Simulation of the passive part of an active car muffler

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Abstract. Active noise reduction is a new method for solving acoustic noise problems. Acoustic and thermal measurements of the passive system were performed to determine the design requirements for the future active system for the internal engine. Further, the study of this system by the finite element method was carried out. The article presents the results of measurements and analysis of the system during modelling.

1. Introduction

One of the most important global problems of modern civilization is noise. It has a negative impact on people's performance and health. The main sources of noise are power plants, namely internal combustion engines. The creation of machines with a reduced level of vibration and noise is an important scientific and technical problem.

Methods of dealing with noise has been divided into two types: active and passive. Passive methods are represented by reactive silencers made in the form of partitions and pipes, and resistive silencers, the channel of which is lined with sound-absorbing material. These types of silencers are valued for their high attenuation over a wide frequency range. However, they have a number of disadvantages, such as a large volume of construction and a low level of efficiency at low frequencies, as well as the creation of back pressure by the reactive part of the muffler, which often makes a passive approach to noise reduction impractical.

The second method in the fight against acoustic noise is active suppression systems. An active silencer contains an electroacoustic device that dampens unwanted noise, creating an audio signal with the same amplitude, but inverted in phase. Using the principles of active noise reduction, light and compact exhaust systems can be manufactured that effectively reduce the weight of the car. The main advantage of the active system is to reduce the hydraulic resistance of the exhaust system.

During the development of the active system prototype, acoustic and thermal tests of the passive exhaust system were performed to determine the requirements for the future design. Based on the test results, a simulation of a passive Bank was performed. This article discusses the simulation of the Gazelle Next silencer by the finite element method.

2. Sound tests

To determine the requirements for the future active noise reduction system, experimental measurements of the main operating parameters in the exhaust path of the internal combustion engine of the Gazelle NEXT car were carried out.
A microphone was used to measure the noise level in dB at various engine speeds. Previously, the car was warmed up to operating temperature. The measurements were performed according to the acoustic measurement standards.

Measurements were made in the idle mode of the engine (800 rpm), and then in steps of 1000 rpm. All measurements were performed without load. The measurement results are shown in figure 1.

![Figure 1. The results of the acoustic measurements at different values of engine speed](image)

3. Thermal test

The temperature of the muffler was measured with a thermal imager. The measurement data were used to determine the boundary conditions for further modeling. The measurements were taken at various locations in the exhaust system. The results are shown in Figure 2.

![Figure 2. Thermal measurements of passive systems](image)

4. Simulation of a passive silencer

A typical automobile exhaust system is a hybrid design consisting of a combination of reflective and diffusing silencers. The reflective parts are usually tuned to remove the dominant low-frequency harmonics of the motor, while the scattering parts are designed to eliminate high-frequency noise.
The silencer analyzed here is an example of a complex hybrid silencer in which the scattering element is created entirely by flow through perforated pipes and plates.

When developing the silencer model without fibrous materials, the following aspects were taken into account:

- **Geometry.** Very similar to commercially available car silencers, namely the Gazelle NEXT car silencer (figure 3);

![Figure 3. Gazelle Next Silencer](image)

- **Average flow distribution** - the Mach number in the exhaust system is usually less than 0.3. This means that in silencers with flow expansion, the average Mach number is quite small (less than 0.1). In such cases, the effects of convective flow can be ignored, and the only important effect of the average flow is its effect on the resistance of perforated pipes/plates. This model applies to the case when there is no medium flow in the silencer;

- **Temperature distribution** - when the engine is running, the air temperature inside the muffler is usually in the range of 300-400° C. The silencer also has a temperature gradient. However, the acoustic effect of this gradient is small, and the average temperature is usually used to calculate the speed of sound. In this case, the experiments were performed at room temperature (20° C). Therefore, the model assumes a constant temperature in the silencer and uses the default values for air density and sound speed at 1 ATM and 20° C.

In General, a silencer can be represented as a kind of system for converting sound vibrations coming to its input. A grid is defined as the process of discretizing an infinite geometric domain into a finite number of elements and nodes. In this study, the use of tetrahedral mesh elements. The maximum element length is calculated based on the wave number and wavelength. For finite element analysis, 10 elements per wavelength were used for the maximum scan frequency, which is 1 kHz.

Calculating the wavelength:

\[ \lambda = \frac{c}{f} \]  

where \( \lambda \) - sound wavelength; \( c \) - Speed of sound (343 m / s), \( f \) - Maximum frequency (1000 Hz)

\( \lambda = 0,343 \) m.
The maximum element length is 0.0343 m. This corresponds to one tenth of the shortest wavelength. The final grid of the calculation model is shown in figure 4.

