A review of experimental investigations into the acoustic black hole effect and its applications for reduction of flexural vibrations and structure-borne sound

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In this paper, a review of experimental investigations into the damping of flexural vibrations and the reduction of radiated sound power using the acoustic black hole effect are presented. The acoustic black hole effect damps flexural vibrations by reducing edge reflections from structures' free edges via the use of wedges or tapered circular indentations of power-law profile. Wedges of power-law profile materialise one-dimensional acoustic black holes for flexural waves that can absorb a large proportion of the incident flexural wave energy. Tapered circular indentations of power-law profile act as two-dimensional acoustic black holes for flexural waves. The results of experimental investigations into the damping of flexural vibrations in turbofan blades incorporating a power-law profile are described along with the incorporation of two-dimensional acoustic black holes into smooth surfaced composite panels, and composite honeycomb sandwich panels. Finally, the results for multiple indentations (arrays) of two dimensional acoustic black holes and the associated reduction in structure-borne sound are given. The reported results demonstrate that the acoustic black hole effect can provide an effective damping of flexural vibrations in the aforementioned blades and panels, as well as an effective reduction of sound radiation from structures.

1 INTRODUCTION

Damping of structural vibrations remains a challenging problem for different branches of engineering, especially for aeronautical and automotive applications. Passive damping of structural vibrations is usually achieved by adding layers of highly absorbing materials to the structure in order to increase energy dissipation of propagating (mostly flexural) waves[1-3]. A common means of damping resonant flexural vibration in a structure is to reduce the reflection of flexural elastic waves from the structures free edges[4]. The acoustic black hole effect[5-7] utilizes

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this method of damping via wedges of power-law profile. The power law profile mentioned above is illustrated in Fig. 1(a, b) and is defined by

$$h(x) = \varepsilon x^m \ , \tag{1}$$

where $h(x)$ is the local thickness, m is the power exponential (non-dimensional positive constant), ε is a dimensional constant, and x is a coordinate measured along the wedge axis.

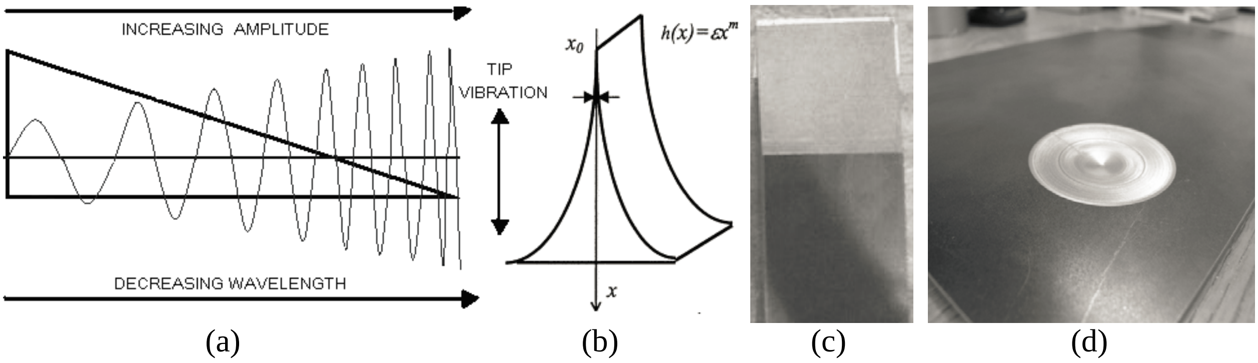


Fig. 1 - (a) Wave behaviour in wedge of power-law profile, (b) Wedge of power law profile, (c) 1D ABH (wedge of power-law profile), (d) 2D ABH (circular indentation of power-law profile) in steel

This method has been applied in the form of one-dimensional (1D) and two-dimensional (2D) acoustic black holes (ABH) based on power-law wedges and circular indentations respectively (see Fig. 1(c, d)). 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[5-8], which implements an 'acoustic black hole' (ABH). However, ideally sharp wedges do not exist in reality; therefore the presence of a small strip of damping layer at wedge tip is paramount to obtain very low reflections from these free edges, which constitutes the acoustic black hole effect[5-7]. It has been established theoretically[5, 6] and confirmed experimentally[7] that this method of damping structural vibrations is very efficient even in the presence of edge truncations and other imperfections.

This paper will give an overview of four investigations carried out at Loughborough University, UK into the ABH effect, the aim of which was to facilitate the integration of this effect into real world applications: Fan blades, Composite panels, Honeycomb sandwich panels, and Sound radiation from arrays of 2D black holes. This paper will also review the results of investigations into the vibration and sound radiation responses of engine covers containing acoustic black holes obtained at the MBtech Group, Germany. Accreditation will be given in each section so the ownership of results can be clearly seen.

