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Comparison of various algorithms for improving acoustic attenuation performance and flow characteristic of reactive mufflers

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ABSTRACT

The parametric optimization of the reactive mufflers is researched by numerical analysis, regarding the performance of the acoustic and flow fields synthetically. The finite element method, based on the Helmholtz equation and the Navier–Stokes equation respectively, is utilized in the analysis of the acoustic and flow fields. And the initial and boundary conditions are set up in the physical fields respectively. The weighting multi-objective function about acoustic and flow fields is formulated. In addition, the optimization results of multidisciplinary, obtained by the Nelder Mead algorithm (NMA) based on the sensitivity analysis, the Monte Carlo algorithm (MCA) and Genetic Algorithm (GA) based on the random sampling, are analyzed comparatively. The optimization results indicate that the NMA can maximize the transmission loss (TL) and minimize the pressure drop with the given weight factor. Finally, numerical optimization examples confirm the validity and reliability of the proposed optimization method in the acoustic-flow field.

1. Introduction

Mufflers are widely used to attenuate the intake and exhaust noise. A reactive muffler is preferred to reduce the low-frequency noise generated by internal combustion engines. And the main goal of the exhaust muffler design is to reduce pressure loss and TL of target frequency band. Due to the space limitation of the structure, structural parameters of the prototype muffler is usually determined according to engineering experiences, and then through the method of orthogonal experiment to achieve muffler optimization. However, this method is often difficult to obtain global optimal solution.

In recent years, many researchers have put forward the methods to optimize the performance of muffler. An Integrated one-dimensional approach using the transfer matrices of simple acoustic elements are proposed by Munjal [1]. Barbieri et al. [2] combined with improved quad-pole parameters method and the finite element analysis and shape optimization, aiming to maximize TL values of the target frequency. Related numerical calculation of the resistance muffler was carried out without regard to the pressure loss. In the recent research, they also involved the optimization of the muffler and acoustic horn with Particle Swarm Optimization method for some linear acoustic problems.

Chiu et al. [3,4] predicted the TL of the multi-cavity muffler using GA and simulated annealing algorithm, based on the theory of one dimensional plane wave and four port transmission matrix method. High professional requirements must familiar with the muffler performance are essential for designers to determine the parameters to be optimized according to the experiences in the design and research of the existing. Bilawchuk [5] comprised the four pole transmission matrix method and the three point method to predict and analyze the TL of the muffler. The research revealed that the three point method had a faster calculation speed and was easier to test and validate by measurement applications. Lee et al. [6–9] detailed a gradient optimization algorithm (the method of moving asymptotes), and combined with the topology optimization to search the layout of the internal structure of the muffler. Yoon et al. [10,11] put forward a new acoustic topology optimization framework a mixed finite-element formulation. By means of the solid isotropic material with penalization interpolation functions for fibrous material, the effect of the pressure attenuation as well as the acoustic performance on the muffler layout for fibrous materials was investigated. Seungjae Oh et al. [12,13] utilized the acoustic input impedance of a suction muffler to improve the energy efficiency of a fluid machine. The objective function including noise reduction, pressure drop and energy efficiency was considered in the suction muffler design of refrigerator's compressor. Theoretically speaking, better acoustic and fluid performance may be obtained by topology optimization method

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[11,14–19] than that of parameter optimization [3,4,20,21] and shape optimization [20,22–24]. However, as far as the engineering application and manufacturing restraint are concerned, the parameter optimization still has an irreplaceable position.

In this study, to consider TL and pressure drop simultaneously, a weighted convex sum technique [7,10] was used. For simplicity, the interaction between fluid and acoustic was not considered and the fluid model was governed by turbulent flow due to the large inlet flow velocity.

2. Analysis model development

2.1. Acoustic model

2.1.1. Governing equation

The governing equation used in an acoustic analysis is the Helmholtz equation, which is shown in [9]:

$$\nabla^2 p(\mathbf{r}) - k^2 p(\mathbf{r}) = -j\rho_0 \omega q(\mathbf{r}) \tag{1}$$

The acoustic pressures are governed by the Helmholtz equation given by Eq. (1). The sound pressure of arbitrary position in the acoustic domain can be approximately obtained through the interpolation of the nodes. Under the assumption of time-harmonic wave, the time varying pressure $p(\mathbf{r}, t)$ is separated as

$$p(\mathbf{r}, t) = p(\mathbf{r})e^{j\omega t} \tag{2}$$

where ω is the angular frequency; $p(\mathbf{r})$ is the complex amplitude function of the sound pressure.

