Composites: Part B 40 (2009) 259–266

Contents lists available at ScienceDirect

Composites: Part B

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

Composite panels for reducing noise in air conditioning and ventilation systems

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A R T I C L E I N F O

Article history:
Received 12 November 2008
Accepted 29 January 2009
Available online 5 February 2009

Keywords:
A. Foam
A. Carbon-fibre
B. Vibration

A B S T R A C T

The plate silencer [Huang L. Broadband sound reflection by plates covering side-branch cavities in a duct. J Acoust Soc Am 2006;119:2628–38] that consists of an expansion chamber with two side-branch cavities covered by a light panel can achieve a desirable noise reduction in broadband theoretically. The concept is similar to drum silencer [Choy YS, Huang L. Experimental studies of drum-like silencer. J Acoust Soc Am 2002;112:2026–35]. To attain optimal noise reduction, either the membrane of the silencer should be of minimal weight while retaining very high tensile strength or the panel should be kept at very high bending stiffness that is dependent on its geometry and mechanical properties. To achieve such goal, various kinds of composite system such as carbon fibres or aluminum were mounted on light core foam to build a noise reflection panel. A design of composite panels which can provide a reduction in panel weight as well as enhance the bending stiffness, is introduced in this paper. Predictions of the new model are to be compared with the normal foam plate in the aspects of noise reflection capability and performance of noise abatement apart from the material properties.

1. Introduction

It is well known that the reduction of duct noise is an area of continuous interest for its practical importance. In a typical heating ventilation and air conditioning (HVAC) system, there are different kinds of noise source, which include the noise caused by bearings, motors and belts; fan noise and aerodynamic noise. Low frequency noise component such as 200 Hz is very difficult to control. Due to the limitation of noise control at the source, various noise reduction methods have been developed. Placing porous absorption materials inside the duct lining is one of the commonly used methods recently. It is a reliable, economic and mature method that enables satisfactory noise reduction from medium to high frequency ranges but it renders bad performance at low frequencies. Recently, many researchers have used panel absorbers to absorb low frequency noise. For example, Kiyama et al. [3] found that the panel absorbers provided good noise absorption at low frequency range and the thickness of the insulating materials is inversely proportional to the frequency of sound. Nevertheless, the frequency range for noise control by this kind of absorber is still narrow. To effectively control the low frequency noise, a relatively new approach of using reflection of sound by membrane or plate silencer is more desirable [2,4,5]. The plate silencer consists of an expansion chamber with two side-branch cavities covered by a light core panel. There is a strong coupling between the acoustic wave inside the duct, cavity and vibration of the light panel. By adopting the panel with an appropriate stiffness and an optimized depth of the cavity, the noise can be greatly reduced in wide frequency range especially at low frequency condition. The noise abatement is mainly attributed to the effective sound reflection due the vibration of the panel. The condition to achieve this is to use a very light but high bending stiffness panel. The panel with high bending stiffness can promote the dominant of the first and second mode of vibration [6] which are very crucial to reflect the sound effectively at low frequency. It is difficult to search a plate made of raw material possessing very light mass and the sufficiently high bending stiffness. Recently, a honeycomb sandwich panel, which consists of hexagonal aluminum honeycomb cores and two thin plates, is adopted in many engineering applications such as automotive and spacecraft [7,8]. This kind of panel has superior characteristics in bending stiffness and light weight compared with the traditional metal plate. However, the requirement of the mass in the current project is extremely light compared with the traditional panel adopted in the structure of automotive or aircraft. The manufacture of very light honeycomb plate by using very thin metal membrane and honeycomb structure remains difficult. Therefore, we attempt to design the configuration of composite plate by different materials such that the noise reduction in wide frequency range can be achieved. Apart from the investigation of the design of the composite panel, an attempt is also made here to investigate the design with different combinations of plates and different geometrical cavity. In all previous studies on drum-like or plate silencers, two rigid cavities are equipped on two oppo-

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doi:10.1016/j.compositesb.2009.01.004

site sides of a rectangular duct [2,5]. The plates, which are flush mounted with the duct walls, are covered by the rectangular cavities completely. Such a configuration is in fact two-dimensional (2D), say on x–y plane. Wang [9,10] extended this into so-called 4-cavity-plate silencer which is three-dimensional (3D) designs theoretically. There is a need to validate the experimental study and have a good design for practical applications.

In what follows, theoretical outline of plate silencer is briefly given in Section 2 for the determination of material properties for realistic applications. Section 3 introduces the design of different kind of composite plate for plate silencer. Section 4 present the performance of plate silencers by using different kind of composite plates based on the criterion adopted to assess the performance of drum silencer [11]. Section 5 shows the comparison between two plate silencers and four plate silencers. The conclusions are drawn in Section 6.

2. Theory

Fig. 1 illustrates the working principle of the plate silencer in two-dimensional configuration. It has a two-dimensional duct of height h^* with two plates flush mounted and clamped on the duct wall. The asterisks denote the dimensional variable while the corresponding dimensionless parameters are denoted shortly without asterisks. The plates with the length L^* are backed by a rigid-walled cavity of depth h_c^*. The two lateral edges of the plates are set free and hence the three-dimensional plates are minimized to a two-dimensional beam. A plane incident wave comes from the left hand side of the duct with a unit of amplitude and it excites the plate into the vibration with a transverse displacement of complex amplitude $\eta^*(x)$ and velocity $V^*(x)$. The vibration of plate then radiates sound and induces a radiation pressure on its surface. Therefore, it involves the fully coupling between the acoustics and plate vibration.

