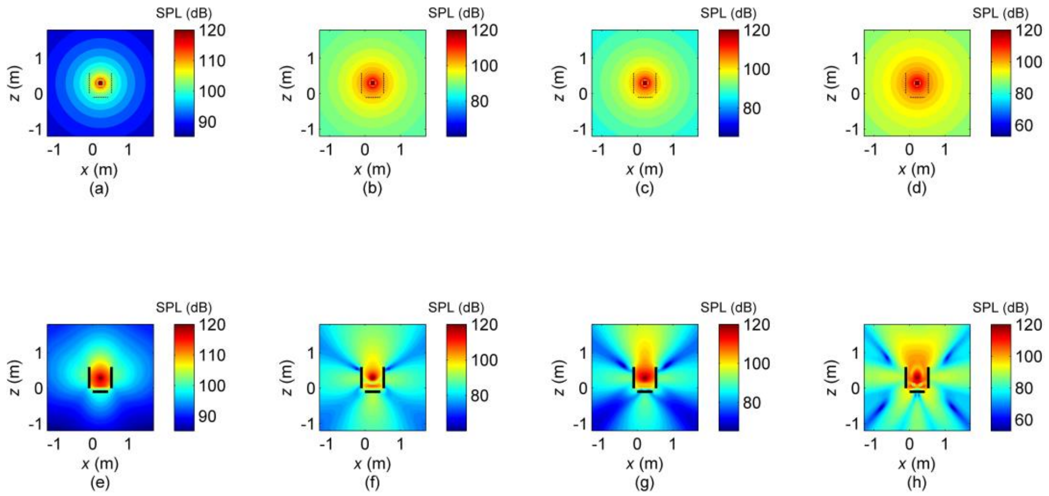


Figure 7: The sound pressure level at plane y = 0.335 m, (a) 250 Hz, in the free field, (b) 500 Hz, in the free field, (c) 630 Hz, in the free field, (d) 1000 Hz, in the free field, (e) 250 Hz, with the enclosure, (f) 500 Hz, with the enclosure, (g) 630 Hz, with the enclosure, (h) 1000 Hz, with the enclosure.

4 CONCLUSIONS

The performance of partial sound barriers is investigated by numerical simulations and experiments are carried out in the anechoic chamber to verify the simulation results. It is found that although there are some gaps in the partial sound barriers which deteriorate its passive noise reduction performance, the partial sound barrier is still effective within some areas. With a proper design, partial sound barriers can achieve global noise reduction and the performance is related to the frequencies and the gaps. Generally, partial sound barriers work more effectively in high frequency range and active noise control can be combined in the future to increase the noise reduction at low frequencies. Such a hybrid control system might have better performance over a wide frequency band. Future work also includes finding out a proper way to design partial enclosures to achieve the best noise reduction performance or even global control.

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