2. EXPERIMENTAL PROCEDURE

The results presented in this paper were obtained using a variety of different experimental techniques. The standard vibration test performed was carried out using a force transducer, accelerometer and shaker, with the sample suspended to allow nearly free vibration of the sample plates (i.e. to eliminate clamping of edges), Fig. 2.

Two further vibration tests techniques were used, the first of which utilized a scanning laser vibrometer, shaker and suspended sample. The second test techniques utilised airflow as the excitation source and was performed in a closed circuit wind tunnel, Fig. 2. Flow visualisation tests were also performed in the wind tunnel; with a max flow speed of 34 m/s. The final investigation was into the sound radiation of suspended plates with arrays of ABH's, and it was performed in a fully anechoic chamber in accordance with ISO 3744.

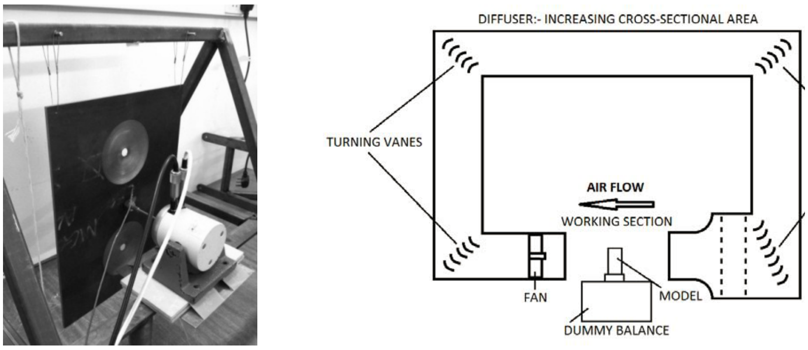


Fig 2 - Standard Vibration set up with suspended sample, and a Closed circuit wind tunnel schematic.

3 RESULTS

3.1 Fan Blades

This section looks at the introduction of a wedge of power-law profile to a fan blade[9] and examines whether this could produce noticeable damping of structural vibrations due to the acoustic black hole effect[5, 6], as seen in experimental steel wedges of power-law profile[7]. To test the applicability of a 1D ABH on a fan blade, a straight reference blade was compared to a straight blade with a 1D Acoustic black hole with the attached damping layer.

This results of the comparison are shown in Fig. 3. As seen in the previous experimental work for steel wedges of power-law profile[7], the addition of a wedge of power-law profile to the end of an aluminium fan blade shows the same trends. Above 1.4 kHz, an increase in the reduction of the resonant peaks is seen up to a maximum reduction of 12 dB from the reference sample at 4.2 kHz. Above this frequency the response is smoothed, with resonant peaks heavily damped if not completely removed.

A twisted blade more accurately represents the real world engine fan blades these experimental samples are emulating. The effect of a twist (11 degrees) on the blade needs to be considered for comparison with real fan blades. When the twisted reference blade is compared to the twisted blade with a wedge of power-law profile, a damped response similar to that observed for the straight blades is clearly visible. Between 1.4 – 6.8 kHz, there are reductions in the resonant peaks of 3-10 dB, with some resonances damped completely. A maximum reduction of 10.5 dB from the reference plate by the profiled sample can be seen at 4.1 kHz. After 6.8 kHz, the response is smoothed with resonant peaks heavily damped if not completely removed.