For convenience, parameters are normalized by three basic quantities: air density ρ_0, speed of sound c_0, and the duct height h. The normalization scheme for the four most important parameters such as frequency$f$, plate-to-air mass ratio m, bending stiffness B and plate displacement η are given as follows:

$$f = \frac{f^* h^*}{c_0^*}, \quad m = \frac{m^*}{\rho_0^* h^{*2}}, \quad B = \frac{B^*}{\rho_0^* c_0^{*2} h^{*3}}, \quad \eta = \frac{\eta^*}{h^*}.$$

The dynamic of the lower plate vibration is governed by

$$B\frac{\partial^4 \eta}{\partial x^4} + m\frac{\partial^2 \eta}{\partial t^2} + (p_i + \Delta p) = 0 \tag{1}$$

where m is mass ratio, B is the plate bending stiffness, p_i is the dimensionless incident wave and Δp is the dimensionless sound pressure difference acting on the plate and it is attributed to the fluid loading on the upper and lower sides of the plate induced by the plate vibration itself. In the theoretical study, the damping is not considered because the sound reflection is dominant in the plate silencer. From the Eq. (1), the vibration of the plate, which depends on the mass ratio m and the bending stiffness B, controls the sound radiation. The total sound pressure transmitted p_t to the downstream of the plate silencer is found by adding incident wave to the far-field radiation wave. The transmission loss can be found as

$$TL = 20\log_{10}\frac{|p_i|}{|p_t|}. \tag{2}$$

And the reflection coefficient β as well as absorption coefficient α are

$$\beta = \left|\frac{p_r}{p_i}\right|^2 \text{and}\, \alpha = 1 - \left|\frac{p_r}{p_i}\right|^2 - \left|\frac{p_t}{p_i}\right|^2, \text{respectively}. \tag{3}$$

Wang and Huang [9] proved that the higher transmission loss can be achieved by higher bending stiffness and lower mass of the plate. The theoretical transmission loss for the plate silencer with the mass ratio $m = 1$ and $B = 0.0698$ and the duct height of 100 mm is shown in Fig. 2. There are three sharp peaks attributed to the dominant of the first and second mode of membrane vibration. We follow the approach adopted by drum silencer [11] to assess the performance of plate silencer. The performance of the drum silencer [11] has been evaluated by the logarithmic width of its stopband, defined as the frequency range in which the transmission loss (TL) is everywhere equal to or greater than a suitable criterion level (TL_{cr}). Expressed symbolically,

$$TL|_{f\in[f_1,f_2]} \geqslant TL_{cr}, \quad r_f = f_2/f_1.$$

where r_f is the ratio of frequency limits of the stopband, or the logarithmic bandwidth. The value of r_f thus becomes the cost function in design optimization, and it is one way of emphasizing the importance of low frequency noise control. In the previous studies [2], the typical geometric limitation of cavity volume is set at $10h^3$, where h is the duct height, and the optimal $r_f = 2.61$ is achieved for $TL_{cr} = 10$ dB. Therefore, in the present study, the criterion value is also chosen to be 10 dB. Therefore, the target to design the composite panel for such kind of panel silencer is to achieve as high as possible for r_f.

Different optimal bending stiffness B_{opt} is required for different mass of the plate in order to achieve the widest bandwidth r_f. The optimized parameters are summarized in Table 1. For example, if the mass ratio of panel is 2, the required optimal bending stiffness is 0.1 which is very high. We are going to verify the theory by

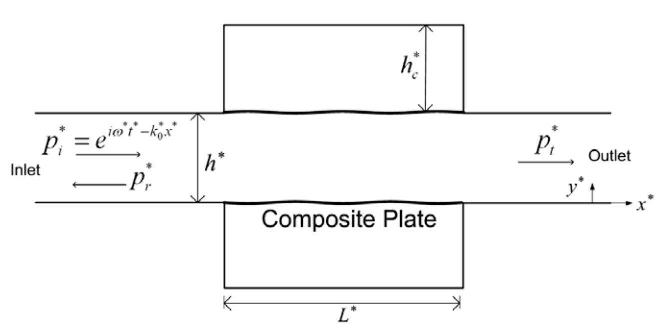

Fig. 1. Working principle of the plate silencer in two-dimensional configuration.

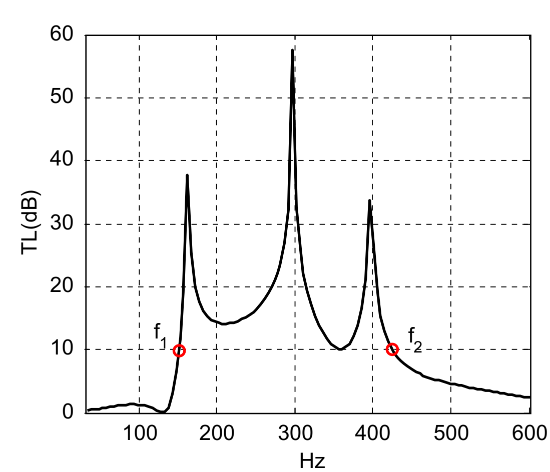

Fig. 2. Transmission loss spectrum for the plate silencer when $m = 1$ and $B = 0.0698$.

Table 1
Optimized parameters for the optimized performance of plate silencer.

m	B_{opt}	$r_f = f_2/f_1$
0.5	0.065	3.2
1	0.0698	2.81
1.5	0.085	2.6
2	0.1	2.45
2.5	0.13	2.2
3	0.14	2.05

producing panel for the plate silencer with different mass and different bending stiffness.

