On the retrofitted design of a truck muffler with cascaded sub-chambers

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Mufflers are widely used in engine exhaust systems for providing sound attenuation in a specific frequency range. In this case study, an engineering problem on reducing the noise radiation from a truck is investigated. The original muffler used by the truck consists of four reactive sub-chambers with similar geometries. Both experiments and numerical simulations revealed the narrow band transmission loss (TL) characteristics of the muffler which limit its attenuation performance, resulting in noise leakage at some particular frequencies. With an aim to broadening the attenuation bandwidth, a recently proposed sub-chamber design method is implemented to retrofit the muffler. The inner structure of the muffler is modified under the guidance of numerical simulations, in order to create extra acoustic stop-bands for an overall broadband performance. The retrofitted muffler is experimentally verified with a better TL response in the design frequency of interest. Analyses demonstrate the effectiveness of using the proposed simulation-based design method, which also show potentials to a wide range of muffler design applications.

Primary subject classification: 34.2; Secondary subject classification: 13.2.4

1 INTRODUCTION

Commercial mufflers are usually designed in multiple-chamber configurations in order to cater for particular applications with specific noise signatures[1]. To maximize the attenuation performance, many studies have been devoted to developing various muffler design methods. For example, Lee and Kim[2] tuned the layout of partitions inside an expansion chamber using finite element method (FEM) with a topology algorithm, where the transmission loss (TL) at a few selected frequency points is set as the design target. Chiu and Chang[3] attempted to optimize the shape of a multi-chamber muffler with perforated inlet extension using transfer matrix method (TMM) with a genetic algorithm. Aiming at a broadband performance, Seo and Kim[4] showed that combining an array of Helmholtz resonators leads to a broadband attenuation in the low frequency range. Similar analysis by using an array of narrow cavity resonators was conducted by Tang[5]. A review of different shape optimization techniques for reactive mufflers was given by de Lima et al.[6,7].

Recently, a sub-chamber method for silencer design was proposed[8], which theoretically pointed out that a broadband TL design can be achieved by cascading a number of sub-chambers, each with optimized TL for different frequency regions. The fact that the previous study was only verified numerically gives us the motivation of the present paper, that is, to demonstrate the effectiveness of the proposed method through a real-life case study on a retrofitted truck muffler.

Noise emission from trucks causes environmental concerns in the surrounding areas of a surface mining site. As shown in Fig. 1, the existing muffler used by the truck comprises a simple elliptical expansion chamber design with offset inlet/outlet, which is divided into four sub-chambers by three rigid cross-sectional partitions. In the first and the fourth sub-chambers, perforated ducts are added to regulate the air flow, with short overshoots into

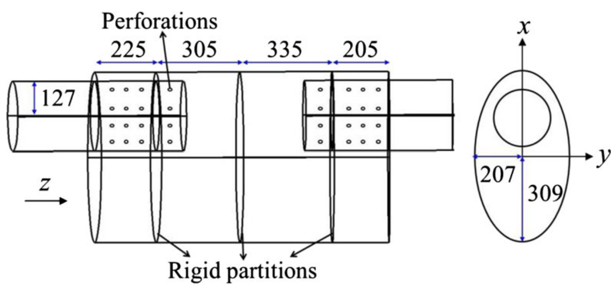

Fig. 1—Geometry and dimensions of the original muffler, in mm.

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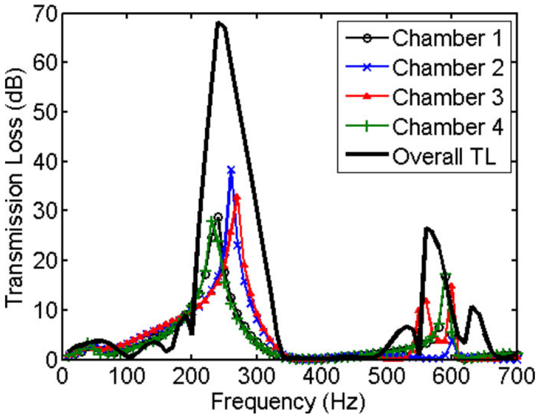

Fig. 2—Predicted TL characteristics of the sub-chambers and the overall muffler TL.

the second and the third sub-chambers. The acoustic performances of elliptical mufflers have been investigated by Selamet and Denia[9] and Sohei et al.[10].