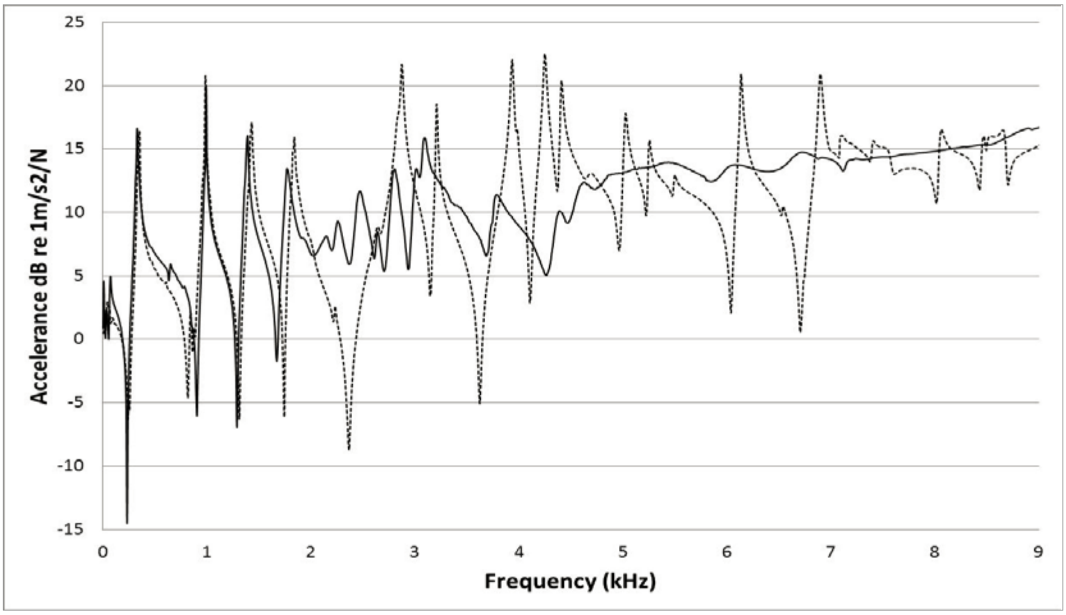
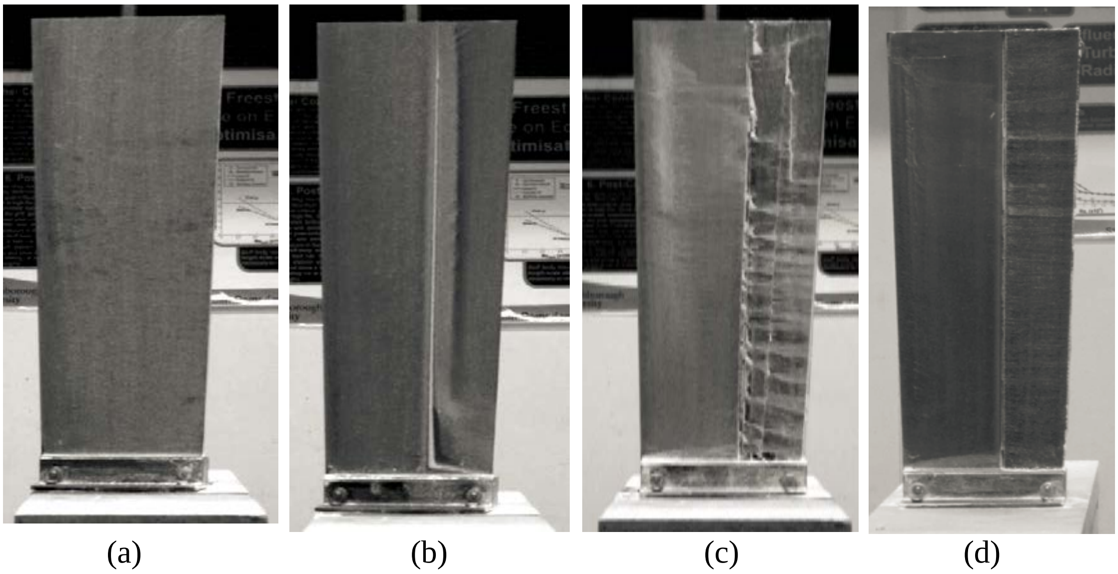


Fig. 3 - Accelerance for Reference blade (dashed line) compared to blade with 1D ABH and damping layer (solid line)[9]

The following section shows the results of the flow visualisation tests for the straight fan blade. The fan blade is at an arbitrary incline of 10 degrees to the airflow. The aim of this investigation was to prove that, with adaptation of the damping layer attached to the wedge of power-law profile, the airflow over the underside of the blade could be returned to a similar state as that seen for the reference blade[9].



(a) (b) (c) (d)

Fig. 4 - Flow visualization diagram for: (a) Reference fan blade, (b) Fan blade with power-law wedge, (c) Fan blade with power-law wedge with single damping layer, (d) Fan blade with power-law wedge and shaped damping layer[9]

Figure 4 shows the progression of the flow visualisation tests from a reference fan blade to a fan blade with a wedge of power-law profile and a specifically shaped damping layer. The flow visualization for the reference fan blade, Figure 4(a), shows laminar flow across the blade surface with no separation. The effects on the airflow of the presence of the wedge are immediately obvious, Figure 4(b). A clear transition line, lamina separation bubble, and then reattachment of the flow towards the trailing edge of the blade are seen.

A piece of damping layer identical to that used in vibrations tests with steel wedges[7] was then attached. From Figure 4(c), a clear line of transition can be seen between the upstream laminar flow and the turbulent flow after the start of the damping layer. This flow is too turbulent to reattach to the blade. It is worth noting that with the damping layer attached in this way there is a step between the blade surface and the damping layer. This step is responsible for the increased turbulence of the airflow in the wedge area[9].

