DOI: 10.61782/fa.2025.0053

FORUM ACUSTICUM EURONOISE 2025

ABSORPTION AND SCATTERING OF BENDING WAVES BY AN ACOUSTIC BLACK HOLE EMBEDDED IN A HONEYCOMB SANDWICH PANEL

Alex Besse[1*] **Omar Aklouche**[2] **Frédéric Ablitzer**[1]
Adrien Pelat[1] **François Gautier**[1]
[1] Acoustic Laboratory of Le Mans University (LAUM), Le Mans University, France
[2] National Engineering school of Le Mans (ENSIM), Le Mans, France

ABSTRACT

Acoustic Black Hole (ABH) is a passive vibration control technique that involves introducing a local reduction in the thickness of a panel, combined with a thin viscoelastic coating in the weak area. This approach increases vibration damping with no additional mass, making it particularly attractive for lightweight structures such as honeycomb sandwich panels used in aerospace applications. The vibration field of ABH is mainly controlled by complex localised modes, classified according to their circumferential order. The reflection coefficient of an incident bending wave converging on the ABH, defined for each circumferential order, is analysed in the complex frequency plane. It is shown that each mode corresponds to a pair of poles and zeros symmetrical about the real frequency axis. The viscoelastic ABH coating allows the pairs to rotate about the origin, bringing the zeros closer to the real axis, creating interesting dips in the reflection coefficient as a function of real frequency. A wave-based model is used to identify local modes, study the role of the ABH profile and the effect of the damping layer. The approach provides a list of rules for optimal design of the ABH absorber.

Keywords: *sandwich panel, vibration mitigation, local modes, acoustic black hole, critical coupling*

****Corresponding author**: alex.besse@univ-lemans.fr.*

1. INTRODUCTION

Acoustic Black Holes (ABHs) are an effective passive technique for vibration and noise reduction in engineering structures. An ABH consists of a tapered region within a beam or plate that traps and dissipates flexural waves. Introduced by Mironov and Krylov [1, 2], it relies on a power-law thickness profile with a thin viscoelastic layer for enhanced wave absorption. Initial studies on one-dimensional (1D) ABHs analysed wave trapping using geometrical acoustics [3, 4] and impedance models [5], highlighting the role of damping in ABH performance. Later research accounted for evanescent waves and imperfections [6]. More recently, ABHs have been integrated into sandwich structures to improve vibration attenuation despite challenges emerging from frequency-dependent bending stiffness [7–9]. The influence of local modes on ABH performance was first demonstrated in [10]. These modes create local minima in the reflection coefficient, with zeros aligning on the real frequency axis under optimal damping (critical coupling). This study extends the concept to 2D configurations, using a semi-analytical model of a honeycomb sandwich plate with a circular ABH to analyse local modes and reflection coefficients. Inspired by prior wave-based modelling [11, 12], the study compares identical ABHs in sandwich and solid panels to assess modal density effects. The ABH profile is then adjusted to examine effects of positioning of zeros and establish design guidelines for improved ABH performance.

2. EFFECT OF BENDING STIFFNESS FREQUENCY DEPENDANCE ON ABH PERFORMANCE

2.1 Configurations

In order to investigate the effect of the sandwich's bending stiffness decreasing with frequency, an adaptation of the wave based model, described in [9], of an ABH embedded in a infinite panel is used. The local behaviour is studied for two configurations. The first configuration corresponds to a sandwich panel modelled as an uniform Kirchhoff's panel based on Nilsson's sandwich theory [7]. This theory propose modelling of a sandwich structure composed of two identical skin as a uniform material panel with a bending stiffness dependent on frequency. The second configuration, corresponds to the "solid panel", which is identical to the sandwich panel, in geometry and material properties, except for a uniform bending stiffness set to the static bending stiffness of the sandwich panel. The mechanical properties of the skins and the core are given in table 1 with an identical loss factor $\eta = 0.04$. An ABH is embedded in these two configurations. This ABH is composed of 2 main zones, a tapered zone in which the honeycomb core of the sandwich is decreasing with the skins staying of constant thickness and a zone made of a central plateau of a single skin. At the connection of these zones, there is a small region where the two skins connect. The ABH can be characterised with $a = 0.05$m, the radius of the central plateau and $b = 0.15$m the radius of the ABH. A viscoelastic layer is added to the ABH using the RKU model [13]. The ABH placed in the two configurations lead to a bending stiffness profile depending on the radius as demonstrated in Figure1 (a) for the solid panel and in Figure 1(b) for the sandwich panel.