It is found that the TL of the existing muffler has only one peak centered at around 250 Hz (shown in Fig. 2) and the sound attenuation below 200 Hz and of 330 to 500 Hz was very weak. Hence a modification for the existing muffler is needed to tackle the strong noise emission at those frequencies. In this paper, the inner structure of the muffler is modified under the guidance of the proposed sub-chamber design[7], in which numerical method using FEM analysis and experimental measures are employed. The designing process proves to be clear and concise. The retrofitted muffler is experimentally verified showing improved silencing performance in the design frequency of interest. The presented results and discussions demonstrate the practical significance of the proposed design, showing potentials to a wide range of industrial muffler applications.

2 ANALYSIS ON THE EXISTING MUFFLER

The muffler TL is predicted by numerical simulations using commercial FEM software COMSOL Multiphysics. The acoustic module is used, where the sound pressure field within the muffler domain is obtained by solving the Helmholtz's wave equation:

$$\nabla\left(\frac{1}{\rho}\nabla p\right) - \frac{1}{\rho c^2}\frac{\partial^2 p}{\partial t^2} = 0, \qquad (1)$$

where p is the sound pressure; c is the sound speed of air and ρ is the air density, taken as c = 343 m/s and ρ = 1.25 kg/m^3 in the simulation. A plane incident wave with pressure amplitude p_0 is assumed at the inlet and the muffler is terminated by applying a perfectly matched layer (PML) at the outlet. The muffler domain is discretized by free tetrahedral element whose size is smaller than 1/10 of the shortest wavelength. For a typical analysis reaching up to 700 Hz, around 5×10^4 degrees of freedom are generated to solve the pressure field response.

The perforated duct in Fig. 1 is described by applying an interior perforated plate boundary condition. The semi-empirical impedance formula, which neglects the flow effect, is given as[11]:

$$Z = \frac{1}{\sigma}\left[\sqrt{\frac{8\mu k}{\rho c}}\left(1+\frac{t}{d}\right) + ik(t+\delta)\right], \qquad (2)$$

where μ is the dynamic viscosity; $k = 2\pi f/c$ is the wavenumber; t is the depth of the hole, which equals to the plate thickness; d is the hole diameter; σ is the perforation ratio, or the percentage of area occupied by the holes and δ is the end correction factor. The measured perforation parameters as the input to the simulation model are d = 4.5 mm, t = 2 mm, σ = 10% and δ = 0.45 t. The TL of the muffler is evaluated by:

$$\mathrm{TL} = 10\log(\Pi_i/\Pi_t), \qquad (3)$$

where $\Pi_i = \int_{S_i} p_0^2/(2\rho c)dS_i$ is the incident sound power from the inlet and $\Pi_t = \int_{S_t} p^2/(2\rho c)dS_t$ is the transmitted sound power to the outlet, assuming that the frequency range of interest is below the duct cut-off frequency. The cut-off frequency for the inlet and outlet duct as shown in Fig. 1 is around 780 Hz[12].

Using the developed FEM model, the acoustic characteristics of the individual sub-chambers constituting the whole muffler in Fig. 1 and the overall muffler TL are studied. In Fig. 2, the TLs of the sub-chambers are presented separately, showing similar attenuation responses. The distinct TL peak for the sub-chambers between 200 and 300 Hz comes from the destructive interference between the open-cavity modes. After connected, the overall muffler TL exhibits the combined behavior of the sub-chambers with a stronger TL peak centered at 250 Hz. The overall TL may be slightly lower than the sub-chamber TLs below 180 Hz because of their strong mutual coupling in the low frequency range, as explained in Ref. 8. This numerical analysis correlates with the measured noise spectrum at 10 m away from the truck, facing directly to the muffler outlet, as shown in Fig. 3. To measure the sound pressure level (SPL), the truck is located on a concrete ground surface which radiates nearly freely into a semi-infinite space. It can be seen that an SPL valley between 200 and 330 Hz appears at nearly the same location as