3. Material selection

The performance of the plate silencer greatly depends on the material properties. As shown in Table 1, a very light mass and moderately high bending stiffness of the panel is required to contribute higher transmission loss at low frequencies for the plate silencer. However, in reality, it is difficult to find such raw material which owns such kind of properties: light weight and moderately high bending stiffness. One of the solutions is to produce composite material for such requirements. Advanced composite materials have been well developed for aerospace, automotive and ocean engineering applications due to their extra-ordinary mechanical properties. Their light-in-weight with high strength properties, lead them to be ideal materials for designing different kind of structural components. Aforementioned, the design of silencers requires the structure to be light while at the same time to provide certain degree of bending stiffness, advanced composite materials would therefore be the best choice for this application. To optimize the weight and stiffness of the system, various materials and designs were used in this project. There are two approaches that were proposed to construct composite plates, they are (i) the use of two light 3K carbon fibre tows (hereafter called "the composite panel/plate") and (ii) the use of a sheet of thin aluminum foil that were glued on the surface of the foam panels (hereafter called "the aluminum foil covered panel/plate"), respectively. As carbon fibre is light and possesses very high mechanical properties (its tensile strength reaches to 3000 MPa), attaching this fibre with tension into the foam surface would substantially increase the bending strength of a resultant composite panel. The major concern in this study was that, the tensile modulus and hardness of the carbon fibre are much higher than that of the core foam material, which eventually may cause local damage to the foam when the composite panel is subject to bending motion. However, by the theoretical sense, there is no high degree of bending on the panel and this method should be able to achieve the purpose of noise reduction if the composite panel is used inside the duct. For the aluminium foil covered panel, the bending stiffness is basically governed by the tensile properties of the aluminium.

3.1. The composite panels

The first type of material used to construct the panel silencers is carbon fibre tows supported by the foam plate. In this method, carbon fibre tows were inserted and laid into two pre-milled grooves of the foam plate longitudinally. To study the mechanical strength of the panels, two different designs were constructed: one sample involved 2 carbon fibre tows inserted on each side of the foam as shown in Fig. 3(a) (we call it 2-2 composite panel/plate) and another sample involved 4 carbon fibre tows inserted on one side while 4 tows remained on another (we call it 4-4 composite panel/plate) for comparison.

3.2. The aluminum foil covered panels

The aluminum foil that was used as the surface reinforcement in this project was made from 99% commercially pure, 1100 series, H-18 full hard, tempered aluminum with brinell hardness of 44. To study the stiffness enhancement of the panels, three different reinforcement designs were used, they are (i) an aluminum foil covered on the whole surface of the foam uniformly as shown in Fig. 4(a) and it is called "completely covered plate"; (ii) an aluminum foil covered on the area of about 60% of the foam such that two end of the foam possess weaker bending stiffness as shown in Fig. 4(b) and it is called "partially covered panel/plate", and the last one is (iii) two pairs of aluminum foil with the width of about 20 mm covered on the top and bottom edge of the foam as shown in Fig. 4(c) and it is so called "Zebra panel/plate".

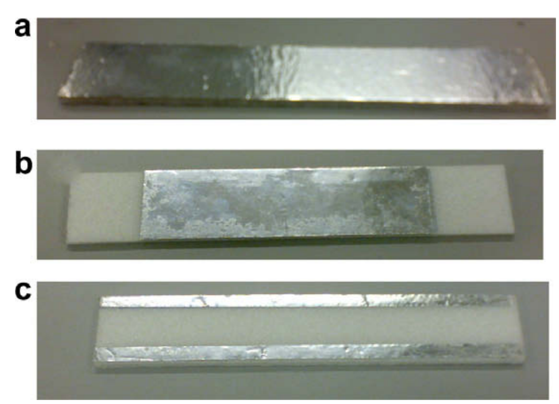

Fig. 4. The aluminum foil covered panels: (a) completely covered; (b) partially covered and (c) Zebra strip arrangement.

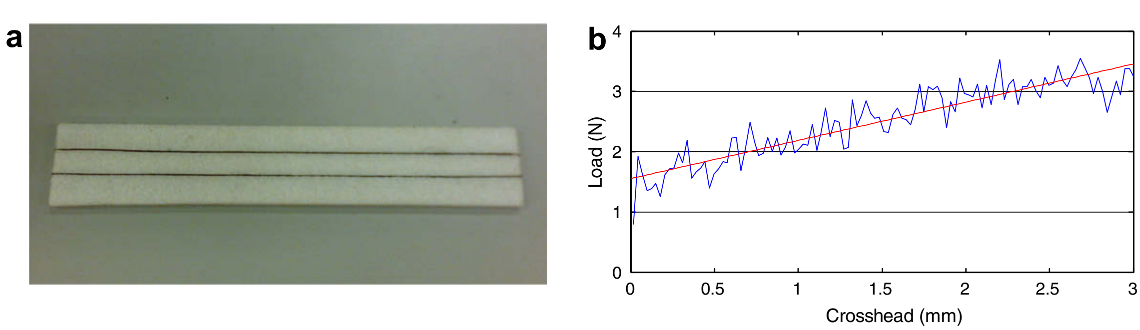

Fig. 3. The composite panel with two carbon fibre tows on the surface (a) is the configuration of the arrangement of the fibre tows on the plate, (b) and the result extracted from the flexural bending test.

Apart from the consideration of the approach of enhancement of bending stiffness, the composite panel should be designed as light as possible. Therefore, panels with different mass by covering the plate with aluminum foil completely, partially in the middle of the panel and partially on the edges are taken to be tested.

Fig. 5 shows the comparison of the bending stiffness of two types of composite panels with the thickness of 4 mm and the original foam panel without any reinforcement. There are two values showing in each bar in Fig. 5. One is the dimensional bending stiffness and another one in the blanket is the mass ratio of the plate. The bending stiffness of the original foam is about 0.0027 Nm^2. The carbon fibre glued on the foam surface can enhance the bending stiffness but the mass is relatively high due to the inherent high mass of resin (Araldite AY 103-1/HY 991). The bending stiffness of the foam with two number of carbon fibre glued is 0.0479 Nm^2 and that with three number of carbon fibre glued is 0.0878 Nm^2. We found that the completely covered panel exhibited a higher bending stiffness although the weight is relatively higher. The bending stiffness is as high as 0.325 Nm^2. The bending stiffness of "Zebra plate" is about 0.311 Nm^2 and the mass ratio of reinforcement to the core material (foam in this project) is about 1.8. All reinforced panels are also taken into the measurement transmission loss. The transmission loss is the indicator to assess the noise reduction of any silencer without the effect of the loading at the termination.

