We are IntechOpen, the world's leading publisher of Open Access books Built by scientists, for scientists

6,300

Open access books available

172,000

International authors and editors

190M

Downloads

Our authors are among the

154

Countries delivered to

TOP 1%

most cited scientists

12.2%

Contributors from top 500 universities

WEB OF SCIENCE™

Selection of our books indexed in the Book Citation Index
in Web of Science™ Core Collection (BKCI)

Interested in publishing with us?
Contact book.department@intechopen.com

Numbers displayed above are based on latest data collected.
For more information visit www.intechopen.com

Periodic Acoustic Black Holes to Mitigate Sound Radiation from Cylindrical Structures

Jie Deng and Nansha Gao

Abstract

Acoustic black holes (ABHs) achieved by progressively diminishing structural thickness have been proved a very efficient approach for wideband vibration reduction, sound suppression, energy harvesting, and wave manipulation. In this chapter, the focus is placed on mitigating the sound emitted from cylindrical shells with embedded ABHs. In the applications of aeronautics, astronautics, and underwater vehicles, cylindrical shells are very common yet the vibroacoustic problems in such structures are very challenging. Even the researches on ABHs for straight beams and flat plates are boosting in recent years, the ABH effect is unclear for curved structures thus it deserves further investigations to push forward their applications. Since cylindrical shells are usually long in, for example, airplanes and rockets, periodic ABHs are designed to alleviate the acoustic emission from them. The Gaussian expansion method (GEM) is employed to recover the vibration field on the cylinder and, based on that, the sound radiation model is developed to determine the emitted sound power level (SWL). The band gaps (BGs) are shown for infinite periodic ABH shells, followed by the vibroacoustic level for a finite periodic shell. Particularly, axial stiffeners are introduced and the influences of their quantity and width are carried out.

Keywords: acoustic black holes, acoustic radiation, cylindrical shells, band gaps, stiffeners

1. Introduction

An acoustic black hole (ABH) is usually realized by reducing structural thickness following a power law $h(x) = \varepsilon x^m$ ($m \geq 2$), as illustrated in **Figure 1**. When an incident flexural wave impinges at the edge of the ABH, its wavelength and wave speed get gradually decreased. Meanwhile, the wavenumber is however increased and the amplitude is intensified. In the ideal case, the thickness at the ABH tip decays to zero, where the wave velocity vanishes as well, such that the traveling time to its center becomes infinite. In other words, the wave will never reach the tip. In analogy with cosmology, the termination behaves like a “Black Hole” in which nothing can escape from it. This is the story of how the term “Acoustic Black Hole” was coined [1]. However, in real applications, generally, an ABH is imperfect. Namely, there exists a truncation near the ABH tip, which results in obvious reflection because of the residual thickness [2]. Fortunately, attaching a thin viscoelastic

Figure 1.

Illustration of ABH effect: the incident wave is localized in the ABH tip as it propagates toward the ABH.

layer at the ABH tip, where the energy is highly concentrated, can alleviate this problem [3]. Recently, constrained viscoelastic layers have been suggested to enhance the damping effects, by changing the normal tensile and compressive deformation of damping material into the shear one [4].

It has been shown that the ABHs are very efficient to reduce vibration from straight beams [5, 6] and flat plates [7, 8]. The shapes and lengths of the damping layers have been extensively investigated [4, 9]. Also, different ABH designs have been proposed for the purpose of enhancing energy consumption [7, 10–12]. Thanks to the vibration reduction because of highly efficient damping, the sound radiation from ABH structures is accordingly reduced [13]. Not only that, recent studies have shown that the ABHs can also impair the sound radiation efficiency because of the thickness reduction [14, 15]. Particularly, for cavity noise, the ABH profile can destroy the coupling strength between structural and acoustic modes, which is the third underlying mechanism of the ABHs for reducing room noise [16, 17]. Furthermore, periodic and gradient-index ABHs are investigated for steering waves [18–20]. It is also worthwhile mentioning that ABHs can also enhance energy harvesting due to wave focalization [21], using piezoelectric layers rather than viscoelastic ones.

The state-of-the-art reviewed above are mainly centered on flat structures. However, in aeronautics, astronautics, and underwater vehicles, cylindrical shells are very common. There, the vibroacoustic problems are very critical to determine their comfortability and safety, thus it is very demanding to apply ABH features on them. Our previous efforts have been focused on the vibration of cylindrical beams [22] and shells [23, 24], together with the sound radiation from a finite cylinder [25]. In this chapter, we continue this topic but analyze the sound radiation from periodic ABH shells.

