STRUCTURAL CONFIGURATION AND ACOUSTICAL PERFORMANCE OF ENGINE EXHAUST MUFFLERS FOR CONSTRUCTION EQUIPMENT AND MACHINERY

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Abstract
The main objective of the present work is to characterize numerically the link between the geometrical parameters and the functional noise reduction properties of typical engine exhaust mufflers for construction equipment and machinery. To this regard, the acoustical modeling of these devices is first presented using the finite element approach. Then, we investigate the effects of both the geometrical parameters and the shape configuration of some typical mufflers (e.g., chambered, turbo, reactive, absorptive…) on their sound transmission loss. In terms of acoustics, for a given total length (∼600 mm), mufflers show a better acoustic performance (up to ∼50 dB) with a high number of chambers/partitions or using absorbing layers, especially micro-perforated panel with hole size of 2×4 mm and its fraction 4×8% can improve the sound transmission loss up to ∼100 dB at low frequency less than 1 kHz. The obtained results show clearly that the noise reduction target could be successfully achieved over an individual octave band or the whole frequency range by tuning structures of appropriate engine exhaust mufflers.

Keywords: Noise emission; engine exhaust muffler; construction equipment and machinery; sound transmission loss.

1. Introduction
Dealing with noise problems in the construction/mining industry is gained more and more technical experts as well as acousticians. Mufflers or silencers having expansion chamber shapes have been extensively in the exhaust systems of internal combustion engines for noise control purpose [1], [2]. As demonstrated in a number of reference studies, the acoustic behavior of exhaust mufflers depends strongly on their geometrical aspects such as the chamber shape [3] - [6], the area ratio between ducts and their relative position [2], [5], [7]. Three main approaches have been proposed in the literature to characterize the relationship between the geometrical parameters of exhausted mufflers and their acoustical performance: analytical, numerical and experimental approaches. The first category focuses on finding a theoretical solution, leading to better understand the mathematical and physical meaning of the macroscopic
equations governing acoustic dissipation phenomena in muffler domains with simple shapes (single inlet/outlet [3], [4], or single-inlet/double-outlet [1]). The second category of studies, based on numerical techniques such as finite element method (FEM) [8] - [10] and boundary element method (BEM) [11], [12], has also been successfully used to predict the acoustic attenuation performance of expansion chambers with a variety of geometrical properties. The third category characterizes the acoustical performance of actual mufflers by measuring their transmission loss. Two alternative measurement approaches are often performed [13] either the two-load method [14] or the two-source method [15].

In this paper, the wave propagation problem in chamber mufflers is considered by the finite element approach. The physical model of wave traveling is first introduced by the governing equation in the muffler chamber domain, and the mathematical description of its acoustical performance is provided. After the validation step, the influence of the geometry characteristics on the acoustic behavior (e.g., sound transmission loss) of some typical exhaust mufflers is studied.

2. Construction machinery and noise problem

Noise pollution in the surrounding mining areas or construction sites within variety of equipment causes by several sources of noise such as processing plants [16], blasting and machinery [17] among others. The characteristics of machinery sound (i.e., sound levels associated with heavy construction equipment range from 80 to 120 dB and power tools commonly used in construction produce sound levels up to 115 dB [18], also see the noise level of some equipment shown in Tab. 1) is important factor to consider when impact assessment is developed in the stage of seeking noise generating points [19], [20].

Tab. 1. Typical noise level measurements at operator’s position for construction equipment [17]

