

Structural Noise and Acoustic Characteristics Improvement of Transport Power Plants

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Abstract. Noise reduction generated during the operation of various machines and mechanisms is an urgent task with regard to the power plants and, in particular, to internal combustion engines. Sound emission from the surfaces vibration of body parts is one of the main noise manifestations of the running engine and it is called a structural noise. The vibration defining of the outer surfaces of complex body parts and the calculation of their acoustic characteristics are determined with numerical methods. At the same time, realization of finite and boundary elements methods combination turned out to be very effective. The finite element method is used in calculating the structural elements vibrations, and the boundary elements method is used in the structural noise calculation. The main conditions of the methodology and the results of the structural noise analysis applied to a number of automobile engines are shown.

1. Actuality of acoustic characteristics research of power plants with internal combustion engines

Power plants with internal combustion engines are used in various industries: in transport, in agriculture, in power engineering, in different manufactories. Modern power plants used in transport have a significant negative impact on the environment and human beings. In addition to toxic substances emitted from power plants and, above all, internal combustion engines, there is also an acoustic effect to the person (noise), which creates discomfort and damages the health of people close to the operating power plants with piston engines. Vehicle acoustic characteristics are strictly regulated by various standards. Their improvement is important for obtaining the necessary consumer properties of newly-created engines with various types and purposes.

Currently, the sources of noise generated by internal combustion engines are divided into aerodynamic ones (when gas fluctuations are the cause of noise) and structural ones. In the latter case, the surfaces oscillations of the structural elements are the cause of noise. Experimental studies [1] show that structural noise creates about half of the total noise of an automobile diesel engine. Thus, the ability to assess the level of structural noise created with consideration for the characteristics of a particular design and the evaluation of design parameters in order to reduce structural noise during engine operation is very relevant. Thanks to the computer technology development, there are now many opportunities to use mathematical modeling to improve the design of basic components. These



possibilities are aimed at improving the acoustic characteristics of power plants.

2. Studying methods of the power plants acoustic characteristics with internal combustion engines

The vibrations determination of complex structures outer surfaces during the analysis of their forced oscillations is performed using numerical methods [2]. In practice, two numerical methods are used to calculate the sound field of an object: the finite element method (FEM) and the boundary element method (BEM). Each of them has advantages and disadvantages and its own application area. Advantages of FEM are widely known. As far as the BEM is concerned, first of all, it is necessary to note the reduction in the unit size of the task being solved. And also we have the lower computation costs when obtaining the results of one-accuracy, in comparison with the FEM. At the same time, the smaller computational costs applied to structural noise calculations of complex structures are the decisive argument in favor of the BEM use. BEM advantages are also joined with the frequency range where in practice it is necessary to perform calculations in evaluating structural noise. For example, for piston engines this is 0.3 - 4 kHz. The maximum permissible size of the elements is $l_{elm} < \lambda/6$ proceeding from the spatial approximation condition of a harmonic wave with the length λ [3]. The size of the finite element grid (that is at the upper boundary of the indicated frequency range in the structural noise calculation) exceeds more than 1000 times the grid size of the boundary elements. When using FEM with reference to the problem of forced vibration of a mechanical system generating structural noise, the equations system to be solved has the following form:

$$[-\omega^2[M] - i\omega[C] + [K]]\{\tilde{u}\} = \{\tilde{F}\}, \quad (1)$$

where $[M]$, $[C]$, $[K]$ are the matrix of mass, damping and stiffness, respectively, $\{\tilde{u}\}$ is a vector of complex displacement amplitudes, $\{\tilde{F}\}$ is a vector of complex amplitudes of the disturbing forces harmonic components.

The finite element models used in practice have hundreds of thousands of freedom degrees, so the complete solution of the system (1) for a great number of harmonics seems to be very resource-intensive. In connection with this, the method of eigenmodes superposition of oscillations has found application in solving similar problems [4]. At the same time, time costs are significantly reduced when obtaining an acceptable accuracy of the solution (1). Complex amplitudes of forced oscillations are represented by an n-forms superposition of natural oscillations:

$$\{\tilde{u}\} = \sum_{j=1}^n \tilde{\lambda}_j \{u_j\}, \quad (2)$$

where $\tilde{\lambda}_j$ is a complex factor of participation in forced oscillations of the j -th form of natural oscillations $\{u_j\}$ at ω_j frequency.

Complex coefficients $\tilde{\lambda}_j$ are determined by the formula:

$$\tilde{\lambda}_j = \frac{\{\tilde{F}\}^T \{u_j\}}{\omega_j^2 + 2i\omega\omega_j\xi_j - \omega^2}, \quad (3)$$

where ξ_j is a coefficient of relative (modal) damping.

Modal damping coefficients are used, when applying the superposition method of proper forms, without specifying the damping law [5].

$$\xi_j = \frac{\Psi(\omega_j)}{4\pi}, \quad (4)$$

where $\Psi(\omega_j)$ is an energy loss factor during the oscillations.