As shown in **Figure 2a**, an infinite periodic ABH shell is considered, with each unit cell having radius R and length L_{cell} . An ABH plus a thin viscoelastic layer (see the green layer) is laid in the center of the cell. The geometries of the ABH and the damping layer are detailed in **Figure 2b**. Here, the profile of the ABH is defined by $h(x) = \varepsilon|x|^m + h_c$, where $\varepsilon = (h_{uni} - h_c)r_{abh}^{-m}$ stands for the ABH slope and, r_{abh} , h_c , and m respectively are the ABH radius, residual thickness, and order. We will characterize the band gaps (BGs) for infinite periodic ABH shells and their dependence on the ABH geometry. Next, a finite periodic ABH shell containing five cells will be characterized, under a ring excitation acting at x_f (see **Figure 2c**). The translational springs k_i , $i = 1, 2$ and rotational ones p_i , $i = 1, 2$ are intended for boundary conditions (distributed circumferentially). As one could expect, the appearance of ABHs weakens the stiffness of the whole structure. This may deteriorate the structural problems. To partially solve this, we can introduce N stiffeners for each cell (see **Figure 2d**), with each width being W . The effects of the stiffeners will also be investigated at the end of this chapter.

Figure 2. Illustration of the periodic ABH cylindrical shell. (a) A unit ABH cell having a thin damping layer (green). (b) The geometrical detail of the ABH profile. (c) A supported finite ABH cell having five cells under a ring excitation. (d) Illustration of the stiffener for enhancing the structure (with N stiffeners in the circumferential direction).

2. Vibration characteristics of infinite periodic ABH shells

2.1 Gaussian expansion for the vibration of infinite periodic ABH shells

The goal of this section is that of developing a semi-analytical model for characterizing the vibration of the infinite periodic ABH shell. Let us consider three variables, u , v , and w , which are the displacements in the axial, circumferential, and radial directions, respectively. They can be decomposed by

$$u(x, y, t) = \sum_i a_i(t) \psi_i(x, y) = \mathbf{a}^\top \boldsymbol{\psi} = \boldsymbol{\psi}^\top \mathbf{a}, \quad (1)$$

$$v(x, y, t) = \sum_i b_i(t) \xi_i(x, y) = \mathbf{b}^\top \boldsymbol{\xi} = \boldsymbol{\xi}^\top \mathbf{b}, \quad (2)$$

$$w(x, y, t) = \sum_i c_i(t) \varphi_i(x, y) = \mathbf{c}^\top \boldsymbol{\varphi} = \boldsymbol{\varphi}^\top \mathbf{c}, \quad (3)$$

where

$$\mathbf{a} = \hat{\mathbf{A}} \exp(j\omega t), \mathbf{b} = \hat{\mathbf{B}} \exp(j\omega t), \mathbf{c} = \hat{\mathbf{C}} \exp(j\omega t), \quad (4)$$

are the coefficient vectors to be determined, while $\boldsymbol{\psi}$, $\boldsymbol{\xi}$, and $\boldsymbol{\varphi}$ are the shape function vectors with entries $\psi_i(x, y)$, $\xi_i(x, y)$, and $\varphi_i(x, y)$, respectively. With the aid of Kronecker product, the vectors $\boldsymbol{\psi}$, $\boldsymbol{\xi}$, and $\boldsymbol{\varphi}$ can be factorized as

$$\boldsymbol{\psi}(x, y) = \boldsymbol{\alpha}^\psi(x) \otimes \boldsymbol{\beta}^\psi(y), \quad (5)$$

$$\boldsymbol{\xi}(x, y) = \boldsymbol{\alpha}^\xi(x) \otimes \boldsymbol{\beta}^\xi(y), \quad (6)$$

$$\boldsymbol{\varphi}(x, y) = \boldsymbol{\alpha}^\varphi(x) \otimes \boldsymbol{\beta}^\varphi(y), \quad (7)$$

where $\boldsymbol{\alpha}^{(i)}$ ($i = \psi, \xi, \varphi$) are column vectors containing basis functions depending on the x direction yet $\boldsymbol{\beta}^{(i)}$ ($i = \psi, \xi, \varphi$) the ones on the y direction. To accurately capture the localized displacements in the ABH portion, the entries of $\boldsymbol{\alpha}$ and $\boldsymbol{\beta}$ are selected as Gaussian functions

$$\alpha_i(x) = 2^{s_x/2} \exp \left[-\left(2^{s_x}x - q_{xi}\right)^2/2 \right], \quad (8)$$

$$\beta_i(y) = 2^{s_y/2} \exp \left[-\left(2^{s_y}y - q_{yi}\right)^2/2 \right], \quad (9)$$

in which s_x and s_y are the scaling parameters, while q_x and q_y are the translational parameters, in the x and y directions, respectively. For brevity, readers are referred to our previous works [4, 26] to thoroughly comprehend the detailed process of how to produce acceptable basis.