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Noise level (dB)</th>
<th>Equipment</th>
<th>Noise level (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crane</td>
<td>78 - 103</td>
<td>Compactor</td>
<td>90 - 112</td>
</tr>
<tr>
<td>Backhoe</td>
<td>85 - 104</td>
<td>Grinder</td>
<td>106 - 110</td>
</tr>
<tr>
<td>Loader</td>
<td>77 - 106</td>
<td>Chainsaw</td>
<td>100 - 115</td>
</tr>
<tr>
<td>Dozer</td>
<td>86 - 106</td>
<td>Concrete saw</td>
<td>97 - 103</td>
</tr>
<tr>
<td>Scraper</td>
<td>97 - 112</td>
<td>Sand blasting nozzle</td>
<td>111 - 117</td>
</tr>
<tr>
<td>Trencher</td>
<td>95 - 99</td>
<td>Jackhammer</td>
<td>100 - 115</td>
</tr>
<tr>
<td>Pile driver</td>
<td>119 - 125</td>
<td>Compressor</td>
<td>85 - 104</td>
</tr>
</tbody>
</table>
Here, Terex RH200 hydraulic shovel (left part of Fig. 2), a hydraulic mining excavator (525 ton, 2500 HP, 26 m³) used in pre-stripping operations, is demonstrated as an example. This heavy shovel is powered by two 12-cylinder engines (Cummins K1500-E). The original engine exhausts system of this equipment has no noise solution/control (e.g., muffler/silencer) that results in the untreated noise source sound power levels at different locations shown in Tab. 2. However, by installing the muffler, the treated engine exhaust sound power levels having an engine exhaust noise reduction of 20.9 dB are displayed in the right part of Fig. 2. This example indicates that the noise environment pollution is mainly produced mining operation and the noise reduction target could be successfully achieved with appropriate mufflers.

**Tab. 2. The untreated noise source sound power levels measured in Terex RH200 shovel [21]**

<table>
<thead>
<tr>
<th>Location</th>
<th>Octave band center frequency (Hz)</th>
<th>Sum (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>31.5</td>
<td>63</td>
</tr>
<tr>
<td>Hydraulic radiators</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(side)</td>
<td>102.0</td>
<td>106.4</td>
</tr>
<tr>
<td>(top)</td>
<td>102.1</td>
<td>99.8</td>
</tr>
<tr>
<td>Engine exhausts</td>
<td>112.4</td>
<td>121.0</td>
</tr>
<tr>
<td>Engine room</td>
<td>104.9</td>
<td>106.6</td>
</tr>
<tr>
<td>Engine radiators</td>
<td>104.7</td>
<td>108.0</td>
</tr>
<tr>
<td>Hydraulic boom</td>
<td>98.4</td>
<td>99.3</td>
</tr>
</tbody>
</table>

**Fig. 1. (Left) Terex RH200 with silencer installation (inset graph corresponds the machine without noise control); (Right) The sound power level from engine exhausts without (circle) and with (square) the muffler. The materials are reproduced from [21]**

In terms of noise reduction resource guideline for construction machinery, the amount of noise reduction that can be obtained from an individual noise control or suite of controls is dependent on a large number of factors (e.g., type of machines and duties,
quality and expensive of used materials, see [22] for a survey). However, installing exhaust mufflers for the engine system is always a primary and potential solution.

3. Geometrical designing and acoustical modeling

The exhaust mufflers can be cataloged in four grades [23]: industrial, residential, critical and super critical levels of insertion loss (i.e., the level of sound reduction after attaching mufflers). In addition, this level of sound reduction is strongly depending on the general geometrical factors (e.g., body/pipe diameter length/pipe diameter…) and chamber configuration (e.g., shape and connection of acoustic chambers).

Fig. 2. (A) Structure configuration of typical exhaust muffler. (B) Illustration of one dimensional stream line (thick dash line) for (a) chambered (CF6) and (b) turbo (CF3) muffler

For practical applications, there are many muffler structures applied in engine technologies. As seen in Fig. 2(A), the simplest muffler is a single inlet/outlet chamber (CF1) or with two outlets (CF2). Considering the number of chambers, mufflers can be classified: single chamber (CF1, CF2) or multi-chamber (CF3-6). Fig. 2(B) shows a detailed structure of CF3 and CF4-6 type mufflers, for late use, several geometrical parameters of these mufflers are also demonstrated. To improve the acoustical performance for some specific cases, several added components can be applied to control the acoustical performance of mufflers such as dividing/tuning components (e.g., V-blade,
resonant chamber/tube/wall) and absorbing structures (e.g., porous materials, microperforated tube/wall). These characteristics can provide an interesting acoustical property of mufflers (see Sec. 4).

