Vehicle structural analysis calculation method development in order to improve noise-vibration-harshness characteristics

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Abstract. The article presents a calculation method of structural analysis of a vehicle through the example of a body, 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 body, local dynamic stiffnesses and frequency response are provided. Also, the body characteristics in natural frequencies are improved through topological and topographical optimization of the body element.

1. Introduction
Today a vehicle is the well-balanced and complicated high-tech object, for which an attempt is made to combine the contradictory properties (inconsistent with each other): safety, environmental friendliness and comfort. The main driving forces for development of the modern vehicle components are steadily growing requirements from the customers and competitive market nature.

NVH (Noise-Vibration-Harshness), which is the comprehensive analysis of vehicle vibration and acoustics characteristics, is one of the most important attributes for development within the automotive manufacturing. The NVH analysis defines the vehicle quality level and applies to the mechanical engineering area focused on measurement and optimization of noise and vibration (NVH) characteristics of mechanical designs or structures. The current trends in the NVH analysis of the vehicle design are as follows: rise of the role of virtual prototypes of the vehicle elements to reduce the development cycle and costs while preserving the quality; large-scale use of a great number of vehicle variants developed on a small number of the “vehicle platforms”. NVH becomes particularly important due to the following customer and society requirements to vehicle manufacturers: customers have become more demanding and exacting about vehicle NVH characteristics; requirements for fuel saving force to develop lighter vehicles that leads to noise and vibration issues and problems becoming more obvious and critical.

Conducting the structural noise and dynamic stiffness assessment research is a part of the NVH analysis and is focused on creation of the vehicle with perfect end-user performance. The main sources of the structural noise of modern vehicles are: powertrain, chassis, suspension, exhaust system mounting points, etc [1-6].

The vehicle structural noise is the following in nature – the source operational load influences the body and causes its forced oscillations. The forced oscillations are "felt" by the customer primarily with the vehicle controls. And also, the forced oscillations excite the body panels, and they, in turn, are sources of acoustic radiation. The sound power of radiating surfaces can be determined by the following formula described in chapter 4 [7]:

\[ W(f) = \sigma(f) \rho c S \Psi^2(f) \]  

(1)
where: $\sigma(f)$ - source radiation coefficient, $\rho \cdot c$ – acoustic resistance of the medium, $S$ – radiator area, $\dot{V}^2(f)$ – radiating surface vibration speed mean-square value.

Based on the above, in order to minimize the structural noise, it is necessary:

1. To achieve the target values for the lowest global natural bending and torsion frequencies of the body. The lowest global natural frequencies of the trimmed body shall be above 20 Hz (excitation from the road) and shall not coincide with the second engine harmonic of the internal combustion engine (ICE);

2. To achieve the target values of the local dynamic stiffnesses of the mounting points of the structural noise sources;

3. To achieve the target values for the vibration acceleration level on the controls;

4. To achieve the target values for the lowest local frequencies and vibration speed levels of the body panels (to reduce the vibration activity of the body panels, stamped reinforcements and stiffeners are applied, based on topological and topographical optimizations);

5. To achieve the target values for the "filtration" of the vibration acceleration levels (10 times ideally) in order to ensure the incoherence of the body and sources oscillations;

6. To diverge the global natural frequencies of the body within the vehicle and oscillation frequencies from the sources.

The target values shall be obtained by comparison with the previous generations of vehicles and further analysis or by benchmarking of this vehicle class.

In view of the above, the article presents theoretical and experimental research as follows:

• Lowest natural frequencies of the vehicle body in white tested for bending and torsion;

• Increase of the lowest frequency of the body in white global torsion by means of optimization of the luggage compartment partition;

• Local dynamic stiffnesses of installation areas of main sources of vibration;

• The vibration transfer function (VTF) from the main sources of excitation to the cross-member frame (the connection point with the steering column, control element);

2. Definition of the lowest global modes of the vehicle body

A. Calculation research of the lowest global modes of the body

When calculating the natural frequencies and oscillation modes in a free state, there shall be modes with zero frequencies that are called the modes of motion of an absolutely rigid body. The motion of the rigid body of all or part of the design represents the motion of the design in the absence of stresses. They can also indicate the modeling errors or incorrect setting of boundary conditions.

Based on the finite element mesh generation procedure, a model of the vehicle body was created shown in Figure 1.
The calculation model is based on the body within the Body in White, including the front subframe, front end panel assy, dashboard crossmember, rear transverse bar and the rear window.

For identification of the first natural bending frequency of the body, it is loaded by means of the force couple at the frequency range from 0 to 60 Hz, as shown in Figure 2.

![Figure 2. Force couples: a – front end bending load, b – rear end bending load.](image)

Then, the response (vibratory displacements) is calculated in the body load-bearing structure reference points. The frequency, at which the movements will be the maximum ones, is the first natural bending frequency of the body. All the tasks were solved in the MSC.Nastran software. When determining the transfer functions, the root-mean-square averages were calculated for all the output points.

For identification of the first natural torsion frequency of the body, it is subject to torsion by dynamic torque by means of the force couple at the frequency range from 0 to 60 Hz, as shown in Figure 3. Then, the response (displacements) is calculated in the body framing reference points. The frequency, at which the displacements will be the maximum ones, is the first natural torsion frequency of the body.