	ρ	ν	E	G
Skin	1541 kg/m^3	0.35	19.3 GPa	7.1 GPa
Core	37 kg/m^3	0.3	55 MPa	21 MPa

Table 1. Mechanical parameters of a honeycomb sandwich panel.

2.2 Reflection coefficient of ABH in sandwich and solid plates

The two configurations previously described are used to investigate the performance and local dynamics of an ABH embedded in an infinite panel composed of either a solid material or a sandwich material modelled with Nilsson theory. In order to do so, the value of the reflection coefficient, R, is evaluated in a complex frequency plane and plotted along the real frequency axis for the circumferential order $n = 0$ (Figure 1(c)). The results show a number of local minima identified by the zeros in the complex frequency plane. It appears that each zero corresponds to a local modes of the ABH. In the sandwich case, an increased number of zeros is observed due to the softening of the structure which increases the modal density of the ABH. When looking at the reflection coefficient value along the real axis, each zero corresponds to a dip, leading to an overall value of $|R|$ being lower in the case of a sandwich panel (for f=1500 Hz, $|R|_{sandwich} = 0.3$ while $|R|_{solid} = 0.5$. An increased number of zeros ,i.e number of local modes, leads to an ABH being more effective. When designing an ABH it is crucial to carefully consider the local dynamics of the scatterer.

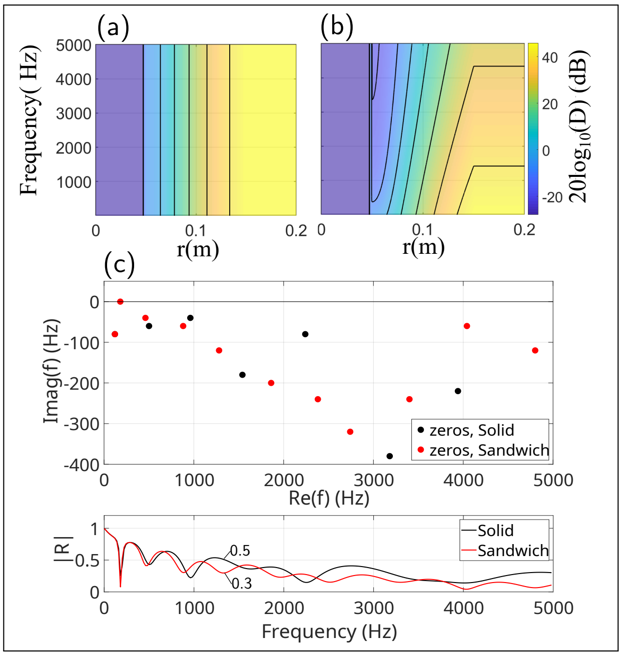

Figure 1. (a) Bending stiffness profile of an ABH placed in a solid panel. (b) Bending stiffness profile of an ABH placed in a sandwich panel panel. (c) Positions of zeros in the complex frequency plane and value of $|R|$ along the real frequency axis for the two configurations at circumferential order $n = 0$.

3. ZERO POSITIONING TO IMPROVE ABH PERFORMANCE

3.1 Selected ABH profiles

As previously shown in section 2, the performance of an ABH is dependent on local modes which create zeros in the reflection coefficient reducing its overall value. The aim of this section is to improve the performance of an ABH by modifying the placement of zeros through geometry variations. Two geometries are compared and correspond to an ABH placed in the same sandwich panel as the previous section with $\eta = 0.01$. In these panels, the two ABH geometries are placed. The first, called 'ABH-1', is identical to the one used in section 2 and its cross-section is shown in Figure 2(a). The second, named 'ABH-2', corresponds to an ABH of radius $b = 0.15$m, with the same power law profile as ABH-1 but with a central plateau twice as large, $a = 0.10$m, its cross section is visible in Figure 2(b).