One could expect that any deviation in profile from the original design specification of the blade will not only have the increased turbulence and increased drag, as seen above. It will also result in lower efficiency and will also affect the airflow into the next stage of the engine. An obvious possible solution to the flow turbulence problem seen in Figs. 4(b) and (c) would be to recreate the flow pattern seen in Fig. 4(a), i. e. the original profile of the blade has to be restored. One method of partly achieving this is to shape the damping layer in order to recreate the original profile. This was achieved by building up layers of the damping material which, when covered by a layer of damping material of the same width as the wedge, would reproduce the original profile. The final diagram (Fig. 4(d)) shows the resultant flow over the blade with this shaped damping layer. There is still a clear line of transition but the flow quickly reattaches to blade. This line of transition will always be seen with the ridge at the edge of the damping layer. This result shows that if the damping layer could be more effectively blended into the blade the line of transition would disappear and a laminar flow would cover the blade.

The above results show[9] that 1D acoustic black holes are effective dampers for fan blades. One should note though that for jet engine applications an integrated damping layer is necessary.

3.2 Composite Panels

A 2D acoustic black hole has an exposed delicate tip that is prone to damage. In order to make ABH’s more applicable in the real world they can be enclosed in a smoothed surface panel[10]. This section considers the production of such a panel in glass fibre composites. Each panel contains two ABH’s

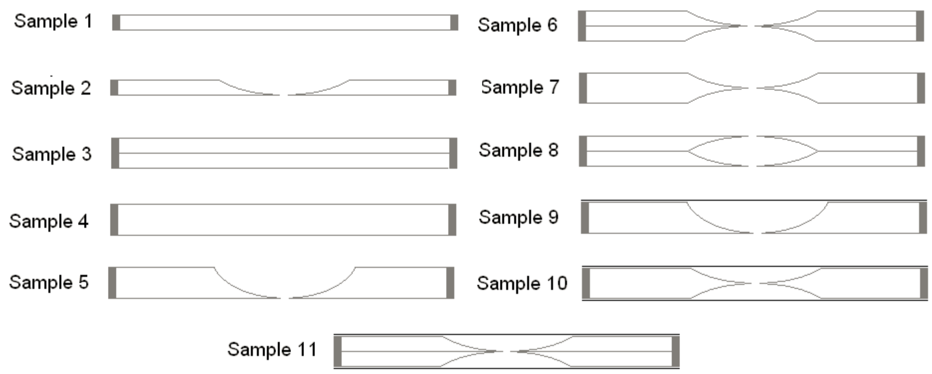


Fig. 5 - Cross-section view of Sample plates 1-11

First the effectiveness of a 2D ABH was tested in 3 mm composite panels (Samples 1, 2), see Fig. 5. 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[10].

To be able to enclose 2D ABH's into a smooth surfaced panel the tip of the indentation is required to be in the centre of the thickness of the panel, see Fig. 5 (Samples 6, 7). A conventional ABH plate (Sample 5) and a plate with only the central hole visible on the outside edge of the plate (Sample 8) was also considered[10]. There was very little difference in the response of samples 5, 6, 7 and 8. Reductions of approximately 10 dB from reference in each case were seen.

Samples 5, 6 and 7 where then enclosed beneath a layer of composite to form a smooth surfaced panel (Samples 9, 10 and 11). Again there was little difference in the responses of these plates. Sample 11 slightly (1-3dB) out-performed the other samples. The response of Sample 11 compared to a reference plate is shown in Fig. 6. After 450 Hz, 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 effective damping of flexural waves in smooth surfaced composite panels, e.g. internal aircraft skins[10]

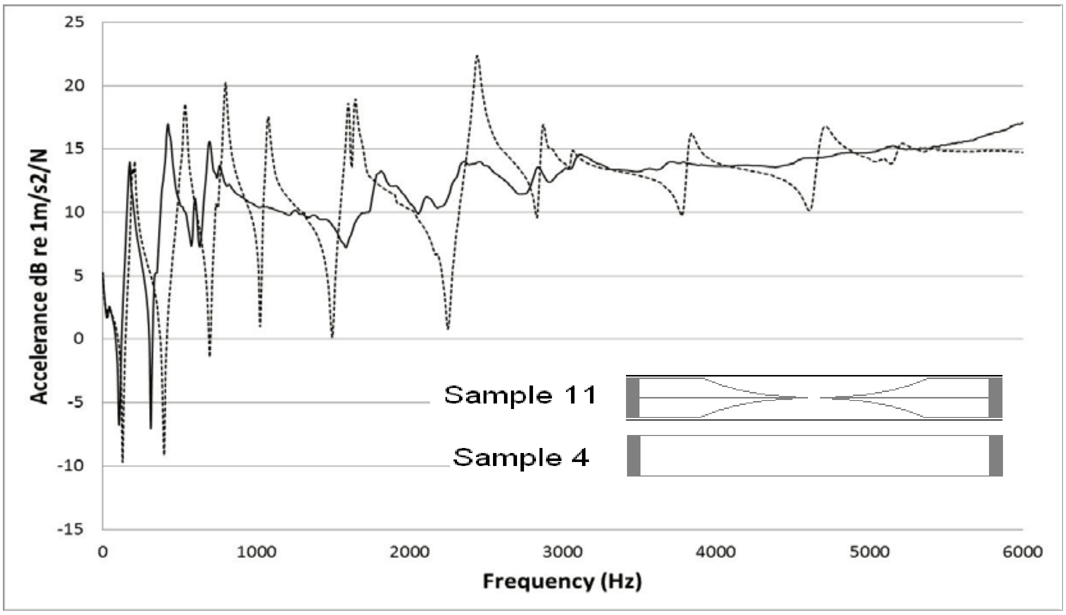


Fig. 6 - Accelerance for Sample 11 (solid line) compared to Sample 4 (dashed line). Insert; cross-section of samples[10]