Unlike finite and flat plates, bear in mind that the displacements, on the one hand, must be continuous in the circumferential direction ($\mathbb{C}^0$ and $\mathbb{C}^1$ are enough, see [22]), which requires

$$\mathbb{C}^0 : u(x, -\pi R) = u(x, \pi R), \quad v(x, -\pi R) = v(x, \pi R), \quad w(x, -\pi R) = w(x, \pi R), \quad (10)$$

$$\mathbb{C}^1 : \frac{\partial u}{\partial y}(x, -\pi R) = \frac{\partial u}{\partial y}(x, \pi R), \quad \frac{\partial v}{\partial y}(x, -\pi R) = \frac{\partial v}{\partial y}(x, \pi R), \quad \frac{\partial w}{\partial y}(x, -\pi R) = \frac{\partial w}{\partial y}(x, \pi R). \quad (11)$$

On the other hand, Bloch-Floquet periodic boundary conditions must be imposed in the axial direction for a unit cell ($\mathbb{C}^0$ and $\mathbb{C}^1$, too), namely

$$\mathbb{C}^0 : u(0, y) = u(L_{cell}, y)\lambda, \quad v(0, y) = v(L_{cell}, y)\lambda, \quad w(0, y) = w(L_{cell}, y)\lambda, \quad (12)$$

$$\mathbb{C}^1 : \frac{\partial u}{\partial x}(0, y) = \frac{\partial u}{\partial x}(L_{cell}, y)\lambda, \quad \frac{\partial v}{\partial x}(0, y) = \frac{\partial v}{\partial x}(L_{cell}, y)\lambda, \quad \frac{\partial w}{\partial x}(0, y) = \frac{\partial w}{\partial x}(L_{cell}, y)\lambda, \quad (13)$$

where $\lambda = \exp(jk_x L_{cell})$, k_x is the axial wavenumber in the irreducible Bernoulli zone [18]. Via implementing the reconstruction process in [18], the continuity in the circumferential direction (Eqs. (10) and (11)) and the periodicity in the axial one (Eqs. (12) and (13)) can be satisfied.

Provided the kinetic energy, K , and the potential one, U , are presented in terms of u , v , and w [27], the Lagrangian of the whole system can be built

$$L = K - U = \frac{1}{2} \dot{\mathbf{q}}^\top \mathbf{M} \dot{\mathbf{q}} - \frac{1}{2} \mathbf{q}^\top \mathbf{K} \mathbf{q}, \quad (14)$$

where

$$\mathbf{q} = [\hat{\mathbf{A}}^\top, \hat{\mathbf{B}}^\top, \hat{\mathbf{C}}^\top]^\top \exp(j\omega t) \equiv \hat{\mathbf{Q}} \exp(j\omega t), \quad (15)$$

represents the assembled undetermined time-dependent vector related to admissible shape functions. $\mathbf{M}$ represents the mass matrix and $\mathbf{K}$ the stiffness one.

Finally, applying the Euler–Lagrange equations $\partial_t(\partial_{\dot{q}}L) - \partial_qL = \mathbf{0}$ to Eq. (14) yields the equations of motion in the frequency domain,

$$(-\omega^2\mathbf{M} + \mathbf{K})\hat{\mathbf{Q}} = \mathbf{0}, \tag{16}$$

whose solution permits calculating the dispersion curves and eigenmodes for infinite periodic ABH shells.

2.2 Numerical results

2.2.1 Dispersion curves and band gaps

The dispersion curves of an ABH cell, whose geometry and material are detailed in **Table 1**, have been carried out and plotted in **Figure 3a**. For the purpose of validation, the result from a reference FEM model has also been included in **Figure 3a**. From the figure, it is seen that the two results are very close at each

Geometry parameters	Material parameters
$m = 2.8$	$\rho = 7800 \text{ kg/m}^3$
$L_{cell} = 1 \text{ m}$	$E = 210 \text{ GPa}$
$R = 5 \text{ m}$	$\eta = 0.005$
$h_{uni} = 0.03 \text{ m}$	$\nu = 0.3$
$r_{abh} = 0.25 \text{ m}$	
$\epsilon = 1.3581 \text{ m}^{-1.8}$	$\rho_v = 950 \text{ kg/m}^3$
$h_c = 0.002 \text{ m}$	$E_v = 5 \text{ GPa}$
$r_v = 0.125 \text{ m}$	$\eta_v = 0.5$
$h_v = 0.008 \text{ m}$	$\nu_v = 0.3$

ρ , shell density; ρ_v , damping layer density; E , shell young modulus; E_v , damping layer young modulus; η , shell loss factor; η_v , damping layer loss factor; ν , shell Poisson ratio; ν_v , damping layer Poisson ratio.

Table 1.
Geometry and material parameters of the ABH cylindrical shell.

Figure 3.
(a) Dispersion curves together with band gaps calculated with GEM and FEM. (b) Transmission from the left end to the right for a finite shell having five cells with and without damping layers, carried out with the GEM.

wavenumber, indicating the correctness of the present GEM model. Most importantly, four-band gaps (BGs) are observed within 1000 Hz, in which the first one is very small. Using the GEM without axial periodic boundary conditions for a finite shell having five cells, we can compute the transmission from one end to the other. As shown in **Figure 3b**, the transmission is very low at BG frequencies. For the shell without damping layers (undamped), the transmission is strong in the passbands. However, this situation can be ameliorated after implementing the damping layers, with a maximum reduction of up to ~ 25 dB.

