Optimization of Noise Reduction Performance for the Muffler in a Heavy Vehicle

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Abstract. The noise reduction performance of the muffler is further optimized in order to study the acoustic performance of the high-performance composite muffler. On the basis of reasonable assumptions, the boundary values of the entrance and exit, which of the inner wall surface of the muffler are set up. The internal acoustic finite element model is established through HYPERMESH, and the transmission loss (TL) is calculated by the SYSNOISE module in Virtual.lab. Then, the effect on noise reduction performance by the main structural parameters of muffler, such as the hole diameter of the perforating tube, the number of expansion chamber, and the position of the main airflow passage are discussed. The experiment proves that the noise reduction performance of the muffler is improved effectively, proves the practicability of this method, and also gives a solution for the design and optimization of complex muffler.

1. Introduction
The noise from engine exhaust system takes the largest portion of the noise sources as most of the environmental noise comes from vehicles on the road. It has been proved that installing a muffler is most helpful for effective reducing the noise of the exhaust system. The principle of noise reduction by installing perforated tube is as follows. Under the action of acoustic pressure, the gas makes a piston-like reciprocating motion in the holes on the perforated plate when the airflow circulates inside the perforation of the muffler; the hole wall is continuously rubbed against the acoustic wave, so that part of the acoustic energy is converted into heat energy and the noise reduction is achieved[1]. It is difficult to accurately predict the noise reduction performance by adopting traditional methods for a complicated muffler, although it works well by that methods to predict and improve the noise reduction performance of a simple muffler. The use of three-dimensional numerical simulation on the noise reduction performance of the muffler has become an effective method for predicting and optimizing the noise reduction performance as it can overcome the problems of the above-mentioned muffler of complicated structure.

In this paper, a heavy-duty vehicle square muffler project is illustrated: acoustic impedance is used to simulate a large number of perforations on the muffler perforated plate, and the parameter curve of the muffler’s transmission loss evaluation is simulated. The paper also focuses on discussing the effect of the main structural parameters such as the hole size of the perforated tube, the number of expansion chambers, and the position of the main airflow passage on the noise reduction performance. The simulated data of noise reduction performance under different structural parameters is compared with the simulated curve of the original structure. The relationship between the noise reduction
performance and the structural parameters of the muffler is summarized, and then the optimal muffler structure model can be obtained.

2. The creation of finite element model

2.1. Basic assumptions

(1) Assuming that the medium is ideal gas, that means there is no energy loss during the acoustic transmission or no viscosity in the medium. It is assumed that the initial velocity of the medium is zero in a static, uniform density environment, the static pressure and static density is supposed to be constant values.

(2) It is assumed that the acoustic wave transfer is under adiabatic condition and there is no heat exchange in the process.

(3) When the acoustic wave is transmitted inside the muffler perforated tube, no energy is radiated to the outside, that is, the wall surface of the perforated tube is assumed to be rigid.

(4) Acoustic waves are transmitted in small amplitude in the air. The description of the acoustic wave can refer to the linear wave equation, and the parameters are all first order.

2.2. Wave equation

For the one-dimensional wave equation, the relationship between the particle velocity and the acoustic pressure at the entrance of the muffler pipe can be expressed as:

\[ p_{in} = (p_1 e^{-jkx} + p_4 e^{jkx}) e^{j\omega t} \]  
\[ v_{in} = \frac{j}{\rho \omega} \frac{\partial p_{in}}{\partial x} = \frac{1}{\rho c} (p_1 e^{-jkx} - p_2 e^{jkx}) e^{j\omega t} \]  
\[ (1) \]

Here, \( p_1 \) represents the incident acoustic pressure at the entrance; \( p_2 \) represents the reflected acoustic pressure at the entrance.

The acoustic pressure and particle velocity can be represented respectively as follows:

\[ p_{out} = (p_3 e^{-jkl} + p_4 e^{jkl}) e^{j\omega t} \]  
\[ v_{out} = \frac{1}{\rho c} (p_3 e^{-jkl} - p_4 e^{jkl}) e^{j\omega t} \]  
\[ (3) \]

Here, \( p_3 \) represents the incident acoustic pressure at the exit; \( p_4 \) represents the reflected acoustic pressure at the entrance. The muffler transmission loss[2] can be obtained by the above formulas derivation.

\[ TL \ (dB) = 10g\left(\frac{W_{in}}{W_{out}}\right) = 10g\left(\frac{(p_m + \rho c)^2}{4p_3^2} \frac{A_{in}}{A_{out}}\right) \]  
\[ (5) \]

2.3. Principles of molding

It is supposed that the acoustical field inside the muffler is uncoupled, and the finite element modeling only for the envelope region through which the gas passes is established. The fineness of the finite element mesh for acoustic must be adequate to clearly distinguish the highest principle frequency, that means there are at least three quadratic units or six linear units in each acoustic wave wavelength range.

