MODEL VIBRATION STUDY OF MACPHERSON TYPE SUSPENSION USING EFFECTIVE MASS FACTOR

Zlatin Georgiev
Burgas State University "Prof. d-r Assen Zlatarov", Bulgaria
zlaty.georgiev@gmail.com

Abstract. In most cases, model vibration study of a system requires a large amount of computing resources for calculation. The solution itself takes a lot of technological time. If it is necessary to change the boundary conditions and initial parameters, the design process becomes even longer and more difficult. At the rapid pace of automotive development, this is unacceptable. This article proposes a method for analysing the vibration behaviour of the MacPherson strut suspension separating its elements. The method uses the so-called effective mass participation factor, which indicates what percentage of the entire mass of the element is involved in the vibrations for a given natural frequency mode. A mode with a high effective mass factor will make a significant contribution to the dynamic response of the element to disturbance in a stated direction. This method is characterized by fast calculations and requires low computing resources for solution. A comparison between the results of the proposed method and the results of the dynamic study of the entire system is made. The solution times of the two approaches are also compared. The object of the research are the elements with natural frequencies in the range from 30 to 150 Hz. These are the coil spring, the shock absorber piston rod and the lower arm. This frequency range is characterized by the presence of the first radial modes of the pneumatic tyre. They have the greatest influence on the structure-borne noise in the car interior. That is why the main task of noise, vibration and harshness engineers is to avoid coupling of the tyre modes with those of the suspension elements. In the present study for the specific MacPherson strut suspension, an overlap of the natural modes of the piston rod and the lower control arm is found at a frequency of about 111 Hz.

Keywords: structure-borne noise, tyre noise, MacPherson suspension, comfort, vibrations.

Introduction

Model study is indispensable in automotive development or any other branch of mechanical engineering. It is the most convenient way to establish how different input data or boundary conditions would affect the system [1-6]. With the increasing model range offered by each automotive manufacturer, the mandatory experimental work should be minimized. This significantly reduces the development process. The complexity of the model also leads to difficulties in its calculation and prolongs the process [7-9]. To facilitate the calculation, the model used to study a given system is broken down into separate models [10]. Co-simulations are carried out by different teams or directly by the supplier of the specific part (Figure 1).