3.4 Honeycomb Sandwich Panels

The natural progression of this research was to investigate composite honeycomb sandwich panels incorporating the smooth surfaced composite panels containing 2D ABH's described above. Figure 7 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. 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 and 3.4 kHz. A composite honeycomb panel with enclosed circular indentations of power-law profile shows a good damping performance[10].

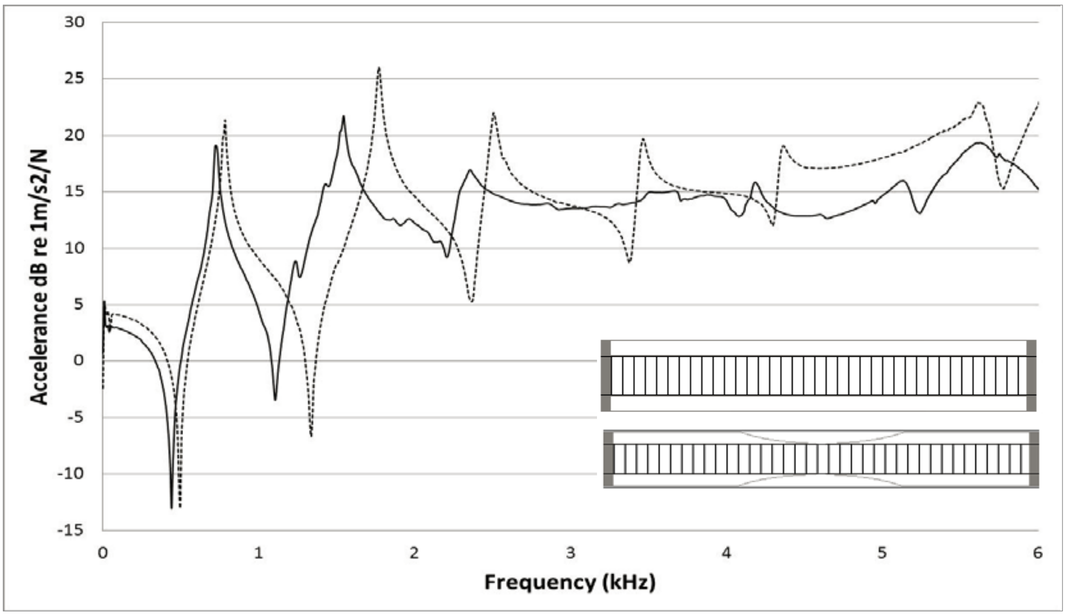


Fig. 7 - Accelerance for a composite honeycomb sandwich reference panel (dashed line) compared to the same style panel with enclosed indentations of power-law profile (solid line). Insert; cross-section of samples[10]

3.5 Sound Radiation from a 2D Array

It has been previously found[11] that a hole need to be present at the centre of the indentation of power-law profile in order for the acoustic black hole effect to occur. That investigation also looked at increasing the number of indentations in order to increase the damping performance of the plate. Arrays of 2 to 6 ABH’s were tested, and it was found that the greater the number of indentations the greater the damping performance of the plate. A plate with six 2D black holes showed the greatest damping[11].

Figure 8 shows the response of the six indentation plate when compared to a reference plate[11]. It can be seen that after 4 dB almost all peak responses are flattened and a maximum damping of 14 dB occurs at 6.5 kHz.

One of the main aims of this research was to explore further practical applications for acoustic black holes. Therefore, sound radiation from structures containing these black holes was considered[12]. Figure 9 shows the results for a plate containing six profiled circular indentations compared to a reference plate. Below 1 kHz there is little to no reduction in the sound pressure level, as was the case with the reduction in vibration response. Between 1-3 kHz the sound power level response is reduced from the reference plate response by 10-18 dB, with the maximum reduction in the sound radiation occurring at 1.6 kHz.

