

Numerical and Experimental Study of Sound Power Reduction Performance of Acoustic Black Holes in Rectangular Plates

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ABSTRACT

Global attenuation of structural velocities is one of the most effective approaches in order to reduce noise emitted by shell structures such as a car roof or aircraft fuselage panels. This global reduction can be achieved by the application of passive damping treatments like constraint layer damping on large fractions of the vibrating surface. The main disadvantage of this approach arises from the fact that it leads to increasing total cost and weight of the structure. To overcome this problem, acoustic black holes can be used to create locations with high vibration amplitudes and low bending waves velocity in order to dissipate the energy of structure borne sound by very limited application of damping treatments. Acoustic black holes are funnel shaped thickness reductions that attract sound radiating bending waves and allow a global vibration reduction by an acceptable use of additional damping. This paper presents the results of a numerical and experimental study of acoustic black holes located on a rectangular plate. The presented work is focused on the influence of size, position and number of acoustic black holes on the acoustic performance. A large number of different configurations of these parameters is studied by finite-element-analysis and evaluated in terms of vibration amplitude and sound power level. In order to confirm simulation results the most efficient configuration is implemented in laboratory setup and characterized in terms of vibrational and acoustic performance.

CITATION: Unruh, O., Blech, C., and Monner, H., "Numerical and Experimental Study of Sound Power Reduction Performance of Acoustic Black Holes in Rectangular Plates," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(3):2015, doi:10.4271/2015-01-2270.

INTRODUCTION

Due to the current ecological and economic development, the use of high performance lightweight structures in all fields of transportation is constantly increasing. Low mass and high stiffness of these structures lead to a lower coincidence frequency, higher vibration amplitudes and subsequently to increased noise levels. In order to maintain or even increase the acoustic comfort of passenger aircrafts, automobiles and trains, passive or active noise and vibration reduction technologies can be used. Active strategies such as active structural acoustic control (ASAC) show promising potentials in lower frequency range [1], but require high hardware and costs effort at the current stage of development. The result is that their current use is limited to some specific applications. Passive damping treatments such as constraint layer damping (CLD) or in composite layup embedded elastomer or thermoplastic layers are much more common due to their simplicity, robustness and good performance in a wide frequency range. Nevertheless, to achieve considerable noise reduction especially in lower frequency range, large fractions of the surface of vibrating structures must be treated by passive damping.

As a result total weight and cost of these high performance lightweight structures can significantly increase. One of the interesting and promising solutions for this problem is a novel, passive concept of acoustic black holes (ABH). The idea behind this concept is to slowdown sound radiating bending waves in the ABH and to dissipate them there by a very limited use of damping materials. ABHs are characterized by continuous decrease of structural stiffness, which "focuses" vibration energy inside the black hole and leads to high local velocity amplitudes that can be efficiently reduced by passive damping treatments like CLD.

First studies dealing with the focusing of structure-borne sound waves are published by Krylov [2], [3], [4] and other authors [5], [6] in the 1990s. In these works beam structures with wedge-shaped free edges are studied. It is shown that with optimal definition of the geometric parameters of the wedge, the reflection coefficient may become zero. This means that instead of standing waves, travelling waves dominate the dynamics of the vibrating beam. Furthermore, it is demonstrated, that additional damping inside the wedge

significantly reduces global vibration amplitudes of the beam. Ten years ago these elastic wedges, absorbing vibration energy were officially introduced by Krylov [7] as “acoustic black holes”.

Some recent publications [8], [9], [10] address mainly the vibration reduction of plate structures using a two-dimensional black hole. In these papers some experimental results for elliptical plates with free boundary conditions are shown and it is demonstrated that significant global vibration reduction can be achieved using the ABH approach. In the latest studies [11], [12] an array of multiple black holes with equal geometrical parameters is used to investigate the impact on structural velocities and radiated sound power of vibrating plates.

As it was mentioned in the literature review, most available studies [8] - [10] deal with elliptic plates with free boundary conditions, affected by a single ABH. In cases where multiple black holes are investigated, the regular array of equal ABHs is considered. For that reason, the work presented in this paper is focused on the numerical and experimental investigation of the attenuation performance of ABHs with varying size, position and number in clamped rectangular plates. This type of boundary condition is more common than free support and it corresponds to many technical applications. The lower frequency range below 2000Hz stands in focus of the study. The reason is the fact, that in particular for lower frequencies a significant amount of damping treatments is required in order to achieve considerable attenuation of vibration amplitudes and radiated sound power. Furthermore, by the definition of the ABH the attention is paid to feasibility in practical applications. This means that ABH addressed in this study have a non-zero thickness at its center in order to provide residual stiffness and strength.