To reveal the mechanism of the BGs, we have computed the first six eigenmodes of the unit cell, at wavenumber $k_x = 0$ for λ . As illustrated in **Figure 4**, for most orders, the vibration is very strong in the ABH area. While this is not the case for the 1-st order (see **Figure 4a**). This is because of the ring frequency (173 Hz), below which the cylindrical shell is almost not vibrating in the radial direction. However, for the 2-nd to 6-th orders (see **Figure 4b–f**), the wave is gradually concentrating in the ABH portion, belonging to the locally resonant effect. That is, the BGs shown in **Figure 3** are locally resonant ones, similar to the periodic ABH beams reported in [28].

2.2.2 Parametric analysis: effects of the ABH order, central thickness, and radius

The ABH profile is generally controlled by three parameters, m , h_c , and r_{abh} , which represent the ABH order, central thickness, and radius, respectively. It is worthwhile testing how these parameters affect the BGs.

Let us first look at the influence of the ABH order. As shown in **Figure 5a**, the 2-nd BG starts to gradually decrease as m increases. The 3-rd BG is however distinctive because the BG inverses near $m = 3$, then the upper bound almost keeps still but its lower bound drops. For the 4-th one, the width of the BG first becomes larger then turns out smaller as m goes up. In general, the total width of the four BGs almost remains the same, but they will be more compact and converge to lower frequencies. Note that we have also included the changes of the BGs for the ABH shell with a damping layer (damped). It is seen that the phenomenon is close to the

Figure 4. The first 6 eigenmodes for infinite periodic cells at wavenumber $k_x = 0$. The shaded area stands for the ABH portion. (a)-(f) respectively correspond to the 1-st to the 6-th orders.

Figure 5. Band gaps changing with (a) the ABH order, (b) the central thickness, and (c) the ABH radius.

former case. Due to the added stiffness of the layer, the BGs however occur at higher frequencies.

Different from the ABH order, the BGs are very sensitive to the central thickness. As graphed in **Figure 5b**, generally, the BGs become very narrow as the central thickness increases from 0^+ to $h_{uni} = 0.03$ m. It seems that the central thickness of the ABH needs to be small enough for wide BGs, whose local resonance is more significant. However, it is not realistic because truncation always exists. Moreover, we observe that for the 4-th BG there is an inflection point near $h_c = 0.002$ m, indicating that the optimal thickness can be found in this range. For comparison, the effects of the damping layer are also characterized in **Figure 5b**, but it seems not important to the BGs. However, for very small r_{abh} the added mass of the damping layer is more dominant than its added stiffness. This is why the BGs for the damped shell are lower than those for the undamped shell.

Finally, the effects of the ABH radius are computed and illustrated in **Figure 5c**. We consider the ABH radius varying from 0 to $L_{cell}/2 = 0.5$ m. From the figure, we can see that the BGs move to low-frequency range very fast as r_{abh} grows. Particularly, the BGs are prone to inverse when the radius is not very large ($r_{abh} < 0.4$ m). The 2-nd BG inverses at $r_{abh} = 0.14$ m, while the 3-rd BG has double inversions respectively at $r_{abh} = 0.1$ m and $r_{abh} = 0.32$ m, whereas the 4-th BG contains at least two inversions at $r_{abh} = 0.22$ m and $r_{abh} = 0.37$ m. When $r_{abh} > 0.4$ m, the BGs become stable and progressively move to lower frequencies. Again, the influence of the damping layer is also characterized in **Figure 5c**. On the one hand, it is inspected that the damping layer has significance merely for a large ABH radius. On the other, the BGs locate at higher frequencies because of added stiffness, as reported in **Figure 5a**.

3. Sound radiation from finite periodic ABH shells

3.1 Radiation theory for cylindrical shells

Once the radial displacement $w(x', \theta')$ (here $\theta' = y'/R$) is obtained, the normal velocity to the surface can be further expressed as $v_w(x', \theta') = j\omega w(x', \theta')$. For a baffled cylindrical shell, the radiated sound pressure of an arbitrary external point (r, θ, x) can be analytically obtained by [29, 30].

$$p(x, \theta, r) = \frac{j\rho_a\omega}{4\pi^2} \int_{-\pi}^{\pi} \int_{-a}^a v_w(x', \theta') \sum_{n=-\infty}^{+\infty} \cos[n(\theta - \theta')] \times \int_{-\infty}^{+\infty} \frac{\exp[jk_x(x - x')]}{k_y R} \frac{H_n^{(1)}(k_y r)}{H_n^{(1)'}(k_y R)} dk_x dx' R d\theta', \quad (17)$$

where ρ_a represents the density of air, k_x and $k_y = (k^2 - k_x^2)^{0.5}$ symbolize the wavenumber components in the x and the y direction, respectively, and k stands for the total sound wavenumber. Here $\theta = y/R$ stands for the circumferential angle. $H_n^{(1)}$ indicates the n -th Hankel function of the first kind, and $H_n^{(1)'}$ denotes its first derivative with respect to the argument $k_y R$.