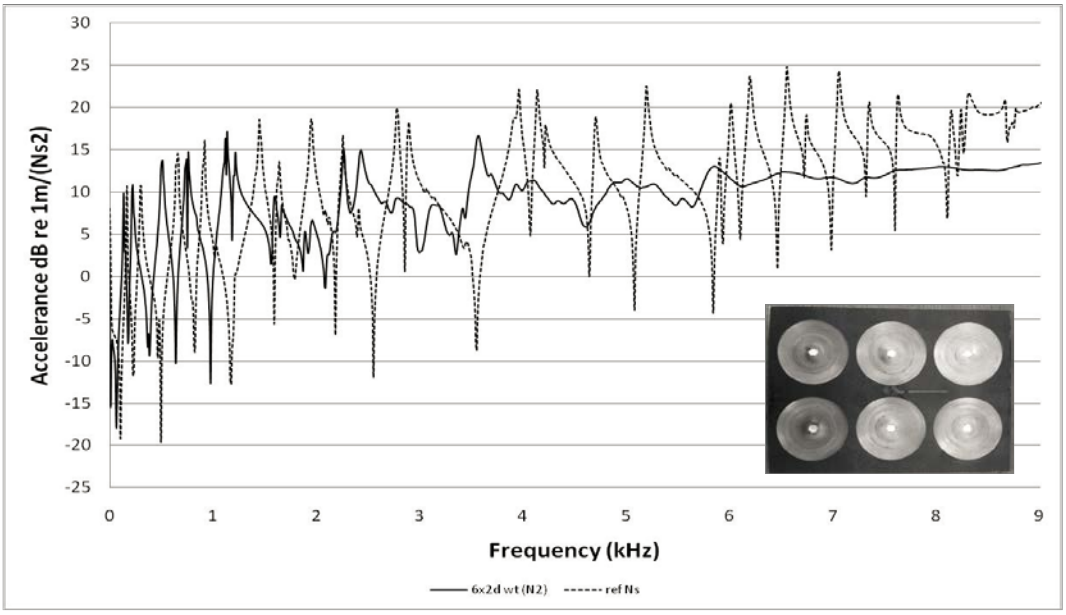


Fig. 8 - *Accelerance for a plate containing six profiled circular indentations with 14 mm central holes and additional damping layers (solid line), as compared to a reference plate (dashed line)*[11]

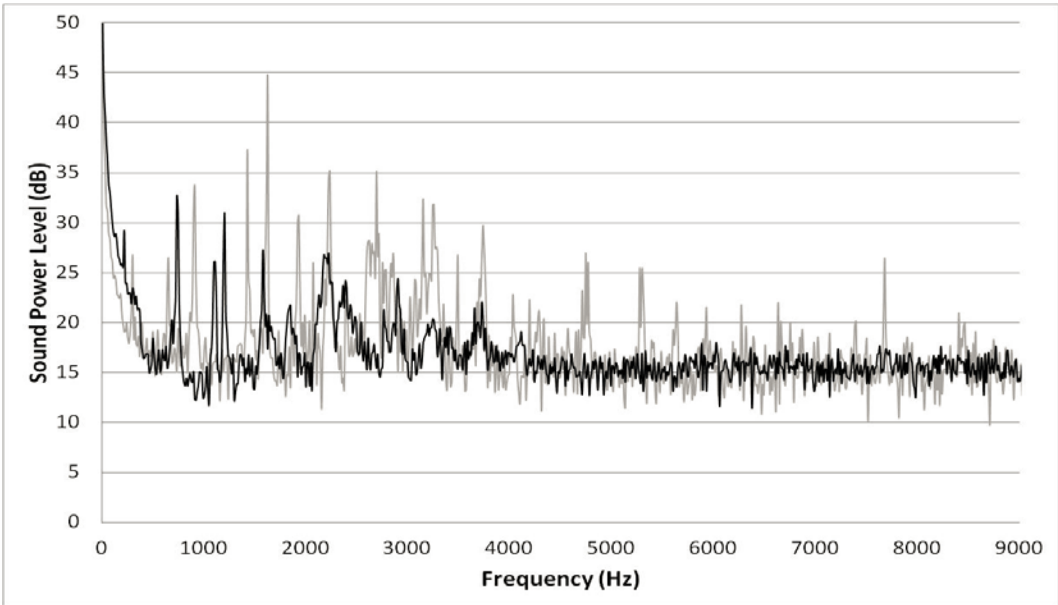


Fig. 9 - *Sound power level of a plate with six indentations of power-law profile with a damping layer (black line) compared to a reference plate (grey line)*[12]

This following section considers the amplitudes of the plate's vibrational response in comparison to the amplitudes of the associated sound radiation. As the frequency increases the amplitude of deflection over the constant thickness section of a plate containing circular indentations of power-law profile tends to zero. At lower frequencies, where no reduction in sound radiation or vibration response is seen, the plate behaves as a constant thickness plate, with a little difference from the plate without indentations.