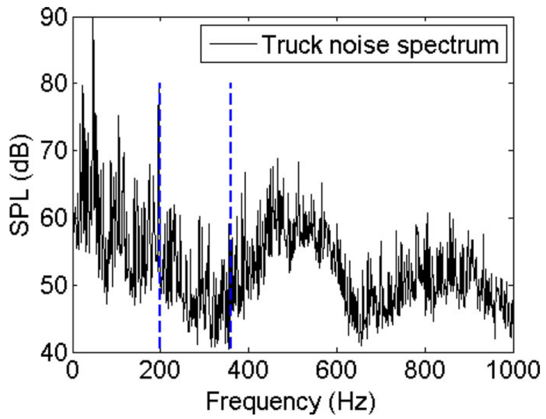

Fig. 3—Noise spectrum measured 10 m away from the muffler, with the truck in high idle condition.

the TL peak of the muffler, whereas the distinct SPL peaks at frequencies below 200 Hz and from 330 to 500 Hz are shown to be caused by the narrowband behavior of the muffler TL. The distance between the surface mining site, the operating location of the truck and the neighboring residences is more than 1 km, meaning that the high-frequency noise components are less important compared to the low-frequency ones. Therefore, frequencies below 200 Hz and of 330 to 500 Hz are chosen for noise elimination.

3 RETROFITTED MUFFLER DESIGN

The engineering constraint for the muffler retrofitting work requires the outer dimension of the elliptical chamber to be retained. In seeking a broader attenuation performance, the TL characteristics and the design limits of sub-chambers are considered. The original muffler involves only two types of sub-chambers, namely the empty elliptical chamber with varying width w and

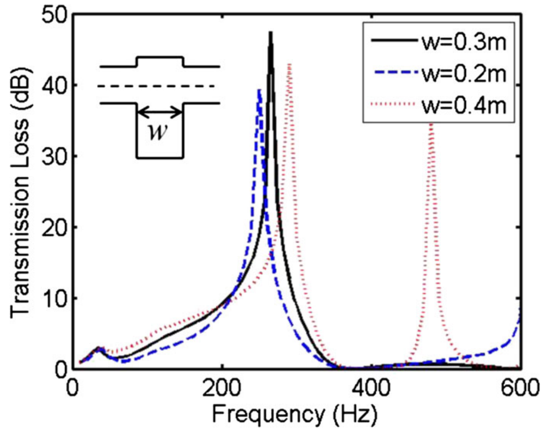

Fig. 4—Effect of varying width, w, of an empty sub-chamber.

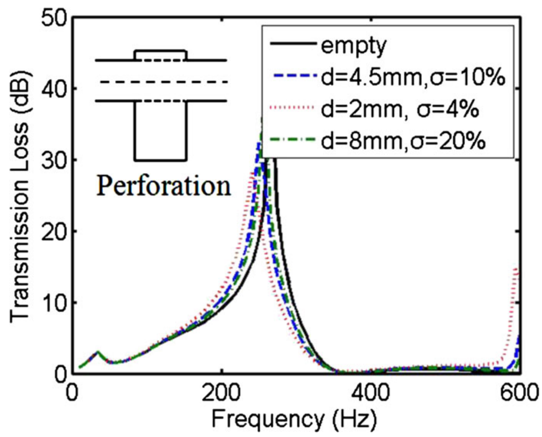

Fig. 5—Effect of varying parameters of the perforated duct, chamber width, w = 0.3 m.

those with a perforated duct. In Fig. 4, the TLs of the empty chambers with varying width w from 0.2 to 0.4 m are presented. Those with different perforation diameter d and ratio σ under a fixed chamber width $w = 0.3$ m are plotted in Fig. 5. It can be seen that these TL curves only show slight variations in the design frequency range and the stop-band defined as TL greater than 10 dB can hardly be tuned beyond 190 to 310 Hz by varying chamber width or inserting perforated duct. Therefore, the simulation results suggest that: (a) for the current elliptical chamber design with offset inlet/outlet, the TL resonance is mainly controlled by the depth of the chamber (vertical to the axial direction), not the width; (b) the acoustical effect of the perforated duct is less important compared to the shape factor. Instead, it can be used to guide the exhaust flow to reduce the pressure drop.

It is clear from the preceding analyses that in order to achieve a broadband attenuation, the existing muffler needs a retrofitted inner structural design. The sub-chamber design proposed in Ref. 7 theoretically demonstrated that, reactive silencer comprising cascaded sub-chambers with TL designed in different frequency regions can possess an overall broadband TL. Therefore,

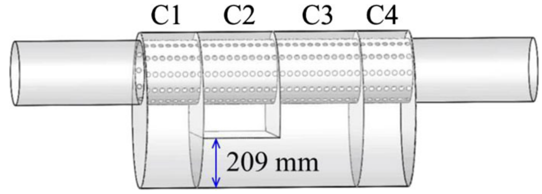

Fig. 6—Configuration of the retrofitted muffler design. The outer dimension of the elliptical chamber and the diameter of the inlet/outlet duct remain unchanged, the same as in Fig. 1.