4. Transmission loss measuring of the plate silencer

4.1. Experimental set-up

As shown in Fig. 6, the test rig was built with a square duct with cross-section of 100 mm × 100 mm. The duct wall was made of 15-mm-thick acrylic, which is considered as acoustically rigid. The

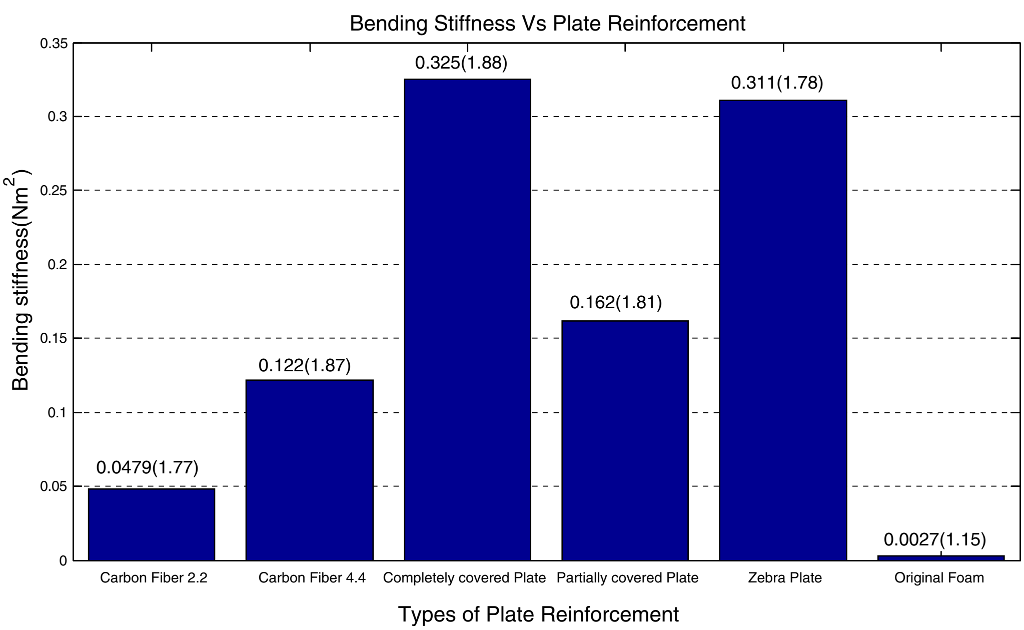

Fig. 5. Bending stiffness of six types of panel.

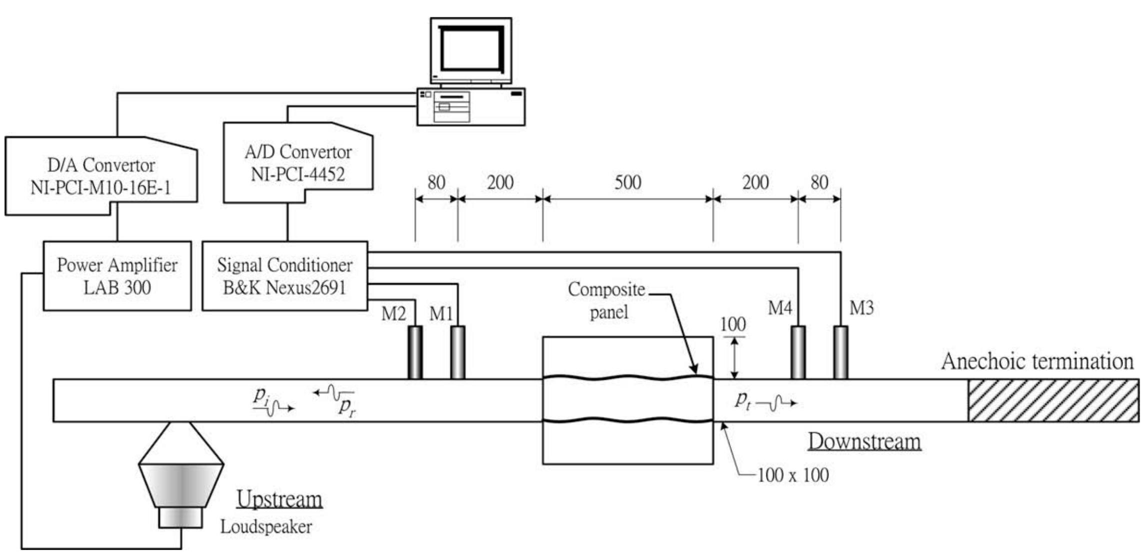

Fig. 6. Experimental set-up for transmission loss measurement.

first cut on frequency in the rigid duct is 1700 Hz. The composite panels were installed flush with the duct wall and they are covered with the cavities with the length of 500 mm and depth of 100 mm.

The panel size is about 510 mm × 102 mm × 4 mm. The leading and trailing edges of the panels were clamped and as a result, the effective length of the panel is 500 mm. The transmission loss is measured by the four-microphone, two-load method that is similar to the approach described by Munjal and Doige [12]. The four-microphone method attains simultaneous measurements of the two standing wave patterns, both upstream and downstream. By changing the impedance at the end of downstream, it is possible to attain a different standing wave pattern. The magnitudes of the incident p_i and transmitted waves p_t in perfect anechoic condition can be measured and used to calculate the transmission loss (*TL*). As previous measurements found that the aluminium foil covered panels gave better compromised solution, in terms of stiffness and also weight concern, the noise test was only focused on the aluminum panels to study the noise reduction effect but the panel with 2 carbon fibre tows on each side will also be test for the comparison.

