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ADDITIVE ACOUSTIC BLACK HOLES TO SUPPRESS ACOUSTIC ENERGY IN A RESONANT CAVITY

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ABSTRACT

Conventional acoustic black hole (ABH) periodic arrays have been extensively investigated for their effective vibration reduction performance. However, embedded ABHs are not suitable in many problems due to the loss of structural stiffness. That is avoided with additive ABHs, which have also proven to be very valuable in suppressing vibrations. In this paper, we explore their potential for noise reduction. An additive ABH plate is connected to a uniform base plate through the corners, and then coupled to a resonant cavity filled with air. In the first configuration, the base plate is facing the cavity, while in the second one, the ABH plate is oriented towards the cavity. It is shown that, in the first configuration, the vibrations in the base plate are significantly reduced and, consequently, the acoustic energy in the cavity decreases as well. For the second configuration, moreover, the coupling force at the fluid-structure interface diminishes, so that the acoustic energy in the cavity is even lower.

Keywords: *acoustic black hole, fluid-structure interaction, cavity*

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1. INTRODUCTION

Acoustic black holes (ABHs) are very efficient in reducing the vibration and noise from thin-walled structures. However, due to the thickness reduction, the structural stiffness may weaken, limiting its applicability in many situations. Recently, additive ABHs have become a promising method to avoid such limitation. The idea is to treat the ABH components as external vibration absorbers, as done in beams [1–3] and plates [4–6].

However, most existing studies on additive ABHs focus on vibration reduction in structures. In this paper, we explore their potential for suppressing noise within a closed sound cavity, which is interesting for applications such as high-speed trains and airplanes. We consider a periodic ABH plate, with the design proposed in [7], which can be easily connected to the host plate through the corners. When the composite plate is assembled, we can couple it to the resonant cavity. Here, we consider 2 cases: i) the host plate faces the cavity (Config. I) and ii) the ABH plate faces the cavity (Config. II). It will be shown how the acoustic energy in the cavity isreduced when an external force is applied to the plate.

2. MODELING

As shown in Fig. 1a, the composite plate in Fig. 1b is placed on the top surface of a rectangular cavity, with dimensions $L_x \times L_y \times L_x = 0.8 \times 0.6 \times 0.5$ m^3, filled with air with density $\rho_0 = 1.21$ kg/m^3 and sound speed $c_0 = 343$ m/s. The host plate (see Fig. 1b), is made of steel with Young modulus $E_p = 210$ GPa, density

$\rho_p = 7800$ kg/m^3, Poisson ratio $\nu_p = 0.3$, and a thickness of $h_p = 5$ mm. On the other hand, the geometry details of the additive ABH plate are depicted in Figs. 1c and 1d, where the thickness reduces from $h_a = 3$ mm to $h_c = 0.2$ mm following the power laws $h = \frac{h_a - h_c}{r_{abh}^m}|x|^m + h_c$ and $h = \frac{h_a - h_c}{r_{abh}^m}|y|^m + h_c$ in the x and y directions, respectively, where $r_{abh} = 9$ cm, and $m = 2$. The ABH plate is also made of steel. The yellow component in Figs. 1c and 1d is the damping layer, with thickness $h_d = 2$ mm, and half width $r_d = 4.5$ cm. The damping layer has Young modulus $E_d = 5$ GPa, density $\rho_d = 950$ kg/m^3, Poisson ratio $\nu_d = 0.33$, and loss factor $\eta_d = 0.5$.

To characterize the coupled system, we employ the Gaussian expansion method (GEM) [8, 9] to model the host plate, the ABH plate (plus the damping layer) and the resonant cavity, separately. This results in the uncoupled equation of motion,

$$(\boldsymbol{K} - \omega^2 \boldsymbol{M})\boldsymbol{A} = \boldsymbol{0}, \tag{1}$$

where $\boldsymbol{K} = \mathrm{diag}(\boldsymbol{K}_p, \boldsymbol{K}_a, \boldsymbol{K}_c)$ and $\boldsymbol{M} = \mathrm{diag}(\boldsymbol{M}_p, \boldsymbol{M}_a, \boldsymbol{M}_c)$ are respectively the stiffness and mass matrices, and $\boldsymbol{A}$ is the coefficent vector of Gaussian functions.

Next we consider the coupling between substructures, and discretize the constraints for $\boldsymbol{A}$ also with Gaussian functions. Finding the the nullspace basis of the constraint matrix we get $\boldsymbol{A} = \boldsymbol{Z}\boldsymbol{k}$, where $\boldsymbol{Z}$ contains the basis vectors and $\boldsymbol{k}$ is the superposition coefficient vector [10]. Taking this relation back to Eqn. (1) results in the following coupled equation of motion,

$$(\boldsymbol{Z}^\top \boldsymbol{K}\boldsymbol{Z} - \omega^2 \boldsymbol{Z}^\top \boldsymbol{M}\boldsymbol{Z})\boldsymbol{k} = \boldsymbol{0}. \tag{2}$$

To obtain Eqs. Eqn. (1) and Eqn. (1) we have used a displacement formulation to describe all problem unknowns. Details can be found in [9].