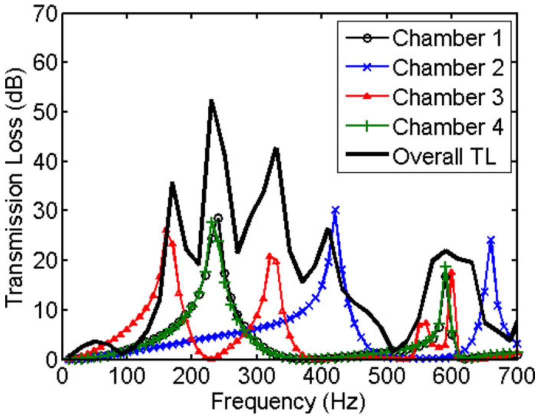

Fig. 7—Simulated sub-chamber TLs and overall TL of the retrofitted muffler.

to create extra stop-bands below 200 Hz and from 330 to 500 Hz, the following configuration as sketched in Fig. 6 is considered. Compared with the original muffler, the outer dimension of the elliptical chamber, annular inlet/outlet ducts and the first and fourth sub-chambers (including the perforated ducts) remain unchanged, whereas the second and third sub-chambers are modified to create a longer and a shorter characteristic chamber depth. Such modification is to deliberately split the TLs of the four sub-chambers which originally overlap at 250 Hz. Meanwhile, a perforated duct is added through the muffler to reduce the possible pressure drop.

Using the developed FEM model, tuning of the partition parameters is performed by simple trial and error process until an acceptable performance is obtained. Certainly, more rigorous optimization can be carried out by prescribing the objective function and selecting an optimization algorithm. An example has been demonstrated in our previous theoretical studies[8]. In Fig. 7, the TLs of the modified sub-chambers are presented, which are seen to provide extra TL bands from 370 to 460 Hz by the second sub-chamber with shorter characteristic depth and from 130 to 190 Hz and 310 to 350 Hz by the third sub-chamber with a longer characteristic depth. As can be seen from the simulated overall TL of the revised muffler, the original pronounced TL peak centered at 250 Hz is maintained, while a much wider stop-band is obtained, with TLs greater than 10 dB covering from 140 to 460 Hz.

4 EXPERIMENTAL SETUP

Experiments are conducted to determine the actual TL of both the original and retrofitted muffler. With frequency of interest lower than the cut-off frequency of the first duct mode, the four-microphone, two-load method[13] is employed for the measurement. Figure 8 presents the modification on the basis of the original muffler. The second partition is cut from the middle and an extra plate is added horizontally to reshape the second and third chamber.

Figure 9(a) shows the experimental test-rig. The incident noise is produced by a loudspeaker, where white signal is generated and amplified successively by a signal

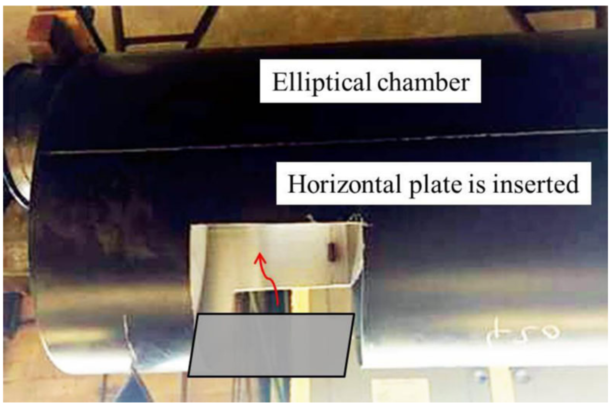

Fig. 8—Modification on the existing muffler: the second sub-chamber is reshaped by inserting a horizontal plate (refer to Fig. 6).

