Development of vehicle noise-vibration-harshness analysis calculation method in order to improve NVH characteristics

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Abstract. The article presents a calculation method of Noise-Vibration-Harshness (NVH) analysis of a vehicle, the purpose of which is to reduce noise pollution. The results of calculations and experimental validations of calculations, natural frequencies and oscillation modes of the trimmed body internal volume, local dynamic stiffnesses and frequency response are provided. Also, contribution of the body panels to forming internal noise is defined.

1. Introduction
The general perception of the vehicle quality significantly depends on its internal noise characteristics. Therefore, it is important to find the right balance between "pleasant" and "dynamic" sound meeting the customer's requirements. The vehicle internal noise is regulated by GOST 33555-2015 "Motor vehicles. Internal noise. Permissible levels and methods of tests", according to which the acceptable sound level shall not exceed 77 dBA.

As a rule, a considerable part of the vehicle internal noise is formed by vibrations from the main sources such as the powertrain, transmission, suspension and steering equipment transmitted by structural means, due to which, in the last years, vehicle NVH analysis has become an integral part of designing and refinement.

On the whole, NVH analysis defines the vehicle quality parameter and falls within the engineering mechanics based on differential equations accurate solving. Analytical solving seems to be impossible, and its accuracy is not sufficient for a complex structure of a vehicle body due to high complexity & challenging of differential equations. Therefore, numerical methods of design calculation gained traction [1-6].

2. Physical and mathematical bases and finite element modeling for studying noise transfer function (NTF)
Such methods include the variational methods, variational and differential methods and the finite element method. In automotive industry challenges, body vibration is studied in most cases with an effective matrix finite element method (FEM) having a number of advantages. The finite element method assumes replacement of a design by a structural model consisting of a set of simply-shaped elements — beams, rods, etc. — with the known elastic properties. Properties of separate elements define properties of the design on the whole within the specified boundary conditions.
Software packages (MSC Nastran, Ansys, Abaqus) developed for the finite element method are universal. However, when solving particular tasks, special studies are required relating to selecting a finite element type, boundary conditions, loading modes, etc.

The FEM is used for different tasks of deformable solid mechanics, hydrodynamics and gas dynamics. The final task of defining a stress-strain state (SSS) of a mechanical design is finding stresses, strains (deformations) and displacements in each point of the design, which appear therein as a result of exposure of the design to mechanical, gas-dynamic and hydrodynamic, thermal and other loads in the course of its real operation within the vehicle. In the three-dimensional setup, defining a displacement field amounts to defining three-dimensional displacement components along the x, y, z coordinate axes in all the design points. As the number of points in a body is infinite, the number of unknown variables is also infinite. Therefore, the solution shall be defined as functions expressed in terms of equations. Even for primitive solids exposed to a simple system of forces (rectangular or round plate loaded by a center-concentrated force or a uniform load, cylindrical shell exposed to two concentrated forces, etc.), derivation of equations for defining the displacement field is a great challenge. Also, it is practically impossible to draw analytical dependencies for real complex spatial structures.

The basic idea of the FEM is as follows:

1. Any complex space structure (design) may be divided into elementary volumes (finite elements), for which rigidity characteristics can be calculated based on their elementary geometry and known material properties, by imaginary surface lines;
2. A finite number of nodes is fixed on the elements and the finite elements are considered to be connected among themselves in these nodes. The nodes and elements are numbered. This operation is often called the finite element mesh generation;
3. The displacement values are considered unknown in these nodes only. Thus, the number of the unknown variables is reduced from infinity to a specific number. For the elements, the predefined approximation laws are set in the form of polynomials (linear, quadratic, etc.). After determination of the displacements in the nodes, the displacement within any element can be defined by approximation by means of the set polynomial;
4. Based on the elementary geometrical shape of the finite elements and physical properties of the materials, the element stiffness matrices are calculated and all actual loads result in the nodal ones;
5. Augmented matrices are built from the element matrices, and then global stiffness and force matrices are formed by means of summing up the augmented element matrices. Then the boundary conditions are set;
6. The system \([K]\{\delta\} = \{R\}\) is solved, from which the vector of displacements in the nodes is found, where \([K]\) is a stiffness matrix; \(\{\delta\}\) is a displacement column vector; \(\{R\}\) is an external force vector;
7. According to the accepted approximation laws, the displacements within the elements are defined (in the points of interest);
8. Based on elasticity theory, the deformations in each element are determined from the displacements;
9. Based on the physical equations, the stresses in each element are determined from the deformations.

