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A two-dimensional approach for sound attenuation of multi-chamber perforated resonator and its optimal design

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ABSTRACT

A two-dimensional (2D) method is investigated to predict the acoustic performances of single-chamber perforated reactive resonators. The effect of non-planar wave propagation on the acoustic performance of acoustically short and long resonator is studied. A desirable resonance behavior appears in acoustically short chamber substantially below the cut-off frequency due to non-planar wave propagation. Adding inlet/outlet extensions has similar effects to that of reducing perforation rate on acoustic performances for short-length perforated resonators. Based on the 2D approach, a 2D transfer matrix method (TMM) is developed through solving the acoustical continuity functions under two outlet boundary conditions to predict the acoustic performances of multi-chamber perforated resonators (MCPRs) which can attenuate broadband noise. Comparisons between the calculations and tests show that the 2D TMM is much more accurate than one-dimensional approach within entire frequency range. In order to evaluate its engineering applicability, a new optimization procedure including a targeted transmission loss curve and a reasonable objective function is introduced to optimize the structure parameters of a MCPR with three chambers using a genetic algorithm. The result can meet the target well at desired frequencies under space constraint. The theoretical method developed in this work can be used for the calculation and optimization of MCPRs in various applications.

1. Introduction

Air intake noise is one of the main vehicle noises, especially for turbocharged engine in transient conditions, which seriously affects the driving comfort [1]. The overall noise level of turbocharged engine is normally 2 dB(A)–3 dB(A) higher than naturally aspirated engine, resulting from the additional air borne noise which commonly distributes in 1.5–3 kHz [2]. However, traditional resonators (Helmholtz resonator, expansion muffler, quarter-wavelength tube etc.) cannot efficiently attenuate such noise within a broadband frequency range.

A multi-chamber perforated resonator (MCPR) is a typical kind of silencer, which can attenuate broadband noise [3]. In addition, the advantages of a compact structure and a low pressure drop enable the resonator to meet the strict installation requirements. As shown in Fig. 1, an ideal acoustic performance can be achieved through adjusting the width, diameter and number of chambers or the diameter, number, thickness of perforated holes. The transfer matrix method (TMM) based on plane wave theory is to obtain the four-pole parameters of a resonator, which can be used to predict the transmission loss (TL). Guo [4] extended the TMM from a single chamber perforated resonator (SCPR) to a MCPR through multiplying the transfer matrixes of connected sections. The one-dimensional (1D) TMM is widely used for the design and optimization of perforated and micro-perforated resonators. Shi [5] investigated the wave propagation in a periodic array of micro-perforated tube mufflers through 1D TMM. Chiu [6] used 1D TMM to evaluate the acoustic performance of MCPRs and obtained the best shape of a MCPR under space constraint through optimization with a genetic algorithm (GA). But the 1D method is limited to the cut-off frequencies of the resonator chambers. Whereas the finite element method (FEM) is able to predict the acoustic performances of resonators more precisely [7]. Yet as the number of chambers increases, the geometry becomes more complicated, hence the structure modeling and finite element calculation become inefficient. In addition, the structure optimization will be inconvenient if the acoustic performance is unsatisfying. Therefore, it is necessary to develop a theoretical method, which is simultaneously accurate and efficient to calculate and optimize the acoustic performance of MCPRs.

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Nomenclature

S_n	modal amplitudes in region S (*A*, *B*, *C*, *D*, *E*)
c	sound speed in air
hd, d_h	diameter of the perforations
ht	thickness of the inner tube
d_1	diameter of the inlet tube
d_2	diameter of the outlet tube
D_o	diameter of the chambers
f	frequency
J_0, J_1	Bessel functions of the first kind of order 0 and 1
k_0	sound wave number in air
$k_{x,S,n}$	axial wave number in region S (*A*, *B*, *C*, *D*, *E*)
$k_{r,S,n}$	radial wave number in region S (*A*, *B*, *C*, *D*, *E*)
li	length of inlet extension
lm	length of perforation area
lo	length of outlet extension
L	length of a resonator chamber
OBJ	objective function
P	acoustic pressure
r_1	radius of the inner tube
R	radius of the chambers
T	transfer matrix
U	particle velocity
v	air viscosity
Y_0, Y_1	Bessel functions of the second kind of order 0 and 1
α	end correction coefficient
ζ	perforation impedance
σ	porosity
ρ	air density
$\varnothing_S^n(r)$	eigen functions in region S (*A*, *B*, *C*, *D*, *E*)

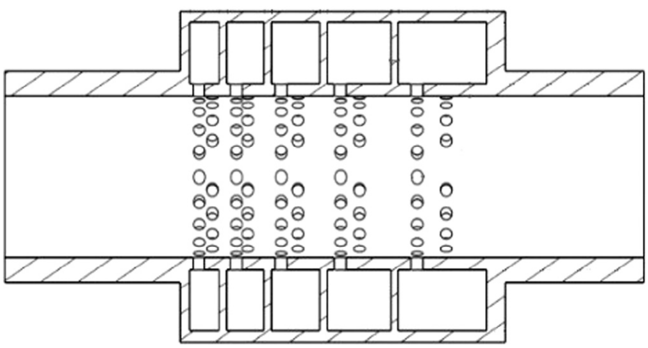

Fig. 1. Configuration of a MCPR.