The effect of ABH's with varying size, position and number on structural velocities and radiated sound power is investigated by finite element method (FEM) and validated for a selected parameter set by experiments in anechoic chamber. It is shown that the size of ABH has the most important influence on sound reduction performance in the addressed frequency range.

NUMERICAL MODELLING

ABH's Design Parameters

In this part of the paper the basic principle of ABH and the numerical model used for the parameter study are discussed. Most important design parameters that affect the sound and vibration reduction performance are defined for later considerations.

The propagation velocity of bending waves is a function of elastic modulus E , thickness h , density ρ , poisson's ratio ν and frequency ω and is given by following equation:

$$c = \left(\frac{Eh^2}{12\rho(1-\nu^2)} \right)^{1/4} \sqrt{\omega} \quad (1)$$

As it has already been mentioned, the physical principle of ABH is based on the continuous reduction of structural bending stiffness, for example by a decreasing thickness h of the vibrating surface.

Corresponding to equation (1) it leads to a reduction of propagating velocities and as a consequence to increasing vibration amplitudes A . Following Figure 1 exemplarily shows the basic principle of ABH and most important design parameters.

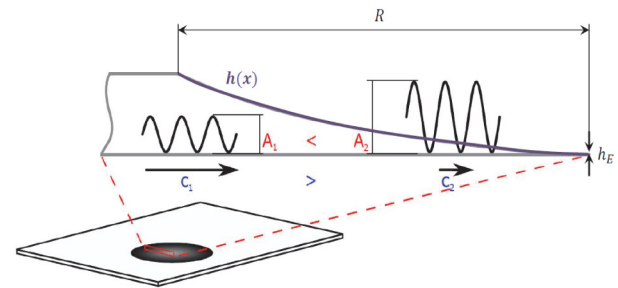


Figure 1. Basic principle of acoustic black hole in rectangular plate with main design parameters

The reflection coefficient of the ABH, or in other words, its capability to attract vibration energy is dependent on parameters such as radius R , minimal thickness h_E and the shape function $h(x) = \varepsilon x^m$. The shape function can be linear with $m = 1$, quadratic with $m = 2$ or any other order m . In [13] the influence of important parameters on reflection coefficients is given. To summarize these results it can be noticed that lowest frequency dependent reflection coefficients and highest ABH performance can be achieved by maximizing the radius R and order of the shape function m and minimizing the thickness h_E .

Unfortunately, the realization of ABH's according to these criteria is difficult for the majority of practical applications. In most cases the order of the thickness function should not be too high to provide a certain degree of structural residual strength inside the ABH. Furthermore it is difficult to reduce the minimal thickness to zero, because of manufacturing or, for example, sealing requirements. For that reason, this study focuses on ABH configurations that build a compromise between high sound power reduction performance and technical feasibility. Therefore, the order of the shape function $m = 2$ is defined for all considered configurations. The minimum thickness of $h_E = 1\text{mm}$ is chosen to have one fifth of the initial 5mm thickness of the panel.

It is well known from the literature [8] that continuous stiffness reductions, shown in Figure 1, will become an ABH only in combination with additional damping. In order to ensure comparability of different ABH configurations, constant amount of additional damping material with radius of 100mm is used. All plates in this study are treated by constraint layer damping of type 3M® CLD Damping foil 2552, which consist of a 0.25mm thin aluminum foil and acrylic viscoelastic polymer layer with 0.13mm thickness.

Finite-Element-Model

In order to investigate a large number of different configurations a suitable FE-model must be provided. This model should be able to deliver accurate predictions of overall vibration and sound power reduction with low computational effort.

In the literature [12] it is common to use 3D models to analyze structures with ABH. These models are probably able to reproduce the dynamics of ABH's more in detail, but require a significant

amount of computation time. Due to the fact, that large number of configurations should be analyzed, an alternative approach using 2D shell elements is investigated in the first part of the numerical study. The accuracy of the simplified modelling using shell elements is demonstrated in comparison to the 3D solid model.

FE-modeling of the reference plate and plates with ABH's is conducted using ANSYS®-Software version 15.0. The aluminum plate with $900\text{mm} \times 600\text{mm} \times 5\text{mm}$ dimensions is discretized by Shell281 elements (2D) with sufficient rate to enable simulation of harmonic response up to 2000 Hz. This rate is verified considering deviations of eigenfrequencies of high vibration modes, which should be lower than 5%. In order to compare the results of computationally efficient shell model to a solid model, Solid186 elements are used in the second numerical simulation. In order to improve accuracy of the reference solid model finer meshing is used, which results in 4 times more elements than in case of shell model. Figure 2 exemplarily shows the FE-model of a plate with a single ABH located at its center, discretized by Shell181 elements.