4.2. Results

The purpose of using panel with high bending stiffness is to promote the first and second mode of vibration and consequently the noise radiation from the panel or the noise reflection to the upstream is very effective. The noise reduction or transmission loss is controlled by the capability of sound reflection by the panel. Therefore, we are looking at the reflection coefficient β as well as absorption coefficient α and they are defined in Eqs. (2) and (3), respectively.

The purpose of this silencer is to increase of the transmission loss by minimizing the absorption but maximizing the reflection. Apart from the comparison between the new composite plate and the original foam plate, how to enhance the performance of such kind of panel silencer at low frequency range is also investigated in the current paper. Fig. 7 depicts the comparison of transmission loss among 2-2 carbon fibre composite plate (solid line with circle), 4-4 carbon fibre plate (solid line with triangle) and original foam plate (dashed dotted line) without any reinforcement material. The transmission loss spectrum in Fig. 7(a) shows that the performance of 2-2 carbon fibre composite plate is slightly better than the original foam at very low frequencies. The frequency range for *TL* greater than 10 dB is 180–195 Hz. It is too narrow which is not desirable enough for realistic applications. It is attributed to the low bending stiffness. The 4-4 carbon fibre composite panel improves the performance of the panel silencer. The transmission loss is marginally greater than 10 dB criterion value when the frequency range is between 130 and 260 Hz. As a result, the stopband is $r_f = 260/130 = 2$. This is attributed to the reflection which is proved in Fig. 7(b). Therefore, the carbon fibre reinforcement is one of the possible ways to increase the bending stiffness of the foam plate and the mass can be very light. To certain extent, the mass of the whole composite plate by using carbon fibre is higher than the expected value due to the increased mass of the resin during the manufacturing process.

Fig. 8(a) depicts the transmission loss of the completely covered plate by using aluminum foil with clamped support at two ends of the plates and original foam without any reinforcement while Figs. 8(b) and (c) shows the reflection coefficient and absorption coefficient, respectively. One more *TL* spectrum to describe the performance of single expansion chamber with expansion ratio of 3 is observed as dashed line in Fig. 8(a). Briefly, expansion chamber means that when the two panels are removed, there is an area expansion from the duct height of h–$3h$. Such kind of design causes tremendous pressure drop and increase the power consumption of engine. Many researchers adopted the idea of area expansion which is integrated with the concept of the resonator to design different kind of flow-through silencer or muffler for the applications in vehicle.

The completely covered panel can render the highest bending stiffness but the mass is relatively high according to the result in Fig. 5. As shown in Fig. 8(a), for the frequency range 120–270 Hz which is the low frequency range, the transmission loss of the com-

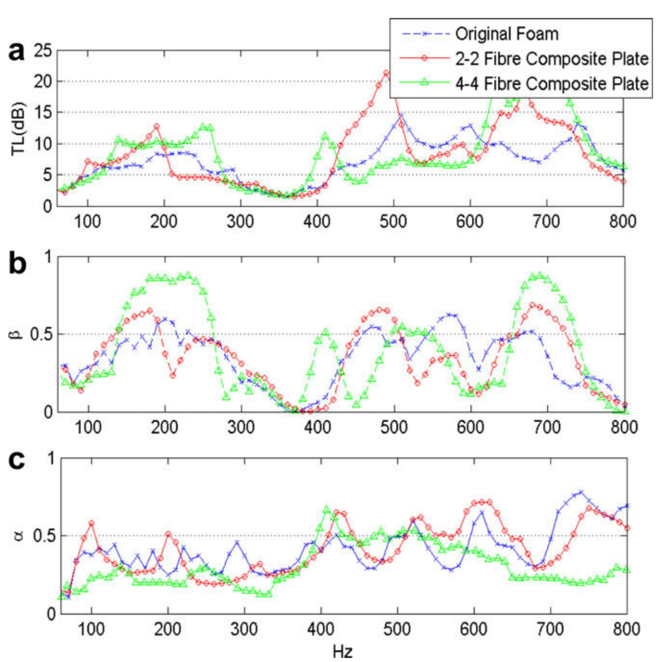

Fig. 7. Comparison between original foam, 2-2 carbon fibre and 4-4 carbon fibre composite plate (a) is the transmission loss, (b) is the reflection coefficient and (c) is the absorption coefficient.

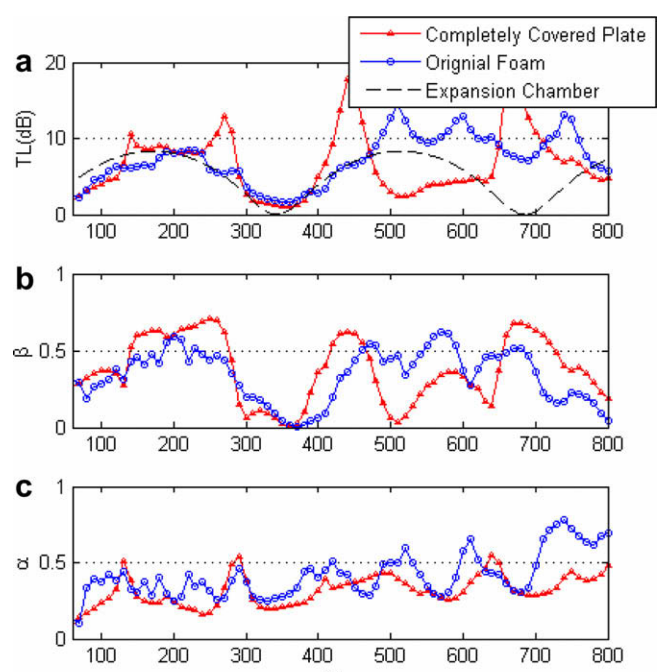

Fig. 8. Performance of the completely covered plate: (a) is the transmission loss, (b) is the reflection coefficient and (c) is the absorption coefficient.