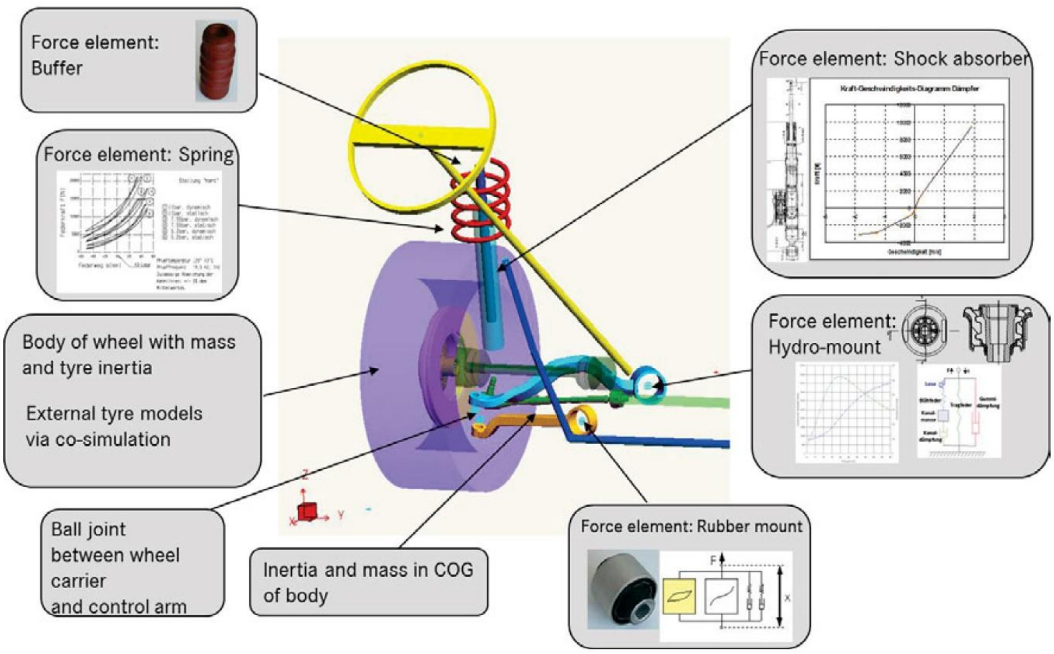

Fig. 1. **Chassis simulation by means of co-simulation [10]**

The car chassis is made up of many different elements – pneumatic tyres, hydraulic shock absorbers, rubber mounts and various metal elements. Each of them has a different vibration behaviour. Overlapping of the natural modes of the individual elements will increase the structure-borne noise that is transmitted to the body panels and the vehicle cabin. That is why the main task of noise, vibration and harshness engineers is to avoid coupling of the tyre modes with those of the suspension elements [11-13].

In this work the vibration behaviour of individual metal elements that have their natural modes in the frequency range 30-150 Hz is investigated. This range is characterized by the fact that the first radial and transversal modes of the tyre are located in it [14]. These modes are characterized by high levels of vibrations that are transmitted to the wheel centre (hub). Passing through the suspension elements, they reach the body panels and windows, which in turn create the structure-borne noise in the passenger compartment [15].

The suspension parts are modeled with finite elements. Finite element models are suitable not only for vibration studies but also for many durability load cases (special events). Knowing the vibration behaviour of individual parts, a conclusion can be drawn about the suspension system as a whole.

Materials and methods

In this work the vibration behaviour of a MacPherson strut suspension is studied using the finite element method (FEM). The model study of the individual suspension parts is expressed in calculating their natural modes and the effective mass participation factor. The effective mass participation factor shows what percentage of the entire mass of the element (system) participates in the vibrations for a given natural frequency (mode). A mode with a high effective mass factor will have a significant contribution to the dynamic response of the element (system) [16]. The objects of study are the parts with natural frequencies in the range from 30 to 150 Hz. These are the coil spring, the piston rod of the shock absorber and the lower arm. For each part is conducted frequency study in SolidWorks Simulation software [16].

In Figure 2 is shown a finite element model of the whole suspension. This model is confirmed to be correct by comparing its results with the experimental ones in [17]. The parts of interest and the corresponding boundary conditions are extracted from the model.

