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Experimental comparison of noise dissipation effects of single- and double-layer acoustic liners

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ABSTRACT

Noise damping performances of 12 perforated liners in the configurations of single- or double-layer are studied in this work. Both experimental and numerical investigations are conducted on these perforated liners with different porosities η, thus enabling the open-area-ratio (also known as porosity η) effect being studied. To simulate practical applications for example gas turbine engines and mufflers, adjustable bias and grazing flows are simultaneously applied to these liners. This enables the role of joint grazing-bias flow being evaluated. These liners' noise mitigating performance is characterized by acoustic power absorption coefficient Δ. And Δ is measured over the frequency range from 200 to 800 Hz. Increasing the Mach number M_g of the grazing flow is found to deteriorate the liners damping performance, while increasing the bias flow results in the maximum Δ_{max} being increased. Additionally, Δ is found to change harmonically with the noise frequency. Furthermore the local maximum of Δ is reduced with increased forcing frequency. Compared to the single-layer perforated liner, the double-layer one shows a higher Δ_{max} and a broader effective frequency range. On the other hand, when the porosities $\eta_{i,o}$ of the outer and inner perforated liners are about 1.1%, the noise absorbing performances are found to be dramatically reduced, especially when the forcing frequency is higher. Increasing the porosities $\eta_{i,o}$ gives rise to Δ_{max} being dramatically increased and so the effective frequency range. To gain insight on the noise absorbing mechanism and 10% more power absorption capacity associated with the double-layer liners than that of single-layer liner, 2D lattice Boltzmann simulations of in-duct perforated orifices are performed in time-domain. The calculated acoustic power absorption coefficients from these 2 different configurations of perforated orifices are compared. Finally, optimum design of a single-layer acoustic liner is conducted using a frequency-domain model of a lined duct. It is shown that the orifice thickness $\mathscr{T}$ and the porosity η_i play critical roles in determining the optimum noise damping performance.

1. Introduction

As one of the typical passive noise control devices, perforated liners are widely applied in industries [1,2]. They are generally a metal sheet perforated with many millimeter diameter orifices [3,4]. Perforated liners are implemented around a combustor to dissipate noise generated by thermoacoustic instabilities [5–8], which are associated with unstable combustion in a lean premixed gas turbine and other engine systems [9–11]. For this, such liners must be designed to be able to dampen unwanted sound over a broad frequency range [12,13]. To protect the liner, a cooling air flow passing through its perforated orifices is needed [14,15]. This cooling air stream is also known as 'bias flow'. In addition, these liners [16,17] are typically exposed to a mean flow along the combustor (also known as 'grazing flow'). The joint bias-grazing flow plays important role in determining the liner's aeroacoustics damping performance [18,19], since unwanted noise is superimposing on the joint flow and propagating along the combustor and/or in a cavity behind a single-liner or a gap between double-layer liners. Currently, there is a resurgence of interest in maximizing the noise dissipation performance of double-layer liners [20–22] over a broad frequency range to meet increasingly stringent regulations on noise emission. This gives rise to a resurgence of acoustic liner-focused engineering research in Europe, USA and Asia [23–25].

During the past half century, extensive numerical, theoretical and experimental studies are performed globally to shed insight on the noise damping mechanism and on improving the performance of perforated

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Nomenclature

Nomenclature

a	orifice radius, m
A_c	cross-sectional area of the annular cavity containing the perforated liners, m^2
A_d	cross-sectional area of the perforated pipe, m^2
$\mathscr{A}_n$	cross-sectional area of the perforated orifice, m^2
$\mathscr{A}_{tot}$	total area of the perforated orifices
$\bar{c}$	mean speed of sound, m/s
$C_{1,2}$	the circumferences of the outer and inner perforated liner, m
$\mathscr{D}$	diameter of the orifice, m
D	diameter of the pipe, m
H_1	stagnation enthalpy, W
L_d, L_u	downstream and upstream pipe lengths of the lined section, m
L, L_l	the length of the lined section, m
M_b	bias flow Mach number
M_d, M_g	Mach number downstream and upstream of the lined section
$\mathscr{N}_i$, $\mathscr{N}_o$	total number of the perforated orifices of the inner and outer perforated pipes.
$\hat{p}_d$, $\hat{p}_u$	frequency-domain pressure disturbances downstream & upstream, Pa
$\hat{R}_d$	downstream reflection coefficient
Re	Reynolds number with D characteristic diameter and U the mean flow velocity
$\mathscr{T}$	thickness of the perforated orifice, m
$\bar{u}$, u'	mean flow velocity and fluctuating velocity in time domain in the pipe, m/s
$\hat{u}$	fluctuating velocity in frequency domain, m/s
$\overline{U}$	mean flow velocity through the orifice, m/s

