Keywords: structure-borne noise; tyre noise; MacPherson suspension; comfort; vibrations

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VIBRATION ANALYSIS OF A MACPHERSON TYPE SUSPENSION SYSTEM

Summary. The road surface disturbs the pneumatic tyre in the contact patch with different frequencies. The disturbing frequency depends on the speed of the vehicle and the wavelength of the road irregularities. In the current study, the propagation of the vibrations created at the contact patch through the suspension is presented. The oscillations are transmitted to the car body through the rubber mounts and the support joints. When these vibrations reach body panels (roof and floor panels, windows, etc.), they are transmitted to the air and, thus, the structure-borne noise in the passenger compartment is created. The structure-borne noise defines the vehicle comfort for the frequency range up to 400 Hz. The main task of noise, vibration, and harshness engineers is to avoid coupling between tyre and chassis elements' natural modes, which significantly increases the level of structural noise in the passenger compartment. The accelerations of the oscillations were measured for different points of the suspension and the car body. The measurements were taken for two sets of tyres (summer and winter) of the same size and mounted on the same rims. The disturbing frequency was in the range of the tyre's natural modes. The results for the two sets of tyres were compared. The measurements were taken for different air pressure in them. This comparison shows the effect of the tyre type on the structure-borne propagation and determines whether increased noise levels are due to the resonance of the tyre or a suspension element. In addition, a finite element analysis (FEA) was conducted. The model points out the exact element responsible for the high levels of vibrations. Thus, the coupling between the tyre and suspension natural modes can be avoided.

1. INTRODUCTION

A tyre's mechanical characteristics affect all the main operating properties of a car, such as acceleration, braking, fuel economy, directional stability, road holding, and comfort. The pneumatic tyre has had to meet a large number of diverse and often conflicting requirements. This has complicated its construction and production technology [1].

In current research, the tyre is no longer considered as an independent object but is taken as a part of one or another system of the vehicle [2-5]. Therefore, the characteristics of the tyre are used as input data for the design, construction, and calculation of different vehicle systems. As for comfort, the frequency response of the wheel hub is the most important [6]. The frequency response functions (FRFs) can be obtained either by model or experimental study. Models constructed with finite elements give the most accurate results [7]. Large and expensive vibration stands are required to experimentally determine the tyre characteristics [8]. The FRFs of the wheel axis must be coupled with the vibration characteristics of different types of suspension, such as MacPherson [9], multi-link [10], and torsion bar [11]. The main task of noise, vibration, and harshness engineers is to avoid overlapping the natural

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modes of the tyre and the suspension elements [12]. The correct selection leads to a reduction of noise levels in the passenger compartment [13] and a reduction of vibration levels of the body panels [14]. Resonances of various elements of the chassis lead to vehicle design changes [15]. Studies show that finite element models provide the most accurate indications of how structural changes affect structure-borne noise in the passenger compartment [16].

Depending on the speed of the vehicle and the wavelength of the road irregularities, the disturbing oscillations in the tyre contact patch have different frequencies [17]. When the profile of the road surface is known, road experiments can replace the use of expensive vibration stands [18, 19].

In this study, road experiments were conducted with a passenger car on a paved surface. The experiments were carried out with two types of tyres (winter and summer) with different internal pressures. The tyres were the same size and were fitted to the same rims. When driving at a speed of 40 to 70 km/h, the disturbances created in the contact patch had frequencies between 80 and 140 Hz. In this frequency range, the tyre has its first radial mode [20, 21]. For this mode, the tyre's transfer function (the ratio between tyre response and excitation) peaks, which means the load transmitted to the rim centre is maximized. Thus, the tyre's first radial mode has the greatest effect on structure-borne noise. During the experiment, vertical accelerations for various points of the suspension and body were measured. This indicated how the disturbances propagated along the chassis of the vehicle. The comparison between the measured values for the two types of tyres provided information about the extent to which the vibrational behaviour of the tyre affects the structure-borne noise in the vehicle body.

