Experimental investigation of damping flexural vibrations in glass fibre composite plates containing one- and two-dimensional acoustic black holes

PLEASE CITE THE PUBLISHED VERSION

http://dx.doi.org/10.1016/j.compstruct.2013.08.011

PUBLISHER

Elsevier © the authors

VERSION

VoR (Version of Record)

Composite Structures 107 (2014) 406–415

Contents lists available at ScienceDirect

Composite Structures

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

Experimental investigation of damping flexural vibrations in glass fibre composite plates containing one- and two-dimensional acoustic black holes

E.P. Bowyer, V.V. Krylov *

Department of Aeronautical and Automotive Engineering, Loughborough University, Loughborough, Leicestershire LE11 3TU, UK

ARTICLE INFO

Article history:
Available online 24 August 2013

Keywords:
Vibration damping
Composite plates
Composite panels
Acoustic black hole effect
Wedges of power-law profile
Circular indentations

ABSTRACT

In this paper, the results of the experimental investigation into the addition of indentations of power-law profile into composite plates and panels and their subsequent inclusion into composite honeycomb sandwich panels are reported. The composite plates in question are sheets of composite with visible indentations of power-law profile. A panel is a sheet of composite with the indentations encased within the sample. This makes a panel similar in surface texture to an un-machined composite sheet (reference plate) or conventional honeycomb sandwich panel. In the case of quadratic or higher-order profiles, the above-mentioned indentations act as two-dimensional acoustic black holes for flexural waves that can absorb a large proportion of the incident wave energy. For all the composite samples tested in this investigation, the addition of two-dimensional acoustic black holes resulted in further increase in damping of resonant vibrations, in addition to the already substantial inherent damping due to large values of the loss factor for composites. Due to large values of the loss factor for composite materials used, there was no need in adding small pieces of absorbing layers to the indentations to achieve desirable levels of damping.

1. Introduction

Composite materials and structures are found in an increasing variety of applications in aeronautical, automotive, and marine industries, where they replace many parts and components traditionally made of metals and metallic alloys. The main advantages of composite structures over their metallic counterparts are their higher stiffness-to-weight ratio and better resistance to corrosion. Another useful feature of composite materials is higher values of their intrinsic energy loss factors, which results in lower levels of undesirable structural vibrations under the same operational conditions. Nevertheless, structural vibrations in composite materials and structures do occur and are a key factor in delamination and crack propagation, therefore their reduction remains an important engineering problem that continues to be investigated (see e.g. [1–3]).

Usually, passive damping of structural vibrations is achieved by adding layers of highly absorbing materials to the structure in order to increase elastic energy dissipation of propagating (mostly flexural) waves [4–6]. The main disadvantage of this approach is the necessity to attach rather thick layers of absorbing materials to structural surfaces in order to achieve desirable levels of damping.

An alternative approach, which is applicable to damping of resonant structural vibrations, is based on reduction of reflections of structural waves from free edges of structures, rather than on increasing energy losses in the process of wave propagation (see e.g. [4]). To implement this approach in a more efficient way, a new method of damping flexural vibrations based on the so-called 'acoustic black hole effect' has been recently developed and investigated [7–9]. This method has been initially applied to one-dimensional metal plates of power-law profile (wedges) that had to be covered by narrow strips of absorbing layers near sharp edges [9]. Ideally, if the power-law exponent is equal or larger than two, the flexural wave never reaches the sharp edge and therefore never reflects back [7–10]. Thus, such a sharp edge of power-law profile materialises the 'acoustic black hole' (ABH) for flexural waves, with no reflection and with total energy absorption in the attached piece of absorbing layer. It has been established theoretically [7,8] and confirmed experimentally [9] that this method of damping structural vibrations is much more efficient than traditional damping methods.

In addition to the above-mentioned wedges of power-law profile, which materialise one-dimensional (1D) acoustic black holes, a serious attention has been paid also to circular indentations of

* Corresponding author. Tel.: +44 1509 227216.
E-mail address: V.V.Krylov@lboro.ac.uk (V.V. Krylov).

0263-8223/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.compstruct.2013.08.011

power-law profile made inside plate-like structures [11–16]. Such circular indentations materialise two-dimensional (2D) acoustic black holes. Like 1D black holes, such 2D black holes have also proven to be efficient means of damping structural vibrations. The advantage of two-dimensional acoustic black holes over their one-dimensional equivalents is that such holes, as well as their combinations (arrays), can be situated inside plate-like structures, which eliminates sharp edges from constructions and thus makes them safer, while reducing the risk of damage to the indentation tips.

The main practical advantage of the method of vibration damping using the acoustic black hole effect is that it requires very small amounts of absorbing materials to be attaches at the sharp edges, which is especially important for light-weight structures. In the same time, the acoustic black hole effect is robust enough to remain effective even for structures with geometrical and material imperfections resulting from manufacturing [17].

The aim of the present paper is to investigate vibration damping in glass fibre composite plates and panels using 1D and 2D acoustic black holes. Note in this connection that a composite panel with smooth outer edges is one of the most commonly found composite structures. The construction of this type of a composite panel that can incorporate the damping abilities of an acoustic black hole forms the ultimate aim of this paper. Five types of composite structures containing acoustic black holes and their effect on vibration damping have been investigated. These structures are: a composite strip with a wedge of power-law profile (1D acoustic black hole), a composite plate with a circular indentation of power-law profile (2D acoustic black hole), combinations of composite plates containing circular indentations of power-law profile, smooth surface composite panel configurations with enclosed circular indentations of power-law profile, and a honeycomb structure with added plates containing circular indentations of power-law profile.

2. Manufacturing of experimental samples

Fifteen glass fibre composite samples were manufactured for this investigation; three strips of dimensions 250 × 50 mm and a thickness of 6 mm; the additional wedge being 50 mm long and of power-law profile with $m = 2.2$. A wedge of power-law profile with $m = 2.2$ was also produced in order to be attached to a steel strip, dimensions being the same as for the composite strip. The eleven glass fibre composite plates were of dimensions 310 × 185 mm and consisted of two 3 mm thick plates and nine 6 mm thick plates. The circular indentations of power-law profile with $m = 4$ had a diameter of 110 mm with a central hole of 10 mm, leaving a profile length of 50 mm.