3.2 ABH performance

These configurations are compared by studying the zeros of the reflection coefficient in the complex plane and the mobility values. The ratio $\chi = im(f)/re(f)$ describes the proximity of the zero to the real axis. The zeros of the reflection coefficient for ABH-1 and ABH-2 are shown in Figure 2(c) for $n = 0$. In this figure, it appears that the zeros of ABH-2 are closer to the real frequency axis ($\chi_{ABH-2} = 2\%$ vs $\chi_{ABH-1} = 10\%$). The proximity of the zeros to the real frequency axis leads to an improvement in the performance of an ABH due to deeper dips in the reflection coefficient value.

To confirm the improved absorption performance of ABH-2, the quadratic mobility averaged over a finite circular sandwich plate is computed using the model presented in [9] and is shown in third-octave band representation in Figure 2(d). In this figure, the mobility of a panel without ABH is compared to the mobility of the panel in the presence of ABH-1 and ABH-2. A mobility reduction of up to 5 dB is observed between the uniform panel and ABH-1, whereas ABH-2 achieves a reduction of up to 8 dB. It appears that the geometry of ABH-2 allows for a greater mobility reduction, particularly between 500 and 2000 Hz. This result correlate with the positions of zeros in the complex frequency plane. These results highlight the importance of the positioning of zeros in the complex plane to enhance the performance of an ABH.

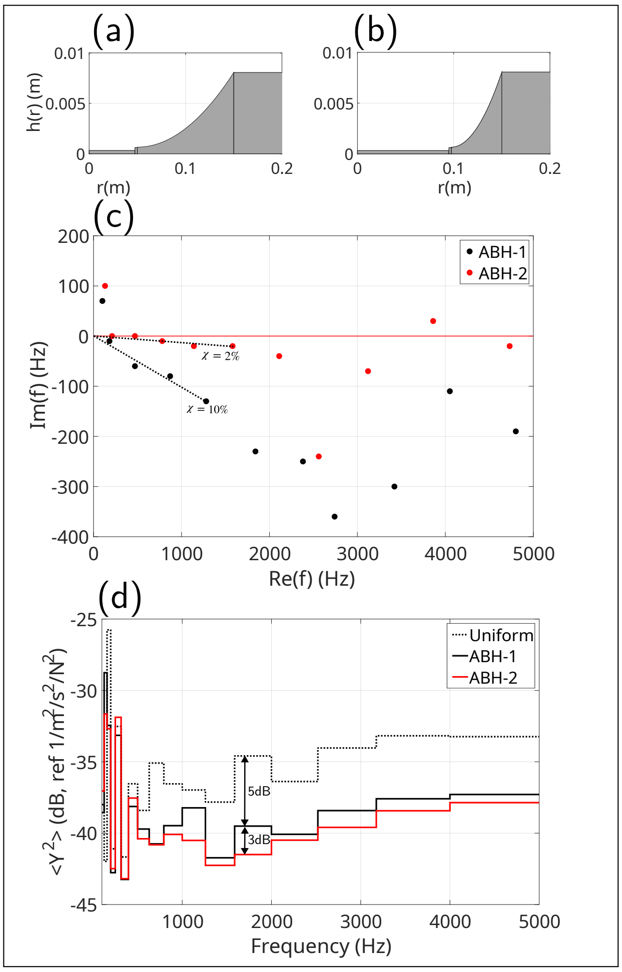

Figure 2. (a) ABH-1 profile with $a = 0.0475$m, $b = 0.05$m and $c = 0.15$m. (b) ABH-2 profile with $a = 0.095$m, $b = 0.0975$m and $c = 0.15$m. (c) Positions of zeros in the complex frequency plane for the two configurations at circumferential order $n = 0$. (d) Spatially averaged quadratic mobility for a plate with no ABH (Uniform), ABH-1 or ABH-2.