Thus, in the case of a molded block-crank design of an automobile engine casing, factor $\Psi(f)=172/f$, where $f=2\pi\omega$ (that is forced oscillations frequency, Hz). Figure 1 shows the empirical dependence of oscillations energy losses Ψ and relative damping coefficient ζ on the frequency [6].

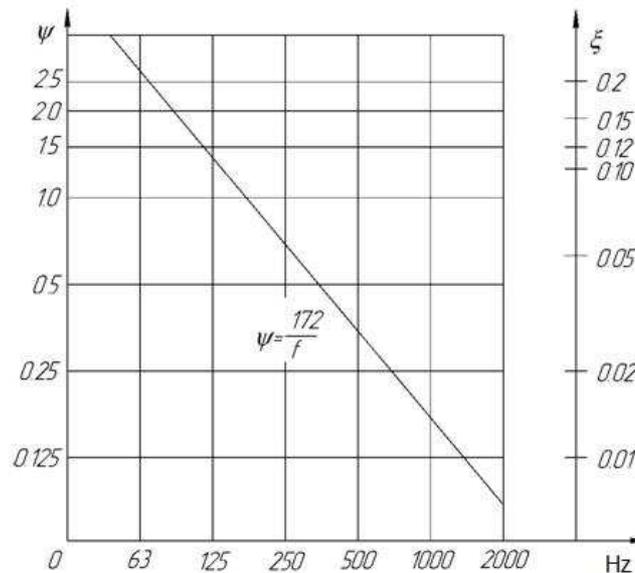


Figure 1. Dependences of the energy loss coefficient of the oscillations $\Psi(f)$ and the relative damping coefficient ζ on the frequency.

Frequencies ω_j and forms u_j of natural oscillations of the conservative system radiating structural noise, are determined by the FEM for solving the matrix system of free-oscillation equations.

$$[-\omega^2 [M] + [K]] \left\{ \tilde{u} \right\} = 0 \quad (5)$$

The problem of sound radiation by external surfaces of the structure (body parts) that perform oscillations, reduces to determining the field of sound pressure $\tilde{p}(x, y, z)$. The field is obtained in the solution of the Helmholtz equation:

$$\Delta \tilde{p}(x, y, z) + K^2 \tilde{p}(x, y, z) = 0, \quad (6)$$

where $K=\omega/c$, c is the sound velocity in the environment. Boundary conditions on the external surface of the design are set:

$$\frac{\partial \tilde{p}}{\partial n}(x, y, z) = -i\omega\rho \tilde{v}_n(x, y, z), \quad (7)$$

where $\tilde{v}_n(x, y, z)$ is the complex oscillation velocity of the outer surface in the direction of the n -normal, ρ is the environmental density.

The previously obtained vibration values of the structural surfaces are used when assigning boundary conditions (7). As noted above, the method of boundary elements is the most promising when solving the radiation problem of structural noise. Three- and four-cornered boundary elements are used. The value of the unknown function is assumed constant within the element. In the case under

consideration, a variant of the direct boundary element method (DBEM) [7] is realized. Under this condition the complex amplitudes of sound pressure $\tilde{p}(\xi_j)$ at control points ξ_j inside the boundary element are unknown.

3. An example of the acoustic characteristics research of power plants with an internal combustion engines

As an example, the problem of the structural noise determination of an 8-cylinder V-shaped 8CN 12/13-type automobile diesel engine is considered. Figure 2 and 3 presents the finite element and the boundary element models respectively. They contain 282 thousand of finite elements and 4.4 thousand of boundary elements. The models included the main body parts of the diesel engine, which had a significant outside surface area.

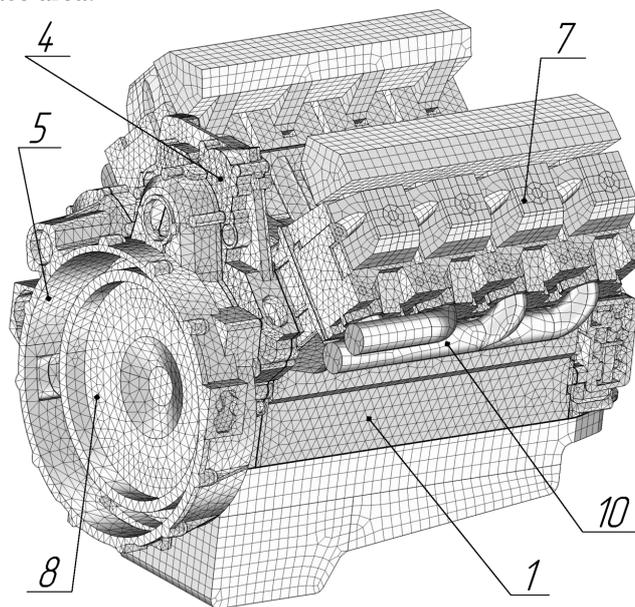


Figure 2. Finite element model of 8CN 12/13 diesel body parts.