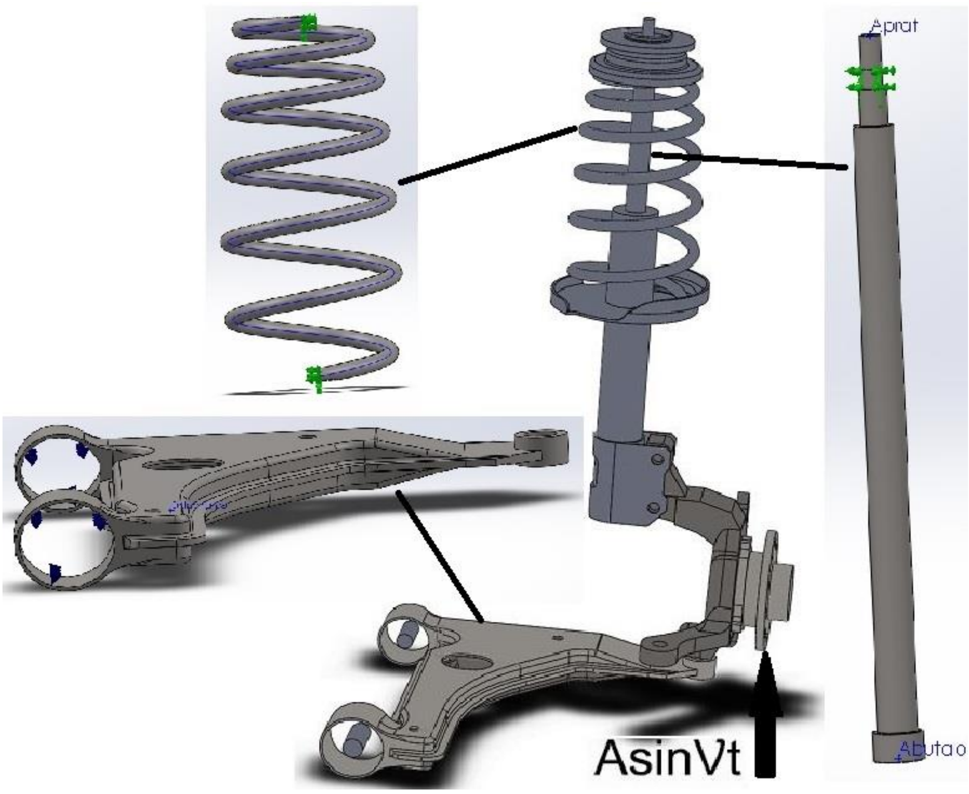

Fig. 2. **Extracting the parts of interest of the finite element model of the whole suspension**

In work [17], a harmonic study is conducted in which the input parameter is harmonic vertical disturbance applied at the wheel flange. Frequency response functions (FRFs) for different points of the suspension are calculated. The results show significant levels of vibrations at the frequency of 41, 120 and 133 Hz. The structure-borne noise is determined by the vibrations obtained at the upper support of the MacPherson strut and those in the lower arm. The former are transmitted to the vehicle body panels, and the latter to the subframe to which the engine is mounted. The vibrations in the subframe also reach the vehicle body through its supports. Thanks to various measurement points in the model, it is established that the high vibrations at 41 Hz are due to the resonance of the coil spring, those at 120 Hz are due to the piston rod and those at 133 Hz frequency are due to the lower arm. In the present work, the vibration behaviour of these parts is studied. The approach is different and the parts are analyzed independently. The solution takes insignificant time and requires low computational resources, while for the harmonic analysis of the suspension system a large computational resource and long corresponding calculation time are required.

The steel chosen for the elements had an elastic modulus $E = 2.1\times10^{11}$ N·m^{-2}, Poisson's ratio $\mu = 0.28$, and density $\rho = 7800$ kg m^{-3}. Tetrahedral elements with a size of 10 mm were used to generate the mesh of finite elements. A subsequent numerical solution with finite elements with a size of 5 mm was carried out to remove the influence of the so-called artificial stiffness (stiffness that depends on the size of the mesh element) on the results.

Results and discussion

The coil spring is modelled in a compressed state, limited by the distance to its upper support when installing in the MacPherson strut. The boundary conditions are expressed in fixing (removing all degrees of freedom) the both ends of the wire. Figure 3 presents the first ten calculated natural modes in hertz (on the X axis) and their corresponding effective mass participation factors in percent (on the Y axis). Of greatest importance for the vibration behaviour of the spring are the natural modes at 50.28 Hz, 57.05 Hz and 58.08 Hz. The first of these mode shapes is characterized by oscillations of the wire in the vertical direction. The other two natural frequencies are relatively close, and the mode shapes are characterized by oscillations in the longitudinal and transverse directions.