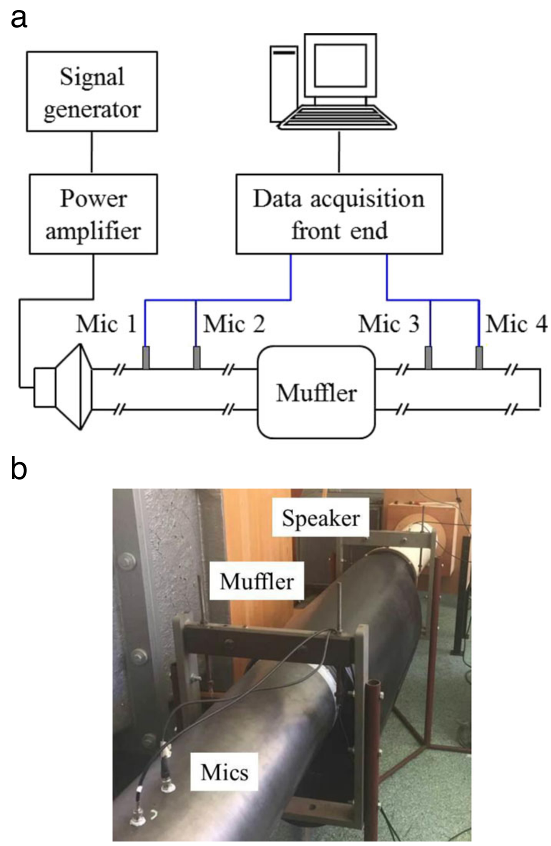

Fig. 9—(a) Test-rigs based on four-microphone method to measure the muffler TL; (b) The actual muffler configuration.

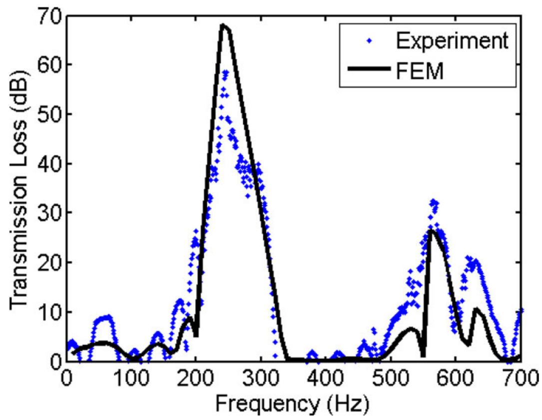

Fig. 10—Comparison between measured and simulated TLs for the original muffler in Fig. 1.

generator (HP 8904A) and a power amplifier (Bruel & Kjaer 2706). Two pairs of phase-matched microphones (BSWA Tech, ½″) are installed on upstream and downstream ducts respectively, with exact locations shown in Fig. 9(a). Signals from the four microphones are acquired by Bruel & Kajear acquisition front end hardware at the sampling rate of 6.4 kHz, whereas the data acquisition and post processing are controlled on PC equipped with Pulse Labshop. The TL of the muffler is obtained by conducting two independent experiments with different downstream loadings[13].

5 RESULTS AND DISCUSSIONS

The TLs of the original and retrofitted muffler are measured by following the description in Sec. 4, which are presented in Figs. 10 and 11, respectively. The good agreement between the predicted and measured TLs validates the accuracy of the simulation. By comparing the experimental TLs of the retrofitted muffler to that of the original one in Fig. 12, it can be seen that the effective attenuation band is greatly enlarged. Experimental results show that the retrofitted muffler provides a stopband with TL greater than 10 dB ranging from 140 to 420 Hz, as compared to 190 to 330 Hz for the original one. The attenuation bandwidth is doubled from 140 to 280 Hz, which is an appreciable improvement for the exhaust noise control.

While the present paper concentrates on a retrofitting case study, a general, multi-level design method is summarized to guide a comprehensive muffler design. As shown in Fig. 13, the flow chart involves the target setting, design and verification stages. The starting point requires a clear identification of the noise problem (noise SPL), noise generating mechanism and other factors such as exhaust temperature and flow speed (although in this work the effect of flow and temperature variance on the TL was not considered). A design target is set as the required noise reduction as a function of frequency, with other design constraints such as geometric and flow criteria taken into account.