Figure 4. Mesh model of a car muffler

The Helmholtz equation is used for problems in the frequency domain, and the classical scalar wave equation is used for research in the time domain.

The model equation is a modified Helmholtz equation for acoustic pressure $p$:

$$\nabla \cdot \left( - \frac{\nabla p}{\rho} - \frac{\omega^2}{c^2 \rho} \right) = 0$$

(2)

where $p$ - density; $c$ - Speed of sound; $\omega$ - Angular frequency.

Density should be included in the equation in cases where there are differences in density in different materials. The model assumes that reactive damping prevails in the low-frequency range.

There are three different types of boundary conditions. Rigid (wall-mounted) boundary conditions are used on all solid boundaries, which include the external walls of the silencer, the partitions between the resonator chambers, and the walls of pipes:

$$\nabla \cdot \left( - \frac{\nabla p}{\rho} \right) \cdot n = 0$$

(3)

A combination of incoming and outgoing plane waves is assumed at the input boundary:

$$\nabla \cdot \left( - \frac{\nabla p}{\rho} \right) \cdot n = \frac{i \omega}{\rho c^2} p - \frac{2i \omega}{\rho c} p_0$$

(4)

In this equation, $p_0$ denotes the applied external pressure, and $i$ is an imaginary unit. An outgoing plane wave is defined at the output boundary:

$$\nabla \cdot \left( - \frac{\nabla p}{\rho} \right) \cdot n = \frac{i \omega}{\rho c} p$$

(5)

The following equation defines the transmission loss in the silencer:

$$TL = 10 \log \left( \frac{P_{in}}{P_{out}} \right)$$

(6)

Here, $P_{in}$ and $P_{out}$ denote the acoustic effect at the input and output, respectively. The acoustic effect is calculated using the following equations:

$$P_{in} = \int_{\partial\Omega} \frac{P^2}{2\rho c} dA, \quad P_{out} = \int_{\partial\Omega} \frac{|P_{out}|^2}{2\rho c} dA$$

(7)
5. Simulation result
In this experiment, a CAD model of a car muffler was developed and imported into a computational program for numerical simulation. The problem was solved in the frequency domain using the pressure Acoustics, Frequency Domain interface. The model geometry consists of four separate resonator chambers separated by thin walls (figure 5).

The inlet and outlet correspond to the connection in the direction of the engine and free air, respectively.

![Resonator chambers](image1)

Figure 5. The geometry of the computational model

Figure 6 shows the result of studying the amount of transmission loss when changing the frequency. This graph shows that damping is better at higher frequencies, with the exception of a few deep dips across the entire frequency range. The simulation results will be used in the design of the active exhaust system of the Gazelle Next car.

![Transmission loss](image2)

Figure 6. Damping (dB) in the silencer as a function of frequency (Hz)
6. Conclusion
Acoustic and thermal measurements of the Gazelle NEXT car were made. The measurement results were used to determine the technical characteristics of the active type system. The measurement results were also used for modelling the exhaust system. The simulation was performed using the finite element method, using a slightly modified Helmholtz equation for the calculation. A graph of transmission losses of a passive silencer in the frequency range from 100 to 1000 Hz was obtained. Measurements were made for the resistive part of the silencer.

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7. References
[1] Mantserov S A, Kocherov A V and Okunev A V 2019 Development of a model of active noise reduction in internal combustion engines IEEE Xplore
[2] Nelson P A, Elliott S J 1992 Active Control of Sound Academic Press. San Diego. CA p 432
[3] Kuznetsov A N and Polivaev O I 2010 Prospects for the use of active noise cancellation systems VESTNIK 1 p 46
[4] Wu J D and Bai M 2000 Digital signal processor implementation of active noise control systems for broadband noise cancellation in engine exhaust systems Jpn. J. Appl. Phys 39 pp 4982-4986
[5] Kuo S M and Morgan D R 1999 Active noise control: A tutorial review Proc. IEEE 87 - 6 pp 943–973
[6] Simranjit S 2013 Implementation of Active Noise Cancellation in a Duct p 82
[7] Shaw J 2002 Design and Control of Active Muffler In Engine Exhaust Systems Taipei p 6
[8] Boonen P S 1998 Development of an active exhaust silencer for internal combustion engines Heverlee p 8
[9] Gonzalez A et al. 2006 Multichannel active noise equalization of interior noise IEEE Trans. Audio, Speech, Lang. Process. 14 – 1 pp 110–122