The aim of the present work is to provide a two-dimensional (2D) TMM for the acoustic modeling and optimization of MCPRs. Theoretically, the 2D approach can be used to calculate the TLs of axisymmetric resonators. Selamet [8,9] predicted the TLs of both a single-chamber and a dual-chamber circular expansion muffler with extended inlet/outlet using a 2D weighted integral method. However, the equations to be solved in this approach will be complicated when many chambers are connected, since the transfer matrixes of the silencer elements are not considered and all continuity equations must be solved at one time. For resonators with perforations, the 2D approach was mostly applied to mufflers with a single chamber [10–12]. An effective and efficient 2D approach has still not been developed for the TL prediction of perforated reactive resonators especially MCPRs.

In this study, a 2D analytical method using direct integral is applied to calculate the TL of a single-chamber perforated reactive resonator with extended inlet/outlet. The effect of non-planar wave propagation on the acoustic performance of acoustically short and long SCPRs is studied through the 2D approach. The transfer matrix of a perforated resonator is derived through solving the continuity equations under two different outlet conditions. Compared to the tests, the accuracy of 2D TMM is validated at both low and high frequencies. In addition, considering the engineering application, a new optimization procedure is introduced, including defining a targeted TL curve in a wide frequency and a reasonable objective function for the MCPRs' design. At last, based on 2D TMM, the geometry of a MCPR with three chambers are optimized applying a genetic algorithm (GA) under space constraint.

2. Two-dimensional approach for sound attenuation of a SCPR

2.1. Acoustic modeling of a SCPR with extended inlet/outlet

As shown in Fig. 2, a SCPR with extended inlet/outlet can be divided into five sections: inlet *A*, extended inlet chamber *B*, perforated tube *C*, extended outlet chamber *D* and outlet *E*. The total length L is divided into an extended inlet of length li, a perforated tube of length lm, and an extended outlet of length lo. The radius of the inner tube and outer chamber are r_1 and R.

The Helmholtz equation of sound wave in axisymmetric tube is expressed as [13]

$$\begin{cases} \frac{\partial^2 P}{\partial r^2} + \frac{1}{r}\frac{\partial P}{\partial r} + \frac{\partial^2 P}{\partial x^2} + k_0^2 P = 0 \\ k_0 = \omega/c = 2\pi f/c \end{cases} \tag{1}$$

where P is the sound pressure; k_0 is the wave number; f is the sound frequency; c is the sound velocity; r is the distance to the axis. Upon making use of the separation method of variables, the sound pressure is assumed as

$$P(r,x) = \sum_n R_n(r) X_n(x) \tag{2}$$

Then, Eq. (1) can be divided into two independent wave equations [13]

$$\begin{cases} \frac{d^2 X(x)}{dx^2} = -k_x^2 X(x) \\ \frac{d^2 R(r)}{dr^2} + \frac{1}{r}\frac{dR(r)}{dr} + k_r^2 R(r) = 0 \end{cases} \tag{3}$$

Solutions of Eq. (3) is expressed as

$$\begin{cases} X_n(x) = S_n^+ e^{-jk_{x,S,n}x} + S_n^- e^{jk_{x,S,n}x} \\ R_n(r) = \phi_S^n(r) \end{cases} \tag{4}$$

Substituting Eq. (4) to Eq. (2) yields

$$P_S = \sum_{n=0}^{\infty} (S_n^+ e^{-jk_{x,S,n}x_S} + S_n^- e^{jk_{x,S,n}x_S}) \phi_S^n(r) \tag{5}$$

Combing the linear momentum equation, the axial particle velocity is derived as

$$U_{x,S} = \frac{j}{\rho\omega}\frac{\partial P_S}{\partial x} = \frac{1}{\rho_0\omega}\sum_{n=0}^{\infty} k_{x,S,n}(S_n^+ e^{-jk_{x,S,n}x_S} - S_n^- e^{jk_{x,S,n}x_S})\phi_S^n(r) \tag{6}$$

where S stands for the five sections *A*, *B*, *C*, *D*, *E*; $S_n{}^+$, $S_n{}^-$ are the nth modal amplitudes corresponding to positive and negative x