For numerical estimation, the cylindrical shell can be segmented into N elementary radiators, with each surface area ΔS . The surface velocity can be assembled as a vector $\mathbf{v}_w$, which can be further used to calculate the sound pressure vector on the cylindrical surface

$$\mathbf{p}_{N \times 1} = \mathbf{Z}_{N \times N} \mathbf{v}_{N \times 1}, \quad (18)$$

where $\mathbf{Z}$ represents the acoustic impedance matrix, with entries

$$Z_{ij} = \frac{j\rho_a\omega\Delta S}{2\pi^2} \sum_{n=0}^{+\infty} \varepsilon_n \cos[n(\theta_i - \theta_j)] \int_0^{+\infty} \frac{\cos[k_x(x_i - x_j)]}{k_y a} \frac{H_n^{(1)}(k_y r)}{H_n^{(1)'}(k_y R)} dk_x, \quad (19)$$

here ε_n is a normalized coefficient, and it is given by

$$\varepsilon_n = \begin{cases} 1, & n = 0, \\ 2, & n > 0. \end{cases} \quad (20)$$

Next, we can write the sound power as

$$W_s = \mathbf{v}^H \mathbf{R} \mathbf{v}, \quad (21)$$

where the superscript H stands for the Hermite transpose and $\mathbf{R} = \frac{\Delta S}{2} \text{Re}(\mathbf{Z}|_{r=R})$ is the radiation resistance matrix (real symmetric and positive-definite). The sound radiation efficiency can be further obtained by

$$\sigma = \frac{W_s}{\rho_a c_a N \Delta S \langle v_w^2 \rangle_{\text{overall}}}, \quad (22)$$

where $\langle v_w^2 \rangle_{\text{overall}}$ represents the mean square velocity (MSV) over the whole surface of the ABH cylindrical shell.

3.2 Sound radiation from unstiffened ABH shells

Now we investigate the sound radiation from a finite unstiffened shell containing five ABH cells. Based on the reconstructed GEM presented in Section 2.1, we cancel the axial periodic conditions, namely Eqs. (12) and (13), such that the vibration field of the finite ABH shell can be characterized. It is well-known that there is a cut-on frequency for an ABH, which is mainly determined by its size. Only beyond this frequency (wavelength smaller than the ABH size, $2r_{abh}$), the wave can be trapped and the ABH effect can be triggered. According to **Table 1**, the radius cut-on frequency $f_r = \frac{\pi h_{uni}}{4r_{abh}^2} \sqrt{\frac{E}{3\rho(1-\nu^2)}} = 282$ Hz.

As plotted in **Figure 6a**, the radial mean square velocity (MSV) on the surface of the ABH shell is compared to that of the uniform (UNI) shell having the same damping layer configuration. From the figure, it can be seen that in the BGs, the vibration of the ABH shell is very low. In the passbands, the vibration beyond $f_r = 282$ Hz is stronger but never exceeds the MSV of the reference shell. Particularly, the average reduction reaches $\Delta MSV = 10$ dB. Looking at **Figure 6b**, the sound power level (SWL) is also very small in the BGs. Even for the passbands, the SWL is effectively suppressed, with an average reduction approaching $\Delta SWL = 15$ dB. Then, we examine the radiation efficiency of the ABH shell. As shown in **Figure 6c**, the radiation efficiency of the uniform shell grows to the maxima in the vicinity of the critical frequency $f_c = 390$ Hz. After embedding the ABHs, however, the radiation efficiency is significantly impaired, almost in the whole frequency band of

Figure 6. Comparison of the (a) mean square velocity (MSV), (b) sound power level (SWL), and (c) radiation efficiency, between the ABH and uniform shells. The shaded areas stand for the BGs.

interest. Careful readers may notice that the radiation efficiency is very small in the 2-nd BG (the 1-st BG is not shown here because it is too narrow), but in the latter BGs, it becomes larger. This is due to the weakening of the local resonance effect at higher frequencies where the uniform portions start to activate (see **Figure 4**).

