Features of the use of quartz sand as a vibration damping spacer for internal combustion engine muffler housing

Igor Deryabin¹, Larisa Gorina² and Aleksandr Krasnov³
Togliatti State University, Belorusskaya St., 14, Togliatti, 445667, Russian Federation

E-mail: iglen19@yandex.ru¹, gorina@tltsu.ru², kaw@yandex.ru³

Abstract. Evaluation of acoustic characteristics of an internal combustion engine (ICE) takes into account its capability to reduce gas dynamic noise from exhausted gases as well as the level of structural (housing) noise emitted by the dynamically excited housing. The muffler housing is dynamically excited in a mechanical way from an ICE vibrating on the suspension as well as by a pulsating flow of exhaust gases transmitting vibration energy via hard support ties of rubber metallic suspension supports on the vehicle frame or body and emitting sound to the surrounding space. One of the methods to reduce the level of muffler housing noise is making the housing double-layered with internal and external walls made of metal, or making it triple-layered containing a vibration damping spacer between the internal and external walls. An efficient solution is the use of a loose vibration damping substances represented by quartz sand as a vibration damping spacer. The article considers the study results of structural sound levels emitted by triple-layered walls of an ICE muffler depending on the bulk density of quartz sand used as a vibration damping spacer between muffler housing walls.

1. Introduction
A high noise level of the ICE exhaust gas system has a negative effect on external noise of a vehicle raising acoustic pollution of the environment. Currently, there are legislative solutions undertaken to restrict external noise levels emitted by vehicles. Usually, designs of ICE mufflers represent a frequency-tuned chamber with an inlet and outlet pipe. To increase efficiency of sound-muffling characteristics, the muffler chamber may contain a set of various partitions as well as sound-isolating materials [1-14]. The muffler chamber housing is generally cylindrical (with a circular or elliptical guide) formed from metallic sheet. Evaluation of acoustic characteristics of a muffler must take into account its capability to reduce gas dynamic noise from exhausted gases as well as the level of structural (housing) noise emitted by the dynamically excited housing. The muffler housing is dynamically excited in a mechanical way from an ICE vibrating on the suspension as well as by a pulsating flow of exhaust gases transmitting vibration energy via hard support ties of rubber metallic suspension supports on the vehicle frame or body and emitting sound to the external environment. In its turn, vibration-excited elements of the bearing frame and body panel of the vehicle also emit respective structural noise into the cockpit space increasing noise loading formed by other sources and ways of transmitting vibro-acoustic energy (hard and air paths), which additionally impairs acoustic comfort in the vehicle cabin. One of the methods to reduce the level of muffler housing noise is making the housing double-layered with internal and external walls made of metal, or making it triple-layered containing a vibration damping spacer between the internal and external walls [1-4, 6, 11, 15]. An efficient solution to improve acoustic characteristics
of the muffler housing design is the use of a loose vibration damping substances represented by quartz sand as a vibration damping spacer [3, 15]. Quartz sand has high dynamic elasticity modulus \( E_d \) \( (E_d \approx 75 \text{ MPa}) \) and increased damping properties caused by losses of energy dissipated in case of dynamic strain. Moreover, quartz sand is characterized by high sound-insulating capability. Vibration energy transmitted from the ICE via an inlet pipe to the muffler housing and from gas flow dynamic pulsations in the muffler chamber will be efficiently dissipated in the quartz sand layer reducing the level of housing noise emitted by the muffler. As we know [2], the drop in the level of structural (housing vibration) noise emitted by a regular metallic plate and a plate containing means of vibration damping can be evaluated using the following formula:

\[
\Delta L = 10 \log \left( \frac{\eta_2}{\eta_1} \right),
\]

(1)

where \( \eta_1 \) is the loss coefficient of a plate without vibration damping, \\
\( \eta_2 \) is the loss coefficient of a plate with vibration damping, \\
\( j_1 \) is the emission coefficient of a plate without vibration damping, \\
\( j_2 \) is the emission coefficient of a plate with vibration damping.