![Figure 3. Force couples: a – front end torsion, b – rear end torsion.](image)

The results of the modal analysis calculations are shown in Figure 4 – the 1st torsion one – and Figure 5 – the 1st bending one.

![Figure 4. First (1st) torsion mode at the frequency of 42.5 Hz (a), the first (1st) bending mode at the frequency of 49 Hz (b).](image)
B. Calculation experimental validation

The reliability of the finite element modeling calculation results for such complex systems as a modern vehicle body shall be evaluated experimentally. In view of the above, the experimental research of natural frequencies and modes of oscillations was conducted resulting in definition of the first torsion and the first bending ones.

The measured data were processed within 10 – 115 Hz frequency range.

The body assy (composition of the body is identical to the calculation model) [8] was installed on 4 pneumatic cylinders. The body excitation was performed via two electrodynamic vibration generators located diagonally (one generator was in the mounting point of the front right strut of the shock absorber; another one was on the left side in the mounting point of the rear subframe), shown in Figures 5 – 6.

![Figure 5. The electrodynamic vibration generator in the mounting point of the front right strut of the shock absorber.](image)

The measurements of the vibration accelerations were performed in 78 points of the body with simultaneous data registration from 12 points (Figure 6). Figure 6 b presents the body points layout for the modal analysis.

![Figure 6. Example of measuring points (a) and Geometrical model of measuring points on the body for modal analysis (b).](image)

The measured data were analyzed using the LMS Modal Analysis software for creation of modal shapes of the body and for calculation of its natural frequencies and damping. The coherence of the input and output signals was checked in the course of measurement. For the calculation of copy oscillation modes, the range of 10-115 Hz was chosen in the Polymax Plus menu.

Figure 7 shows the result of experimental research, the 1st torsion mode is at frequency of 42.6 Hz
The measured data were analyzed using the LMS Modal Analysis software for creation of modal shapes of the body and for calculation of its natural frequencies and damping. The coherence of the input and output signals was checked in the course of measurement.

Figure 7. First (1st) torsion mode at frequency of 42.6 Hz.

Figure 8 shows the 1st bending mode at frequency of 49.27 Hz.

Figure 8. First (1st) bending mode at frequency of 49.27 Hz.

The natural frequencies obtained during the modal analysis are given in Table 1 in comparison with those obtained in the FEM calculation.

| Mode name         | Calculation | Experiment | Difference |
|-------------------|-------------|------------|------------|
| 1st torsion mode  | 42.5 Hz     | 42.6 Hz    | 0.1 Hz     |
| 1st bending mode  | 49.1 Hz     | 49.27 Hz   | 0.17 Hz    |

Good correlation of results (natural oscillation frequencies) obtained using the modal analysis and the FEM is obtained. The first natural global torsion frequency is lower than the target ones, and therefore the optimization of the luggage compartment partition was performed.

C. Structural optimization of the body for achieving target natural frequencies

Topological optimization was conducted to achieve the target values. The optimization target function is body torsion oscillation frequency increase. The luggage compartment partition is substituted by a solid plate of ~ 0.2…4 mm thickness. Topological optimization was conducted, as shown in Figure 9 (a); the partition model taking into account packaging and shown in Figure 9 (b) was created based on the results of the topological optimization.
Further, the topological optimization was conducted on the partition model which takes into account packaging peculiarities of the vehicle. Based on the topological optimization results, the areas of the material exclusion were defined, as shown in Figure 10.

As seen in Figure 10, the basic material of the part is concentrated on the lower part of the partition (near the floor transverse member). The maximum thickness is 4 mm (marked in red), the minimum thickness is 0.2 mm (marked in red). Moreover, the topological optimization shown in Figure 11 was performed to reduce the partition surface vibration speed.

Due to the optimization works, the first torsion frequency of the body was increased up to 4 Hz, while the first natural frequency of the luggage compartment partition was increased from 32 up to 129 Hz (Figure 12).
Figure 11. Process of stiffness increase of the luggage compartment partition aimed at decrease in the surface vibration speed level.

Figure 12. The first oscillation mode of the optimized design of the luggage compartment partition at the frequency of 129 Hz.

3. Determination of the local dynamic stiffnesses of the body excitation points
A. Physical and mathematical bases for research of the local dynamic stiffnesses

As it is known, the local dynamic stiffness is the value of the dynamic load (stress) causing a single displacement in this load application point, N/mm. And the accelerance is the reciprocal of the dynamic stiffness expressed as the accelerations, mm/N*s².