The glass fibre composite used for these samples was SE84LV-Low Temperature Cure Epoxy Prepreg System. This composite has a high compressive strength and it is widely used in large heavily loaded components, such as yacht hulls, spars, aviation panels, and also in non-structural applications. SE 84LV is also widely used in sandwich structures with honeycombs. Each sheet had a thickness of 0.2 mm. The composite plates and strips were layed-up to the required thickness and then cured using the vacuum bagging process.

Fourteen of the profiles were created in the traditional way: A CNC (Computer Numerically Controlled) milling machine operating at a cutter speed of 1200 rpm with a carbide cutter was used to produce the wedges and circular indentations. The main problem encountered when utilizing this method of manufacturing was that it is not possible to construct directly a panel with internal cavities while utilizing a 'vacuum only processing' method. The outer layer needs to be cured separately and attached using epoxy resin. This can lead to a more lengthy manufacturing time.

These fourteen samples consisted of a reference strip and strip with an additional wedge (Fig. 1(a)); examples of the other three types of plates can be seen in Fig. 1(b–d). Fig. 2 displays the cross-sectional view of the plate samples when viewed from the narrow end. The average profile tip thickness for each of the samples is given in Table 1.

Two types of glass fibre composite honeycomb sandwich panels were created for this investigation, see Fig. 3: a reference composite honeycomb sandwich panel and a composite honeycomb sandwich panel containing two acoustic black holes in each of the composite plates.

There are several methods that could have been employed in the manufacture of these honeycomb sandwich structures: For this investigation, a method that was specific to composites and that would allow the integration of acoustic black holes into the structure was required. The standard method has four stages, the first being the curing of the outer composite plates. This was done using the same vacuum curing technique described above. The composite plates, aluminium honeycomb and the adhesive sheets are then cut to size. Each composite plate is then cured in turn onto the aluminium honeycomb, Fig. 4(a).

In addition to the method used above, two further stages were added for the construction of the sandwich panel incorporating the acoustic black holes (ABHs). The first being the manufacture of the ABHs, a CNC (Computer Numerically Controlled) milling machine operating at a cutter speed of 1200 rpm with a carbide cutter

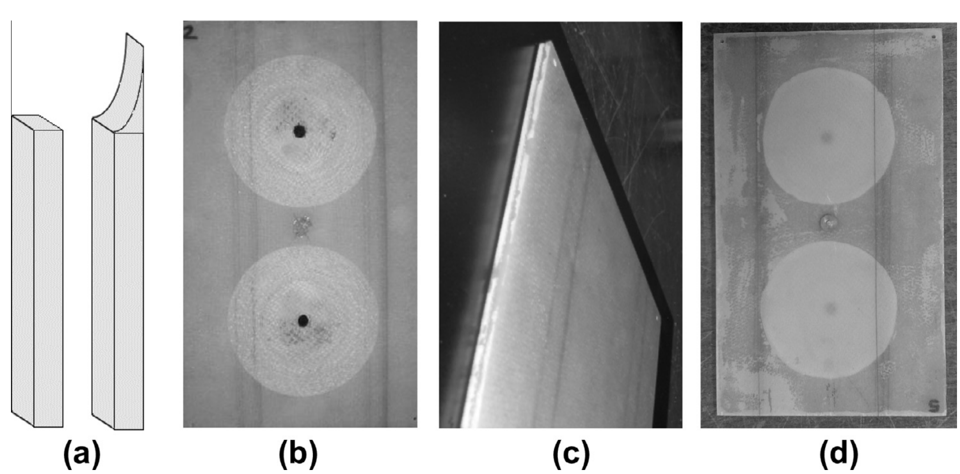


Fig. 1. (a) Strip with and without a wedge, (b) plate (6 mm) containing two 2D acoustic black holes, (c) combined composite plate (6 mm), and (d) composite panel (6 mm).

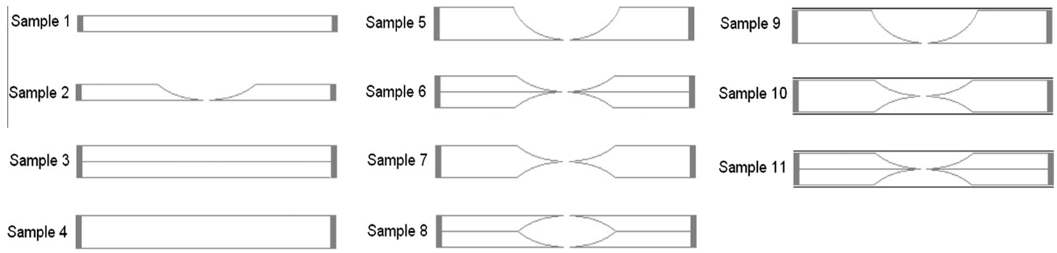


Fig. 2. Cross-section views of sample plates 1–11.

Table 1
Average profile tip thickness for each sample (in mm).

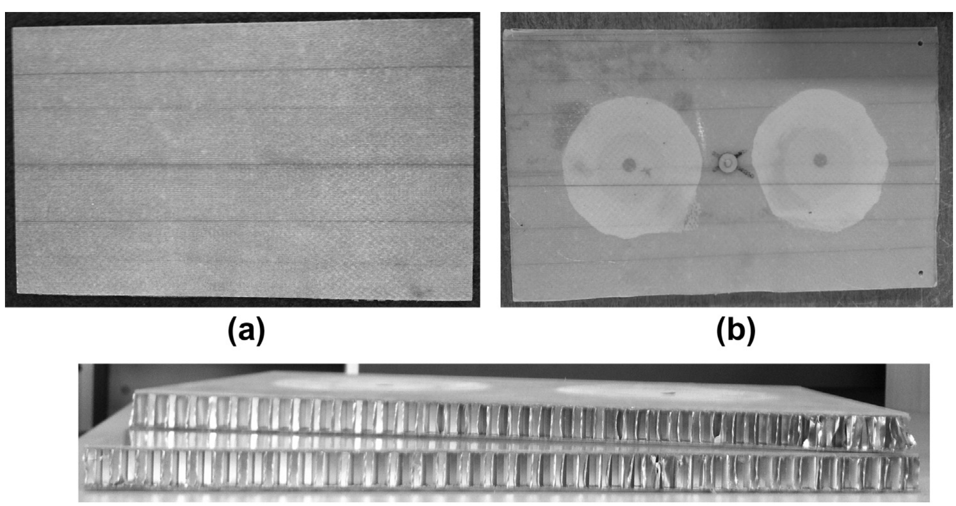
Sample	5	6	7	8	9	10	11
Tip thickness (mm)	0.10	0.11	0.10	0.10	0.10	0.10	0.11

was used to produce the circular indentations in the composite. This stage was introduced prior to the bonding of the composite sheets to the honeycomb, resulting in a honeycomb sandwich panel with exposed indentations; Fig. 4(b). The second being the fixing of the single sheet of composite to the outer surfaces of the panel to enclose the ABHs, leaving a smooth surfaced panel. This stage was added to end of the manufacturing procedure. A cross-section view displaying the composition of the sandwich panel as seen from the narrow end of the samples can be seen in Fig. 5.