2.4. Boundary hypothesis

(1) Boundary at the entrance

Based on the assumption that the incident wave is a plane wave, a unit velocity is added at the entrance.

(2) Boundary of the inner wall
It is assumed that the inner wall surface is rigid, wall surface absorption is regardless, the normal velocity of the medium on the wall is zero[3].

3) Boundary at the exit
The full sound absorption property is performed at exit end boundary perform.

2.5. Equivalent structure to the perforation tube
There are a large number of holes on the perforation tube, and the diameters of which are small. In order to simplify the simulation calculation process of the large number of holes, a set of continuous equivalent acoustic impedances are employed to simulate the holes, that is, a set of boundaries of impedances is added on the equivalent wall inside and outside the perforated tube instead of the perforations on the tube. A lot of complicated calculation process can be saved[4].

The impedance of perforated plate $Z_p$ can be expressed as:

$$ Z_p = \frac{\Delta P}{V} = R_p + jX_p $$ (6)

Here, $\Delta P$ is the acoustic pressure difference between the two sides of the perforated tube; $V$ is the average particle vibration velocity in the perforated tube, $R_p$ and $X_p$ are the real and imaginary parts of the impedance ratio simulating the perforation tube.

Here, when the thickness of the perforated plate is much less than four times the radius of the holes, there is $\Delta l = -1$. The real part and the imaginary part can be expressed by the following formula:

$$ R_p = \frac{1}{\varepsilon} \sqrt{8 \cdot \omega \cdot \eta \cdot \rho_0 (1 + \frac{l}{2 \cdot a})} $$ (7)

$$ X_p = \frac{1}{\varepsilon} \cdot \omega \cdot \rho_0 (l + 2\Delta l) $$ (8)

In the equation: $\omega$ is the angular frequency; $\varepsilon$ is the perforation rate; $\rho_0$ is the gas density in the perforated tube; $\eta$ is the viscosity coefficient; $l$ is the wall thickness of the perforated plate; $\Delta l$ is the correction coefficient of the small hole distribution; $a$ is the perforation radius, and the small holes are arranged in a square on the surface of the perforated tube, $0 < \frac{a}{d} < 0.25$, Therefore, the holes are arranged in a regular quadrangle on the surface of the perforating tube, $\Delta l = 0.85a(1 - 2.34 \frac{a}{d})$,

$d$ is the spacing between the holes on the perforated tube.

3. Numerical Analysis
There are two chambers in a square muffler on heavy-duty vehicle. The transmission loss analysis of its noise reduction performance is demonstrated in Fig. 1. It shows that the original model has a poor noise reduction performance in the range of 850 Hz and (1300-2000) Hz. The effects of the parameters of small holes on perforated plate, effects of the number difference of expansion chambers and position difference of main airflow passage upon the noise reduction performance are the main issues discussed in this paper. Then, based on the original frequency spectrum with muffling, the internal structure design is improved, and the poor noise reduction performance of some frequency band is optimized. The structure diagram of muffler and finite element mesh model are shown in Figure 2. In figure2 (a), The internal density of the muffler is set to be $\rho = 1.225 \text{kg} / \text{m}^3$, the medium to be $R600a$, and the speed of sound to be $c = 340 \text{m} / \text{s}$. 
Figure 1. Transmission loss curve of muffler

Figure 2. (a) Structure of muffler, 1-cylinder, 2-head I, 3-head II, 4-partition I, 5-partition II, 6-partition III, 7-partition IV, 8- Inlet pipe, 9-outlet pipe, 10-coaming plate, 11-air pipe, 12-outlet elbow, 13-inlet connector, 14-outlet connector, 15- bracket.; (b) Finite element mesh model of muffler.

3.1. Equivalent structure of the perforation tube hole
In Fig. 3, a comparison curve of the muffler performance is represented when the perforation rate is constant, and the hole sizes of the perforated tube are different. In the original model, the muffler perforated tube has a hole diameter of 3 mm, a hole spacing of 20 mm, and a plate thickness of 1 mm. On the basis of the original scheme, the effect on the transmission loss caused by the change of the hole diameter is analyzed provided that the perforation rate is ensured to be constant.
From Fig. 3, when the perforation rate is constant, the peak of the noise reduction curve shifts to the right, that is the direction of high frequency, as the diameter of the perforation increases. In the middle and high frequency bands, the peak value appears again at 1700 Hz by increasing the diameter of the perforation. Generally, in the interval of 0 Hz to 2200 Hz, a fine performance of noise reduction can be obtained by changing the diameter of the perforated tube without changing the perforation rate; below 700 Hz, changing the diameter of the perforation has little effect upon the transmission loss of the muffler[5]. The size change has the most significant effect on the noise reduction performance in the medium frequency band; when the diameter of the perforation on the perforated plate is 1.5 mm, the peak value is reached when at 1275 Hz, and the noise reduction amount of 67 dB can be achieved.