The method of normal oscillation modes (the alternative name is the normal mode method) was used for calculation of the local dynamic stiffness. This method uses the natural oscillation modes of the design to decrease the task order, untie the motion equations (which is possible when using damping according to the modes) and make the numerical integration more effective. As the natural modes are usually calculated at determination of the design dynamic properties, the analysis of the unsteady oscillations by the normal mode method is the natural extension of the natural frequencies analysis. The first step of the unsteady oscillations analysis by the normal mode method is conversion of variables from physical coordinates {u} to generalized coordinates {ξ} according to the formula:
The natural modes \([\phi]\) are used to transform the task from the terms of unit displacements to the terms of generalized displacements. Equation (2) is valid if all the natural modes are used; however since the natural modes are used rarely, the equation is usually some approximation. Moreover, when using structural damping, it is necessary to remember that the damping matrix \([B]\) is used to provide the properties of energy diffusion in the system. As a rule, the damping matrix comprises several matrices

\[
[B^1] + [B^2] + (G/W_3)[K] + (1/W_4)\sum G_E[K_E]
\]  

(3)

- \([B^1]\) - damping elements;
- \([B^2]\) - direct input matrix;
- \(G\) - global structural damping coefficient;
- \(W_3\) - characteristic frequency in rad/s for conversion of the global structural damping into equivalent viscous damping;
- \([K]\) - global stiffness matrix;
- \(G_E\) - elements structural damping coefficient;
- \(W_4\) - characteristic frequency in rad/s for conversion of elements structural damping into equivalent viscous damping;
- \([K_E]\) - elements stiffness matrix.

Linear analysis of unsteady oscillations does not allow using complex coefficients. Therefore, the structural damping is considered by its substitution with the equivalent viscous damping. To determine the impact of such substitution on the solution, it is required to define the relation between the structural damping and equivalent viscous damping.

The viscous damping force is a damping force which is the function of damping coefficient \(b\) and velocity. This force is represented in the motion equation through matrix \([B]\) and velocity vector.

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

(4)

The structural damping force is the damping force depending on displacement. The structural damping force is the function of damping coefficient \(G\) and is included in the stiffness matrix as a complex-valued component.

\[
[M][\ddot{u}(t)] + (1+iG)[K][u(t)] = \{P(t)\}
\]  

(5)

Supposing that response amplitude of the system with one degree of freedom is constant, then both damping forces are equal if:

\[
Gk = b\omega
\]  

(6)

or

\[
b = Gk/\omega
\]  

(7)

Therefore, if simulating structural damping \(G\) by equivalent viscous damping \(b\).

B. Local dynamic stiffness calculation research

Based on the above, equation 4 shall be solved for local dynamic stiffness calculation definition taking into account the structural damping. The calculations and experimental research of local dynamic stiffnesses were performed in 120 points.

Figure 13 shows an example of location of the accelerance investigation point.
Figure 13. LDS upper front arm suspension attachment.

The results of the calculation and experimental research of the accelerance of the body excitation point from the front suspension upper arm right mounting are shown in Figure 14.

Figure 14. LDS upper front arm suspension attachment

The insignificant deviation of the calculated values accelerance amplitudes is associated with the structural damping, which was set at 0.03 in the calculation. And when the structural damping was increased up to 0.05, the error amounted to less than 1%. There is an experimental method of damping determination based on exponential decay, but in this paper, we used only the calculated values of 3% and 5% of the structural damping recommended by. Good correlation of the calculation and experimental research results for the front suspension mounting point accelerance is obtained. For result analysis, the average local dynamic stiffnesses (LDS) or accelerance are usually used. As an example, an excitation point accelerance averaging based on the calculated/measured local dynamic stiffnesses is considered.

4. Defining frequency response from the body excitation points

A. Body frequency response calculation research

In order to understand the full vibration transfer function process, the frequency response in our points of interest (controls, seat attachment, body panels, etc.) shall be also considered. The calculations and experimental research of the frequency response were performed at 30 points. As an example, let us consider only one excitation point of the frequency response. The excitation point is located at the suspension upper arm right mount with the response situated at the dashboard cross-member (Figure 15).
Figure 1.5. a - the point of excitation and response in the calculation model and b – the point of excitation and response under the experimental research.

Figure 16 shows the results of calculation and experimental research. Due to the fact that the maximum vibration acceleration amplitudes are found in the frequency range up to 50 Hz, the amplitude spectral characteristics are given in this frequency range.

![Figure 16. Result of the calculation and experimental research of the frequency response.](image)

It is evident from the provided diagrams that at suspension front upper right arm mounting point excitation, there is an increased vibration acceleration in the dashboard cross member in the vertical direction. This phenomenon is caused by the coincidence of the dashboard framing natural frequency and the excitation frequency. In order to decrease the vibration acceleration amplitude, two solutions are supposed:

- Increasing local dynamic stiffness of the excitation point;
- Increasing stiffness of the dashboard framing.

According to the frequency response results, good correlation of the calculation and experimental research results is recorded, the error is less than 3%.

5. Conclusions
The developed structural analysis calculation method allowed as follows:

- Assessment of an increase in the lowest first global torsion frequency of the body by 4 Hz based on the topological optimization;
- Assessment of the local dynamic stiffnesses of the installation points of the structural noise sources;
- Assessment of an increase in the natural frequencies of the luggage compartment partition based on the topographical optimization, as well as allowed a decrease in the vibration activity after the topographical optimization;
- Assessment of the vibration acceleration levels on the controls.
The error of the calculation and experimental research was no more than 7%, and upon that special attention shall be paid to the structural damping.

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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