The dimensions of the final samples were 310 × 185 mm and a thickness of 20 mm; the circular indentations of power-law profile with $m = 4$ had a diameter of 110 mm with a central hole of 10 mm, leaving a profile length of 50 mm. The composite plates had a thickness of 1.2 mm each. The 5052 aluminium honeycomb centre had a thickness of 12.7 mm, a cell size of 3 1/16 in, a foil thickness of 15 μm and a mass of 70 kg per m^2. A single profile was chosen for this investigation rather than a double profile. There was only a small difference in the damping performance of the two profiles, so the above profile was selected to ensure accurate results.

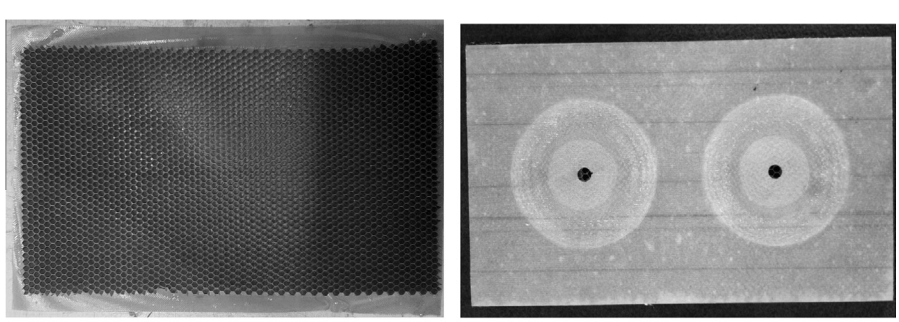
3. Experimental set up

The experimental set-up has been designed to allow nearly free vibration of the sample plates (i.e. to eliminate clamping of edges),



(a) (b) (c)

Fig. 3. (a) Composite honeycomb sandwich panel – reference plate, (b) composite honeycomb sandwich panel with two acoustic black holes, and (c) side view of the panels.



(a) (b)

Fig. 4. (a) Composite plate cured onto the aluminium honeycomb and (b) honeycomb sandwich panel with two exposed indentations.

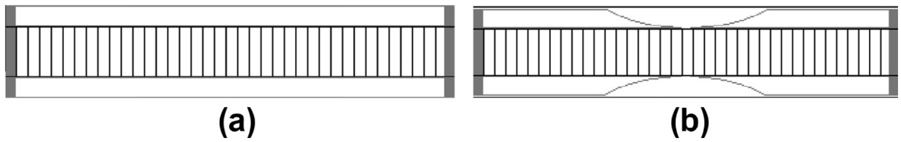


Fig. 5. Cross-section view of (a) composite honeycomb sandwich panel – reference plate and (b) composite honeycomb sandwich panel with acoustic black holes (not to scale).

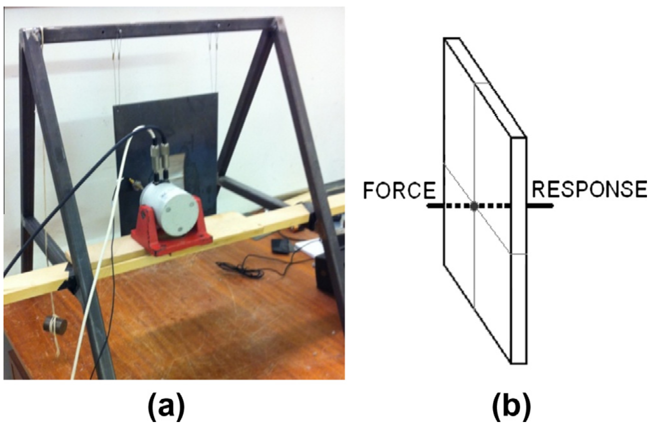


Fig. 6. (a) Experimental set up and (b) locations of the shaker (force) and of the accelerometer (response) on an experimental sample.

take the weight off the plate edges and introduce minimal damping to the system, see Fig. 6(a). The excitation force was applied centrally on the plate via an electromagnetic shaker attached to the plate using glue and fed via a broadband signal amplifier. The response was recorded by an accelerometer (B&K Type 4371) that was attached to the one surface, directly in line with the force transducer (B&K Type 8200), also attached using glue, Fig. 6(b). The acquisition of the point accelerance was utilised using a Bruel & Kjaer 2035 analyser and amplifier. A frequency range of 0–6 kHz was investigated; above this range no discernable response could be detected.

4. Results and discussion

4.1. Introduction of a wedge of power-law profile to a composite strip

In the first instance it seemed prudent to first ascertain whether the introduction of a wedge of power-law profile to a composite strip could produce the acoustic black hole effect as seen in previously tested steel samples [9,13,17]. In this section, two types of sample were tested: a reference strip and a strip with a machined wedge of power-law profile (1D acoustic black hole). It was found during initial testing that a composite sample, unlike the steel samples, required no additional damping layer to be attached to the wedge tip to produce the acoustic black hole effect. This is primarily due to the increased loss factor of the composite material itself (~0.1 to 0.2). For this reason, all of the following results show the samples without any additional damping layers.

A comparison of the frequency responses for a strip with and without a power-law profiled wedge is shown in Fig. 7. There is no difference between the two samples below 250 Hz. After this point, an increase in the reduction of the resonant peaks is seen up until a maximum reduction of 3.5 dB from the reference sample, which is seen at 2.2 kHz. At 2.6 kHz the two peaks match and no reductions are seen. Beyond this point the sample with the wedge has damped all remaining peaks.