For the investigated MacPherson type suspension, a finite element model is built. The model examines the individual elements of the suspension and determines which element has the greatest effect on the measured high vibration levels. Knowing the output characteristics of the tyre and the suspension design makes it possible to avoid coupling their vibration modes. This, in turn, results in lower structure-borne noise levels in the passenger compartment.

2. MATERIALS AND METHODS

2.1. Choice of road surface

Pavement was chosen as the road surface for conducting the experiment. The distance between the tops of two adjacent pavers is approximately 14 cm; the profile of the road is shown in Fig. 1. This type of surface produces structure-borne noise in the vehicle body. The road surface disturbance frequency f can be calculated from the following formula:

$$f = \frac{zV_a}{3.6} \tag{1}$$

where z is the number of pavers in one linear meter and V_a is the vehicle speed in km/h.

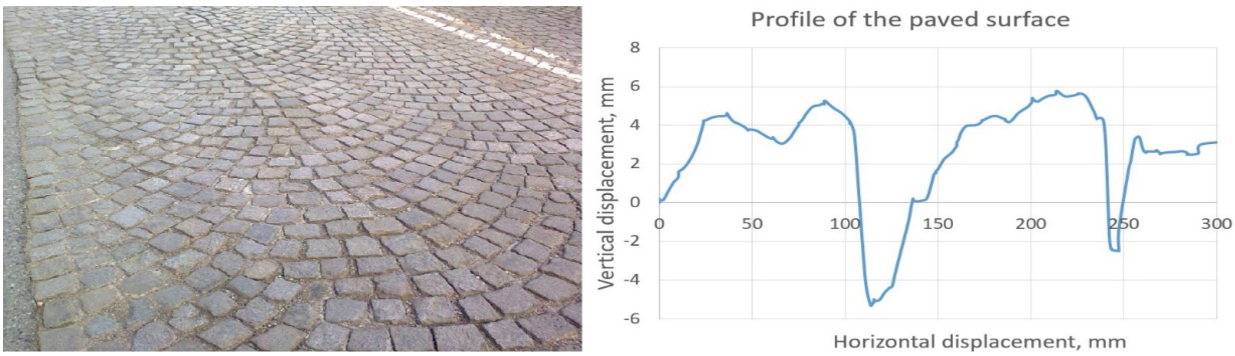

Fig. 1. The pavement surface (left) of "King Boris the Third Boulevard" in Sofia and its profile for two adjacent pavers (right)

2.2. Locations of the measuring sensors

Four points of the vehicle's suspension and body were selected for positioning the accelerometers, as shown in Fig. 2. The first point was from the front right wheel hub carrier (the suspension knuckle). The second point was on the lower control arm, located next to the rear mount connected to the subframe. The third point was located on the vehicle body, right next to the upper support of the front MacPherson strut. The fourth was located on the roof of the car. These measuring points provided sufficient information about the propagation of structure-borne noise created by the road surface to the vehicle body.

Fig. 2. Location of the measuring points: 1 – on the body, 2 – on the roof, 3 – on the control arm, and 4 – on the wheel hub carrier

2.3. Setup of the measuring system

An accelerometer was mounted at each measurement point. A total of four uniaxial accelerometers were used – one capacitive-type PJR and three piezoelectric-type (Bruel & Kjaer). They were used to measure accelerations in the vertical direction. An exception was made for the measurement point on the body, next to the upper support of the MacPherson strut. Due to the complex shape of the local body structure, the mounting of the accelerometer deviated from the vertical axis. The piezoelectric accelerometers were coupled with the appropriate charge amplifiers. The accelerometers were connected to a DAQ system (HBM DQ401), which enabled the oscillograms to be recorded by a computer. The setup of the measuring system is shown in Fig. 3. The error of the measuring equipment was less than 1%. A reference device (Bruel & Kjaer Type 4291) was used to calibrate the piezoelectric-type accelerometers, which produced a vibration acceleration with an amplitude of 1 g and a frequency of 79.6 Hz (500 rad/s).

