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Estimating acoustic transmission loss of perforated filters using finite element method

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ABSTRACT

Finite element method is well established in estimating acoustic transmission loss (TL) of passive reactive acoustic filters. However, the acoustic impedances derived from the established empirical models are being used to model the perforated elements present in these acoustic filters. Such empirical models have been established with known restrictions such as position, shape, size, orientation and evenness of the perforations. In the present work, finite element analysis in frequency domain has been demonstrated to circumvent the necessity of such empirical models in estimating acoustic TL of reactive filters having perforated elements at zero mean flow condition. In order to achieve so, a three-pole measurement based simulation has been carried out which exactly replicates the experimental transmission-loss tube test setup. The essential necessity of simulating anechoic termination to perform three-pole measurement and the associated complexity has been resolved. The constraint of desired meshing for estimating the acoustic TL of a perforated plate has been quantified. The strength of the proposed methodology has been exploited by analyzing reactive acoustic filters with various shapes of perforated components. Further, the challenge of analyzing the reactive filters with external perforation has been considered. In order to simulate the perforation facing to atmosphere, an additional domain or volume with non-reflecting boundary attached to perforation, has been proposed. The proposed methodology has been verified by evaluating the TL of a Helmholtz resonator with a leak and of a perforated tube. Summarizing above, the proposed three-pole based finite element methodology can be used for acoustic analysis of any shape and size of perforated components and reactive filters with perforated elements.

1. Introduction

Perforated filters are a compromise between the reactive and absorptive types of acoustic filters and play a very important role in general filter design. Hence, a precise estimation of transmission coefficient of such filters is an indispensable part in the design process. So far, acoustic impedances of perforated components are approximated empirically or analytically with many assumptions and are used in evaluating the performances of such filters. It is becoming increasingly difficult to ignore the limitations involved in the empirical models or analytical solutions.

In the last three decades, numerous empirical and analytical models have been introduced to estimate the equivalent acoustic impedance and the corresponding acoustic TL of perforated filter components [1–9]. In recent years, finite element method (FEM) and boundary element method (BEM) have been well established in predicting the acoustic TL of filters [1]. From industrial perspective,

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commercially available FEM or BEM based software enable investigators to design complex acoustic filters. A variety of software such as ABAQUS®, SYSNOISE® and VNOISE® have been adopted by many investigators to estimate the acoustic TL of filters [2–5]. Nevertheless, the equivalent acoustic impedance derived empirically or analytically has been used in these software to model the perforated element. Such empirical models have been established with known restrictions such as position, shape, size, orientation and evenness.

In the present paper, the finite element analysis in frequency domain has been demonstrated to circumvent the necessity of empirical model in estimating acoustic TL of perforated reactive filters at zero mean flow condition. One of the commercially available FEM based software, ANSYS® has been used for such purpose. Various issues involved and the precautions desired in simulation have been discussed. Choice of proper element type, corresponding requirements of finite element quality and their effect on the result accuracy has been demonstrated in detail.

After this present section of introduction, Section 2 explains theoretical basis of finite element model for acoustic analysis. In Section 3 benchmarking acoustic analyses have been carried out. Acoustic performances of perforated plates have been evaluated in Section 4. In Section 5, few complex perforated filters have been analyzed. The reactive filters with external perforation have been analyzed in Section 6. The last section summarizes some important observations of the present exercise.

2. Finite element model for acoustic analysis

Acoustic analysis, in our context, is based on the hypothesis such as an enclosed volume of fluid, which is compressible, in-viscous, without mean flow and with uniform mean density and pressure. Along with this, the enclosure walls are assumed to be ideally rigid. Based on the above assumptions, the acoustic wave equation may be written as:

$$\nabla^2 p = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2}, \tag{1}$$

where c is the speed of sound in the medium, p is acoustic pressure, t is time and ∇^2 is Laplace operator [6]. Since the viscous dissipation is neglected, the above expression is also considered as loss-less wave equation for sound propagation. Then, for a small change in pressure, δp, over a finite fluid volume, the wave equation can be represented in the integral form as:

$$\int_V \frac{1}{c^2}\delta p \frac{\partial^2 p}{\partial t^2} dV + \int_V (\{L\}^T \delta p)(\{L\}p) dV - \int_S \{n\}^T \delta p (\{L\}p) dS = \{0\}, \tag{2}$$

where, the matrix operator can be defined as:

$$\begin{cases} \nabla.() = \{L\}^T = \left[\frac{\partial}{\partial x} \;\; \frac{\partial}{\partial y} \;\; \frac{\partial}{\partial z}\right] \\ and, \nabla() = \{L\} = \left[\frac{\partial}{\partial x} \;\; \frac{\partial}{\partial y} \;\; \frac{\partial}{\partial z}\right]^T \end{cases} \tag{3}$$

In above equation, V is the volume of the fluid domain, S is the surface where the derivative of pressure normal to the surface is applied and $\{n\}$ is the unit vector normal to the surface S [7].

In simulating fluid–structure interaction problem, the surface S is treated as the fluid–solid interface. The normal pressure gradient of the fluid and the normal acceleration of the structure at the fluid–structure interface S can be expressed as:

$$\{n\}\cdot\{\nabla p\} = -\rho_0 \{n\}\cdot\frac{\partial^2\{u\}}{\partial t^2}, \tag{4}$$

where $\{u\}$ is the displacement vector of the structure at the interface. Now, the Eq. (4) can be rewritten as [8]:

$$\int_V \frac{1}{c^2}\delta p \frac{\partial^2 p}{\partial t^2} dV + \int_V (\{L\}^T)(\{L\}p) dV + \rho_0 \int_S \delta p \{n\}^T \left(\frac{\partial^2}{\partial t^2}\{u\}\right) dS = \{0\} \tag{5}$$

In finite element modeling, using element shape function, $\{N\}$, for pressure, element shape function, $\{N'\}^T$, for displacements, and nodal pressure vector $\{\ddot{p}_e\}$, the second order derivatives can be represented as:

$$\frac{\partial^2 p}{\partial u^2} = \{N\}^T\{\ddot{p}_e\} \tag{6}$$

$$\frac{\partial^2}{\partial u^2}\{u\} = \{N'\}^T\{\ddot{u}_e\} \tag{7}$$

where $p = \{N\}^T\{p_e\}$ and $u = \{N'\}^T\{u_e\}$. Now, the discretized wave equation can be written in matrix notation as:

$$[M_F]\{\ddot{p}_e\} + [K_F]\{p_e\} + \rho_0 [R]^T\{\ddot{u}_{F,e}\} = \{0\}, \tag{8}$$

where,

$$[M_F] = \frac{1}{c^2}\int_V \{N\}\{N\}^T dV \tag{9}$$

$$[K_F] = \int_V [\nabla N]^T [\nabla N] dV \tag{10}$$

$$[R]^T = \int_S \{N\}\{n\}^T\{N'\}^T dS \tag{11}$$

The $[M_F]$, $[K_F]$, $[R]^T$ are acoustic fluid mass matrix, acoustic fluid stiffness matrix and acoustic fluid coupling matrix, respectively. Assuming time harmonic input, $p = pe^{j\omega t}$, the Eq. (1) can be re-written as:

$$\nabla^2 p + k^2 p = 0, \tag{12}$$

where $k = (\omega/c)$ is wave number, ω is the angular frequency ($\omega = 2\pi f/c$).

3. Benchmarking

In general, the finite element analysis is resorted to when the experimental investigation is not feasible. They are used predominantly during the design stage of the