4.2. Tapered circular indentation of power-law profile in glass fibre composites

The next step, as with the previously tested steel samples [13], was to introduce circular indentations of power-law profile (2D ABHs) into glass fibre composite plates. This section describes the effect of the addition of two 2D ABH's into both a 2.5 mm (Sample 2) and 5 mm thick plate (Sample 5) when compared to a respective thickness reference plate (Samples 1 and 4). This section also looks at the effect of a double profiled indentation of the same power-law profile (Sample 7) compared to the single profiled 5 mm thick sample (Sample 5) and the reference plate. The cross-sections of these profiles can be seen in Fig. 8.

Fig. 9 shows the results for the 2.5 mm samples: Sample 2 when compared to a plain reference plate, Sample 1. One can see that the effect of adding two circular indentations of power-law profile is immediately obvious, with considerable damping of resonant

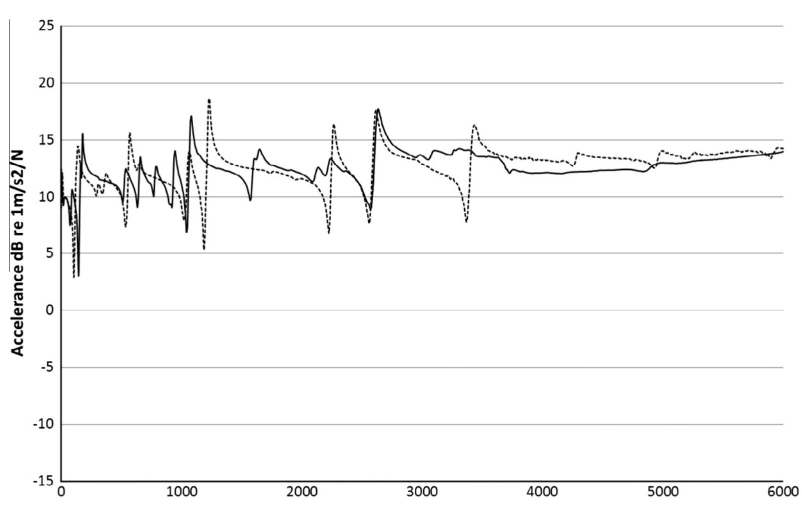


Fig. 7. Measured accelerance for a strip with a wedge of power-law profile (solid line) compared to the case of a reference strip (dashed line).

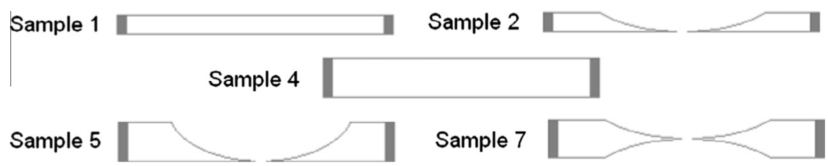


Fig. 8. 2.5 mm reference plate (Sample 1), 5 mm reference plate (Sample 4), cross-sections of samples containing tapered circular indentations (Samples 2, 5 and 7).

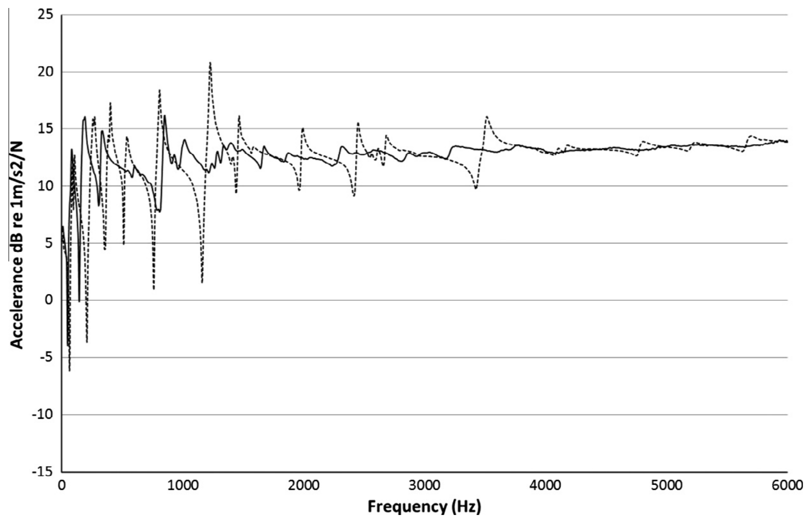


Fig. 9. Measured accelerance for Sample 1 – 2.5 mm reference plate (dashed line) compared to that for Sample 2 – 2.5 mm plate with two 2D acoustic black holes (solid line).

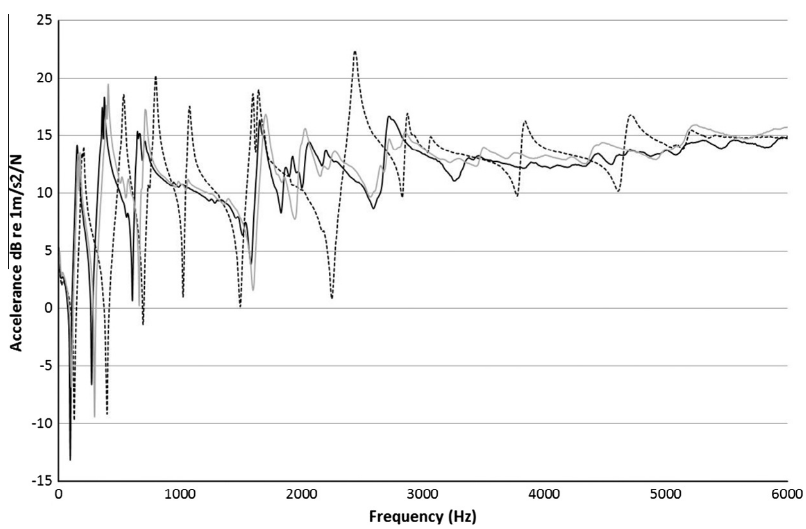


Fig. 10. Measured accelerance for Sample 7 (solid black line) compared to those for Sample 5 – 5 mm plate with two 2D acoustic black holes (solid grey line) and Sample 4 – 5 mm reference plate (dashed line).

peaks easily observed. Below 500 Hz little to no damping is seen. An increase in the reduction of the peak responses of the reference plate is seen in the profiled sample until a maximum reduction from the reference plate of 7.5 dB can be observed at 1.2 kHz. After 2.7 kHz the response is smoothed, with all resonant peaks seen in the reference sample heavily damped if not completely removed.