The wave propagation governed in the muffler chamber domain is written [2]:

\[
\nabla^2 p - \frac{1}{c_0^2} \frac{\partial^2 p}{\partial t^2} = 0
\]

(1)

where \( p \) is the acoustic pressure, \( c_0 \) is the speed of sound in air, \( t \) is the time variable, and \( \nabla^2 \) is the Laplace operator.

Three different boundary conditions are implemented as follows:

\[
\left(-\frac{\nabla p}{\rho}\right) n = \begin{cases} \frac{j \omega}{\rho c} - \frac{2 j \omega}{\rho c} p_0 n & (BC_1) \\ 0 & (BC_2) \\ \frac{j \omega}{\rho c} & (BC_3) \end{cases}
\]

(2)

where \( \omega \) is angular frequency, \( j \) is the imaginary unit, \( p_0 \) is the applied outer pressure, \( n \) is the normal vector of the boundary. An incoming-outgoing plane wave (BC\(_1\)) and an outgoing plane wave (BC\(_3\)) are assumed respectively at inlet and outlet of the muffler. Sound hard (wall) boundary condition (BC\(_2\)) is applied to the outer walls of the muffler, separating wall between the resonating chambers and the walls of the tubular pipes.

Transmission loss (TL) is defined as the difference between the power incident on a muffler and that transmitted downstream into an anechoic termination. This parameter describes the accumulated decrease in intensity of waveform energy as a wave propagates outwards from a source. The transmission loss of an exhausted muffler is calculated as [24],

\[
TL = 10 \log_{10} \frac{W_{in}}{W_{out}}
\]

(3)

where \( W_{in} \) and \( W_{out} \) denote the incident and transmitted sound powers at the inlet and outlet of the muffler, respectively. These powers are deduced by,

\[
W_{in} = \int_{\partial \Omega} \frac{|P_0|^2}{2 \rho c} dS, \quad W_{out} = \int_{\partial \Omega} \frac{|P|^2}{2 \rho c} dS
\]

(4)

From the structure of mufflers and the governing equations over their fluid domain and boundary, the desired field solutions can be obtained by using commercial
finite element packages or in-house programs. Here, COMSOL Multiphysics® v5.2 is used for all finite element computations proposed in the following section.

4. Results and discussion

4.1. Validated examples

A single inlet/outlet muffler with an available analytical model (see Eq. (4.26) in [25]), numerical (BEM) and experimental results of transmission loss [13]. The TL comparison showing in Fig. 3 indicates that our numerical result agrees especially well with the reference predictions as well as the two-source method. The obtained good agreement gives a strong validation for the proposed finite element procedure.

![Fig. 3. Transmission loss comparison of single inlet/outlet muffler configuration. The dimensions (insert graph) are given: \(d_1 = d_3 = 1.375\) in, \(d_2 = 6.035\) in, \(l_1 = l_3 = 1.5\) in, \(l_2 = 8\) in](image)

4.2. Effects of geometrical parameter

![Fig. 4. Transmission loss of various chambered mufflers: Single chamber (line with circle marker) and multi-chamber (lines without marker)](image)
In this section, we focus on investigating the acoustical properties of some multi-chamber mufflers (see Fig. 2B for a detailed geometrical diagram). Here, it is noted that all CF-muffler configurations have the total chamber length of 300 mm and a cross-section expansion chamber of 60×180 mm. The main parameters of the chambered muffler CF4-6 are: \( l_1 = 60 \text{ mm}, \ l_2 = 40 \text{ mm}, \ l_3 = 60 \text{ mm}, \ l_4 = 40 \text{ mm}, \ l_5 = 60 \text{ mm}, \) whereas the CF2-like muffler is 300-length single chamber one with a rectangle cross-section. The location of the inlet and outlet tube is given as: \( h_1 = h/2, \ h_2 = 3h/4. \) It can be seen clearly from Fig. 4 that the three-chamber mufflers have a higher level of transmission loss compared with the single chamber one. There is a slight difference between the acoustic behavior of straight and elliptic chamber shape (see thick solid and thin dashed line). By using tuning component (e.g., V-blade), we can modify the transmission loss trend having a lower TL at frequency range 500÷1250 Hz and a higher TL at frequency range 1250÷2000 Hz (e.g., lines named CF6 and CF4).