The differential equation of motion exposed to the harmonic forces is used for the dynamic tasks that include determination of the vehicle vibration characteristics: in general, the NVH analysis is determined by the parameter:

\[
[M]\ddot{u}(t) + [B]\dot{u}(t) + [K]u(t) = \{P(\omega)\} e^{i\omega t}
\]  

(1)

where \([M]\) is a mass matrix;
\([B]\) is a damping matrix;
\([K]\) is the stiffness matrix;
\{\ddot{u}(t)\} \text{ is the column vector of nodal accelerations as a time function;}
\{\dot{u}(t)\} \text{ is the column vector of the nodal speeds as the time function;}
\{u(t)\} \text{ is the column vector of the nodal displacements as the time function;}
\{P(\omega)\} \text{ is the external harmonic force.}

The harmonic motion implies the harmonic solution in the following form:

\[ \{u(t)\} = \{u(\omega)\} e^{i\omega t} \tag{2} \]

where \{u(\omega)\} is a complex displacement vector

Having taken the first and the second derivatives from (2), having applied the obtained expressions to (1) and reduced by \(e^{i\omega t}\), we obtain the motion equation:

\[ [-\omega^2 M + i\omega B + K]\{u(\omega)\} = \{P(\omega)\} \tag{3} \]

The equations for air volume and the structure can now be combined:

\[
\begin{bmatrix}
M_s & 0 & B_s & 0 & K_s & 0 \\
-A^T & M_f & 0 & B_f & 0 & K_f \\
\end{bmatrix}
\begin{bmatrix}
\ddot{u}_s \\
\ddot{\ddot{u}}_s \\
\end{bmatrix}
+ \begin{bmatrix}
B_s & 0 & 0 \\
0 & B_f & 0 \\
\end{bmatrix}
\begin{bmatrix}
\dot{u}_s \\
\dot{\dot{u}}_s \\
\end{bmatrix}
+ \begin{bmatrix}
K_s & 0 & 0 \\
0 & K_f & 0 \\
\end{bmatrix}
\begin{bmatrix}
u_s \\
p \\
\end{bmatrix}
= \begin{bmatrix}
P_s \\
P_f \\
\end{bmatrix} \tag{4}
\]

This equation of the interaction of the structure and ambient air is solved in MSC.Nastran. This equation is asymmetrical. MSC.Nastran by default solves the symmetric version of this equation, developed by Everstin.

3. Calculation research of the noise transfer function from the main sources

A. Finite elements mesh generation procedure for the NVH analysis

The FEM of the vehicle for the NVH calculations does not include the powertrain, exhaust system and chassis. The FEM of the body in white and the components was built based on the digital mock-up. The "Trim body" assembly is carried out by mounting of the hang-on units to the body in white – the trim body components, which include the front and rear doors, trunk lid, dashboard with console, front and rear seats.

The "Trim body" assembly is schematically shown in Figure 1. The "Trim body" assembly was performed in ANSA software suite of BETA CAE Company, in NVH Console module, according to the procedure described in. The internal acoustic volume of the vehicle was also created in ANSA Software as shown in Figure 2.
B. Calculation research of the natural frequencies and oscillation modes of the vehicle inner volume

One of the significant factors influencing the noise level in the vehicle passenger compartment is the air volume resonance. This phenomenon is caused by coalescence of the natural oscillation frequencies of the air volume and the natural oscillation frequencies of the body panels. The sound pressure level increase at resonance considerably exceeds the peak values caused only by excitation from the body panels vibration. Such phenomena can be observed when driving on uneven surfaces having quite a long distance, for example, cobblestone pavement. Analysis of natural oscillation modes was conducted in the MSC.Nastran software, the first twenty modes were defined.