2.4. Methodology of the experiment

The experiments were carried out on pavement at different vehicle speeds and different air pressures in the tyres. Velocities of 40, 50, 60 and 70 km/h (10 km/h step) were selected. These are typical speeds of road users along the chosen road section. The experiments were carried out for three different tyre pressures: 1.8, 2.1 and 2.4 bar. A minimum of four tests were conducted for the respective speed and pressure. The suspension is coupled with tyres of size 205/55 R16. The tests were conducted for two sets of tyres (summer and winter). All experiments were performed on a dry pavement at an ambient temperature of 25°C.

The root mean square value was used to compare the results of the measured accelerations as follows:

$$\sigma_a = \sqrt{\frac{1}{k}\sum_{i=1}^{k}(M - a_i)^2}\,, \tag{2}$$

where k – the number of elements in the record, a_i – measured accelerations, M – mean value of the accelerations:

$$M = \frac{1}{k}\sum_{i=1}^{k} a_i\,. \tag{3}$$

MATLAB software was used to process the experimental values. The RMS value of the measured accelerations was calculated for each driving speed and each level of tyre pressure.

Fig. 3. Setup of the measuring system: 1 – DAQ system, 2 – charge amplifiers, 3 – piezoelectric accelerometers, 4 – laptop, 5 – 12V battery, 6 – capacitive accelerometer

2.5. Model study

SolidWorks Simulation software was used for the model study [22]. A structural model with finite elements was built for the investigated suspension. The steel chosen for the elements had an elastic modulus E=2.1×1011 N/m^2, Poisson's ratio μ=0.28, and density ρ=7800 kg/m^3. The rubber bushes at the mounting points of the suspension were represented in the model as distributed springs without their own mass (spring connections). The software does not allow the damping of the rubber mounts to be set; therefore, the calculated values of the accelerations in the FRF were inflated. The main purpose of the model study is to point out the source of increased levels of structure-borne noise, not to predict its levels. The frequency of the peaks in the FRF was decisive for this task.

The rubber bushes' locations are shown in Fig. 4a. The mounts between the lower arm and the subframe had a radial stiffness of 1×106 N/m (pos. 1 in Fig. 4a). Normal (axial) stiffness of 1×106 N/m (pos. 2 in Fig. 4a) was set for the ball joint between the control arm and the hub carrier. The upper mount of the MacPherson strut had a normal stiffness of 1.5×106 N/m (pos. 3 in Fig. 4a). The upper support of the strut, which was a part of the vehicle body, was represented as circular sheet metal shaped like a washer. The support was fixed at its outer periphery, which has infinite stiffness when modelled like this. Likewise, two fixed pins were used for supports of the lower control arm (Fig. 4b). A rigid bearing connection was used to connect the wheel hub to the hub carrier. Also, pin connections were used to represent the bolted assemblies in the model.

The shock absorber was modelled as a system consisting of two elements: a rigidly coupled piston and piston rod, as well as an outer cylinder. A spring connection with a total stiffness of 1×107 N/m was set between the two damper elements. This stiffness represents the compressibility of the fluid, which affects high-frequency oscillations [23].

An input parameter is a harmonic vertical disturbance applied at the wheel flange (Harmonic Study). In this case, the disturbance was an acceleration with an amplitude of 1g. FRFs for five points of the suspension were calculated: one point from the lower control arm, one on the top of the piston rod, one from the outer cylinder of the shock absorber, one from the upper support of the coil spring, and one from the coil spring itself (Fig. 4b). The frequency responses of the accelerations for each direction were derived, providing information about the mode shape of the system and the influence of each element on the vibrational behaviour of the suspension.