pletely covered panel is higher than the original foam by about 2 dB in average. The increment of noise reduction by using completely covered panel at low frequency range is not very obvious. The transmission loss given by the completely covered panel is greater than 10 dB criterion value when the frequency range is between 270 and 290 Hz. As a result, the stopband $r_f = 290/270 = 1.07$. In this frequency range, the reflection coefficient is in the range of 0.6–0.7 as shown in Fig. 8(b). Below 200 Hz, the transmission loss is lower than 10 dB. Although the transmission loss is lower than 10 dB, reflection coefficient in the frequency range from 130 to 200 Hz is still about 0.8 while Fig. 8(c) shows that the absorption coefficient about 0.2 in average. This means that the noise reduction is mainly due to the reflection of sound instead of absorption. It is proved that the transmission loss cannot achieve 10 dB criterion level because the plate may be too heavy to vibrate and the response of the panel is not too high to reflect the sound. The transmission loss of the original foam is similar to the expansion chamber. The bending stiffness of the original foam is very small and it vibrates at higher order mode. Consequently, the original foam can be regarded as the transparent for the sound penetrating from the duct to the cavity and this is the reason why the performance of original foam is the same as that of expansion chamber.

Fig. 9 shows the comparison of the performance between the partially covered plate and the original foam. As shown in Fig. 9(a), the transmission loss of the partially covered panel is greater than that of the original foam in the frequency range of 100–210 Hz. The improvement is not very great. By comparing the criterion level, the transmission loss of partially covered panel greater than 10 dB is in the frequency range from 150 to 210 Hz and $r_f = 1.4$. This means that the bandwidth to achieve the criterion level is wider. As shown in Fig. 9(b), the corresponding reflection coefficient is 0.6 in average. Although the absorption coefficient is about 0.4, the absorption capability of such thin plate is not so big to reduce the noise at low frequencies. Therefore, the noise reduction by such kind of plate is totally attributed to the sound reflection. Since two end of the plate possess relatively low bending stiffness but the middle region of the plate have very high bending stiffness, this configuration is similar to the plate with simply supported instead of clamped supported. This is beneficial and enhances the vibration of plate at lower frequency and as a result of this, the sound radiation and reflection to the upstream is greatly increased at the low frequency.

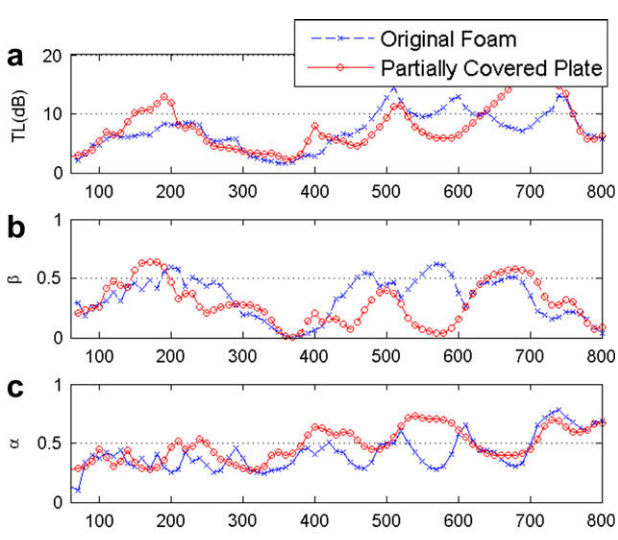

Fig. 9. Performance of the partially covered panel: (a) is the transmission loss, (b) is the reflection coefficient and (c) is the absorption coefficient.

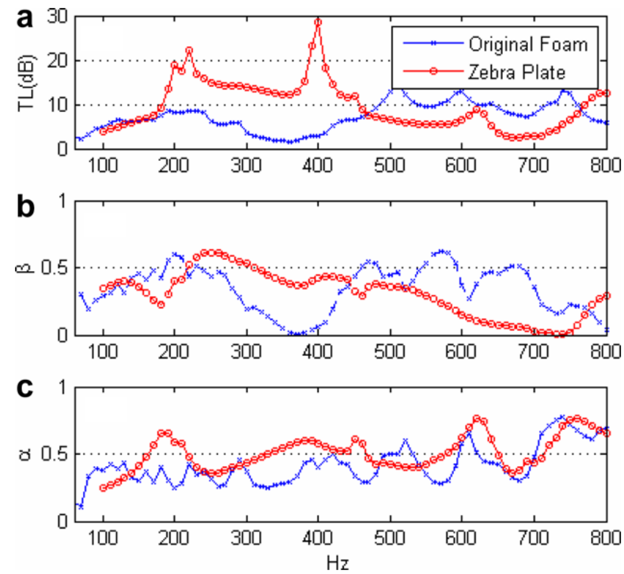

Fig. 10. Aluminum reinforcement – Zebra plate (a) is the transmission loss, (b) is the reflection coefficient and (c) is the absorption coefficient.

Fig. 10 shows the comparison of the performance between the Zebra arrangement panel and the original foam. The Zebra panel is lighter because of less aluminum foil and resin used. It is easily to be excited and vibrate and consequently the transmission loss of Zebra panel is greater than original foam by about 10 dB in the frequency range of 180–450 Hz as shown in Fig. 10(a). There is great improvement in the noise reduction at low frequency range by using Zebra arrangement panel. For the design of silencer, the transmission loss of the Zebra arrangement panel greater than the criterion value 10 dB is in the frequency range from 170 to 450 Hz and $r_f = 450/170 = 2.65$. Although the lower frequency limit f_1 is shifted to higher frequency, the bandwidth is greatly enhanced. This is due to the light panel and easily to be excited. This shows that the use of the configuration of Zebra arrangement panel is very promising.