To clearly manifest the characteristics of the ABH shell, we have further calculated the vibration field at each frequency. As illustrated in **Figure 7**, compared to the uniform shell whose distribution of vibration nodes is very regular to location and frequency (over the ring frequency 173 Hz), the vibration in the BGs is obviously isolated as propagating in the axial direction. For frequencies outside of the BGs, the amplitude of the displacement is also clearly reduced to the right direction. Specifically, we choose 287 Hz and 340 Hz as two representative frequencies in the BG and the passband, respectively, for demonstrating the effectiveness of the ABH shell. In **Figure 8a**, it is clearly seen that the local resonance effect in the ABHs is very strong, such that vibration can be substantially stopped when compared to the uniform shell. While in **Figure 8b** we can see that the wave can be transmitted to the whole shell, but the amplitude is very small because of the highly efficient damping effect by the ABH + damping layer configuration.

Figure 7.

Displacements ($w_{ref} = 1 \text{ m}$) of (a) the ABH shell and (b) the uniform shell, changing with frequency.

Figure 8.

Forced vibration shapes for the finite shell having five cells at (a) 258 Hz (at the center of the second bandgap), (b) 340 Hz (in the passband), compared to a uniform cylindrical shell having the same damping layer configuration. The shaded areas represent the ABH portions.

3.3 Sound radiation from stiffened ABH shells

Note that the embedded ABHs may obviously reduce the structural stiffness of the shell, therefore, the stiffeners (see **Figure 2d**) can be utilized to alleviate this issue. In this section, we first investigate the effects of the stiffener number, then study that of the stiffener width.

For simplicity, let us first consider three cases. That is, the stiffener number $N = \{8, 16, 32\}$, while the width of each stiffener keeps $W = 4h_{uni}$. As shown in **Figure 9a**, when adding eight stiffeners the vibration level is very similar to the ABH shell (without stiffener), except for the frequencies in the BGs. Increasing the stiffener number will further deteriorate the vibration in the BGs, but it seems that the MSV level is similar to that of the unstiffened one. This means that the combination of the ABH and stiffeners not only results in a more rigid structure (compared to the pure ABH one) but also maintains the overall damping effect. Similar results can be found for the radiated SWL (see **Figure 9b**). The existence of the stiffeners almost merely increases the SWL in the BGs, yet in the passbands, the stiffened ABH shell has a similar SWL to that of the unstiffened one. Furthermore, the radiation efficiency is checked (see **Figure 9c**). From the figure, we can see that after inserting stiffeners the radiation efficiency in the 2-nd (the 1-st BG is not shown here) is very small and, that for the case of 32 stiffeners the efficiency is the lowest over 700 Hz. In general, the stiffeners will not obviously degrade the reduction ability of the ABH shell.

Figure 9.
 (a) Mean square velocity (MSV), (b) sound power level (SWL), and (c) radiation efficiency, for the cylindrical shell having a different number of stiffeners with the same width $W = 4h_{uni}$.

Now we keep the 16 stiffeners in the circumferential direction for each cell but modify the width from $W = 4h_{uni}$ to $W = 8h_{uni}$, then to $W = 16h_{uni}$. Similar to the above results, from **Figure 10** it can be observed that increasing the stiffener width mainly intensifies the MSV and SWL in the BGs, while their values in the passbands almost do not change. Particularly, the radiation efficiency of the stiffened shells is obviously lower than the ABH shell at frequencies greater than 650 Hz. Even large stiffener width will enlarge the vibration and the radiated sound power at low frequencies (e.g., at 180 Hz), the high-frequency performance is still plausible and the structure is more rigid.

For the purpose of illustration, we have figured out the normal velocity and sound pressure distributions on the finite shells at 340 Hz (in passband). The former is graphed in **Figure 11** while the latter in **Figure 12**. From **Figure 11**, we can see that the vibration level of the uniform shell is very strong (**Figure 11a**), but the ABHs can help reduce the overall vibration, with only strong vibrations in the ABH portions where the damping layer is very effective (**Figure 11b**). After adding stiffeners (**Figure 11c1–d3**), the vibration in the ABH areas is intensified compared to **Figure 11b**. The sound pressure distributions in **Figure 12** display that the ABH and stiffened ABH shells can also effectively reduce the sound pressure, compared to the reference uniform shell. The existence of the stiffener makes the distributions no longer axially symmetrical, except for **Figure 12c3** where a lot of small and distributed stiffeners are imposed.

Figure 10.

(a) Mean square velocity (MSV), (b) sound power level (SWL), and (c) radiation efficiency, for the cylindrical shell having 16 stiffeners with different widths.

Figure 11. Surface normal velocity distributions at 340 Hz. (a) Uniform shell, (b) ABH shell without stiffener, (c1)-(c3) stiffened ABH shell with the different number of stiffeners of the same width, (d1)-(d3) stiffened ABH shell with 16 stiffeners having different widths. The red circles stand for the ring force.