In the medium and high frequency range, the following empirical formula can be used to evaluate the drop of sound pressure level when using plate vibration damping:

\[
\Delta L = (6 \ldots 10) \log \frac{\eta_2}{\eta_1},
\]

(2)

To improve design and technological properties of ICE mufflers in terms of vibration acoustics, design modeling methods are widely used including advanced hardware and software calculation systems [5-10, 12, 16], experimental study methods in the conditions of road tests within vehicles and in stand conditions in special acoustic test chambers [1, 12, 17]. The study [17] gives the results of experimental studies of dependency between the sound level emitted by the ICE exhaust system and temperature and pressure of exhaust gases. The work [18] studies the effects of a loose vibration damping substances (quartz sand) on the vibration level of a road bed. However, currently the effects of bulk density of sand used as a vibration damping spacer for a triple-layered ICE muffler on the level of structural sound emitted by the muffler housing wall are not yet studied.

A metallic housing of the muffler is a mechanical medium where longitudinal and transverse sound waves propagate. As we know [2], the propagation velocity of longitudinal waves is found by the formula:

\[
c_{long} = \sqrt{\frac{G(2-2v)}{\rho(1-2v)}},
\]

(3)

where \( G \) is the shear modulus, Pa, \\
\( v \) is the Poisson coefficient, \\
\( \rho \) is the mechanical medium density, kg/m³.

The propagation velocity of transverse waves is found by the formula:

\[
c_{trans} = \sqrt{\frac{\mu}{\rho}},
\]

(4)

In a simplified form, the vibration velocity amplitude of a simply oscillatory system represented by a plate can be shown as follows

\[
V = \frac{F}{Z_k},
\]

(5)

where \( V \) is the vibration velocity amplitude, m/s, \\
\( F \) is the excited force amplitude, N, \\
\( Z_k \) is the plate inlet impedance.

The plate inlet impedance is found using the formula:
\[ Z_k = 8\sqrt{Bm} \]  

where \( B \) is the bending stiffness, Nm\(^{-1}\), \( m \) is the weight, kg.

Therefore, according to the above formulas, the density of a hard mechanical structure must be increased in order to decrease the propagation velocity of sound waves in the mechanical system and decrease sound vibration.

Loose density of dry sand depends on the fineness modulus and degree of compaction (material porosity) and can vary within 1200-2000 kg/m\(^3\). The goal of this study is to define dependency between the level of structural sound emitted by the ICE muffler housing and bulk density of sand used as a vibration damping spacer of the housing sandwich panel.

2. Methodology

For the study, six ICE exhaust gas mufflers were manufactured having identical geometric dimensions and a housing of a triple-layered design consisting of external and internal steel walls of elliptical shape 0.8 mm thick, with an air gap of 7 mm thick between them. The void between muffler housing walls was filled with a loose vibration damping substance as quartz sand of various loose density. For comparative analysis, a single layer housing made of a steel plate 0.8 mm thick was also studied. The design of the studied mufflers is given in figure 1.

![Figure 1. Design of the studied mufflers. 1 – muffler housing external wall, 2 – muffler housing internal wall, 3 – quartz sand, 4 – front end wall, 5 – rear end wall, 6 – inlet pipe, 7 – outlet pipe.](image)

Open bevels of the inlet and outlet pipes are located co-axially to minimize hydraulic resistance of the discharge duct at a specific distance from end walls of the chamber equal to half length of the muffler air void minus 0.3 diameter of the outlet pipe and at the distance of one fourth of the air void length minus 0.3 diameter of the inlet pipe. This geometric arrangement of muffler pipes prevents excitation of odd lowest eigen modes and transmission of even lowest eigen modes of the air void to the muffler’s cylindrical chamber.

Housings of the studied mufflers (nos. 1-6) comprised a vibration damping spacer as quartz sand with the following bulk density:

- No. 1 – 1200 kg/m\(^3\);
- No. 2 – 1400 kg/m\(^3\);
- No. 3 – 1500 kg/m\(^3\);
- No. 4 – 1600 kg/m\(^3\);
- No. 5 – 1700 kg/m\(^3\);
- No. 6 – 1900 kg/m\(^3\).

Muffler No. 7 had a single-layered steel housing 0.8 mm thick.