3.2. The effect upon transmission loss caused by the number difference of expansion chambers
As shown in Fig. 4(a), the initial structure model of muffler has only two expansion chambers, and a partition plate is added on the right side for modification. The simulation of the transmission loss of the two-expansion-chamber and the three-expansion-chamber muffler is compared in Fig. 4(b). As the Figure shown, the noise reduction performance is improved significantly in the middle and high frequency range (700~1300) Hz. After forming one more chamber, the volume of each cavity is reduced. Because of the shorter wavelength in the middle and high frequency bands, interference and reflection are more likely to occur on the discontinuous interface, which makes the noise reduction performance better. In addition, there are some passing frequencies of the initial muffler model, and they can cause some frequency noise can’t be reduced. The problem can be solved by the three-expansion-chamber structure, thus better noise reduction can be achieved.
Figure 4. (a) Comparison of the model before and after forming more chambers; (b) The transmission loss before and after forming more chambers.

3.3. Effect upon transmission loss caused by the position difference of main air passage

The position of the inner cannula, that is, the position of the main airflow passage on the partition is as shown in Fig. 5(a). It is indicated the effect upon the transmission loss caused by the position difference of the main airflow passage in Figure 5(b). Between 0 Hz and 2000 Hz, it can be seen that there is an overall decrease of the maximum noise reduction when the Lc value is reduced from 0.4L to 0.23L; while in the 900Hz to 1600Hz range, the noise reduction curve also varies remarkable accompany with the position change of the main airflow passage. When the Lc value increases, that is, the position of the main airflow passage moves upward, the distance between the main airflow passage and the inlet pipe is shortened. When the distance is close enough and the intake cavity can be regarded as a resonant cavity, and a good noise reduction performance for the resonant cavity can be reached when approximate to resonant frequency; when the Lc value is small, that is, the distance
between the intake pipe and the main airflow passage is far, the airflow path of the intake air chamber is increased, and the noise reduction effect in the low frequency band is better, but the number of passing frequencies is increased. As can be seen from Fig. 5(b), taking $L_c = 0.4L$, a good noise reduction performance can be obtained in the low frequency band as well as in the medium and high frequency bands.

![Diagram](image1)

(a)

![Graph](image2)

(b)

Figure 5. (a) The position of main airflow passage on the baffle; (b) Effect upon the transmission loss caused by position difference of the main flow passage.

Based on the simulation of the noise reduction performance of the muffler and qualitatively analysis of the structural parameters of the perforated tube[6], the optimal model is obtained as follows: the muffler has a small hole diameter of 3 mm, a small hole spacing of 20 mm, a plate thickness of 1
mm, and a three-chamber expansion cavity, \( L_c = 0.4L \). Figure 6 is a comparison of the curves of noise reduction performance before and after the improvement of the muffler. It can be seen from the figure that the noise reduction performance of the muffler is significantly increased after the structure optimization. In the middle and high frequency bands, there is no significant improvement on the performance of the optimized muffler. The sound absorption material such as slag wool can be added to the inner wall of the muffler for further improvement; from the whole frequency, the noise reduction performance of the muffler after structural optimization is improved compared with the initial muffler.

![Figure 6. Transmission loss before and after muffler optimization](image)

The optimized muffler is assembled into a heavy-duty vehicle for noise measurement of the whole machine. The measurement results are shown in Fig. 7[7]. It can be seen the results of the finite element analysis agree well with the experimental measurements in most of the frequency bands. The noise reduction performance of the modified muffler has indeed been improved in the experiment. It not only proves the utility of finite element simulation in the noise reduction analysis of the muffler model, but also gives a new solution for engineering design optimization.

![Figure 7. The muffler performance of the whole machine before and after the modification](image)
4. Conclusion
The measurement results of optimized muffler in the heavy-duty vehicle are in good agreement with
the finite element simulation results. The effects upon the noise reduction performance by changing
the parameters of perforation, the quantity of expansion chamber and the position of main airflow
passage are studied in this paper. The results reveal that: a good performance of sound reduction can
be obtained by changing the diameter of the perforated tube without altering the perforation rate; the
diameter change of the perforation in medium and high frequency has greatest effect on the noise
reduction performance; the peak value is reached when the diameter of holes are 3mm. After the
partition is added to the right chamber, the noise reduction performance of the middle and low
frequency muffler is obviously improved. The position of the main airflow passage has an evident
influence on the noise reduction performance; the noise frequency is necessary to be taken into
account. Sound-absorbing materials such as slag wool can also be placed on the inner wall of the
muffler to further improve the noise reduction performance. Finally, the improved structure of the
muffler is obtained in this paper. The experiment proved that the noise reduction performance of the
muffler after comprehensive optimization is improved.

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