To seek the sound pressure at the specified reference point or surface, the FEM equation of e-th finite element can be calculated through the minimal potential energy principle [7]

$$[K^{(e)} - k^2 M^{(e)}]P^{(e)} = F^{(e)} \tag{3}$$

where $K^{(e)} = \int_{V^{(e)}} \nabla N \nabla N^T dV^{(e)}$ denotes the element stiff matrix evaluated in the domain $V^{(e)}$; $M^{(e)} = \int_{V^{(e)}} NN^T dV$ is the element mass matrix; $F^{(e)} = \int_{\partial V^{(e)}} fN^T d\partial V^{(e)}$ is the equivalent load vector of the element node; $\nabla N = [N_1 \quad N_2 \quad \cdots \quad N_n]^T \left[\frac{\partial}{\partial x} \quad \frac{\partial}{\partial y} \quad \frac{\partial}{\partial z}\right]$ is the derivative matrix of the shape function; e is the finite element of the acoustic domain.

We use a finite-element discretization of the Helmholtz equation to simulate the sound propagation in the muffler. With the finite element discrete method of Galerkin, the above equation is discretized to obtain the element matrix. The weak form of the Helmholtz equation is then obtained by integrating over some local volumes. Numerical calculations are performed through COMSOL Multiphysics (versions 5.2) to obtain the solutions of the partial differential equation and combined with MATLAB to utilize a stochastic optimal algorithm GA.

2.1.2. Acoustic boundary and initial setup

In this example, the parametric optimization of the intake muffler is analyzed. The frequency-domain acoustic analysis of the acoustic system reveals that the characterization of the noise source is related with the base frequency of the engine. Thus, the optimal objective is to minimize the acoustic energy at the target frequency, that is, to maximize the reflected energy of the muffler. One of the commonly measurements of performance for a muffler is TL, which can be theoretically simulated with the software and verified by the experiment as well. In this paper, TL is defined as the difference of the sound power levels between the inlet and outlet, which is calculated as

$$TL(f_t) = 10 * \ln\left(\frac{W_{in}}{W_{out}}\right) \tag{4}$$

where $TL(f_t)$ is the TL value at the target frequency f_t, dB; W_{in} is the sound power at the inlet, W; W_{out} is the sound power at the outlet, W.

The sound power in (4) can be calculated as

$$W_{in} = \iint_{S_1} \left(\frac{p_{in}^2}{2 * \rho * c}\right) dS_1 \tag{5}$$

$$W_{\text{out}} = \iint_{S_2} \left(\frac{p_{out}^2}{2 * \rho * c}\right) dS_2 \tag{6}$$

where p_{in} is the plane-wave sound pressure at the inlet, Pa; p_{out} is the sound pressure at the outlet, Pa; ρ is the density of the medium, kg/m^3; c is the sound velocity of the medium, m/s.

Note that the excitation is only one single target frequency lower than the inlet tube cut-off frequency, coupled with the hypotheses that the acoustic source at the entrance of the muffler is approximated to plane wave. The boundary and initial setups of the sound field are shown in Fig. 1, which are depicted as follows:

1. The initial value of the plane-wave sound pressure at the inlet boundary is set as $p = 1$ Pa (TL is independent with the value of incoming sound pressure through a wide range of numerical verifications), which is specified using the real constant function. And a detailed derivation of Dirichlet-to-Neumann (the details of DtN operator in Eq. (7) are in [23]) type boundary conditions can be expressed as follows:

$$\frac{\partial p}{\partial n} - DtN(p) = 2ikf_t \tag{7}$$

$$\int_{S_1} rf_t^2 dS_1 = 1 \tag{8}$$

2. Anechoic boundary conditions [7] are imposed at the outlet boundary to ensure that all outgoing waves are perfectly absorbed and non-reflecting, with the acoustic characteristic impedance satisfying

$$z = \rho c \tag{9}$$

3. As the exterior wall of the muffler and the inner insert pipes are rigid, the sound pressure is such that the normal derivative equals zero, that is

$$\frac{\partial p}{\partial n} = 0 \tag{10}$$

2.2. Fluid model

2.2.1. Governing equation

Substantial significant researches and experiments on flow channels have been conducted. The properties of fluidic pressure and velocity in fluid domain can be described using the incompressible Navier–Stokes equations [25]. The physical field is treated as a linear time invariant system, and the initial conditions

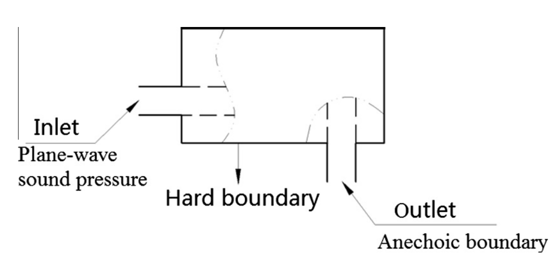

Fig. 1. Boundary and initial setups of the sound field.