The same test as above was repeated for 5 mm thick plate with the addition of a third sample for comparison, Sample 7, a double profiled sample. From Fig. 10 it can be seen that, when the double profiled sample (Sample 7) is compared to a plain reference plate (Sample 4) and a single profile plate (Sample 5), the damping values achieved for Samples 5 and 7 are very similar. Sample 7 performs slightly (1–1.5 dB) more effectively than Sample 5 below 1 kHz. Above this frequency this trend is reversed, with Sample 5 behaving slightly (again 1–1.5 dB) more effectively than Sample 7. As expected, below 450 Hz there is little to no damping. A maximum reduction from the reference plate by both Samples 5 and 7 of 8.5 dB can be seen at 2.4 kHz. After 2.7 kHz both Samples 5 and 7 show a smoothing of the reference plate resonant peaks. Samples 5 and 7 can therefore be classified as interchangeable. Whether a sample has a double profile, as with Sample 7, or a single profile, as with Sample 5, the damping performance obtained is the same.

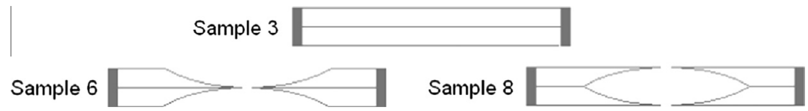


Fig. 11. Cross-sections of combined composite samples, Sample 3 – reference plate, Samples 6 and 8.

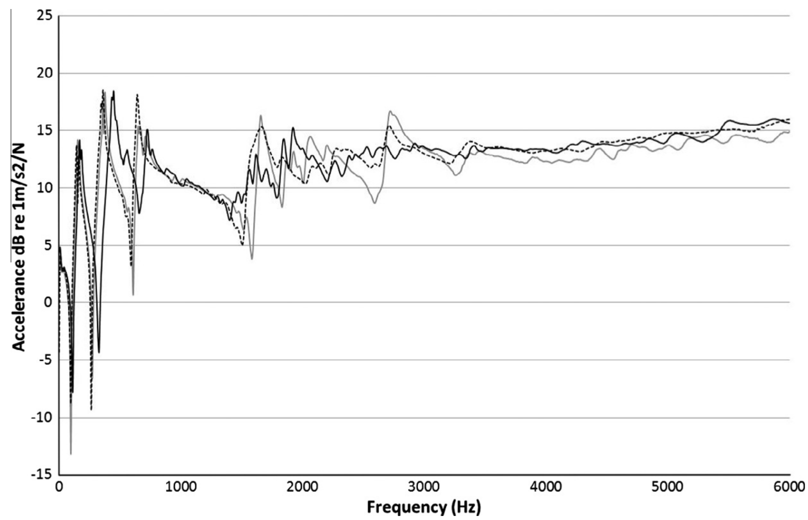


Fig. 12. Measured accelerance for Sample 8 (solid black line) compared to Sample 7 (solid grey line) and to Sample 6 (dashed line).

4.3. Combined plates containing circular indentations of power-law profile

In this section, different ways of combining 2.5 mm plates were considered in order to maximise the damping performance of the composite plates. The initial motivation for this combining of plates (Sample 6) was due to two main drivers: to produce a profile combination that could easily be converted into a panel with a continuous outer surface and also to increase the ease of manufacture of the double profiled indentations. The cross-sections of these profiles can be seen in Fig. 11. The plates were combined using a long cure epoxy resin.

Sample 7 from Section 4.2 was also used for comparison in this section, as it could show a direct comparison between a sample manufactured from two separate composite sheets and then combined to a sample manufactured from a single sheet of composite. Two different styles of reference plate were considered in order to gauge the effect of combining the plates and to determine which should be used for comparison. A plain plate (Sample 4) and a combined plate (Sample 3), see Fig. 10, were considered as both had merit as a reference. Both reference plates were of the same dimensions as the profiled plates, the combined plate consisted of two 2.5 mm plates combined with epoxy resin.

The results showed that the effect of the combination of the two plates and the addition of the epoxy resin had no quantifiable effect on the response of the plates by either increasing, decreasing or shifting the resonant peaks of the plain reference plate. Both samples therefore had merit as a reference plate. However, as the main comparison considered in this paper was the effect of these changes on existing complete panels, Sample 4, the plain reference was chosen to be used for comparison both in this and the following sections.

A comparison of the two combined double profiles, Samples 6 and 8, and Sample 7 is shown in Fig. 12. Below 500 Hz little difference in the responses of the three samples is seen. In general, it can be said that the amplitudes of the resonant peaks for Sample 8 are generally lower than for the other two samples, and therefore it performs more efficiently at damping resonant peaks. Sample 8 shows a maximum reduction from Sample 6 of 3 dB at 640 Hz and the same reduction from both the other two samples at 1.65 and 2.7 kHz. Above 2.9 kHz the response of all three samples is smoothed with resonant peaks heavily damped if not completely removed.

Fig. 13 shows the results for Sample 8 when compared to a plain reference plate. Below 450 Hz little to no damping is seen. The reduction in the amplitude again increases until a maximum reduction from the reference plate of 10 dB can be seen at 2.4 kHz. After this point the response is smoothed with resonant peaks heavily damped. It should be noted that the resonant peak seen at 1 kHz in the reference sample has been completely damped in Sample 8.

4.4. Comparison of composite panel constructions

The present section deals with encasing Samples 5–7 with a single sheet of pre-preg glass fibre composite in order to create a panel with a continuous outer surface, as shown in Fig. 14. Despite the good damping performance of Sample 8, the two 10 mm diameter holes on the either side of the outer surface of the panel meant that this configuration would be impractical to convert to a smooth surfaced panel without interfering with the circular indentations at the tip. Therefore, in this section the responses of the three enclosed panels will be investigated when compared to each other, to a reference plate, and finally when compared to the equivalent sample without the additional casing. This final comparison will show if there are any adverse effects to encasing the sample and whether such an effect, if present, can be off set against the advances gained by encasing the panels.