The boundary condition of the plate is modeled according to the mixed type of support in experimental setup. Despite the fact, that in laboratory setup massive steel frame is used to support tested plates, the boundary condition is not perfectly clamped. For that reason the elastic support is defined using a number of the elements with variable stiffness, located on the plates boundaries. Outer nodes of these elements are fixed in all translational and rotational degrees of freedom. In contrast, the inner nodes are free in rotational displacements. The stiffness of the boundary elements is estimated empirically by initial measurements of a reference plate without ABH with regard to minimal deviations of eigenfrequencies between experimental and simulation results. This approach minimizes modeling uncertainties and improves the prediction of structural velocities and radiated sound power levels.

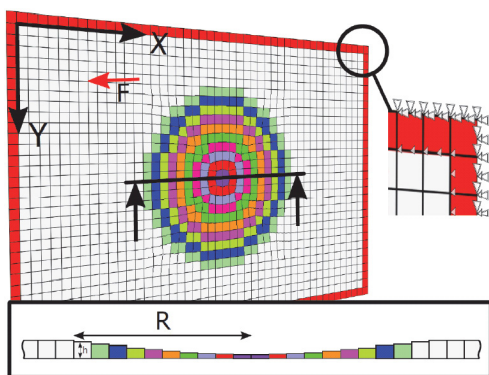


Figure 2. Finite element model of the plate with acoustic black hole using Shell281 elements

The geometry of the ABH is modelled by stepwise variation of the thickness in particular shell elements, according to the shape function with the order of $m = 2$, also shown in the lower part of Figure 2. Different colors in this illustration indicate sections of varying thickness of the corresponding shell elements. This simple approach allows a straight forward definition of multiple ABH's with different

size and position. At the same time it provides a sufficient accuracy in prediction of impacts on structural velocities and radiated sound power, as it is shown in the next section.

For the estimation of structural loss factors inside the black hole the model of Ross-Ungar-Kerwin, also used in [13], is applied. This model allows the calculation of total loss factors of layered systems with variable thickness. Due to the decreasing thickness of the base structure (low loss factor) and constant thickness of additional damping layer (high loss factor) inside the ABH, the total loss factor of the layup must continuously increase towards the center of the black hole. In order to calculate this thickness-dependent damping, structural loss factors of the base structure and additional damping layer are required. Initial measurements with a simple clamped aluminum beam, affected by additional damping, were used for the empirical estimation of these loss factors. Due to the fact that relative comparison of different ABH configurations is addressed in this study, frequency independent loss factors are. Further studies will deal with the improved prediction of the absolute vibration reduction using ABH and include frequency-dependent loss factors.

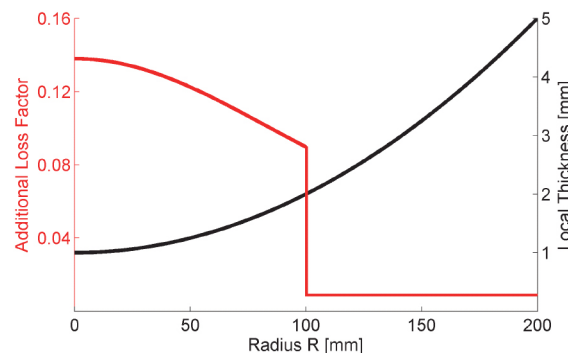


Figure 3. Thickness dependent loss factors inside the ABH

The resulting thickness-dependent loss factors are defined for corresponding shell elements inside the ABH, according to the color scheme, shown in Figure 1. Figure 3 shows this thickness dependent total loss factors for the ABH with $R = 200\text{mm}$ and 100mm additional damping. For the blank aluminum structure a constant loss factor of 0.01 is defined. It can be seen that the loss factor increases abruptly at the boundary of additional damping treatment by $R = 100\text{mm}$. Further increase of the loss factor is continuous with decreasing thickness of the plate and increasing fraction of damping material inside the ABH.

Simulation Results

In order to estimate the dynamic response of the plates in the bandwidth of 10 - 2000 Hz, a full-harmonic analysis is performed with frequency increment of 1 Hz. A point force excitation of $F = 1\text{N}$, located at $x = 135\text{mm}$ and $y = 90\text{mm}$, is used in order to allow comparability to the referenced publications and to enable a manageable and reliable realization in the experimental setup. The location of excitation point is chosen with regard to the high controllability of the majority of structural modes in the targeted frequency bandwidth. The point force is fixed but the effect of varying excitation position can be correlated to the variation of the relative distance to the ABH, which is investigated in this paper.