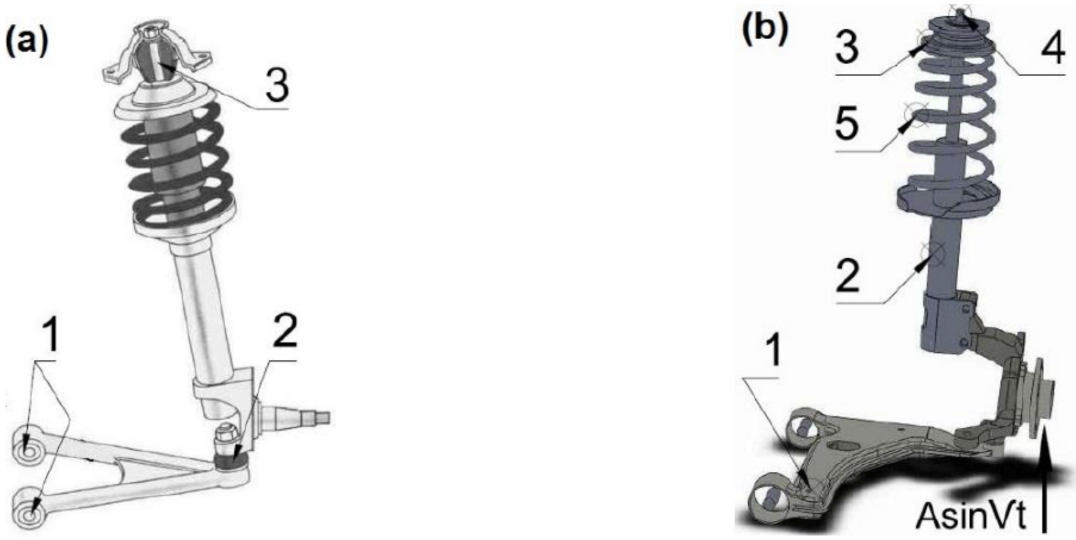

Fig. 4. (a) Location of the rubber mounts and (b) measuring points for the model study

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. A relative damping coefficient of 0.1 was set for the model study.

3. RESULTS FROM THE EXPERIMENTAL STUDY AND DISCUSSION

Figs. 5 to 7 show a comparison between the measured accelerations with summer and winter tyres. In the figures, “ST” stands for the measurements taken with summer tyres and “WT” represents the same measurement for winter tyres. The RMS values are presented as a function of the driving speed and the corresponding disturbance frequency.

The character of the curves for the measuring points on the roof and control arm was the same as that for the hub carrier. This means that the frequency response measured in the wheel axis also gives information about the structure-borne noise in the passenger compartment. At tyre pressures of 2.1 and 2.4 bar, the maximum measured acceleration values occurred at 120 Hz. Also, the accelerations for the control arm and the roof exhibited similar values for both summer and winter tyres. This suggests that the vibration behaviour of the tyre is not the only cause of these peak values.

The measured accelerations in the hub carrier for the different tyres at different internal pressures are presented in Fig. 8. The figure shows that with increasing pressure, the measured values of the accelerations increased, but the character of the curve was preserved. For both types of tyres, at pressures of 2.1 and 2.4 bar, the acceleration peaked at 120 Hz; meanwhile, at a pressure of 1.8 bar, no such peak was observed. This can be explained by the increased enveloping ability and the reduced tyre’s sidewall stiffness at low internal pressure.

Fig. 9 shows the comparison of the measured accelerations on the upper support of the MacPherson strut with different tyre pressures. Due to the very high stiffness of the body in this location, the accelerations were low and no decisive information can be extracted from this graph.