3. RESULTS

We apply a unitary force on the plate and compute the total acoustic energy of the cavity, $E_c = \frac{1}{2\rho_0\omega^2}\int \nabla^2 p \mathrm{d}V + \frac{1}{2\rho_0 c_0^2}\int p^2 \mathrm{d}V$, E_c (dB)$=$ $10\lg(E_c \times 10^{12})$. To better inspect the results, we have converted the spectrum into 1/3 octave bands, see Fig. 2. It can be observed that when the cavity is coupled to the bare plate (i.e., the host plate alone), the cavity energy remains the highest above 150 Hz. When the ABH plate is placed on the top of the host one (Config. I), the sound energy is substantially reduced over the entire frequency range of interest, particularly at high frequencies. The reason for that is that the vibration of the host plate is transferred to the ABH plate where it is efficiently dissipated by the damping layer, resulting in less energy transmitted into the cavity from the

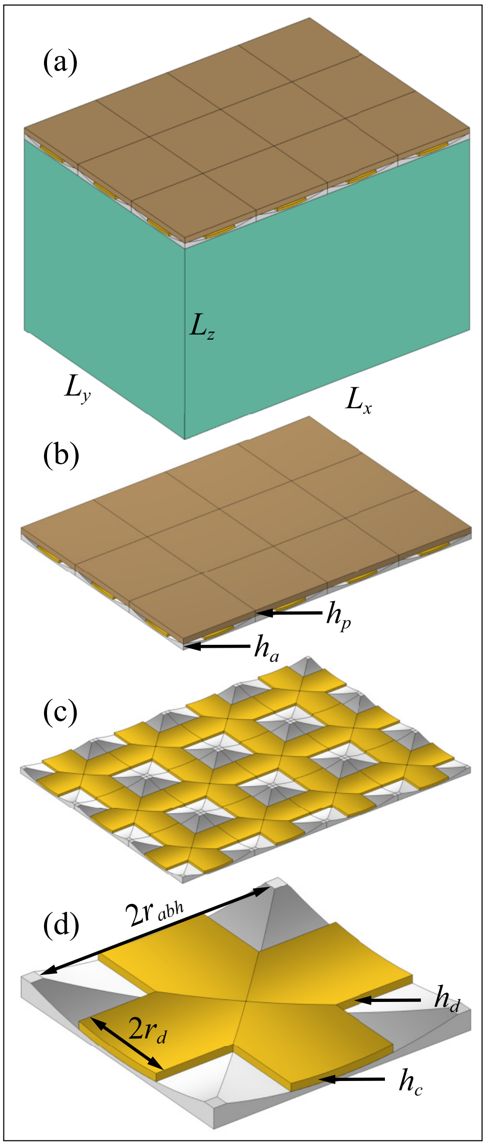

Figure 1. Geometry of the model. (a) The coupled system. (b) The composite plate. (c) The ABH plate with a damping layer (yellow region). (d) A single ABH cell.

host plate. If we change the situation and let the ABH plate face the cavity (Config. II) it is observed that the energy within the cavity is further reduced for $f > 200$ Hz. This is because the radiation efficiency of the ABH plate is lower than that of the uniform plate [11], resulting in a smaller coupling coefficient of the fluid-structure interaction [12]. .

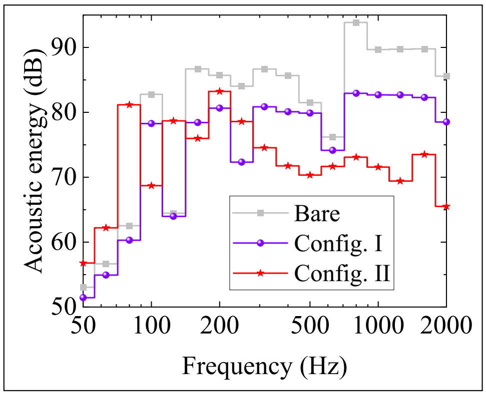

Figure 2. Comparison of the acoustic energy of the cavity of the bare host plate, and the composite plate. Config. I: The host plate is facing the cavity. Config. II: the ABH plate is facing the cavity.

4. CONCLUSIONS

In this paper, we have investigated the potential of additive acoustic black holes (ABHs) to reduce sound energy in resonant cavities. To this end, a semi-analytical model has been developed, incorporating coupling conditions using the nullspace method. Two configurations have been compared. When the uniform host plate faces the cavity, noise is efficiently suppressed due to vibration damping in the ABH plate. Alternatively, when the ABH plate faces the cavity, the sound energy inside the cavity is further reduced because the radiation efficiency decreases, leading to a lower coupling coefficient.

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