From Fig. 15, it can be seen that, when the composite panels, Samples 9–11, are compared, there is very little difference between the three samples. For Samples 9 and 10, there is no quantifiable difference between the resonant peaks. However, as Sample 9

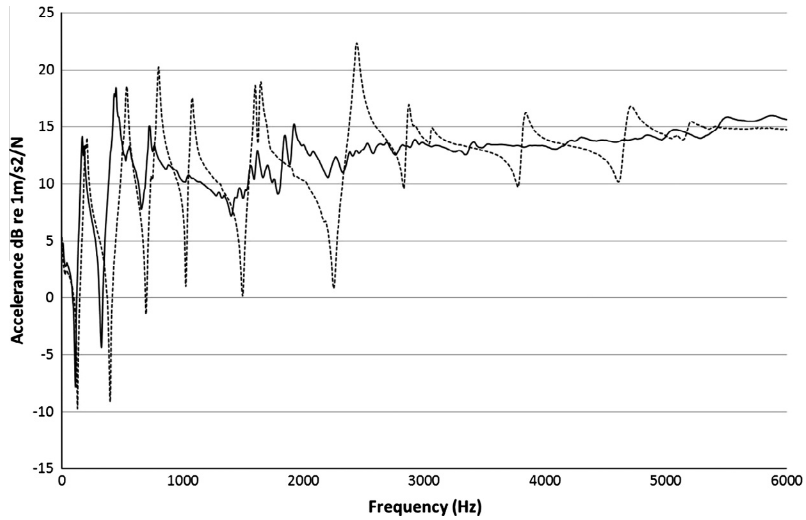


Fig. 13. Measured accelerance for Sample 8 (solid line) compared to Sample 4 – reference plate (dashed line).

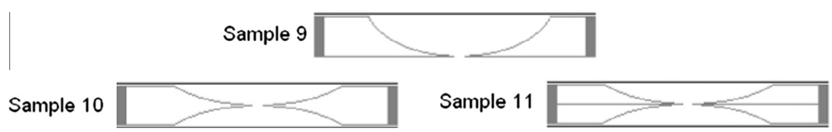


Fig. 14. Cross-sections of Samples 9–11.

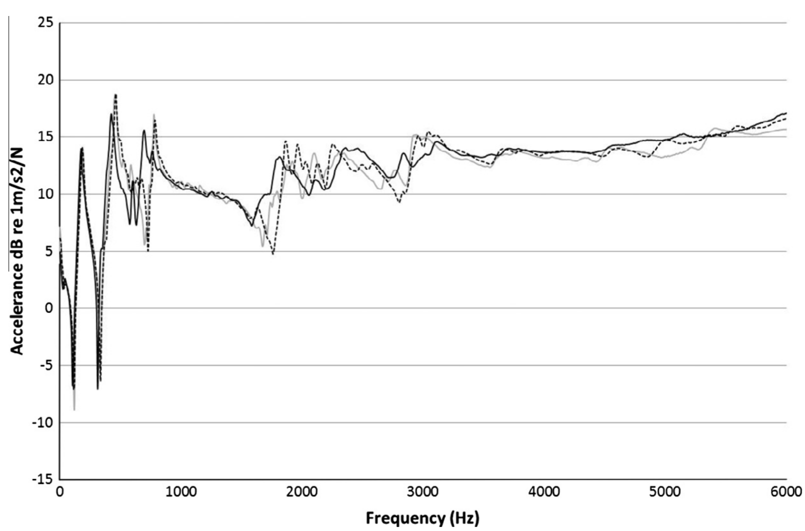


Fig. 15. Measured accelerance for Sample 11 (solid black line) compared to Sample 9 (solid grey line) and to Sample 10 (dashed line).

has two 10 mm diameter holes on one side of the plate, it does not have two smooth surfaces, therefore Sample 10 would best fit the specification. Sample 11 shows a reduction of 0.5–1 dB at 460 and 700 Hz and 2 kHz from the other two samples. Below 460 Hz there is no difference in the response of the three samples, and above 3 kHz the response is smoothed with resonant peaks heavily damped if not completely removed, thus producing a similar response.

A comparison of most effective damping panel; Sample 11, when compared to a reference plate, is shown in Fig. 16. A peak shift to the left from reference is observed, however, in the case of the combined composite plates, this effect occurs at a much lower frequency than previously observed, with the peak shift occurring as low as at 500 Hz. There is no difference between the two samples below 450 Hz. After this frequency, the dB reduction of the peak amplitudes increases until a maximum reduction of 10 dB from the reference plate is achieved at 2.4 kHz. It can also be seen that the reference plate resonant peak at 1 kHz has been damped completely in Sample 11. Thus, the combination of the composite plates and sheets results in an effective method of damping flexural waves in smooth surfaced composite panels.

Finally, from Fig. 17 it can be seen that, when Sample 11 is compared to its exposed indentation equivalent; Sample 6, the damping achieved for Sample 11 is in fact greater (1–2 dB) or equal to that of Sample 6. Below 450 Hz there is little to no difference in the samples. A maximum reduction of 3 dB can be seen at 650 Hz

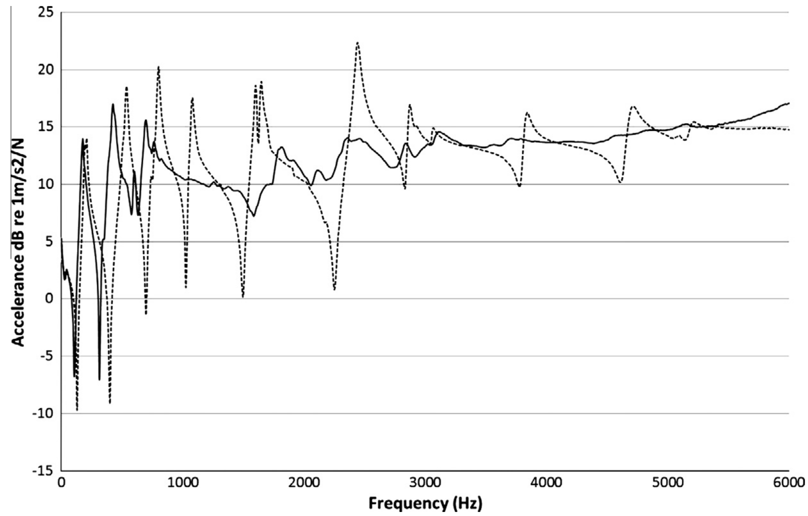


Fig. 16. Measured accelerance for Sample 11 (solid line) compared to Sample 4 – reference plate (dashed line).

