Vibroacoustic debugging technique for vehicle exhaust systems

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Abstract. Influence of the exhaust system on the vibroacoustic comfort of a passenger car with a transverse power unit is considered. The methodology is presented which, when using road tests and calculating the exhaust system on the finite element model, allows to reduce the levels of vibration and noise from the exhaust system when changing to more powerful engines in serial passenger cars.

1. Introduction
Currently, more and more attention is paid to the consumer properties of cars. Fierce competition in the global automotive market forces car manufacturers to look for ways to increase the comfort of new car models. At the same time, one of the most important indicators is vibroacoustic comfort or, in other words, reduction of noise and vibration. As it is known, the main sources of noise and vibration in a car are wheels, external aerodynamic processes, an internal combustion engine, transmission units and an exhaust system. For cars with a transverse arrangement of the internal combustion engine, a high impact on vibroacoustic comfort is exerted by the exhaust system, due to an increase in the oscillation amplitude of it. The external noise and internal noise of the car, as well as local and general vibration in the vehicle interior depend to a large degree on the exhaust system properties. When designing the exhaust system, it is necessary to take into account a number of conflicting requirements and factors (reducing the hydraulic resistance of the entire exhaust path, reducing external noise and vibration transmitted to the body, increasing the durability of the entire exhaust path, etc.).

2. Research methods
Often, car manufacturers to increase the consumer properties of a car as a whole and to expand the model range install more powerful power units on already designed base cars. Thus, the task of adapting the exhaust system for a new power unit arises. When fulfilling this task in terms of increasing the vibroacoustic comfort, it is most convenient to use modern engineering analysis tools based on the finite element method (FEM) [2, 5]. The calculation models developed on the basis of FEM make it possible to take into account the nonlinear properties of suspension shock absorbers, power unit supports, the presence of metal compensators in the exhaust system and the flexibility of its components, the flexibility of the body, etc.

A similar task to improve vibroacoustic performance and adaptation of the exhaust system to a new, more powerful power unit is solved on the example of one of the modern class B cars.
The initial design of the exhaust system of this vehicle is shown in Figure 1.

![Diagram of exhaust system](image)

**Figure 1.** The original design of the exhaust system of a passenger car.

The exhaust system design includes:
1) exhaust manifold;
2) bellows expansion joint (metal compensator of angular and linear vibrations);
3) additional silencer (resonator);
4) muffler main left;
5) muffler main right;
6) six attachment points of the muffler pipes to the car floor (FC_12, FC_21, FC_31, FC_32, FC_41 и FC_42);
7) exhaust pipe.

When installing a more powerful engine on a serial passenger car, the performance of vibroacoustic comfort deteriorated significantly. It should be noted that the serial exhaust system for this engine was finalized by adding the main right muffler and minor modifications to the exhaust tract elements.

To assess the level of vibration and noise when operating a car with a more powerful engine and the initial design of the exhaust system, field tests were conducted in two modes: 1) intensive acceleration in 3rd gear; 2) movement at a constant speed of 100 km/h and 120 km/h in 5th gear. The assessment was made according to the levels of internal noise and vibration transmitted from the exhaust system elements to the car body in accordance with the methodology [1].

Noise measurement using microphones was carried out at the following control points:
1) the zone in the region of the right ear of the driver;
2) the center of the seat of the rear left passenger;
3) the center seat of the rear right passenger;
4) engine compartment (in the area of the intake module);
5) the engine compartment (in the area of the cooling system radiator);
6) the exhaust pipe area of the exhaust system (left);
7) the exhaust pipe area of the exhaust system (right).

Measurement of vibration levels was evaluated using three-component accelerometers at the following control points:
1) brackets of all suspension mountings of the power unit (up to support beds);
2) brackets of suspension mountings of the power unit (after support beds);
3) brackets of suspension mountings of the exhaust system (after support beds at points FC:12, FC:21, FC:31, FC:32, FC:41 и FC:42 in Figure 1);
4) dashboard panel (on the right);
5) the driver’s seat slide (right, rear).

The processed test results are shown in Figures 2-6.

Figure 2. Internal noise (second harmonic) during intensive acceleration in third gear near the driver’s right ear (BT: D1 (B)), near the center of the rear left passenger seat (BT: (A2)) and near the center of the rear right passenger seat (BT: (A4))

Figure 3. The noise of the intake system and the exhaust system (second harmonic) during intensive acceleration in third gear in the area of the intake module (BT:BBA08), the cooling system radiator (BT:MO), the left exhaust pipe of the exhaust system (BT:BBE15_left) and the right exhaust pipe of the exhaust system (BT:BBE15_right)
Figure 4. Vibration at the location of the brackets of the exhaust system suspension support brackets (second harmonic) during intensive acceleration in third gear in the region of the additional silencer (FC_12 and FC_21), the left main muffler (FC_31 and FC_32) and the right main muffler (FC_41 and FC_42)

Figure 5. Noise level in the octave range when driving at 100 km/h in the area of the driver’s right ear (BT:D1(B)), the center of the rear left passenger seat (BT:(A2)) and the center of the rear right passenger seat (BT:(A4))
Having analyzed the test results, we can draw the following conclusions:

1) the noise from the exhaust system reaches the level of 109 dBA in the intensive acceleration mode in 3rd gear (the main contribution from the second motor harmonic);
2) there is a strong vibration in the region of the brackets of the additional silencer (points FC_12 and FC_21) in the range of 3500 ... 5500 min-1;
3) the maximum vibration felt on the slide of the driver’s seat occurs in the region of the bracket of the left main muffler (point FC_31) at frequencies of 160 Hz and 173 Hz — eigen frequencies of the exhaust system;
4) the total noise level for rear passengers in the intensive acceleration mode in 3rd gear (according to the results of 5 drives) is 86 dBA with an acceptable value of 77 dBA [1].