Structural sound emitted by muffler housing walls was evaluated in test stand conditions in an acoustic no-echo chamber with a motor stand installed inside (see photo in figure 2).
Figure 2. Photo of the studied ICE muffler mounted in the acoustic no-echo chamber.

No-echo (fully muffled) test chamber is made as an independent premise installed on an individual foundation vibration-insulated from the primary building. To approximate acoustic properties of the chamber to the free sound field of the wall, the chamber ceiling and floor are lined with volumetric sound absorbers of wedge shape (rockers). The motor stand braking (driving) machine is installed in the bottom under the floor surface of the acoustic no-echo chamber. Braking torque is transferred using a flat-belt transmission. When using this concept of a test stand, the acoustic chamber floor is fully vibration insulated from the independent foundation where the driving (braking) engine is installed. Background values of the sound emitted by driving units and motor stand systems within the speed range of the braking (driving) machine of 1500-5000 min\(^{-1}\) were 46-58 dBA, which proves an opportunity of seamless qualitative objective vibration acoustic studies of the ICE in general and its individual systems, units and assemblies, namely, exhaust gas mufflers. The motor stand is equipped with an efficient low-noise technological system of exhaust gas discharge that was coupled in an air-tight manner to an outlet pipe of the studied muffler. The technological system of exhaust gas discharge comprised a muffler in the form of a stand muffler mounted under the test room floor and having an optimized design in terms of acoustic and hydraulic characteristics. This muffler ensures efficient sound muffling of an exhaust flow from the tested ICE keeping the low hydraulic resistance of communicated pipeline branches of the exhaust suction and high performance of exhaust suction of the fan unit. This prevented intensive emission of gas dynamic sound from the open bevel of the muffler’s outlet pipe, which, therefore, reduced the level of background noise having a negative effect on the quality of studying the structural sound emitted by muffler housing walls. To reduce the background noise of the ECI housing, the outlet system and the neutralizer, the ICE surface was covered with sound-absorbing mats made of non-combustible basalt fiber (see figure 3).

A measuring microphone was installed 15 cm from the geometric center of the muffler housing (see figure 4). Sound levels corrected under scale A were recorded at full loading (fully open throttle) for a four-cylinder eight-valve ICE 1.6L within the range of 1500-5000 min\(^{-1}\).

When studying each design option of the muffler housing, sound level was measured at least 5 times followed by averaging of obtained values. All measurements were taken in identical atmospheric conditions. Air temperature in the test chamber was +24°C, atmospheric pressure was 100-101 kPa, relative humidity was 40-50%.
3. Results and discussion

Graphical dependencies of recorded sound levels $P$ (dBA) from ICE revolutions $n$ (min$^{-1}$) are given in figure 5.

The graphical dependencies given in figure 5 show that the muffler housing design as a triple-layered structure containing quartz sand as a vibration damping spacer is an efficient technical method to reduce the level of structural sound emitted by muffler housing walls. A reduction in the sound level to 5 dBA was registered in the measurement point within the entire ICE speed range. A dependency between the level of structural sound emitted by the muffler housing and sand bulk density was noted. The highest level of sound was registered in the housing design containing sand with a bulk density of 1200 kg/m$^3$. 

Figure 3. Photo of ICE screening.

Figure 4. Photo of the measuring microphone location.

Figure 5. Sound levels in the measurement point.
(No. 1). When the bulk density is increased to 1600 kg/m$^3$, the sound level falls down by 1.3-2.5 dBA. Further increase of sand density does not affect the level of muffler’s housing noise. The below tables give sound level values in the measurement point of the studied muffler designs at ICE revolutions of 2600 and 3700 min$^{-1}$.

The minimal noise level was registered when using sand with a bulk density of 1600-1900 kg/m$^3$ in ICE operating conditions represented in tables 1 and 2 at the speed of 2600 and 3700 min$^{-1}$. When the bulk density is decreased to 1200 kg/m$^3$, the sound level rises by 1.5-2.5 dBA.