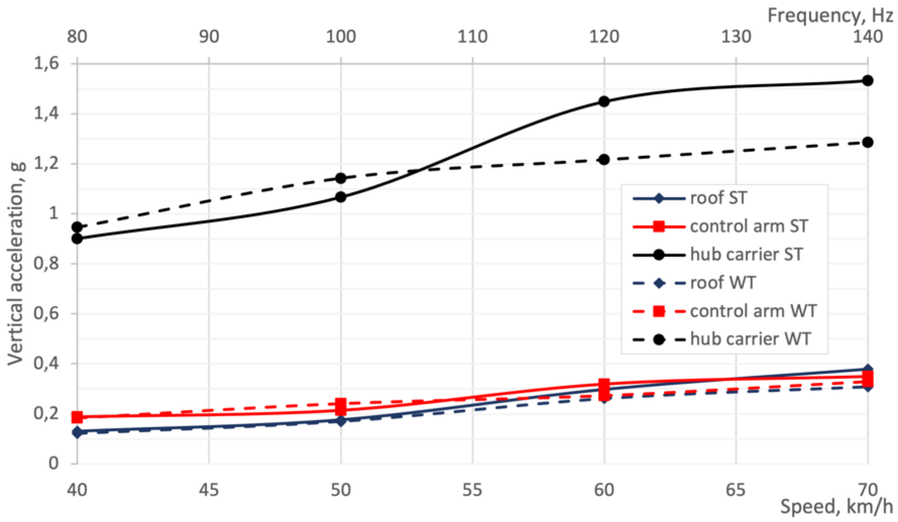

Fig. 5. Measured acceleration with a tyre pressure of 1.8 bar

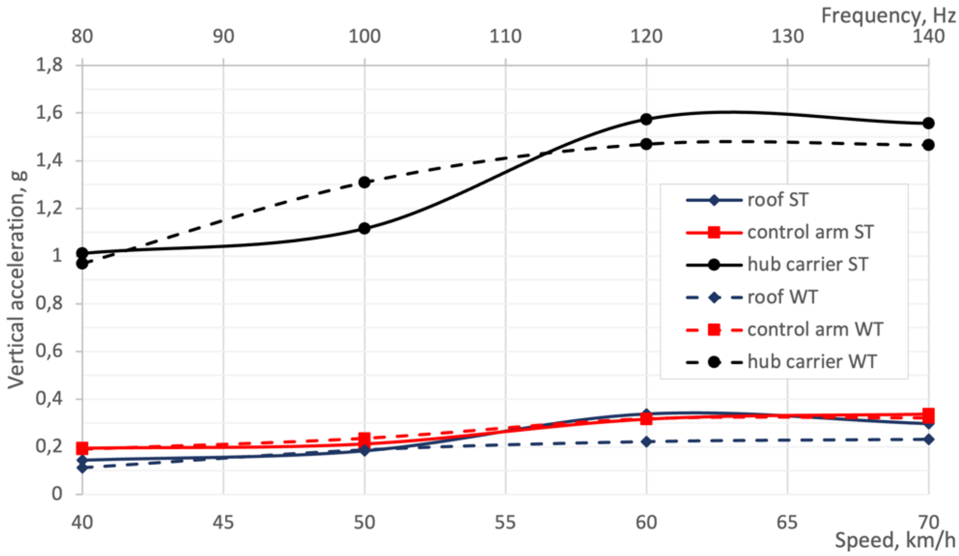

Fig. 6. Measured acceleration with a tyre pressure of 2.1 bar

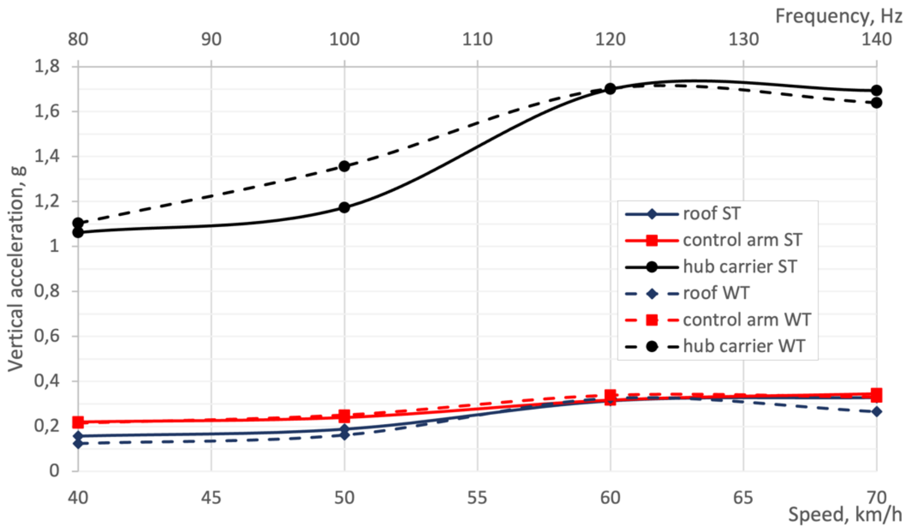

Fig. 7. Measured acceleration with a tyre pressure of 2.4 bar