The design procedure is guided by the sub-chamber method. Given a prescribed frequency range, the entire frequency is divided into several sub-intervals, each being tackled with a specific sub-chamber. Considering the geometric constraints, a systematic parametric study by analyzing the individual TLs of sub-chambers, such as the effective frequency range, TL peak location and bandwidth, is performed to reveal their design limits. The sub-chamber parameters are then tuned and the overall performance of a muffler realized by cascading the selected sub-chambers is then evaluated to see if the targeted broadband attenuation is achieved. If not, further parametric tuning is possible, which forms a closed-loop design with greatly reduced number of trials and thus computational costs, compared with the global optimization over the whole muffler. Before arriving at

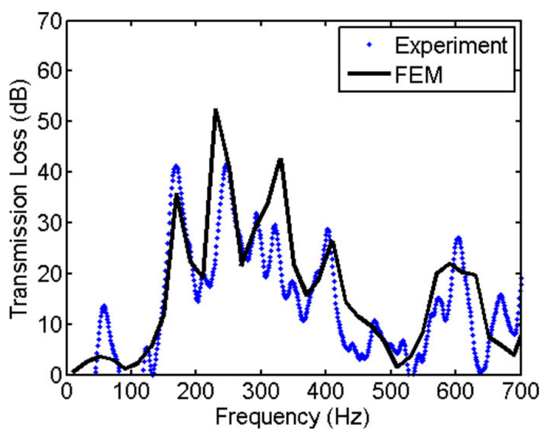

Fig. 11—Comparison between measured and simulated TLs for the retrofitted muffler in Fig. 6.

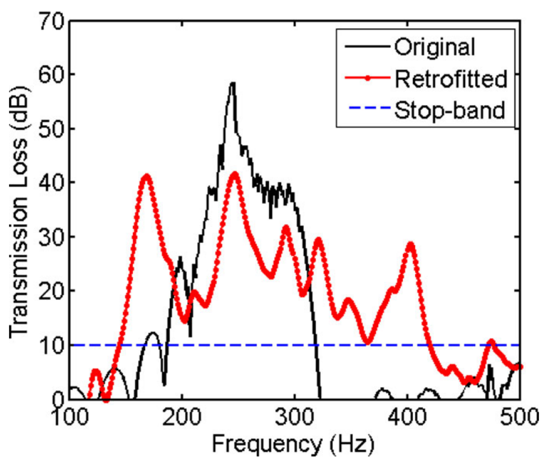

Fig. 12—The measured TLs of the original and retrofitted muffler from 100 to 500 Hz.

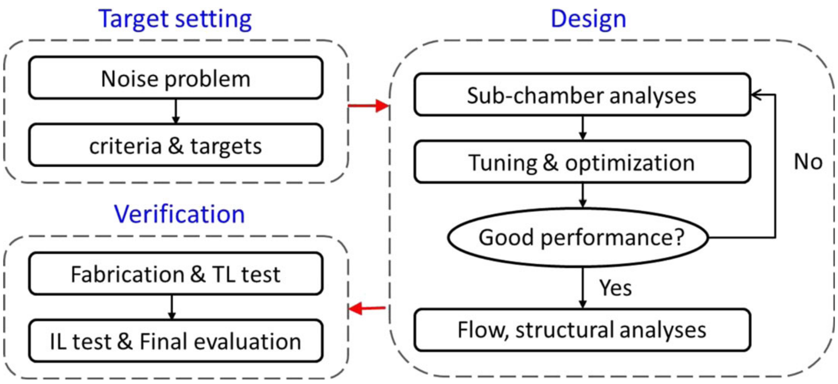

Fig. 13—Flowchart of the cascade muffler design.

a final design, other engineering constraints such as flow, thermal and structural considerations have to be properly addressed. The final verification step will fabricate the muffler prototype and assess its performance using either TL or insertion loss (IL) criterion.

6 CONCLUSIONS

The effectiveness of a proposed sub-chamber method for muffler design has been investigated through a real-life problem on retrofitting a truck muffler. In this case study, a typical muffler showing unbalanced, narrow band TL performance was analyzed.

To improve the TL in the deficient frequency range, the original muffler was retrofitted with a different inner structure, by tuning sub-chamber TLs to cover additional frequency bands. Simulated results on the retrofitted muffler displayed an improved broadband, balanced TL performance, which was experimentally verified. The retrofitted muffler provides a doubled attenuation bandwidth with TL greater than 10 dB.

As a final remark, a general multi-level flowchart raised from the present case study was discussed. It is concluded that the proposed design method based on local tuning or optimization of sub-chambers greatly reduces the design costs compared with global design, which shows great potentials to a wide range of industrial muffler applications.

7 ACKNOWLEDGMENTS

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