Figures 3 show an example of the first three oscillation modes of the inner air volume of the vehicle.
Figure 3. The first eigenmode at 47.5 Hz (a), the second eigenmode at 82.8 Hz (b) and the third eigenmode at 84.1 Hz frequency.

The red marking in Figures 3-5 shows ambient air pressure corresponding to the maximum values; this means that the local maximum of acoustic pressure for this oscillation mode occurs in this area. The blue marking in the figures corresponds to the minimum values of the acoustic pressure inside the vehicle passenger compartment.

The first air volume oscillation mode (47.5 Hz) is longitudinal – the acoustic pressure spreads in the longitudinal direction of the passenger compartment, the maximum sound pressure occurs at the rear part of the vehicle’s passenger compartment.

The second air volume oscillation mode (82.8 Hz) is diagonal – the acoustic pressure from the driver side front end shield spreads to the passenger side luggage compartment.

The third air volume oscillation mode (84.1 Hz) is symmetric as to distribution in the diagonal direction – distribution from the driver side front end shield to the passenger side luggage compartment. The acoustic wave spreads from the passenger side front end shield to the driver side luggage compartment.

C. Main vibration sources of the vehicle body

The vibration sources affecting the vehicle bodies can be divided by excitation type into the sources with periodic, random and compound type of excitation. The first type is as follows: the powertrain, transmission and chassis. The second type is the suspension and hang-on units. The third type is the vehicle exhaust system mounts.

In the powertrain mount vibration frequency spectrum, there are components caused by the overturning moment, unbalance and unbalanced forces, as well as the engine torque, when the vehicle is motionless. When the vehicle is motionless and the engine is at idle, at low crankshaft speed, the general vibration level of the powertrain mounts is defined by the first harmonic intensity of the engine overturning moment. The higher the engine crankshaft speed in vibration spectrums of the powertrain mounts is, the lower the level of components dependent on the overturning moment action is and the higher the level of components caused by unbalance, unbalanced forces and inertia moments is.
The passenger car suspension transfers vibration to the body as a result of kinetic excitation from the road micro profile. The stiffness of tyres and suspension, the sprung weight and its distribution, unsprung weight, friction in suspension, damping and elastic characteristics of rubber vibration isolators connecting the suspension elements define the intensity of low frequency vibration and the sound level under the same disturbance transferred from road microprofile.

In view of the above, 120 points have been considered as the vibration sources, which relate to: the powertrain, suspensions, chassis, exhaust system and hang-on units.

D. Calculation and experimental research of internal noise from the main vibration sources

The sound pressures from the vibration sources were calculated in 10 reference points of the vehicle passenger compartment as follows:

- Driver's left ear;
- Driver's right ear;
- Front passenger's left ear;
- Front passenger's right ear;
- Rear passenger's left ear on the driver's side;
- Rear passenger's right ear on the driver's side;
- Rear passenger's left ear on the passenger's side;
- Rear passenger's right ear on the passenger's side;
- In the middle, between front and rear passengers;

Due to limitation of the paper volume, let us consider one excitation point, which is the powertrain rear mount shown in Figure 5. The sound pressure at the driver's right ear is also shown in Figure 5.

![Figure 4](image1.png)

**Figure 4.** Excitation point position – rear powertrain mount: a – calculation research, b – experimental research.

![Figure 5](image2.png)

**Figure 5.** Acoustic response reference point position.
Figure 6 shows the results of the calculation and experimental research of the sound pressure from the structural excitation along the Z axis.