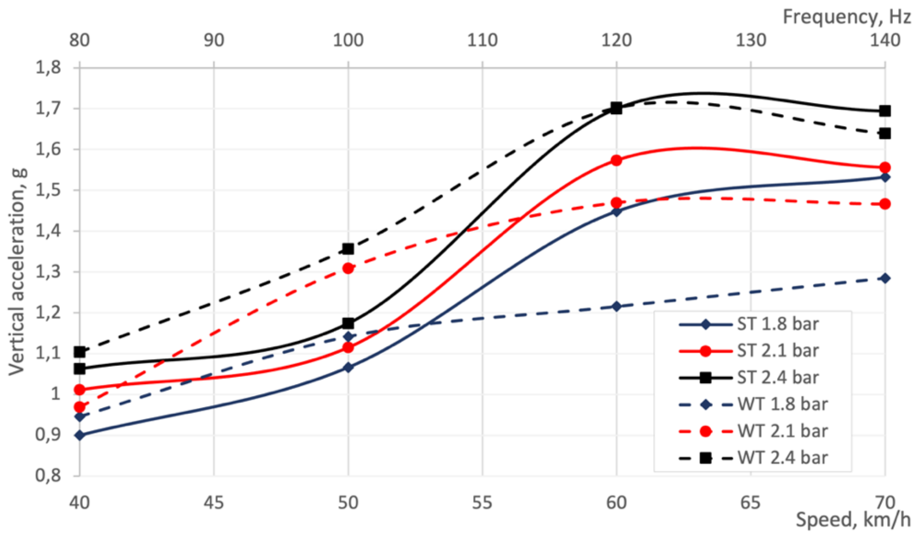

Fig. 8. Measured accelerations at the hub carrier with different tyre pressures

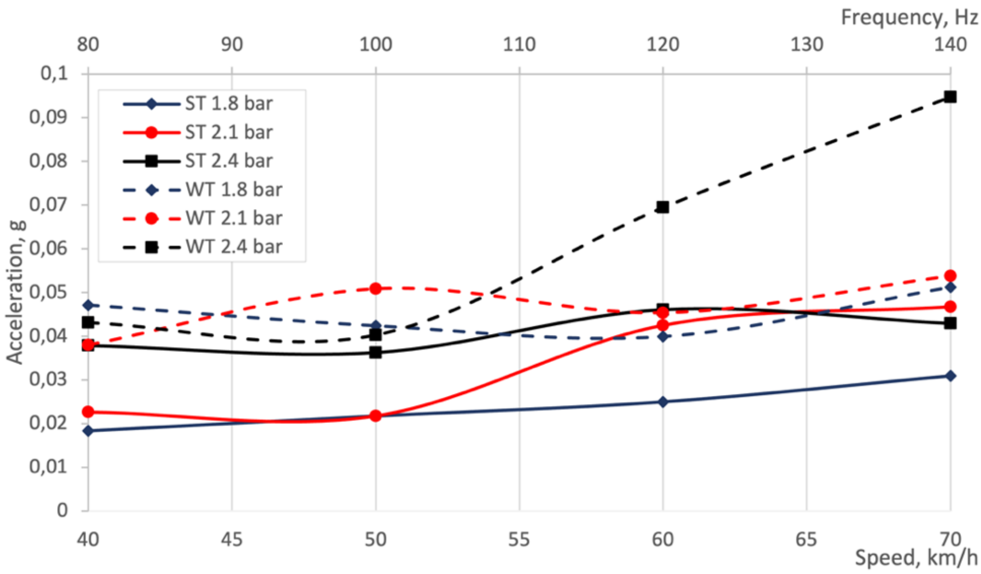

Fig. 9. Measured accelerations on the upper support of the MacPherson strut with different tyre pressures