In the frequency range where reductions in vibration response and sound radiation are seen the plate vibration pattern changes substantially, with a noticeable amplitude reduction outside the indentations[12]. In the higher frequency range the only displacement on the plate is seen in the last 2 cm off the indentation tip (centre). This corresponds to the area of maximum effectiveness of the damping layer.

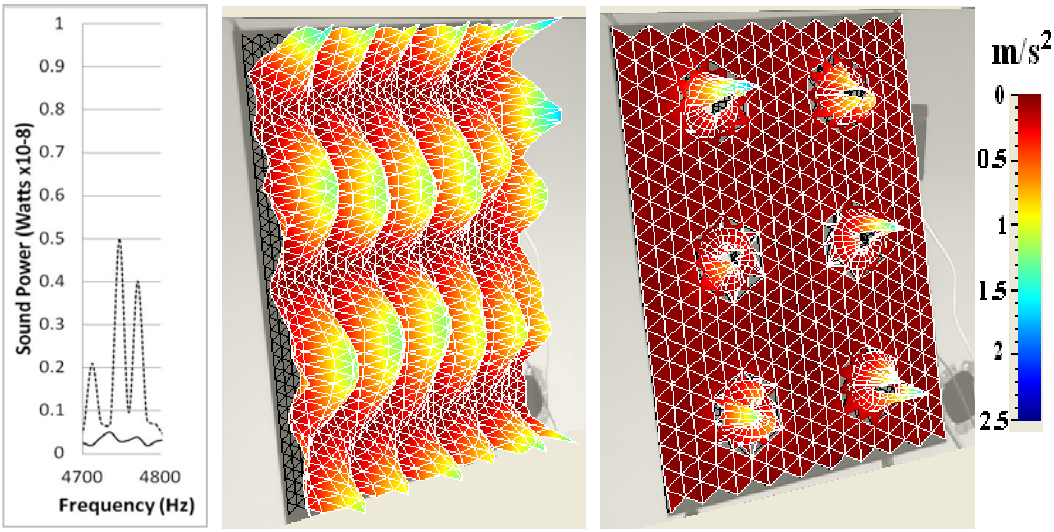


Fig. 10 - Results for resonant peak at 4.75 kHz; (a) Sound power in Watts for reference plate (dashed line) compared to the plate containing six indentations with damping layers, (b) Modal response of the reference plate, (c) Model response of the plate containing six indentations with damping layers, (d) Amplitude of response; key[12].

Figure 10 shows the results for the sound power (in Watts) at 4.75 kHz for a reference plate compared to the plate containing six indentations with damping layers, and for the modal response of the reference plate and the plate containing six indentations with damping layers. It can clearly be seen, see Fig. 10(c), that other than around the centres of the indentations, the amplitudes of the response over the entire plate are almost zero. Although the amplitudes of the response in the ‘active’ area (around the centres) are approximately 1 m/s^2 greater than those seen on the reference plate, this does not affect the sound radiation of the plate, as seen in Fig. 10(a). Thus, acoustic black holes result in a considerable reduction in sound radiation.

4 PRACTICAL APPLICATION

4.1 Experimental Samples and Set Up

The following sections describe the results of the preliminary investigations into an automotive application of acoustic black holes (ABH) undertaken at MBtech Group. Two samples were used for comparison in the initial investigation into ABHs in engine covers. The vehicles standard engine cover was used as a reference sample, and a second sample was constructed to contain two plates with ABH’s, see Fig. 11. The engine covers are made from a PA thermoplastic, and the ABH panels - from a POM plastic.

The Acoustic black holes were machined and then bonded into the engine cover using a two-component glue. The vehicle utilized for the testing of the engine covers was a mid-sized salon vehicle with a 4- cylinder spark ignition engine.

Fig. 11 - ABH engine cover –top view and location of accelerometers on reference engine cover

Two experimental set ups where used; bonnet open and bonnet closed. Two different experiments were conducted for each set up; engine idle, and a steady state 2100 rpm. Four ICP accelerometers were located on the surface of the engine cover (Fig. 11) to record the acceleration of the cover.

Four condenser microphones were located around the external front end of the vehicle; driver side, passenger side, front, and directly above the engine cover. Each test was repeated a minimum of five times to ensure repeatability. The recording and analysis of the vibration and sound pressure response was performed using the Müller-BBM PAK system. A frequency range of 0-2 kHz was investigated.

4.2 Results

The results of the investigations into the sound radiation from engine covers containing ABH's are detailed in this section. Two key results are shown: the response of the ABH engine cover at idle, 2100 rpm, and with the bonnet open and closed.

A comparison of the sound pressure results for an engine cover with and without acoustic black holes at idle with the bonnet open and microphone position 1 was carried out. The results showed that the main reduction in the resonant peak amplitudes was at 30-300 and at 600-1400 Hz. A maximum reduction of 9.7 dB is seen at 190 Hz. The total average reduction in noise from the vehicle was recorded as 3 dB when compared to the reference engine cover. This result was mirrored by the other microphone positions, but to a lesser extent.