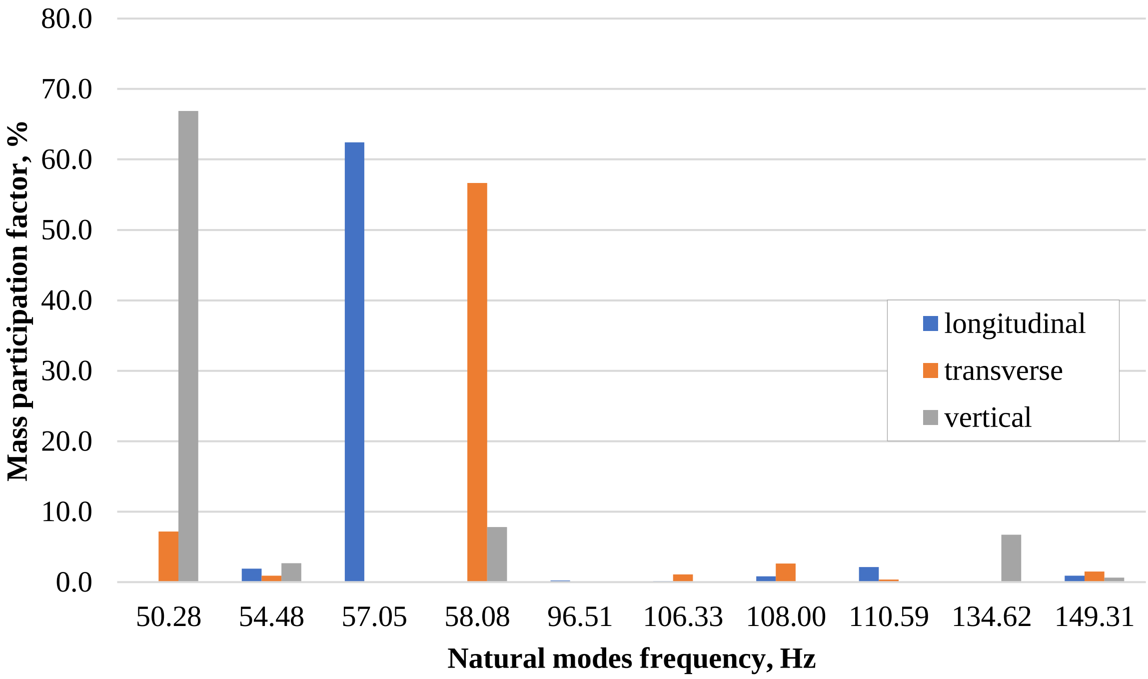

Fig. 3. **Natural modes of the coil spring and the corresponding effective mass factors for a given direction of vibration**

The piston rod and the piston are modelled as a single body. The boundary conditions are expressed in the restriction of all degrees of freedom at the surface of attachment of the piston rod to the upper mount of the MacPherson strut. Figure 4 presents the first four calculated natural modes of the piston rod in hertz (on the X axis) and their corresponding effective mass participation factors in percent (on the Y axis). Only the first two modes are in the frequency range up to 150 Hz. The frequencies are extremely close, 107.14 Hz and 107.19 Hz, respectively, which is due to the symmetry of the element. The mode shapes are characterized by vibrations in the transverse and longitudinal directions (bending shape).