![Figure 6. Calculation and experimental research results.](image)

From the diagram, it is obvious that good correlation of the experimental research results is observed within the frequency range of 20-75 and 90-130 Hz. The maximum sound pressure corresponds with the frequency of the lowest global bending of the trimmed body. The not good correlation of the results of the calculation and experimental research within the bandwidth of over 130 Hz is caused by influence of the absorbing materials used in the experiment.

4. Defining body panel contribution to the general integral noise level

In the course of solving the connected task of defining the sound pressure levels, impact of both air volume natural oscillation modes and separate panel vibrations on the noise level in the vehicle passenger compartment can be defined. Mode impact coefficient can be defined from Equation 4 using the following transformations:

\[
\{u_1\} = [\Phi_s] \{\xi_s\} \tag{5}
\]

\[
\{p\} = [\Phi_f] \{\xi_f\} \tag{6}
\]

where \([\Phi_s]\) – uncoupled and undamped oscillation modes of the body structure;
\([\Phi_f]\) – uncoupled and undamped oscillation modes of the air volume;

vectors \(\{\xi_s\}, \{\xi_f\}\) – natural oscillation mode amplitudes

Inserting these ratios in the equation and multiplying with modal matrices, the following equation is obtained

\[
\begin{bmatrix}
\Phi_j^T \mathbf{M}_s \Phi_j & 0 \\
-\Phi_j^T \mathbf{A} \Phi_j & \Phi_j^T \mathbf{K}_s \Phi_j
\end{bmatrix} \begin{bmatrix}
\{\xi_s\} \\
\{\xi_f\}
\end{bmatrix} + \begin{bmatrix}
\Phi_j^T \mathbf{B}_w \Phi_j & 0 \\
0 & \Phi_j^T \mathbf{B}_r \Phi_j
\end{bmatrix} \begin{bmatrix}
\{\xi_j\} \\
\{\xi_f\}
\end{bmatrix} + \begin{bmatrix}
\Phi_j^T \mathbf{M}_r \Phi_j & 0 \\
0 & \Phi_j^T \mathbf{M}_p \Phi_j
\end{bmatrix} \begin{bmatrix}
\{\xi_s\} \\
\{\xi_f\}
\end{bmatrix} = \begin{bmatrix}
\{\Phi_j^T \mathbf{P}_j \Phi_j\} \\
\{\Phi_j^T \mathbf{P}_f \Phi_j\}
\end{bmatrix}
\]

or

\[
\Sigma \Phi_j^T \mathbf{M}_s \Phi_j \{\xi_s\} + \Sigma \Phi_j^T \mathbf{A} \Phi_j \{\xi_f\} + \Sigma \Phi_j^T \mathbf{K}_s \Phi_j \{\xi_j\} = \Sigma \Phi_j^T \mathbf{P}_j \Phi_j \{\xi_j\}
\]
The following harmonic solution causing frequency $\omega$ is defined:

$$\begin{bmatrix} m_s & 0 \\ -a^T & m_f \end{bmatrix} \begin{bmatrix} \ddot{\xi}_s \\ \ddot{\xi}_f \end{bmatrix} + \begin{bmatrix} b_s & 0 \\ b_f & 0 \end{bmatrix} \begin{bmatrix} \dot{\xi}_s \\ \dot{\xi}_f \end{bmatrix} + \begin{bmatrix} k_s & a \\ 0 & k_f \end{bmatrix} \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = \{Q_f\}$$  \hspace{1cm} (8)

The following is obtained

$$\omega^2[a]^T[\xi_s] + [-\omega^2[m] + i\omega[b]]\xi_f + [k_f]\{\xi_f\} = \{Q_f\}$$  \hspace{1cm} (9)

Then, $[Z_2]$ is defined as follows:

$$[Z_2] = [-\omega^2[m] + i\omega[b] + [k_f]]^{-1}$$  \hspace{1cm} (10)

Then

$$\{\xi\} = -\omega^2[Z_2][a]^T[\xi_s] + \{Q_f\}$$  \hspace{1cm} (11)

The following is obtained

$$\omega^2[a]^T[\xi_s] + [-\omega^2[m] - i\omega[b]]\xi_f + [k_f]\{\xi_f\} = \{Q_f\}$$  \hspace{1cm} (12)

Coefficient of air volume oscillations impact on the noise level is defined as follows:

$$[P_f] = [\Phi_f][\{\xi_s\}]$$  \hspace{1cm} (13)

where $[\{\xi_s\}]$ – a diagonalized vector of air volume modal amplitudes for the selected bandwidth.