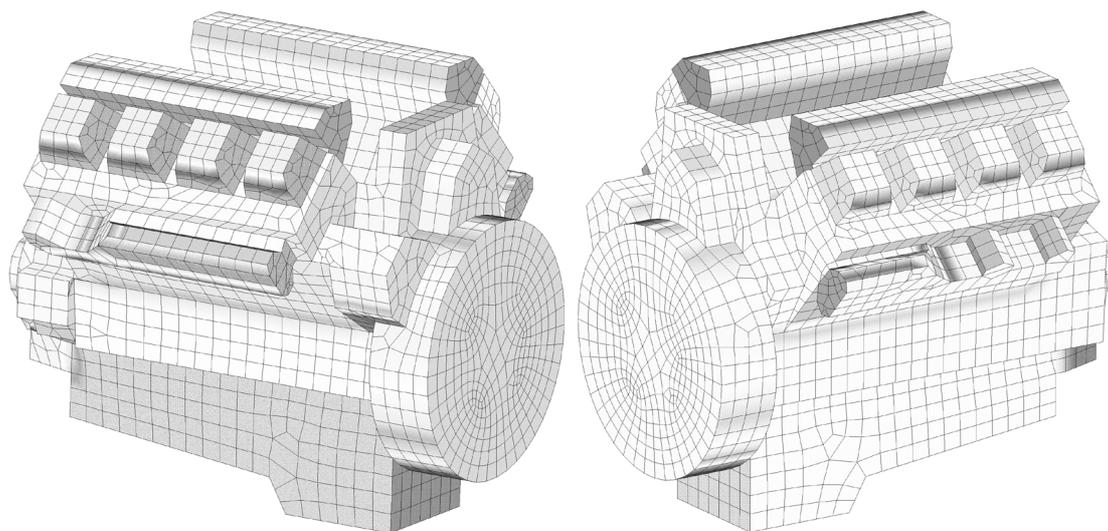


Figure 3. Boundary element model of 8CN 12/13 diesel housing parts for structural noise calculation.

The transmitting function method was used when performing calculations for various design

variants. The method made it possible to exclude the repeated execution of a number of computational operations and accelerate the calculation substantially [8, 9, 10]. Calculations were carried out in acoustic octaves from 63 to 2000Hz. Figure 4 shows a comparison of calculation results with experiment.

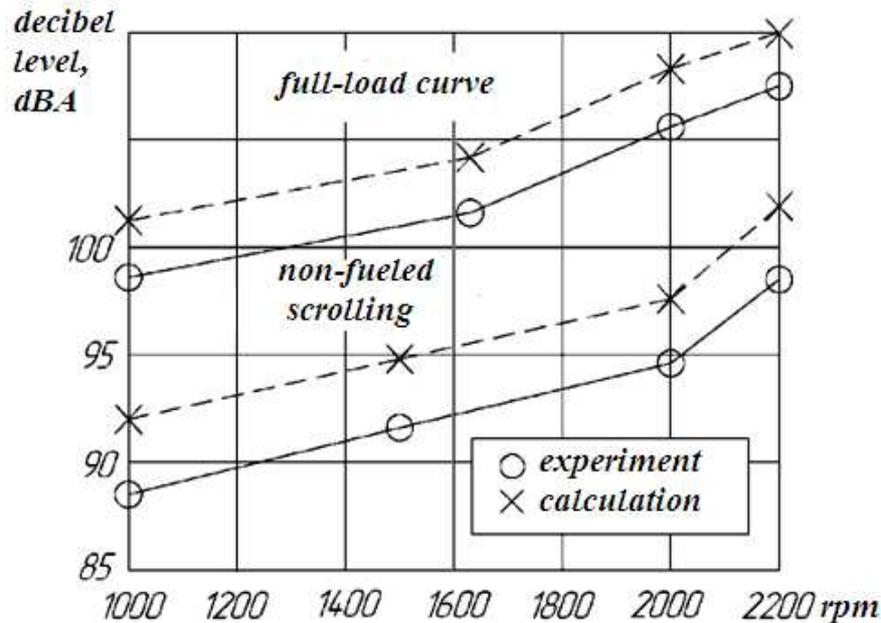


Figure 4. Average sound levels of the diesel engine external noise based on the results of the calculations and the experiment.

The discrepancy between the calculation and experiment results is connected with the experimental data deficit on the damping of the housing parts of the considered construction.

4. Conclusion

The conducted studies confirmed the necessity of studying the acoustic characteristics of power plants with internal combustion engines in order to reduce the structural noise created by them. The considered mathematical model for structural noise definition can be used both at modernization of existing internal combustion engines designs and at creation of new ones. Thus, the structural noise was managed to reduce by 2.4 dBA at the nominal operating mode. It was possible with the introduction of an additional ribbing of the case of an experimental 4CN 10.2/12.2 in-line diesel engine.

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