**Table 1.** Sound level in the measurement point at 2600 min$^{-1}$.

| Muffler no. | Sand bulk density [kg/m$^3$] | Sound level [dBA] |
|-------------|-------------------------------|-------------------|
| 1           | 1200                          | 82                |
| 2           | 1400                          | 81.5              |
| 3           | 1500                          | 81                |
| 4           | 1600                          | 80.5              |
| 5           | 1700                          | 80.4              |
| 6           | 1900                          | 80.3              |
| 7           | single layer, no sand         | 85.5              |

**Table 2.** Sound level in the measurement point at 3700 min$^{-1}$.

| Muffler no. | Sand bulk density [kg/m$^3$] | Sound level [dBA] |
|-------------|-------------------------------|-------------------|
| 1           | 1200                          | 83                |
| 2           | 1400                          | 82                |
| 3           | 1500                          | 81.5              |
| 4           | 1600                          | 81.1              |
| 5           | 1700                          | 81.1              |
| 6           | 1900                          | 81                |
| 7           | single layer, no sand         | 85.5              |

4. Conclusion

A conclusion can be made about a dependency between the sand bulk density and the structural sound level emitted by the triple-layered ICE muffler housing walls containing quartz sand as a loose vibration damping substances. To increase conversion of vibration energy into heat energy and, therefore, to reduce the emission of structural sound, it is reasonable to use fine and medium-grained sand fractions with a bulk density of at least 1500 kg/m$^3$. Further increase of sand density above 1500 kg/m$^3$ has no significant effect of sound levels emitted by ICE muffler housing walls. The study results can be useful for developers of ICE mufflers and in developing devices to reduce sound levels of engineering, process and power equipment.

References

[1] Fesina M I, Krasnov A V, Gorina L N and Pankov L A 2010 *Automobile Acoustic Materials Designing Low-Noise Designs of Automobile Vehicles*, A Monograph 2 (TSU, Togliatti)

[2] Heckl M and Muller H A 1980 *Guide in Technical Acoustics* Translated from German by B.D. Vinogradov and N.M. Koloyartsev (Shipbuilding, Leningrad)

[3] Kochetkov O S and Kochetkova M O 2007 Patent RU2306430

[4] Malkin I V and Grebnev I V 2010 Patent RU102682

[5] Li M et al. 2017 Acoustic Performance Optimization of the Exhaust Muffler for a Car Based on Response Surface Method, Shanghai Jiaotong Daxue Xuebao *Journal of Shanghai Jiaotong*
Ranjbar M, Boldrin L, Scarpa F, Niels S and Patsias S 2016 Vibroacoustic optimization of anti-tetrachiral and auxetic hexagonal sandwich panels with gradient geometry Smart Materials and Structures 25(5) 054012

Ranjbar M and Alinaghi M 2016 Effect of Liner Layer Properties on Noise Transmission Loss in Absorptive Mufflers Mathematical Modelling and Applications 1(2) 46-54

Ranjbar M and Kemani M 2016 A Comparative Study on Design Optimization of Mufflers by Genetic Algorithm and Random Search Method Journal of Robotic and Mechatronic Systems 1(2) 7-12

Zhong S 2020 Research on Computational Data Simulation of Automobile Engine Exhaust Muffler Performance Journal of Physics: Conference Series 1533 042043

Kumar S 2015 Linear Acoustic Modelling And Testing of Exhaust Mufflers, KTH Royal Institute of Technology Master of Science Thesis, Stockholm (Sweden) 2-105

Shishko A I and Krishkevich A V 2006 Patent RU2721394

Wei-Hong T and Mohd Ripin Z 2013 Analysis of exhaust muffler with micro-perforated panel Journal of Vibroengineering 15(2) 558-73

Kovalnogov et al. 2016 Patent RU165516

Karpenko A G 2017 Patent RU2617527

Deryabin I V and Andreyanov S A 2020 Patent RU2721394

Mahadir M et al. 2017 Simulation of automotive exhaust muffler for tail pipe noise reduction Jurnal Teknologi (Sciences & Engineering) 79(7-4) 37-45

Golgotiu E et al. 2016 Experimental Investigation on Functional Parameters of the Engines Exhaust Mufflers Applied Mechanics and Materials 822 224-9

Guangya D et al. 2017 Effect of sand bags on vibration reduction in road subgrade Soil Dynamics and Earthquake Engineering 100 529-37