Coefficient of structure vibration impact on noise level is defined as follows:

$$[P_s] = -\omega^2[\Phi_s][Z_2][a]^T[\{\xi_s\}]$$  \hspace{1cm} (14)

where $[\{\xi_s\}]$ – the diagonalized vector of the structure modal amplitudes for the selected bandwidth.

Acoustic loading impact coefficient is defined as follows:

$$[P_l] = [\Phi_f][Z_2][\{Q_f\}]$$  \hspace{1cm} (15)

where $[\{Q_f\}]$ – a diagonalized vector of the acoustic pressure amplitude for the selected bandwidth.

Coefficient of impact of the separate body panels on the sound pressure level can be defined as follows:

$$[P_p] = -\omega^2[\Phi_p][Z_2][A]^T[\{\xi_s\}]$$  \hspace{1cm} (16)

The abovementioned dependencies allow defining the level of impact of separate calculation model parts (air volume, body panels) on the sound pressure level in the vehicle passenger compartment, which allows unambiguously defining the main structural noise sources in the vehicle, as well as main activities to reduce the noise.

At the same time, the natural oscillations of the air volume are also the reason for emergence of the acoustic pressure peaks in the vehicle passenger compartment. The oscillation modes of the body panels and air volume are complicated enough. It is impossible to define analytically, which of them make the largest contribution to the noise level in different points of the vehicle passenger compartment. In order to define the vibration impact of both structural part (body) and air volume on sound pressure level, the oscillation mode impact was analyzed. The model can be conditionally divided into two parts – a structural model and an air volume model. In turn, for more accurate defining of separate panel vibrations, the bodies can be divided into the separate panels shown in Figure 7.
The boundary conditions are similar to the previous calculation research, but with addition of request for panel output in the Case Control section, it is also necessary to record the SET3 chart and name the panels in the PANEL chart. The calculation result is shown in Figure 8.

The NTF calculation results show the maximum amplitude of the sound pressures from the first global bending. Also, there are increased amplitudes of the sound pressures within the bandwidth of 115-215 Hz, a significant contribution to the general integral level is made by the roof panel and left front floor panel.

Moreover, when analyzing the NTF from 120 excitation points, the largest contribution was defined from the roof panel and rear floor panel. Minimizing the NTF level is achieved by reducing the vibration activity of the body panels of the roof and rear floor. Due to that, topographical optimization of the rear floor was made. The target function of topographical optimization is increasing the natural oscillation frequency. The rear floor panel FEM is shown in Figure 9.
The topographical optimization result is shown in Figure 10.

Based on the optimization results, the first oscillation mode natural frequencies of the floor panel were successfully increased from 46.5 to 70.9 Hz (Figure 11).
After optimization, the NTF was calculated, which showed noise reduction by 2-3 dB from all excitation points.

5. Conclusions

The developed NVH analysis calculation method allowed as follows:

• To perform a comprehensive assessment of the impact of structural excitation on sound pressure formation in reference points. The spectral content and amplitudes of sound pressures from each source of structural excitation were determined;
• To perform an assessment of the impact of the body panels on noise formation in the vehicle;
• To define the panels that make the most significant contribution to the general integral noise level.

To reduce the noise generated by the rear floor panel, the topographical optimization was performed, as a result of which the noise was reduced by 2 dB;

This method can be used in research and academic institutions, as well as in manufacturing enterprises specializing in development and production of complex products including vehicle bodies.

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