Consequently, it was found that the exhaust system resonates at the following frequencies: 80 Hz (2400 min-1), 125 Hz (3750 min-1), 160 Hz (4800 min-1), 200 Hz (6000 min-1). The most critical are resonances at 160 Hz and 200 Hz.

Hence, it is obvious that it is necessary to fine-tune the exhaust system in terms of vibroacoustics. To carry out this work, a finite element model (FEM) of the initial design of the exhaust system was created. Its design is shown in Figure 7. Validation of the FEM was carried out on the basis of the field test results.
FEM consists of the finite elements of the shell type SHELL. The elastic elements of the exhaust system are made using beam elements having 6 degrees of freedom. FEM contains 19071 nodes and 18779 elements.

The work on the FEM-based exhaust system debugging consists of two main stages: 1) modal analysis and amplitude-frequency characteristics analysis (AFC) of the original design and its validation; 2) selection of the optimum design option of the exhaust system for vibroacoustics based on permissible structural solutions and changes. Moreover, when selecting the best option, a modal analysis and a frequency response analysis are carried out.

As a perturbing effect in the AFC analysis, a unit velocity along three axes — X, Y, Z — is fed to the exhaust manifold flange. The range of frequencies under study is from 50 Hz to 350 Hz. The following response is investigated - a graph of the dependence of the output signal speed amplitude on the input signal frequency at the given points shown in Figure 1. In the work, the amplitude response is studied only along the Z axis, as the most dangerous in terms of transmitting vibrations to the car body. The goal in this calculation is to find the optimum design of the exhaust system to reduce noise in the frequency ranges of 80 ... 120 Hz and 180 ... 200 Hz.

The results of a modal analysis of the exhaust system’s initial design are shown in Figure 8. The calculation verification confirmed the test results: three natural frequencies and vibration modes of the exhaust system were revealed at frequencies of 163 Hz, 169 Hz and 200 Hz. The modes of vibration directly correlate with the test results - the highest amplitude of relative vibrations is in the area of the additional silencer and the left and right exhaust pipes of the main mufflers.

In the process of performing the calculations, the FEM parameters were tuned to obtain more accurate modeling results and to take into account a number of factors inherent in the exhaust system’s design.
Figure 8. The results of a modal analysis of the exhaust system’s initial design: a) natural form of oscillations at a frequency of 163 Hz; b) natural form of oscillations at a frequency of 169 Hz; c) natural form of oscillations at a frequency of 200 Hz;

To solve the identified problem areas, 48 designs of the exhaust system were formed, as combinations of several possible constructive solutions in the production environment (for example, the introduction of an additional metal compensator of angular vibrations, the transfer of silencer mounting brackets, etc.). The FEM were designed for each of the options. As a result of modal analysis and frequency response analysis, 4 most optimum options were selected:

- option 42 (triangular amplifiers on two pipe elbows after an additional silencer were added, sheet thickness - 4 mm);
- option 43 (based on option 42). A bellows has been added to the pipe branch immediately in front of the main muffler on the left);
- option 47 (based on option 42). A bellows has been added immediately after exiting the connecting pipe on the pipe branch in front of the main right muffler);
- option 48 (combination of options 43 and 48). Two bellows have been added structurally similar to the bellows used in the original version).

The results of the selected options analysis of the exhaust system are given below.

The maximum amplitudes in the ranges of the frequencies under consideration are observed in the initial version of the exhaust system (see Figure 9) and in version 42 with triangular amplifiers.

The introduction of an additional bellows on the left pipe (option 43) reduces the amplitudes at all points except for the point immediately after the additional bellows (FC_31).

The introduction of an additional bellows on the right pipe (option 48) reduces the amplitudes in the ranges of 80 ... 100 Hz and 180 ... 200 Hz, however, it increases the response in other frequency ranges. The graph for point FC_32 grew especially noticeably - after the main muffler on the left. In the range of 80 ... 100 Hz, its value is even higher than in the original version.

The use of two bellows (option 48) had a positive effect on all points. In the range of 80 ... 300 Hz, the responses at all points do not exceed 3 mm/s (Figure 9). Charts are smooth, without pronounced peaks. Of all the options formed, this design option of the exhaust system is the best.
Figure 9. The amplitude-frequency characteristics of the exhaust system: a) the initial version b) option 48 - the best option

The appearance of the optimum design of the exhaust system based on FEM (option 48) is shown in Figure 10. The natural forms and oscillation frequencies of this design option are shown in Figure 11. The values of the natural frequencies and their forms with respect to the original design of the exhaust system have changed. It was possible to unleash a ‘conditionally rigid’ oscillatory system of the original version and reduce the amplitude of the oscillations at the exhaust pipes of the left and right main mufflers. Thus, the introduction of two additional metal compensators and an increase in the local stiffness of the pipe between the additional silencer and the main mufflers leads to a shift of the resonant frequencies of the system to the outside of the operating range (in particular, this is caused by the division into two or more subsystems with high resonant frequencies). This, in turn, can significantly reduce the levels of vibration of the exhaust system, therefore, improve the performance of the vibroacoustic comfort of the car to the existing standards.
Figure 10. Appearance of the exhaust system’s design (option 48).

Figure 11. Natural modes and vibration frequencies of the exhaust system design (option 48): a) natural vibration mode at a frequency of 91 Hz; b) natural vibration mode at a frequency of 116 Hz; c) natural vibration mode at a frequency of 172 Hz; d) natural vibration mode at a frequency of 195 Hz;

3. Conclusion
The presented methodology for increasing the car’s vibroacoustic comfort by reducing the vibration and noise levels from the exhaust system is suitable for fine-tuning the acoustics of serial front-wheel drive cars with a transverse power unit.

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