4. Conclusions

In the current chapter, the acoustic black hole (ABH) effect is concerned and applied to cylindrical structures. By reducing thickness following the power law, the wave velocity is substantially slowed and the wavenumber is increased when it propagates to the ABH center, where the damping layer is very efficient to consume vibrational energy. The focus is placed on reducing the sound emission from cylindrical shells, which can be found in many fields, via embedding periodic ABHs. First, the reconstructed Gaussian expansion method (GEM) is presented to characterize infinite periodic ABH shells. The band gaps (BGs), induced by the locally resonant effect in the ABH area, are investigated, together with the influence of the ABH parameters. Next, the sound radiation model for finite periodic cylindrical shells is developed. Numerical results show that the periodic ABHs can both reduce the vibration and sound power, relying on two mechanisms—(i) the BGs for

Figure 12. Surface sound pressure distributions at 340 Hz. (a) Uniform shell, (b) ABH shell without stiffener, (c1)-(c3) stiffened ABH shell with the different number of stiffeners of the same width, (d1)-(d3) stiffened ABH shell with 16 stiffeners having different widths. The red circles stand for the ring force.

isolating vibrations and (ii) the damping effects for energy consumption. The inclusion of stiffeners not only strengthens the structural stiffness but also keeps the vibration and sound power level as the pure ABH one in the passbands.

Acknowledgements

This work was supported by the National Natural Science Foundation of China (grant number 11704314) and the China Postdoctoral Science Foundation (grant numbers 2018M631194 and 2020T130533).

Conflict of interest

The authors declare no conflict of interest.

IntechOpen

IntechOpen

Author details

Jie Deng* and Nansha Gao

Key Laboratory of Ocean Acoustic and Sensing, School of Marine Science and Technology, Northwestern Polytechnical University, Xi'an, China

*Address all correspondence to: dengjie@nwpu.edu.cn

IntechOpen

© 2022 The Author(s). Licensee IntechOpen. This chapter is distributed under the terms of the Creative Commons Attribution License (<http://creativecommons.org/licenses/by/3.0>), which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

References

- [1] Krylov VV, Tilman FJBS. Acoustic black holes for flexural waves as effective vibration dampers. *Journal of Sound and Vibration*. 2004;**274**(3): 605-619. DOI: 10.1016/j.jsv.2003.05.010
- [2] Krylov VV. A new type of vibration damper based on flexural wave propagation in laminated wedges of powerlaw profile. *The Journal of the Acoustical Society of America*. 2001; **110**(5):2654-2654. DOI: 10.1121/1.4777028
- [3] Krylov VV, Winward RETB. Experimental investigation of the acoustic black hole effect for flexural waves in tapered plates. *Journal of Sound and Vibration*. 2007;**300**(1): 43-49. DOI: 10.1016/j.jsv.2006.07.035
- [4] Deng J, Zheng L, Zeng P, Zuo Y, Guasch O. Passive constrained viscoelastic layers to improve the efficiency of truncated acoustic black holes in beams. *Mechanical Systems and Signal Processing*. 2019;**118**:461-476. DOI: 10.1016/j.ymssp.2018.08.053
- [5] Zeng P, Zheng L, Deng J, Elsabbagh A, Xiang S, Yan T, et al. Flexural wave concentration in tapered cylindrical beams and wedge-like rectangular beams with power-law thickness. *Journal of Sound and Vibration*. 2019;**452**:82-96. DOI: 10.1016/j.jsv.2019.04.002
- [6] Li H, Touz C, Pelat A, Gautier F, Kong X. A vibro-impact acoustic black hole for passive damping of flexural beam vibrations. *Journal of Sound and Vibration*. 2019;**450**:28-46. DOI: 10.1016/j.jsv.2019.03.004
- [7] Deng J, Guasch O, Zheng L. Ring-shaped acoustic black holes for broadband vibration isolation in plates. *Journal of Sound and Vibration*. 2019; **458**:109-122. DOI: 10.1016/j.jsv.2019.06.017
- [8] Ma L, Zhang S, Cheng L. A 2D Daubechies wavelet model on the vibration of rectangular plates containing strip indentations with a parabolic thickness profile. *Journal of Sound and Vibration*. 2018; **429**:130-146. DOI: 10.1016/j.jsv.2018.04.042
- [9] Tang L, Cheng L, Ji H, Qiu J. Characterization of acoustic black hole effect using a one-dimensional fully-coupled and wavelet-decomposed semi-analytical model. *Journal of Sound and Vibration*. 2016;**374**:172-184. DOI: 10.1016/j.jsv.2016.03.031
- [10] Jeon W, Lee JY. Vibration damping using a spiral acoustic black hole. *The Journal of the Acoustical Society of America*. 2017;**141**(5):3644-3644. DOI: 10.1121/1.4987872
- [11] Tang L, Cheng L. Enhanced acoustic black hole effect in beams with a modified thickness profile and extended platform. *Journal of Sound and Vibration*. 2017;**391**:116-126. DOI: 10.1016/j.jsv.2016.11.010
- [12] Fu Q, Du X, Wu J, Zhang J. Dynamic property investigation of segmented acoustic black hole beam with different power-law thicknesses. *Smart Materials and Structures*. 2021; **30**(5):055001. DOI: 10.1088/1361-665x/abed32
- [13] Bowyer EP, Krylov VV. Experimental study of sound radiation by plates containing circular indentations of power-law profile. *Applied Acoustics*. 2015;**88**:30-37. DOI: 10.1016/j.apacoust.2014.07.014
- [14] Li X, Ding Q. Sound radiation of a beam with a wedge-shaped edge embedding acoustic black hole feature. *Journal of Sound and Vibration*. 2019; **439**:287-299. DOI: 10.1016/j.jsv.2018.10.009