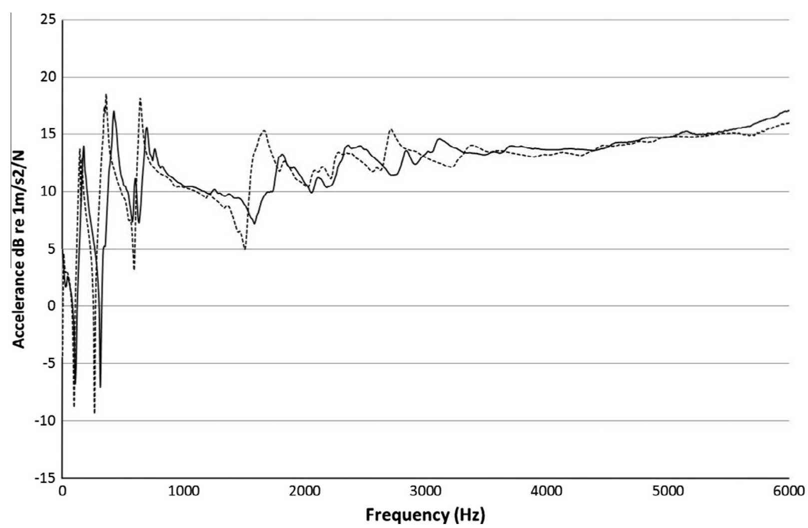


Fig. 17. Measured accelerance for Sample 11 (solid line) compared to Sample 6 (dashed line).

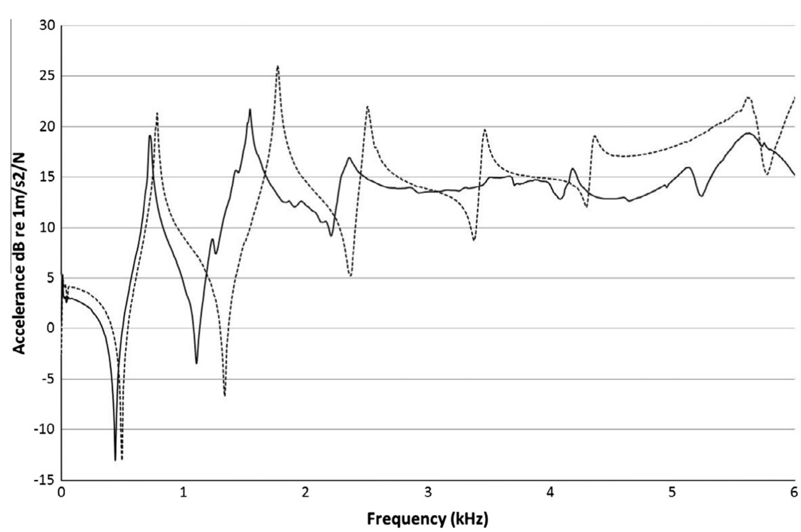


Fig. 18. Measured accelerance for a composite honeycomb sandwich reference panel (dashed line) compared to a composite honeycomb sandwich panel with enclosed indentations of power-law profile (solid line).

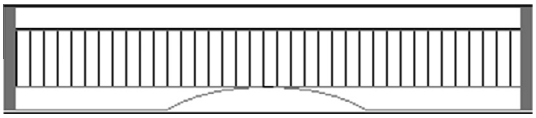


Fig. 19. Composite honeycomb sandwich panel with one 'normal side' and one containing enclosed indentations of power-law profile.

and 1.6 kHz. The most likely explanation for this increase is the thin layer of epoxy resin between the sample and the composite sheets. It is enclosed within, providing a slight increase in the loss factor of the sample. This result shows that circular indentations of power-law profile can be successfully enclosed in a smooth surface panel with a positive effect on the damping performance of the plate. There also appears to be little to no significant 'drum skin effect' over the area of the enclosed indentations. This result was replicated in the other samples tested.

4.5. Introduction of indentations of power-law profile to a honeycomb sandwich panel

This section describes the investigation of a composite honeycomb structure incorporating the smooth surfaced composite panels containing circular indentations of power-law profile designed above.

Fig. 18 shows the measured accelerance for a composite honeycomb sandwich reference panel compared to a composite honeycomb sandwich panel with enclosed indentations of power-law profile. As expected, there is a peak shift to the left as a result of reduced mass and stiffness. Above 1 kHz, the resonance peaks of the acoustic black hole plate show increasing reductions in amplitude compared to the reference sample. Above 2.4 kHz, the response is smoothed, with all resonant peaks seen in the reference sample heavily damped if not completely removed. A maximum reduction of 6 dB is seen at 2.5 kHz and at 3.4 kHz.

One of the configurations discussed in this paper was a composite honeycomb sandwich panel with indentations on both sides of the panel. There may be applications though where this is not suitable, for example on boat hulls. In cases such as these indentations can be placed on only one side of the panel, see Fig. 19. As a result, a reduction in damping performance will occur, but a substantial amount of damping can still be achieved while increasing the strength of the outer surface of the panel.

Theoretically the double profile, Fig. 20, would be the most effective profile to use, not only due to the damping performance, but also because of a greater contact area to be attached to the honeycomb to reduce the implications of the indentation on the structural performance of the panel.

It is important to note that further testing is required into the structural effects of the indentations on the composite and honeycomb panels. One would expect that making power-law profiled indentations in panels will reduce their structural strength in the same way as the introduction of non-profiled holes does as a result of increased mechanical stresses due to the reduction of the effective cross-sections of the panels, e.g. in the case of tensile strength of composite plates with through holes [18]. Generally, the requirement to introduce power-law profiled indentations into structures to be damped may compromise their structural strength. This represents the main disadvantage of the method of damping structural vibrations based on the acoustic black hole effect. Therefore, this method is limited to some specific structures and applications where structural strength is either not so important or can be improved.

5. Conclusions

Glass fibre composite strips containing wedges of power-law profile and plates containing circular indentations of power-law profile, materialising 1D and 2D acoustic black holes (ABHs) respectively, are effective configurations to be used for damping flexural vibrations. A glass fibre composite strip behaves in much the same way as steel when a wedge of power-law profile is added to the end of the strip. A 1D ABH in a strip produces a maximum reduction of 3.5 dB at 2.2 kHz.

A glass fibre wedge can be successfully attached to a steel strip in place of a steel wedge to produce increased reduction of resonant peaks. In theory this method can be also used to create 2D acoustic black hole inserts that can be integrated into existing structures with existing holes/slots.

There is little to no difference in the amplitudes of the responses when compared a machined wedge to a cured wedge. There is however, a difference in the frequency at which the resonances occur. This observation could be useful when a shift and reduction in resonance is required. The 2D indentation can also be easily manufactured via the curing method of manufacture.

