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Experimental and frequency-domain study of acoustic damping of single-layer perforated plates

Gang Wu [a,*], Shen Li [b], He Zhao [b], Xinglin Yang [b], Jiaqiang E [c]

[a] College of Automotive and Mechanical Engineering, Changsha University of Science and Technology, 410114, Yuhua District, Changsha City, Hunan, China
[b] Department of Power Engineering, School of Energy and Power Engineering, Jiangsu University of Science and Technology, Mengxi Road, Zhenjiang City, Jiangsu, 212003, China
[c] College of Mechanical and Vehicle Engineering, Hunan University, Changsha, 410082, China

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ABSTRACT

Perforated plates/liners have been widely applied in aeroengines and gas turbines to dissipate unwanted noise. In this work, numerical simulations and experimental study of acoustic damping performance of single-layer perforated plates are conducted. For this, a frequency-domain numerical model is developed via solving 3-dimensional Helmholtz equations. The model validation is performed first by evaluating and comparing the calculated acoustic damping performances of three perforated plates with the experimental results available in literature. These plates are perforated with a number of circle-shaped orifices, which have different porosities σ. Each plate backed by a resonant cavity is placed at the end of a rectangular-shaped pipe with a loudspeaker implemented on the other end. To simulate real engines, a mean flow (also known as bias flow) is applied to pass through the perforated orifices. The effects of 1) the mean flow Mach number M_a, 2) the resonance parameter Q, 3) the porosity σ and 4) the plate thickness T are studied one at a time. Acoustic damping performance of these plates is characterized by using sound absorption coefficient α and specific acoustic impedance z. It is found that sound absorption coefficient α is increased first and then decreased with increased M_a. The real part of z characterizing acoustic resistance/damping is linearly increased, as M_a is increased. However, the imaginary part of z characterizing acoustic reactance remains almost unchanged. It is also found that the presence of the bias flow greatly increases the maximum sound absorption coefficient α_{max} by approximately 33% in comparison with that of without mean flow. The plate thickness T is shown to shift the frequency corresponding to the maximum sound absorption coefficient. Comparing the present simulation results with the experimental and analytical ones available in the literature reveals that good agreement is obtained at lower frequency range. However, as frequency is increased, the model predicts that the bias flow leads to the local α_{max} reduced by 15%. The plate with a larger thickness is associated with another local α_{max}, which is not observed for a lower thickness plate. Finally, experimental measurements of the acoustic damping of 2 single-layer in-duct perforated plates are performed. The frequency-domain model is then applied to simulate the experiment. Good agreement between our experimental and numerical results is obtained.

1. Introduction

Many modern aero-engine afterburners, rocket motors, boiler, ramjets and gas turbines are more prone to combustion instability [1]. In these combustion systems, unsteady heat release and pressure disturbances [2] can be coupled to produce self-sustained thermoacoustic oscillations under certain conditions. To dampen such instabilities, the coupling must somehow be interrupted [3]. Perforated plates/liners are widely applied to attach to the bounding wall of a combustor. They are usually metal sheets perforated with thousands of tiny orifices and work as acoustic dampers to absorb acoustic disturbances [4–10]. To prevent the perforated liners/plates from being burned in such extremely high temperature combustor, a cooling air flow is needed to pass through the orifices. This cooling flow is also known as bias flow. A review of such acoustic dampers applied in combustion system is recently reported by Zhao and Li [11]. The drive of developing lower-emission engines and 'quieter' HVAC (heating, ventilation and air conditioning) systems leads to a resurgence of perforated plates/liners [12] or porous foam focused research [13], with the aim of improving

* Corresponding author.
E-mail address: gangwucust@126.com (G. Wu).

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the design of effective acoustic liners and/or stabilizing unstable combustion systems [14–18].

Over the past half century, perforated plates/liners have been theoretically, numerically and experimentally studied, aiming to better understand the noise damping mechanism and to predict its damping performance. Experimental investigations [19,20] are typically conducted to measure the acoustic impedance and/or power absorption coefficient of perforated plates/liners. This is due to the fact that these parameters are easier to measure in comparison with the vorticity-involved flow field near the perforated orifice (with a diameter ≤1 mm) edge. Jing and Sun [20] conducted experimental tests on an impedance tube with a perforated plate backed by a cavity in the presence of a bias flow. They found that the plate thickness and the bias flow Mach number play dominant roles in affecting the noise damping performance. To simulate real engines with both bias and grazing flow present, Eldredge and Dowling [19] designed and tested a cold-flow pipe with a double-layer liner implemented. They found that approximately 80% of sound energy can be absorbed with an 'optimum' bias flow Mach number. Flow visualization measurements [21,22] were conducted to gain insight on flow-acoustics interaction near orifices. The amplitude and frequency of incident sound, the diameter and thickness of the orifice were found to affect the induced motion of the fluid.

Frequency-domain theoretical studies have been widely conducted, focusing on the prediction of the noise damping performance of single or multiple orifices. The acoustic resistance and reactance of a single-layer liner was theoretically predicted by Guess in 1975 [23]. Nonlinear effect due to high-amplitude incident sound was successfully considered. The damping behavior of perforated orifices is characterized by Rayleigh conductivity. Howe [24] used Rayleigh conductivity to model the noise damping behavior of a single orifice at a high Reynolds-number. Hughes and Dowling [25] also adopted Rayleigh conductivity to model the damping performance of perforated screen backed by a rigid plane. They showed that almost 100% of incident sound on the perforated screen with a bias flow might be absorbed, if the flow speed and the liner geometry were chosen properly. Eldredge and Dowling [19] developed a 1D duct model via using a homogeneous liner compliance adapted from Rayleigh conductivity.