- [15] Ma L, Cheng L. Sound radiation and transonic boundaries of a plate with an acoustic black hole. *The Journal of the Acoustical Society of America*. 2019; **145**(1):164-172. DOI: 10.1121/1.5081680
- [16] Deng J, Guasch O, Maxit L, Zheng L. Transmission loss of plates with multiple embedded acoustic black holes using statistical modal energy distribution analysis. *Mechanical Systems and Signal Processing*. 2021;**150**:107262. DOI: 10.1016/j.ymssp.2020.107262
- [17] Deng J, Zheng L. Noise reduction via three types of acoustic black holes. *Mechanical Systems and Signal Processing*. 2022;**165**:108323. DOI: 10.1016/j.ymssp.2021.108323
- [18] Deng J, Guasch O, Zheng L. Reconstructed Gaussian basis to characterize flexural wave collimation in plates with periodic arrays of annular acoustic black holes. *International Journal of Mechanical Sciences*. 2021; **194**:106179. DOI: 10.1016/j.ijmecsci.2020.106179
- [19] Deng J, Zheng L, Gao N. Broad band gaps for flexural wave manipulation in plates with embedded periodic strip acoustic black holes. *International Journal of Solids and Structures*. 2021; **224**:111043. DOI: 10.1016/j.ijsolstr.2021.111043
- [20] Deng J, Zheng L, Guasch O. Elliptical acoustic black holes for flexural wave lensing in plates. *Applied Acoustics*. 2021;**174**:107744. DOI: 10.1016/j.apacoust.2020.107744
- [21] Deng J, Guasch O, Zheng L, Song T, Cao Y. Semi-analytical model of an acoustic black hole piezoelectric bimorph cantilever for energy harvesting. *Journal of Sound and Vibration*. 2021;**494**:115790. DOI: 10.1016/j.jsv.2020.115790
- [22] Deng J, Guasch O, Zheng L. A semi-analytical method for characterizing vibrations in circular beams with embedded acoustic black holes. *Journal of Sound and Vibration*. 2020;**476**: 115307. DOI: 10.1016/j.jsv.2020.115307
- [23] Deng J, Guasch O, Maxit L, Zheng L. Vibration of cylindrical shells with embedded annular acoustic black holes using the Rayleigh-Ritz method with Gaussian basis functions. *Mechanical Systems and Signal Processing*. 2021; **150**:107225. DOI: 10.1016/j.ymssp.2020.107225
- [24] Deng J, Guasch O, Maxit L, Zheng L. Reduction of Bloch-Floquet bending waves via annular acoustic black holes in periodically supported cylindrical shell structures. *Applied Acoustics*. 2020;**169**:107424. DOI: 10.1016/j.apacoust.2020.107424
- [25] Deng J, Guasch O, Maxit L, Zheng L. Annular acoustic black holes to reduce sound radiation from cylindrical shells. *Mechanical Systems and Signal Processing*. 2021;**158**:107722. DOI: 10.1016/j.ymssp.2021.107722
- [26] Deng J, Zheng L, Guasch O, Wu H, Zeng P, Zuo Y. Gaussian expansion for the vibration analysis of plates with multiple acoustic black holes indentations. *Mechanical Systems and Signal Processing*. 2019;**131**:317-334. DOI: 10.1016/j.ymssp.2019.05.024
- [27] Sivasubramonian B, Rao GV, Krishnan A. Free vibration of longitudinally stiffened curved panels with cutout. *Journal of Sound and Vibration*. 1999;**226**(1):41-55. DOI: 10.1006/jsvi.1999.2281
- [28] Tang L, Cheng L. Broadband locally resonant band gaps in periodic beam structures with embedded acoustic black holes. *Journal of Applied Physics*. 2017;**121**(19):194901. DOI: 10.1063/1.4983459
- [29] Stepanishen PR. Radiated power and radiation loading of cylindrical

surfaces with nonuniform velocity distributions. The Journal of the Acoustical Society of America. 1978; **63**(2):328-338. DOI: 10.1121/1.381743

[30] Sun Y, Yang T, Chen Y. Sound radiation modes of cylindrical surfaces and their application to vibro-acoustics analysis of cylindrical shells. Journal of Sound and Vibration. 2018;**424**:64-77. DOI: 10.1016/j.jsv.2018.03.004