Greek letters

ρ	air density, kg/m^3
η	effective compliance of the perforated orifices
η_i, η_o	porosities of the inner and outer perforated pipes
η_r	correction factor resulting from resonance effect
Δ	power absorption coefficient
ϵ	transmission loss coefficient
ω	oscillation frequency, rad/s
μ	kinematic viscosity
Ψ^+, Ψ^-	decomposed travelling enthalpy waves, i.e. $\widehat{H} = \Psi^+ + \Psi^-$
γ, σ	real and imaginary parts of Rayleigh conductivity

Subscript

i	inner perforated pipe
o	outer perforated pipe
u	upstream
d,	downstream
g,	grazing flow
max,	maximum
m	m^{th} eigenmode

Superscript

$-$	mean value
$\wedge$	Fourier transform
$'$	fluctuation part

liners [26–31]. 2D DNS (direct numerical simulation) is conducted by Tam et al. [27] and Leung et al. [32] to examine the noise damping mechanism of perforated orifices. They found that the main noise damping mechanism is that vortices are periodically shed around the rims of the orifices [33]. This enables acoustical energy being converted into kinetic vortex energy. Such vorticity-involved noise damping mechanism was visualized in the 3D lattice Boltzmann simulations [34–36] in time domain, which are an alternative computational aeroacoustic tool. The first experimental visualization of vorticity-involved noise damping mechanism was performed by Bechert [37] in 1990s. Sound-vorticity interaction near a rectangular slit was studied by Dai et al. [30]. They applied the discrete vortex (DV) method and developed a model basing on a 3D Green's function. It was found that there are different types of vortex shedding behaviors and convection patterns, which were strongly related to the intensity of the incident sound. To the best knowledge of the present authors, there is no comparison of the vortex-shedding behaviors and the noise damping performances between single- and double-layer liners in the presence of a bias flow. This partially motivated the present work.

In practice, acoustic impedance or power absorption coefficient is typically applied to quantify the noise mitigation effect of in-duct orifices or perforated liners. Lee et al. [31] applied boundary element method (BEM) to predict the acoustic impedance of an orifice via solving the incompressible Euler equation. Zhong &Zhao [38] modelled a lined duct in time domain to calculate power absorption coefficient of perforated liners in the configurations of either single- or double-layer. They found that the power absorption coefficients of the single- and double-liner were dramatically different. The power absorption coefficient describing the fraction of incident sound energy being dissipated is calculated by applying the conventional two-microphone method [39–41]. Hillereau et al. [42] conducted experimental studies of the aeroacoustic impedance of a porous honeycomb with tiny cells. It was found that varying the resistance of the porous honeycomb affected strongly on the acoustic damping performances.

Understanding and studying the acoustic damping mechanism and performance of perforated liners/orifices [43] become more challenging and complex, when a grazing or/and bias flow is present. Such mean flows present in a practical system are confirmed to strongly influence the liners' noise mitigation performance [24,44–46]. The role of a cooling(bias) flow on the noise absorbing behaviors of a single-layer acoustic liner backed by a Helmholtz cavity was experimentally examined by Jing & Sun [44]. Eldredge and Dowling [46] performed experimental measurements of the bias-grazing flow effect on the noise absorbing performance of a double-layer liner. Sun et al. [47] experimentally studied how a joint grazing-bias flow influenced the noise mitigation performances of the perforated orifices. It was found that the orifice's acoustic resistance was increased slowly, as the Mach number of the bias flow was increased due to the presence of a grazing flow. Optimizing the bias flow velocity can maximize the noise absorbing performance of acoustic liners over a broad frequency range [9,38,46,48,49]. This finding was confirmed by Zhao et al. [50] in their recent experimental studies. However, no physical explanation on vorticity-induced damping behaviors was provided.

Previous experimental or numerical studies confirm that a grazing flow does affect perforated liners' noise damping performances [24,26,51]. Tam et al. [24] found that a single-layer liner backed by a Helmholtz resonator-like cavity produced tonal 'sound', as a 'proper' grazing flow was present. The tone is generated due to the Kelvin-–Helmholtz instability. Such hydrodynamic instability was described and discussed in the previous work [26]. Fung et al. [52] applied an impulse technique to evaluate the noise dissipation performance of perforated liners [53] confined in a pipe with or without a grazing flow. Tonon