4. RESULTS FROM THE MODEL STUDY AND DISCUSSION

Fig. 10 shows the resultant accelerations at the measuring points from the model. The results of the model study of the system confirm those of the experiment, which show increased vibration levels at a frequency of 120 Hz. From the vertical accelerations presented in Fig. 11, it can be seen that the coil spring resonates at 41 Hz and that these vibrations are propagated to the other elements even with lower values compared to those on the spring itself. For the control arm, a resonance at a frequency of 133 Hz was observed, characterized by increased vibration levels in the vertical direction. From the calculated accelerations presented in Figs. 12 and 13, it can be seen that the piston rod resonated with a bending mode shape in the transverse direction for a frequency of 120 Hz and in the longitudinal direction for a frequency of 127 Hz. The resonance vibration levels at 120 Hz were higher than those at 127 Hz, and these oscillations even propagated to the lower suspension arm.

5. CONCLUSIONS

This research presented a methodology for analysing the propagation of structure-borne noise generated in the contact patch between tyres and the road. The advantage of the presented methodology lies in the selection of a suitable road surface for creating disturbances over the use of expensive vibration stands. Excitation frequency generated by the longitudinal unevenness of a road depends on the wavelength and the speed of the vehicle. In addition to the experimental study, a model study was also presented, which provided information about the vibrational behaviour of the suspension elements, allowing the main source of the measured vibration peaks to be pointed out. This methodology can be used for different types of suspension and to investigate how the design changes of the elements affect the frequency of the vibration peaks.

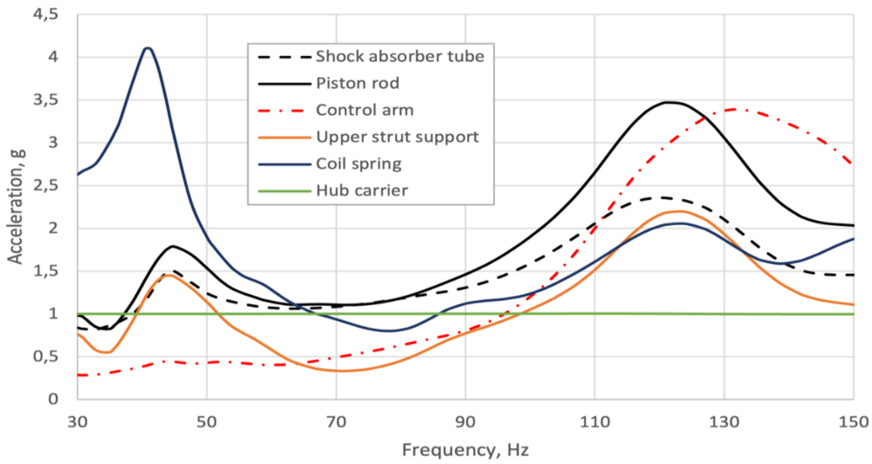

Fig. 10. Resultant accelerations at the measuring points of the model

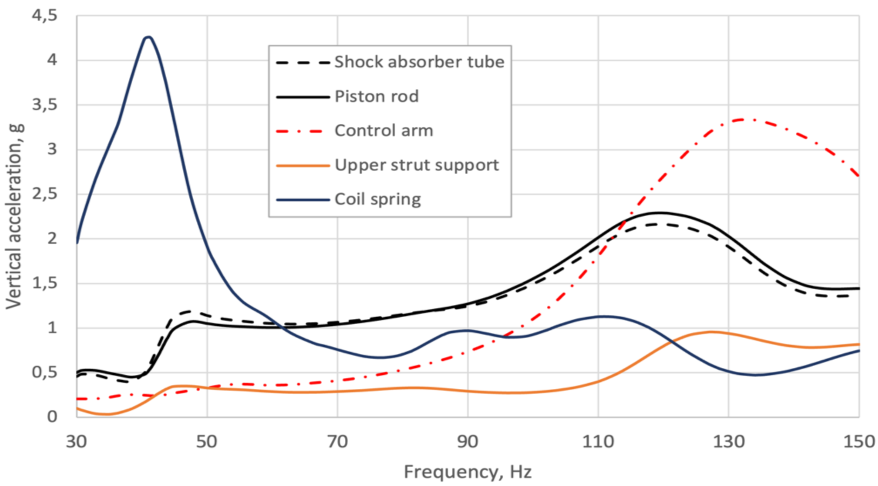

Fig. 11. Vertical accelerations at the measuring points of the model

An experimental study was carried out with both summer and winter tyres with different pressures. For both tyre types, the peak vibration values were obtained at a disturbance frequency of 120 Hz. These vibrations propagate from the hub carrier to the control arm and the roof of the vehicle. When these vibrations reach the body panels, they cause structure-borne noise in the passenger compartment.