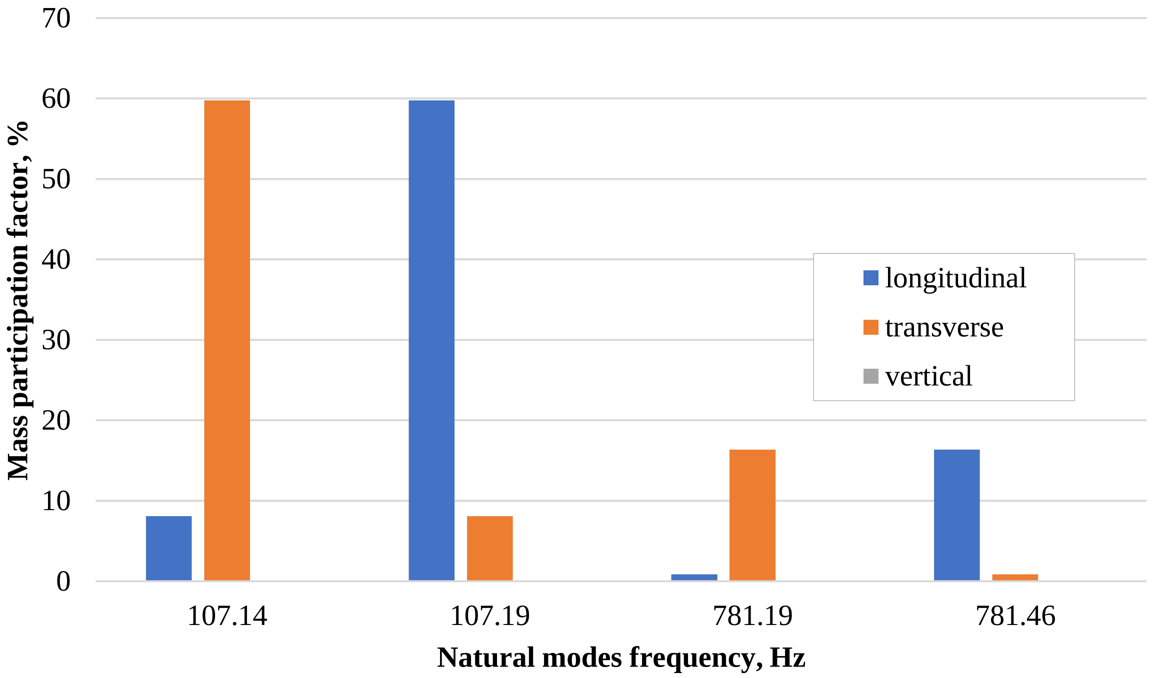

Fig. 4. **First four natural modes of the piston rod and the corresponding effective mass factors for a given direction of vibration**