![Fig. 5. Transmission loss of chambered muffler compared with turbo muffler](image)

Fig. 5. Transmission loss of chambered muffler compared with turbo muffler

Fig. 5 depicts the TL performance comparison of the turbo (CF3) and chambered (CF5) mufflers. The dimensions of CF3 are given as: \( l_1 = 50 \text{ mm}, \ l_3 = 190 \text{ mm}, \ l_5 = 40 \text{ mm}, \ l_2 = l_6 = 10 \text{ mm}, \) and \( h_3 = h/4. \) It is observed that the CF3 shows a good TL behavior in both low (<420 Hz) and high (>2380 Hz) frequency ranges. The higher sharp peak (94.35 dB) of CF3 occurs at a lower frequency (~2020 Hz) in compared with a lower peak of 84.66 dB at a higher frequency 2180 Hz of the CF5 muffler.

The multi-chamber hybrid silence having solid partitions is selected for investigating the influence of the geometrical parameters on its acoustic behavior. Three 500-mm length mufflers have 1, 3, and 5 sub-chambers using solid partitions (see the insert graph of Fig. 6 for a 5-chamber muffler). It can be seen from Fig. 6 that the TL
performance of the simplest empty chamber is rather weak in the entire frequency range (the single and triple chambers), whereas the 5-chamber muffler provides a pronounced sharp peak near 920 Hz. This sharp TL peak is actually due to the accumulative effect of connecting multiple identical unit cells in series, forming a kind of resonator array.

![Graph showing transmission loss vs. frequency for different partitions (0, 2, 4)](image)

**Fig. 6.** Effects of number of sub-chambers or partitions on the transmission loss. Insert graph shows a hybrid muffler with 4 partitions (dimensions are in mm)

### 4.3. Effects of sound absorbing components

![Graph showing transmission loss vs. frequency for muffler with and without sound absorbers](image)

**Fig. 7.** Effects of added sound absorbing materials on the transmission loss of the muffler as shown in Fig. 3 with its dimensions: \(d_1 = d_3 = 25\) mm, \(d_2 = 120\) mm, \(l_1 = l_3 = 60\) mm, \(l_2 = 300\) mm

Sound absorbing structure (e.g., porous material, micro-perforated panel…) is a potential solution for noise shielding, thus using this material class in exhaust mufflers provides an improvement in the cavity of transmission loss. In order to investigate the effect of a chamber cover made of absorbing materials (noted that the outside diameter of the chamber is kept), we use the simple muffler structure as shown in Fig. 3. The
added absorber is characterized by the characteristic impedance and the complex wave number, and these equivalent properties are deduced from the Delany-Bazley model with a given resistivity.

Fig. 7 shows that both the thickness and the properties of absorptive materials produce a steady ability of transmission loss, the poor TL performance in several frequency bands (thin dashed line) is improved (other lines). It is also noted that instead of adding a thicker absorbing layer (thin solid line); we can use a higher resistivity material (thick solid line) in order to obtain the enhanced TL.

![Graph showing transmission loss vs. frequency for different materials and configurations.]