Besides frequency-domain theoretical modeling [19,26], time-domain numerical simulations [27–32] are becoming more popular due to the fact that it offers a way to a deeper understanding of the small-scale flow-acoustics physics. This flow field information is missing about acoustic-vortex interactions in frequency domain. Recently, Zhang and Bodony [29] conducted 3D DNS (direct numerical simulation) to study the acoustics behavior of a circular orifice backed by a hexagonal cavity. They found that the orifice boundary layer played a critical role in determining the orifice discharge coefficient, i.e. the nonlinearity. Mendez and Eldredge [30] conducted 2D compressible large-eddy simulations (LES) on a single perforated orifice. They found that there was a strong relation between the dynamic field and the acoustic response of the system. Tam et al. [31] carried out 3D DNS of a single aperture. They confirmed that vortex shedding was the dominant damping mechanism, when the amplitude of incident sound is high. The numerical simulations described above attempt to solve the Navier–Stokes equations by using high-order finite volume (FV) or difference (FD) method. As an alternative computational tool, lattice Boltzmann method was implemented to gain insight on the noise damping mechanism and performance of perforated plates [32,33]. Both 2D and 3D simulations of perforated orifices with different shapes and thickness were conducted.

In this work, numerical and experimental studies of the acoustic damping performances of single-layer perforated plates are conducted. For this, three-dimensional finite element (FE) simulations are conducted first to predict the acoustic damping perforated of 3 perforated plates [34]. The numerical simulations are conducted in frequency domain, as described in Sec. 2. The 3D numerical scheme and configuration of interest are presented. To evaluate the sound absorption performance of these perforated plates, power absorption coefficient and acoustic impedance are defined and used. This is described in Sec. 3. In Sect. 4, the effects of 1) the mean flow Mach number M_a, 2) the acoustic excitation frequency ω, the porosity σ and 4) plate thickness T are studied and discussed one at a time. Comparison is then made between the present numerical results and the experimental and theoretical ones available in the literature. With the model validated, experimental measurements of acoustic damping effect of 2 in-duct perforated plates are performed. This is described in Sect. 5. The damping performance is measured via power absorption coefficient. Finally, in Sect. 6, the main findings of the present work are summarized.

2. Numerical method and configuration of interest

The basic idea for the proposed methodology is to efficiently model acoustic wave propagation in the presence of a mean flow. In this section, an appropriate form of the governing equations is derived. The scope is acoustic wave propagation in a flow duct with a monopole-like source and a perforated plate present. For the derivation, the starting point is the mass and momentum governing equations for compressible air [36]. These are in dimensional form in a Cartesian coordinate system:

$$\frac{\partial \rho}{\partial t} + \sum_{i=1}^{3} \frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho u_i) + \sum_{i=1}^{3} \frac{\partial}{\partial x_j}(\rho u_i u_j + p_{ij}) = 0 \tag{2}$$

where $\rho = \rho_0 + \rho'$, $p = p_0 + p'$ and $u_i = u_{i0} + u_i'$ are the instantaneous density, pressure and velocity in ith coordinate. They consists of both a mean part denoted by a subscript 0 and a fluctuating part denoted by a prime. $p_{ij} = p\delta_{ij} - \tau_{ij}$ is the stress tensor. τ_{ij} is the viscous stress tensor and δ_{ij} is the identity matrix. Taking the x_i derivative of Eq. (2) and sum over $i = 1, 2, 3$ and then subtracting that from the t derivative of Eq. (1) gives

$$\frac{\partial^2 \rho}{\partial t^2} = \sum_{i,j=1}^{3} \frac{\partial^2}{\partial x_i \partial x_j}(\rho u_i u_j + p_{ij}) \tag{3}$$

Subtracting $c_0^2 \nabla^2 \rho$ from both sides leads to

$$\frac{\partial^2 \rho}{\partial t^2} - c_0^2 \nabla^2 \rho = \sum_{i,j=1}^{3} \frac{\partial^2}{\partial x_i \partial x_j}(\rho u_i u_j + p_{ij} - c_0^2 \delta_{ij} \rho) \tag{4}$$

Linearizing the equation gives

$$\frac{1}{c_0^2}\frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = Q_s(t) \tag{5}$$

where $Q_s = \sum_{i=1}^{3} \frac{\partial^2}{\partial x_i \partial x_j}\left[\rho u_i u_j + p_{ij} - c_0^2 \rho' \delta_{ij}\right]$ is known as the Lighthill stress tensor [35]. Eq. (5) is in the form of acoustic wave equation for sound propagation through a uniform medium. However, the right hand side is non-zero, which corresponds to the classical sound sources: quadrupole, dipole and monopole. Applying the Fourier transform, the well-known Helmholtz equation is obtained as

$$\nabla \cdot \left(\frac{1}{\rho_0}\nabla \hat{p}\right) + \frac{\omega^2}{\rho_0 c_0^2}\hat{p} = -\frac{\hat{Q}_s(\omega)}{\rho_0} \tag{6}$$