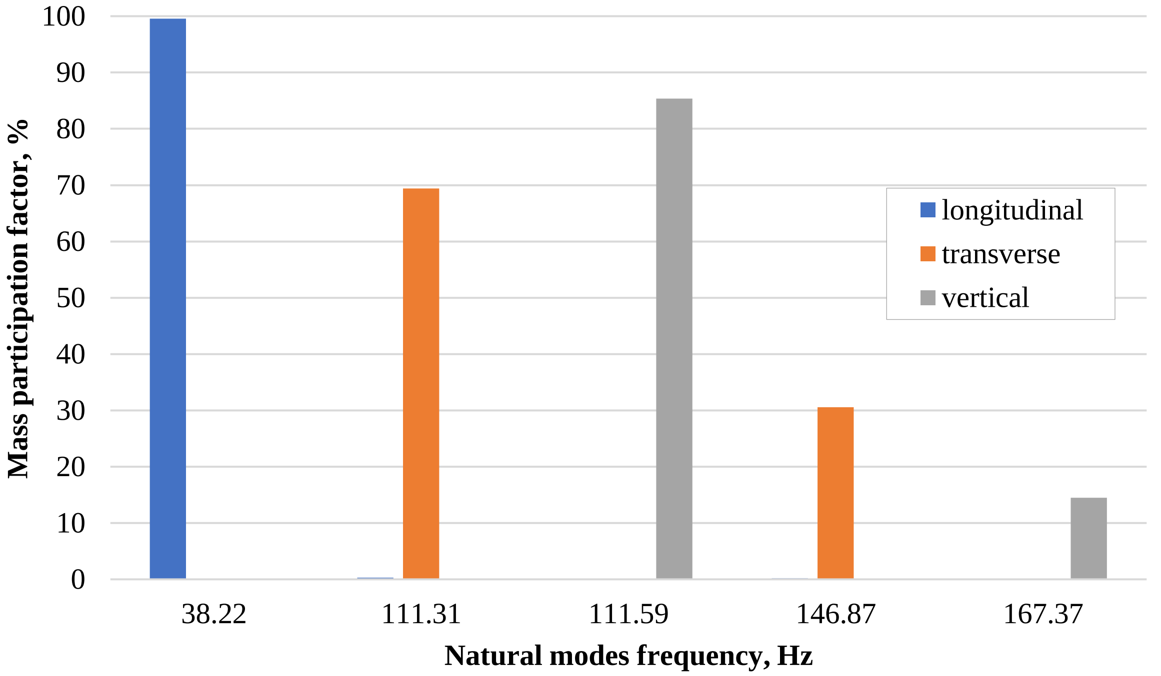

Fig. 5. **First five natural modes of the lower arm and the corresponding effective mass factors for a given direction of vibration**

For the lower arm, the boundary conditions are expressed in the placement of elastic supports in the holes for the rubber mounts. The supports have a radial stiffness of 1×10^6 N·m^{-1}. Figure 5 presents the first five calculated natural modes of the lower arm in hertz (on the X axis) and their corresponding effective mass participation factors in percent (on the Y axis). The first natural mode at 38.22 Hz is due to the axial stiffness of the supports, which in this case has a value of 1×10^5 N·m^{-1}. It is set to limit the movement (degree of freedom) in the longitudinal direction and it is not an object of the study. The next two natural mode shapes at 111.31 Hz and 111.59 Hz are characterized by vibrations in the transverse and vertical directions. They have extremely close values, because both are determined by the radial stiffness in the rubber mounts and the mass of the lower arm. The fourth natural mode at 146.87 Hz is also in the range up to 150 Hz. Its mode shape is characterized by oscillations in the transverse direction.

Table 1 shows the results of the present study, and the results obtained in the work [17]. Comparing the results from the model study of the suspension system and from the experiment, it is noticeable that they have close values. But the results of the frequency study of the individual parts are different. Regardless of this discrepancy, a trend is noticeable in the results – that the modes of the piston rod and the lower control arm are close. Especially in the individual analysis of the parts the frequencies are extremely close – 107 Hz and 111 Hz, respectively. This can lead to overlapping resonances of these elements when the disturbing excitation of the road surface is in that frequency range. Special attention should be paid to this coupling and appropriate stiffness values should be selected for the mounts. Moreover, this frequency range is holding the radial and transverse modes of the radial tyre.

The solution of the individual parts takes insignificant time about a few seconds, while for the harmonic analysis of suspension system, with the same computational resource, the corresponding calculation time is about 12 hours.

Table 1

First natural modes of suspension parts extracted from different study methods

Part of the suspension	**Model study of the individual part**	**Model study of the suspension as a system**	**Experimental study**
Coil spring	50.28 Hz	41 Hz	40 Hz
Piston rod	107 Hz	120 Hz	120 Hz
Lower arm	111 Hz	133 Hz	140 Hz

Conclusions

1. The piston rod and the lower control arm of the MacPherson strut suspension have close frequencies of their first natural modes, with a difference of 3.6% (for the model study of the individual parts).
2. To avoid the overlapping of their natural modes the stiffness of the mounts should be corrected.
3. Coupling of the natural modes of the radial tyre and the suspension elements is possible, which will lead to increasing the levels of structure-borne noise.
4. The suspension must be equipped with radial tyres with suitable vibration behaviour to avoid the coupling of the natural modes.
5. The presented model study in this article of the individual suspension parts can guide the researchers in the right direction, but the accuracy of the results is not satisfactory. The natural modes with high percentage of the mass participation factor would increase the levels of structure-borne noise in the passenger compartment. The smallest difference in results is 10.8% for the first natural frequency of the piston rod. The more complex harmonic study of the suspension as a system leads to more accurate and closer to the experiment results.

Acknowledgements

This research is supported by the Bulgarian Ministry of Education and Science under the National Program “Young Scientist and Postdoctoral Students-2”.