**Fig. 7. Influence of absorptive materials on transmission loss.**

Fig. 8. Influence of the MPP properties on acoustical behavior of hybrid muffler

The improvement in acoustical performance of mufflers having micro-perforated panel (MPP) is examined by considering the hybrid structure with five sub-chambers as shown in the insert graph of Fig. 6. Noted that these sub-chambers are covered by the MPP panel with several configurations of the hole diameter \(d\) and the perforation ratio \(r_p\). Fig. 8 shows the influence of the MPP characteristics by comparing three micro-perforated cases: MPP1 \((d = 2\ \text{mm},\ r_p = 4\%)\), MPP2 \((d = 2\ \text{mm},\ r_p = 4\%)\), and MPP3 \((d = 2\ \text{mm},\ r_p = 4\%)\). In general, for three cases, the sharp peak of TL performance appears in a lower frequency in compared with the point of 920 Hz for the muffler without the MPP. Additionally, the TL performance of the hybrid muffler using MPP depends strongly on the diameter of holes and their perforation ratio. It is seen from the comparison that with decreasing hole diameter, the narrow TL peak becomes flattened and widened, leading to a better broadband attenuation performance.

5. Conclusion

In this study, a three-dimensional finite element modeling for predicting the acoustical performance of exhaust muffler structures is developed. The numerical procedure has been validated by comparison with reference predictions proposed
somewhere in the literature, and then employed to the analysis of the acoustic properties (e.g., transmission loss) of several muffler structures. From the obtained results, the following remarks can be stated: (i) both the individual chamber shape and its connection affect strongly on the transmission loss properties. It seems that the lost energy of sound wave propagating inside mufflers is related to the characteristics of the flow path. Both the chambered and the turbo configurations provide an increase TL over the whole range of considering frequency; (ii) using absorptive components for muffler structures also brings an interesting transmission loss. This solution can solve unwanted resonant at a number of frequencies for the original muffler chambers. We can design the acoustical performance of exhaust silencers by tuning either the properties of absorbing materials or the configuration of MPP; (iii) for a specific class of mufflers even having some fixed parameters (e.g., chamber volume, package sizes...), the acoustical performance of such muffler still can be tailored by tuning its geometrical configuration (e.g., length or area ratio, number of internal partitions, relative location of inlets/outlets...).

As a final remark, it should be mentioned that the numerical modeling procedure allows predicting the acoustical performance of complex muffler design in an effective design process. This can readily be used to optimize muffler-design for higher engine performance and lower noise emission in automobiles and machinery, in which the effects of muffler configuration on the engine efficiency and emission property will be also considered.

References


CÂU TRÚC VÀ HIỆU SUẤT ÂM CỦA ỐNG XÁ ĐỘNG CƠ DỤNG CHO MÁY VÀ THIẾT BỊ XÂY DỰNG

Tóm tắt: Mục tiêu chính của bài báo này là nghiên cứu mối liên hệ giữa các tham số hình học và tính năng giảm tiếng ồn của một số dạng ống xả động cơ lắp trên máy và thiết bị xây dựng. Để thực hiện, trước tiên mô hình âm học của các thiết bị này được trình bày nhờ sử dụng phương pháp phân tử hướng. Sau đó tiến hành khảo sát ảnh hưởng của các tham số hình học và hình dạng của một số bộ giảm âm điển hình (ví dụ dạng phân ngăn, turbo, phản kháng, hấp thụ) đến hệ số suy giảm âm. Về mặt âm học, với tổng chiều dài định trước (~ 600 mm), ống xả có hiệu suất âm tốt hơn (lên đến 50 dB) nhờ có nhiều khoang/phân ngăn hoặc sử dụng lớp vật liệu hấp thụ, cấu trúc tấm đục lỗ với kích thước lỗ 2-4 mm và với mật độ diện tích 4-8% có thể cải thiện tốn thất truyền âm lên tới 100 dB ở dải tần số thấp hơn 1 kHz. Kết quả nhận được cho thấy rõ ràng mục tiêu giảm tiếng ồn có thể đạt được ở những dải tần riêng lẻ hoặc trên toàn bộ dải tần bằng cách tùy chỉnh các thong số cấu trúc của bộ ống xả thích hợp.

Từ khóa: Phát thải tiếng ồn; ống xả động cơ; máy và thiết bị xây dựng; hệ số suy giảm âm.

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