Fig. 10(b) shows that the reflection coefficient attributed by Zebra plate is about 0.6 in the frequency range of 240–440 Hz while Fig. 10(c) shows that the absorption coefficient is about 0.4 in average in the same frequency range. By comparing the reflection coefficient in Fig. 10(b), the difference between the original foam and Zebra plate is about 0.4 in frequency range of 240–440 Hz and this is roughly about 40% increment of sound reflection attributed to the Zebra plate. On the other hand, the increment of sound absorption attributed to the Zebra plate is only about 15% as shown in Fig. 10(c). It is proved that the increase of transmission loss is mainly due to the sound reflection enhancement. Generally speaking, the performance of the clamped plate silencer by using Zebra plate is good because such kind of attachment of aluminum foil on the foam ultimately renders the light mass but high bending stiffness which is very crucial to the first and second mode of vibration of the panel and noise reflection.

5. 2-Plate silencer and 4-plate silencer in two different geometrical cavities

If the design of the plate silencer by using two identical plates covered with two cavities can contribute attractive noise abatement at low frequencies, it is natural to ask the question of whether four plates are beneficial.

When two more modules of plate covering with the cavity are added, theoretically speaking, the noise reduction can be enhanced because of the increase of the total sound radiation from the plates. As shown in Fig. 11(b), such configuration means that four plates covering the square duct are backed with four rectangular chambers of cross-section area. In order to keep the same total cavity volume of 10 which is equivalent to that for two-cavity plate silencer, the side height of the outer rectangular chamber from the middle point of the duct is h. The theoretical result from Wang et al. [10] shows that the stopband $r_f = 6.2$ is found when the plate with mass ratio $m = 1$ owns the optimal bending stiffness of 0.215 in the 4-plate silencer. This design of silencer with four rectangular chambers appears to be too bulky. In order to give more compact design, four rectangular chambers are replaced by one circular chamber as shown in Fig. 11(c). The cavity volume is distributed uniformly around the four sides of the rectangular duct, this size can be reduced and appropriate for the installation. The radius is found to be about 0.95h by keeping the same total cavity volume for the comparison. As shown in Fig. 11(d), the radius of the chamber of the prototype is 90 mm. There are four sets of metal grids and frame, which are made of iron, for clamping the composite plates. The cylindrical chamber is made of 8-mm-thick acrylic. The optimal bending stiffness of each panel required for the 4-plate silencer is higher than that for 2-plate silencer for the same mass according to the theoretical prediction [10]. Therefore, the type of composite panels employed in the 4-plate silencers is the completely covered panel.

Fig. 12 shows the comparison of the experimental result between the 2-panel silencer as shown in Fig. 11(a) and 4-panel plate silencer as shown in Fig. 11(d) with the same dimensions and the same physical parameters of the panels. As shown in Fig. 12(a), the transmission loss of 4-panel plate silencer is greatly higher than that of 2-panel plate silencer. Roughly speaking, the magnitude of transmission loss of 4-panel silencer, which is about 18 dB in

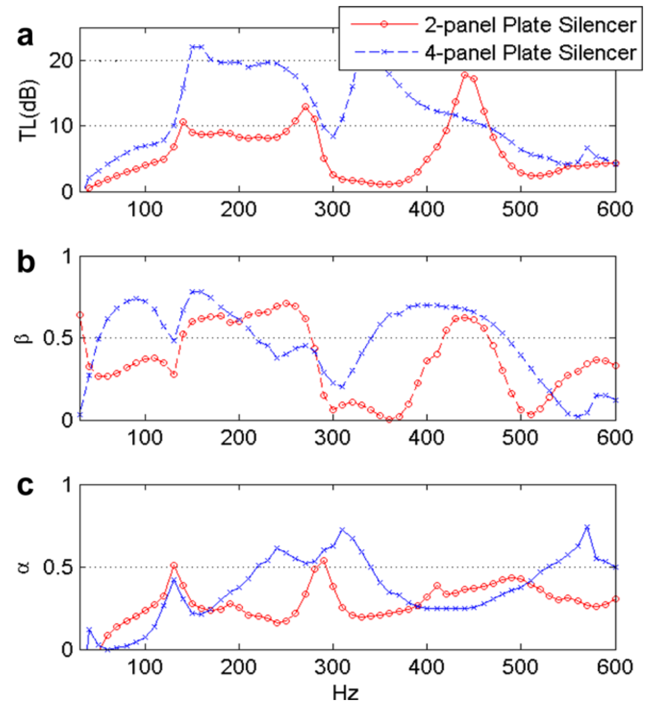

Fig. 12. Comparison between 2- and 4-panel plate silencer by using completely covered panel: (a) is the transmission loss, (b) is the reflection coefficient and (c) is the absorption coefficient.