Author contributions:

Conceptualization, Z.G.; methodology, Z.G.; software, Z.G.; validation, Z.G; formal analysis, Z.G.; investigation, Z.G.; data curation, Z.G.; writing – original draft preparation, Z.G.; writing – review and editing, Z.G.; visualization, Z.G.; project administration, Z.G.; funding acquisition, Z.G. The author has read and agreed to the published version of the manuscript.

References

[1] Tsujiuchi N., Koizumi T., Takenaka T., Iwagase T. An Optimization of Rubber Mounting for Vehicle Interior Noise Reduction. Proceedings – SPIE The International Society for Optical Engineering, Issue 4359, 2001, pp. 275–281.
[2] Donders S., Hermans L., Nauwelaerts E., Chojin S., Pluymers B., Desmet W. CAE Technologies for Efficient Vibro-Acoustic Vehicle Design Modification and Optimization. Proceedings of ISMA 2008 /International Conference on Noise and Vibration Engineering/, 2008, pp. 4127-4142.
[3] Kaldas M., ÇalJGkan K., Henze R., Küçükay F. Optimization of Damper Top Mount Characteristics to Improve Vehicle Ride Comfort and Harshness. Shock and Vibration – Hindawi Publishing Corporation, vol. 2014, Article ID 248129, 2014.
[4] Kosaka F., Mizuno H., Inoue T., Takagi K. Road noise sensitivity analysis with respect to suspension geometry. Inter-noise 2014.
[5] Jund A., Gagliardini L. Noise Control Using Input Power at Low and Midfrequencies: Sensitivity to Structural Design Changes. Proceedings of ISMA 2010, pp. 4141-4148.
[6] Petkov P. Studying The Vibro-Insulators Parameters Impact of Automobile Suspension on High Vibrations Transmissions. Mechanics Transport Communications – Academic Journal, vol. 13, article № 1202, 2015, pp. 23-29 (in Bulgarian).
[7] von Mach A. Identification and Updating of Large FE-Models with an Example from Automotive Industry. Proceedings of ISMA 2004, pp. 1965-1979.
[8] Krylov V. Simplified analytical models for prediction of vehicle interior noise. Proceedings of ISMA 2002, Volume V, 2002, pp. 1973-1980.
[9] van der Linden P. J. G., Wyckaert K. Module Target Setting and Substructuring for Vehicle NVH Development. Proceedings of ISMA 2002, pp. 1921-1938.
[10] Leister G. The Role of Tyre Simulation in Chassis Development – Challenge and Opportunity. Proceedings of the 4th International Tyre Colloquium – Tyre Models for Vehicle Dynamics Analysis, 2015, pp. 1-10.
[11] Kido I., Ueyama S. Coupled Vibration Analysis of Tire and Wheel for Road Noise Improvement. SAE Technical Paper 2005-01-2525, 2005.
[12] Kido I., Nakamura A., Sueoka A. Coupled vibration analysis of tire/wheel and suspension. Proceedings of the 10th Asia-Pacific Vibration Conference, 2003, pp. 265-271.
[13] Kido I., Nakamura A., Hayashi T. and Asai M. Suspension Vibration Analysis for Road Noise Using FE Model. Proceedings of the Noise and Vibration Conference, SAE paper No. 99-01-1788, 1999, pp. 1053-1060.
[14] Kindt P., De Coninck F., Sas P., Desmet W. Measurement of the Tire Dynamic Transfer Stiffness at Operational Excitation Levels. Proceedings of ISMA 2010, pp. 3991-4002.
[15] Société de Technologie Michelin. The tyre – Mechanical and acoustic comfort, 2002, 129 p.
[16] Kurowski P. Vibration Analysis with SolidWorks Simulation. SDC Publications, 2016, 338 p.
[17] Georgiev Z., Kunchev L. Vibration analysis of a MacPherson type suspension system. Transport Problems, Vol. 19, Issue 3, 2024, pp. 193-202. DOI: 10.20858/tp.2023.19.3.16