The results of the model study confirm those of the experiment. A structural model with finite elements was created for the investigated MacPherson type suspension. This model allowed us to determine that the peak vibration values at 120 Hz are due to piston rod resonance. Even with a purely vertical excitation in the wheel hub, a bending modal form of the piston rod in the longitudinal and transverse directions was observed. These modes had frequencies close to that of the control arm. Mutual overlap can be avoided by changing the stiffness of the control arm's mounts.

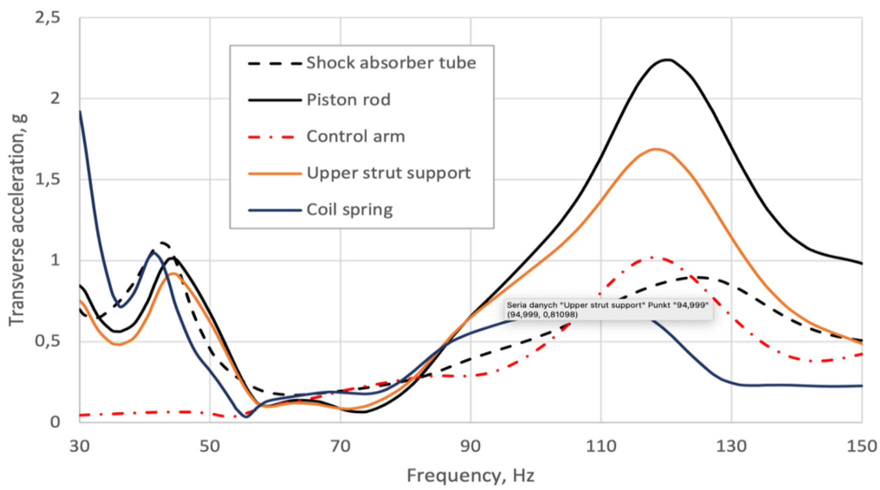

Fig. 12. Transverse accelerations at the measuring points of the model

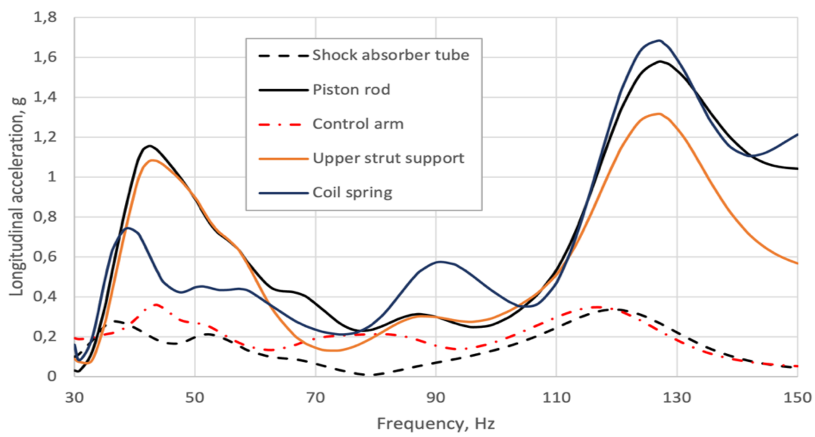

Fig. 13. Longitudinal accelerations at the measuring points of the model

To avoid the membrane effect in the body panels, their construction must be stiffened, or they must be covered with vibration-isolating materials. The action that would lead to the greatest reduction in structure-borne noise would be removing the existing modal coupling between the elements of the suspension and the wheel (tyre and rim). This can be achieved with the available output vibration characteristics of the wheel.

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