The most promising results were found during tests conducted at 2100 rpm with a closed bonnet again from microphone position 1, see Fig. 12, where a total average reduction from the reference sample of 6.5dB was recorded. Here a boarding damping range is observed with substantial reductions of 0-10 dB occurring in the range 70-1400 Hz. A maximum reduction in peak amplitude of 10 dB is seen at 369 and 442 Hz. Again, as expected, this result was mirrored by the other microphone positions but to a lesser extent.

Two other interesting observations were made during this investigation. It was found that at engine idle the ABH engine cover was perceived to be most effective when the bonnet was open and that the measured reductions were less when the bonnet was closed. It is therefore possible that the reductions obtained from the ABH cover are lost due to the sound radiation from the bonnet itself. The second observation is that the opposite of the above occurs when the engine is held at 2100 rpm, with the most substantial reductions occurring with a combination of the acoustic black hole cover and a closed bonnet.

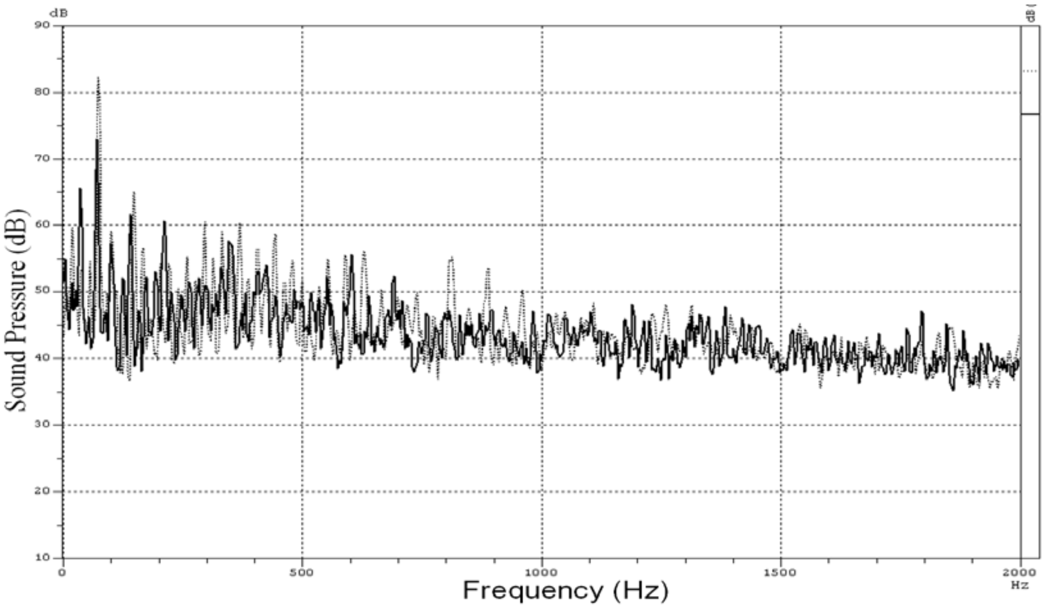


Fig. 12 - Sound radiation response for a reference engine cover (dashed line) when compared to an engine cover containing ABH's (black line) at 2100 rpm with the bonnet closed.

It can be concluded from these results that acoustic black holes in engine covers are an effective method of reducing sound radiation from the vehicle engine at increased rpm from idle with the bonnet closed.

5 CONCLUSIONS

It can be seen from this brief review that both 1D and 2D ABH's are an effective method of damping flexural vibrations in plates, fan blades, composite panels, and honeycomb sandwich panels. It can be concluded that it is possible to manufacture a fan blade with an acoustic black hole and built up damping layer that produces little interruption to the airflow while providing considerable damping to the blade.

Enclosed smooth surfaced composite panels can be manufactured to give the same level of damping of flexural vibrations that can be achieved by plates with exposed indentations. These panels are more versatile as their sharp tips are protected and could be used in real applications, such as interior aircraft panels, to reduce noise and vibration. Composite honeycomb panels with enclosed circular indentations of power-law profile also show a good damping performance.

The greater the number of indentations of power-law profile in a plate the greater the damping performance. Array plates provide a considerable reduction in radiated sound power when compared to a reference plate.

It can be concluded that it is possible to produce an engine cover that can reduce the total noise emitted from the engine compartment by an average of 6.5 dB when panels containing acoustic black holes are integrated into a standard engine cover. It is also possible to damp certain frequencies by as much as 10 dB. This ability ensures that the obtained noise reductions do not result in one dominant frequency and therefore increase the perceived positive psycho-acoustic effect of the vehicle.

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