As in the case of wedges of power-law profile, 2D ABHs perform in composite plates in much the same way as in steel plates, with a maximum reduction of 7.5 dB at 1.2 kHz in a 3 mm thick plate and with a maximum reduction of 8.5 dB at 2.4 kHz in a 6 mm plate, when compared to a reference sample.

The composite plates can be combined to attain more effective damping combination and to achieve a more time/cost and skill effective method of production. A combined plate provides a similar if not better reduction of the resonant peaks as a sample machined out of a single plate of composite. A maximum reduction of 10 dB at 2.4 kHz was achieved by Sample 8 when compared to a reference sample.

When a composite plate is enclosed by a composite sheet to form a smooth surfaced panel, there is relatively little reduction in damping performance when compared to an unenclosed plate. A maximum reduction of 10 dB at 2.4 kHz was achieved by Sample 11 when compared to a reference sample. Little to no drum skin effect is seen over the enclosed ABHs. Therefore, an enclosed smooth surfaced composite panel can be manufactured to give the same level of damping of flexural waves that can be achieved by a plate with exposed indentations.

It can be shown that curing is an acceptable method of manufacture to produce a wedge of power-law profile and to materialise the acoustic black hole effect. This method can be used to mass produce panels incorporating acoustic black holes with relative ease.

A smooth surfaced composite panel with ABHs is an effective method of damping flexural vibrations in composite panels. These panels can be substituted into applications where a non-structural

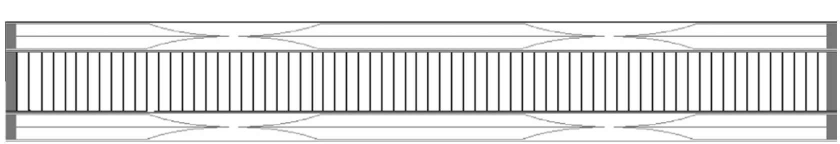


Fig. 20. Composite honeycomb sandwich panel with a different configuration of enclosed indentations of power-law profile.

composite panel is in use and a flexural vibration problem is present as an effective damping method. Such applications would include aircraft internal panels, internal panels in boats and formula one vehicles.

As in the case of composite panels, a composite honeycomb panel with enclosed indentations of power-law profile is an effective configuration for damping flexural vibrations within the structure. A maximum reduction of 6 dB is seen at 2.5 and 3.4 kHz, with heavy damping or elimination of resonant peaks above 2.4 kHz. The circular indentations, as it was seen before, provide substantial reductions in resonant peaks over a large frequency range.

Enclosing the circular indentations of power-law profile within the composite panel results in a surface texture similar to that of an un-machined conventional honeycomb sandwich panel, it also increases the damping performance of the panels. Honeycomb sandwich panels can be made with either both sides or a single side machined, depending on the application and damping requirements.

Acknowledgement

The research reported here has been partly supported by EPSRC Grant EP/F009232/1.

References

[1] Alam N, Asnani NT. Vibration and damping analysis of fibre reinforced composite material plates. J Compos Mater 1986;20:2–18.
[2] Chow ST, Liew KM, Lam KY. Transverse vibration of symmetrically laminated rectangular composite plates. Compos Struct 1992;20:213–26.
[3] Kramer MR, Liu Z, Young YL. Free vibration of cantilevered composite plates in air and in water. Compos Struct 2013;95:254–63.
[4] Heckl M, Cremer L, Ungar E. Structure borne sound. 2nd ed. Springer-Verlag; 1988.
[5] Mead DJ. Passive vibration control. Chichester: Wiley; 1998.
[6] Ross D, Kerwin E, Ungar E. Damping of plate flexural vibrations by means of viscoelastic laminae. In: Ruzicka JE, editor. Structural damping, Pergamon Press, Oxford. 1960. p. 49-87.
[7] Krylov VV, Tilman FJBS. Acoustic black holes for flexural waves as effective vibration dampers. J Sound Vib 2004;274:605–19.
[8] Krylov VV. New type of vibration dampers utilising the effect of acoustic 'black holes'. Acta Acust United Acust 2004;90:830–7.
[9] Krylov VV, Winward RETB. Experimental investigation of the acoustic black hole effect for flexural waves in tapered plates. J Sound Vib 2007;300:43–9.
[10] Mironov MA. Propagation of a flexural wave in a plate whose thickness decreases smoothly to zero in a finite interval. Soviet Phys Acoust 1988;34:318–9.
[11] Krylov VV. Propagation of plate bending waves in the vicinity of one- and two-dimensional acoustic 'black holes'. In: Proceedings of the ECCOMAS international conference on computational methods in structural dynamics and earthquake engineering (COMPDYN 2007), Rethymno, Crete, Greece, 13–16 June, 2007 [CD-ROM].
[12] Georgiev V, Cuenca J, Molerón-Bermúdez MA, Gautier F, Simon L, Krylov VV. Numerical and experimental investigation of the acoustic black hole effect for vibration damping in beams and elliptical plates. In: Proceedings of the European conference on noise control "Euronoise 2009", Edinburgh, UK, 26–28 October, 2009 [CD-ROM].
[13] Bowyer EP, O'Boy DJ, Krylov VV, Gautier F. Experimental investigation of damping flexural vibrations using two-dimensional acoustic 'black holes'. In: Sas P, Bergen B, editors, Proceedings of the international conference on noise and vibration engineering (ISMA 2010), Leuven, Belgium, 20–22 September, 2010. p. 1181–92.
[14] Georgiev VB, Cuenca J, Gautier F, Simon L, Krylov VV. Damping of structural vibrations in beams and elliptical plates using the acoustic black hole effect. J Sound Vib 2011;330:2497–508.
[15] O'Boy DJ, Krylov VV. Damping of flexural vibrations in circular plates with tapered central holes. J Sound Vib 2011;330:2220–36.
[16] O'Boy DJ, Bowyer EP, Krylov VV. Point mobility of a cylindrical plate incorporating a tapered hole of power-law profile. J Acoust Soc Am 2011;129:3475–82.
[17] Bowyer EP, O'Boy DJ, Krylov VV, Horner JL. Effect of geometrical and material imperfections on damping flexural vibrations in plates with attached wedges of power law profile. Appl Acoust 2012;73:514–23.
[18] Mallick PK. Effect of hole stress concentration and its mitigation on the tensile strength of sheet moulding compound (SMC-R50) composites. Composites 1988;19:283–7.