4. CONCLUSION

This study investigated the influence of local modes and their positioning in the complex frequency plane on the performance of Acoustic Black Holes (ABHs) embedded in sandwich panels. Using a semi-analytical wave-based model, the analysis demonstrated that an increase in modal density of the scatterer induces an increased number of zeros in the reflection coefficient. This leads, particularly when they are critically coupled, to enhance vibration attenuation. The comparison between ABH-1 and ABH-2 confirmed that a modified geometry can improve the absorption performance by increasing the proximity of the zeros to the real axis. These findings highlight the critical role of local dynamic behaviour in ABH design and provide key guidelines for optimising ABH performance. To create better ABH, careful consideration must be given to the study of local modes and the positioning of zeros, achieved through precise tuning of damping, geometry and materials when possible.

5. ACKNOWLEDGMENTS

This work is part of the MAMBO project, financially supported by the Direction Générale de l'Aviation Civile (DGAC), 2021-2025.

6. REFERENCES

[1] M. Mironov, "Propagation of a flexural wave in a plate whose thickness decreases smoothly to zero in a finite interval," *Sov. Phys. Acoust.*, vol. 34, pp. 318–319, 1988.

[2] V. V. Krylov, "Localized acoustic modes of a quadratic solid wedge," *Moscow University Physics Bulletin*, vol. 45, pp. 65–69, 1990.

[3] V. V. Krylov and R. E. T. B. Winward, "Experimental investigation of the acoustic black hole effect for flexural waves in tapered plates," *Journal of Sound and Vibration*, vol. 300, pp. 43–49, 2007.

[4] V. Krylov, "New type of vibration dampers utilising the effect of acoustic 'black holes'," *Acta Acustica united with Acustica*, vol. 90, pp. 830–837, 2004.

[5] V. B. Georgiev, J. Cuenca, F. Gautier, L. Simon, and V. V. Krylov, "Damping of structural vibrations in beams and elliptical plates using the acoustic black hole effect," *Journal of Sound and Vibration*, vol. 330, pp. 2497–2508, 2011.

[6] D. J. O'Boy, V. V. Krylov, and V. Kralovic, "Damping of flexural vibrations in rectangular plates using the acoustic black hole effect," *Journal of Sound and Vibration*, vol. 329, pp. 4672–4688, 2010.

[7] A. C. Nilsson and B. Liu, "Prediction of some vibro-acoustic properties of sandwich plates with honeycomb and foam cores," *The Journal of the Acoustical Society of America*, vol. 144, pp. 1600–1614, 2018.

[8] K. Ege, N. Roozen, Q. Leclère, and R. Rinaldi, "Assessment of the apparent bending stiffness and damping of multilayer plates; modelling and experiment," *Journal of Sound and Vibration*, vol. 426, pp. 129–149, 2018.

[9] A. Besse, P. O'Donoughue, O. Aklouche, F. Ablitzer, A. Pelat, and F. Gautier, "Vibration damping of fiberglass honeycomb sandwich panels based on the acoustic black hole effect," *Journal of Sound and Vibration*, vol. 604, p. 118991, 2025.

[10] J. Leng, V. Romero-García, A. Pelat, R. Picó, J. P. Groby, and F. Gautier, "Interpretation of the Acoustic Black Hole effect based on the concept of critical coupling," *Journal of Sound and Vibration*, vol. 471, pp. 115–199, 2020.

[11] D. J. O'Boy and V. V. Krylov, "Damping of flexural vibrations in circular plates with tapered central holes," *Journal of Sound and Vibration*, vol. 330, pp. 2220–2236, 2011.

[12] O. Aklouche, A. Pelat, S. Maugeais, and F. Gautier, "Scattering of flexural waves by a pit of quadratic profile inserted in an infinite thin plate," *Journal of Sound and Vibration*, vol. 375, pp. 38–52, 2016.

[13] D. Ross, E. Ungar, and E. K. Jr., "Damping of plate flexural vibrations by means of viscoelastic laminae," *Structural damping*, pp. 49–97, 1959.