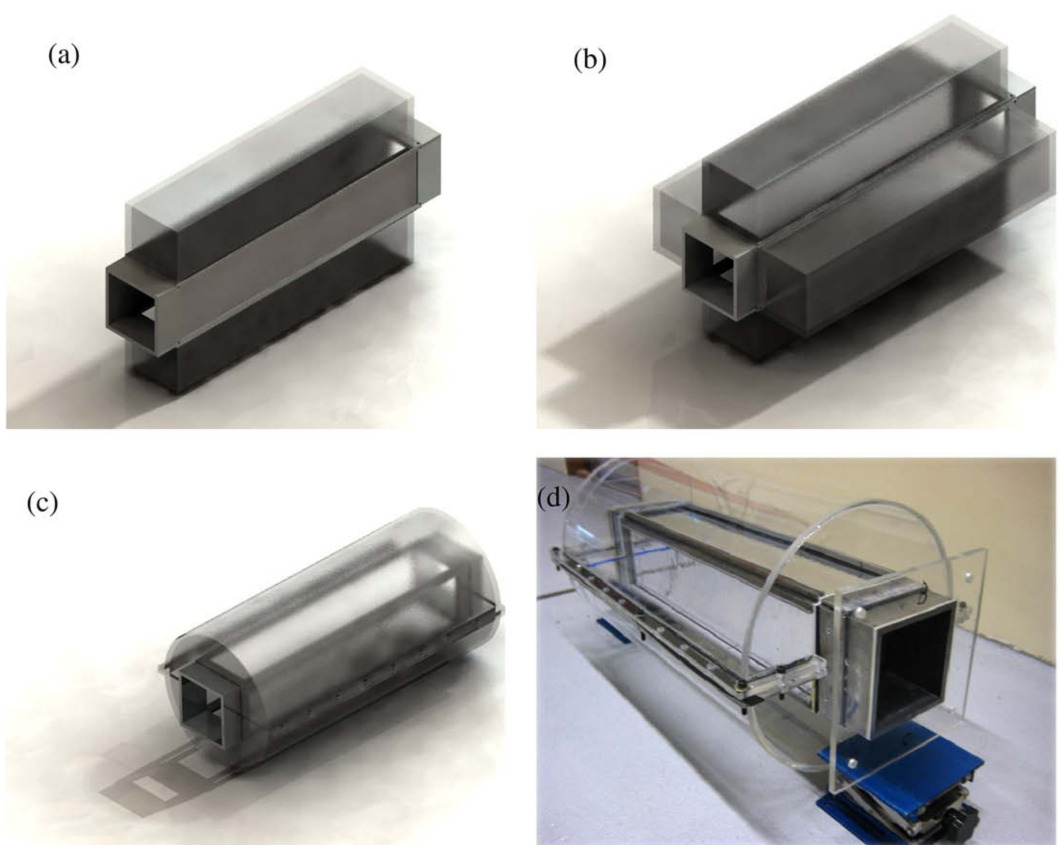

Fig. 11. (a) The model of 2-plate silencer, (b) is the model of 4-plate silencer with four rectangular chambers and (c) is the model of 4-plate silencer with cylindrical chamber and (d) is the prototype of 4-plate silencer.

the frequency range of 140–260 Hz, is about double of that of 2-panel silencer. Although there is a trough point dropping to about 9.1 dB at 300 Hz, the frequency range for the *TL* over 9 dB is about 125–450 Hz which is very wide stopband. As shown in Fig. 12(b), the reflection coefficient is about 0.8 in average. This value is even higher than the result of 2-panel silencers. This can be explained by the increase of the total sound radiation from the panels or sound reflection as the number of panels is increased. Therefore, it is expected that more attractive results can be obtained if 6 panels are used.

6. Conclusion

The general conclusion of this study is that the theoretical prediction of various reinforced panels used for duct noise reduction at low frequency is validated by experimental data. The following are specific conclusions derived from various tests conducted.

1. Both material reinforcement methods by using carbon fibre and aluminum foil provide higher bending stiffness for the panels; thus enhancing the noise reduction by exciting the first and second order mode of vibration. In general, aluminum foil reinforced panels results in higher transmission loss than that the carbon fibre reinforced panels at low frequencies. This may be due to the higher stiffness to mass ratio of the panel.
2. Zebra plate is very promising in the noise reduction at low frequencies. Less resin and aluminum foil are used to construct the Zebra plate so it is relatively light and at the same time it can also render reasonably high bending stiffness. First, the light panel can be easily to be excited and vibrate. Second, the plate with high bending stiffness can easily to be in motion at the *in vacuo* first and second mode that are very crucial for sound radiation and sound reflection.
3. To enhance the transmission loss, the light panel with high bending stiffness is very important. Therefore, resin with lighter mass can reduce the total mass of the panel while retaining high bending stiffness. Therefore, the approach of controlling the resin to construct the composite the panel is very important.
4. The panel silencer by using 4 panels can greatly enhance the transmission loss and widen the stopband. The cylindrical 4-plate silencer can contribute about double transmission loss of 2-plate silencer because of the increase of sound radiations from the four panels. The design of circular chamber can be more compact and it is very promising in realistic applications.

Acknowledgements

This project was supported by the Hong Kong Polytechnic University Grant A-PA7Y. The authors also thank Evonik Degussa Hong Kong Ltd. for providing the foam plates for our testing.

References

[1] Huang L. Broadband sound reflection by plates covering side-branch cavities in a duct. J Acoust Soc Am 2006;119:2628–38.
[2] Choy YS, Huang L. Experimental studies of drum-like silencer. J Acoust Soc Am 2002;112:2026–35.
[3] Kiyama M, Sakagami K, Tanigawa MA. A basic study on acoustic properties of double-leaf membranes. Appl Acoust 1998;54:239–54.
[4] Huang L. A theoretical study of duct noise control by flexible panels. J Acoust Soc Am 1999;106:1801.
[5] Wang C, Han J, Huang L. Optimization of a clamped plate silencer. J Acoust Soc Am 2007;121:949–60.
[6] Huang L. Modal analysis of a drumlike silencer. J Acoust Soc Am 2002;112:2014–20.
[7] Renji K, Narayan SS. Letter to editor: loss factors of composite honeycomb sandwich panels. J Sound Vib 2002;250:745–61.
[8] Saito T, Parbery RD, Okuno S, Kawano S. Parameter identification for aluminum honeycomb sandwich panels based on orthotropic timoshenko beam theory. J Sound Vib 1997;208:271–87.
[9] Wang C, Huang L. Analysis of absorption and reflection mechanisms in a three-dimensional plate. J Sound Vib 2008;313:510–24.
[10] Wang C, Cheng L, Huang L. Realization of a broadband low-frequency plate silencer using sandwich plates. J Sound Vib 2008;318:792–808.
[11] Choy YS, Huang L. Effect of flow on the drumlike silencer. J Acoust Soc Am 2005;118:3077–85.
[12] Munjal ML, Doige AG. Theory of a two source-location method for direct experimental evaluation of the four-pole parameters of an aeroacoustic element. J Sound Vib 1990;141:323–33.