Comparative assessment of vertical vibrations of a vehicle on the road and during the EUSAMA test

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Abstract. The work was undertaken to assess the discrepancies between the vertical vibrations of a motor vehicle examined on an EUSAMA suspension tester and the vibrations that occur during rectilinear motion of the same vehicle with a constant speed on an uneven road. The road surface irregularities were of random type and have been described in accordance with recommendations of the International Organization for Standardization (ISO) specified in the document ISO/TC 108/253: “Reporting vertical road surface irregularities. Generalised vertical road inputs to vehicle”. A largely nonlinear “quarter-car” model with two degrees of freedom (2DoF) was used, where the actual elasticity characteristics of vehicle suspension and tyres, sliding friction in the suspension system, and the phenomenon of tyre separation from the road (“bouncing”) were taken into account. The smoothing properties of the tyres were also represented in the model. The description of these properties exactly corresponded to the results of measurements carried out on a test rig. The calculations were done for various values of the coefficient of viscous damping in the shock absorber. In total, 52 simulations of the EUSAMA test (with both nominal and zero sliding friction in the suspension system) and 26 numerical experiments for an average road (of class “C” according to ISO8608) at a vehicle speed of 90 km/h were carried out. The values of the vertical tyre-road contact force, suspension deflection rate, and suspension damping force were analysed in detail. The results of the tests carried out have been chiefly presented in the form of graphs.

1. Introduction
The most popular diagnostic test intended for checking the damping components of a motor vehicle suspension system is the one carried out in compliance with the EUSAMA recommendations [1]. Pursuant to the recommendations, the shock absorber tester should ensure the applying of a frequency-modulated sinusoidal input with an amplitude of 0.003 m. The specificity of the EUSAMA method lies in the fact that the final test result is usually obtained at a vibrator frequency of 75-125 rad/s. In such conditions, the vehicle suspension system still does not operate too intensively and the suspension deflection rate is rather moderate. In consequence, the forces generated by the shock absorber under test are not as strong as they can be when the vehicle moves on an uneven road surface. As an example: on an average road (of class “C” according to ISO 8608 [2]), road roughness height exceeding ±0.03 m and vibration frequencies higher than 500 rad/s should be expected.

The problem mentioned here has also one more aspect. As it has been shown in previous studies [3, 4, 5], one of the factors that may significantly disturb the final result of the EUSAMA test is sliding friction in the motor vehicle suspension system. The values of the resistance forces of this kind are sometimes close to the extreme viscous damping forces generated in shock absorbers being in their nominal condition, subjected to diagnostic tests. No wonder, therefore, that vehicles with strong sliding
friction in their suspension systems achieve good or even very good results in the EUSAMA test, even if the damping elements of the suspension systems are completely worn out. In real road conditions, in turn, adequate dissipation of the vibration energy can exclusively be ensured by the viscous resistance to motion, where the amplitude of the damping force generated is not constant but depends on the shock absorber compression and extension rates.

In consideration of the above, the work was undertaken to assess the discrepancies between vertical vibrations of a motor car examined on an EUSAMA suspension tester and the vibrations that occur during rectilinear motion of the same vehicle with a constant speed on an uneven road.

2. The EUSAMA method

The EUSAMA test [1] consists in kinematic forcing of vertical vibration of the road wheel with an amplitude of 0.003 m and a frequency declining from about 25 Hz to zero. The examination of the degree of wear of suspension system components includes the following stages:

- measurement of the static force exerted by the tyre on the tester’s vibration plate;
- forcing of the system to vibrate with a frequency of about 25 Hz;
- measurement of the minimum value of the normal reaction of the vibrator with the tester driving system having been switched off and the input frequency gradually declining.

Based on the values measured, a diagnostic parameter is determined, which indirectly defines the technical condition of the shock absorber under test. In Polish literature, this parameter is often referred to as “EUSAMA indicator” (WE):

\[
WE = \frac{N_{\text{min}}}{N_{\text{st}}} \cdot 100\%
\]

(1)

where: \(N_{\text{min}}\) is the minimum normal tyre-plate contact force in the core phase of the test, and \(N_{\text{st}}\) is the normal tyre-plate contact force in static conditions.

The way of interpretation of the test results is strictly defined (table 1) and identical criteria should be adopted for various vehicle makes and models.

| Values of the diagnostic parameter | Final result of the diagnostic test |
|-----------------------------------|-----------------------------------|
| > 60 %                            | very good                         |
| 40-60 %                           | good                              |
| 20-40 %                           | acceptable                        |
| 0-20 %                            | unacceptable                      |

3. Excitation from road surface irregularities

At the second stage of the tests, the input applied corresponded to an average road (of class “C” according to ISO 8608 [2]) with undeformable surface where the vertical irregularities had features of a stationary Gaussian random process.

In the ISO standard [2], the road of a specific class has been defined by the power spectral density (PSD) function \(S_d(\Omega)\) of one longitudinal track parallel to the road centreline:

\[
S_d(\Omega) = S_d(\Omega_0) \cdot (\Omega/\Omega_0)^w
\]

(2)

where: \(\Omega\) is angular frequency of the road profile, \(\Omega_0\) is reference angular frequency (in most cases, \(\Omega_0 = 1.0\)), \(S_d(\Omega_0)\) is road surface roughness index, defining the technical condition of the road (good, average, etc.), and \(w\) is road surface waviness index, which shows whether short or long waves predominate in the road profile spectrum.

Roads of different classes differ from each other in the \(S_d(\Omega_0)\) values, while the exponent \(w\) has a constant value of \(w = 2\). The road roughness wavelength \(L\) values have been assumed as varying between 0.1 m and 100 m. This is referred to as “micro-profile” of road surface irregularities [6, 7].
Based on the known function \( S_d(\Omega) \), a discrete realization of the road irregularities was generated, which was then approximated by cubic splines \([8, 9]\). Such a form enabled the use of realizations with high wavelength values, exceeding the maximum micro-profile wavelength \((L=100 \text{ m})\), thanks to the periodical form of the approximating function.

The radial characteristic curve of the pneumatic tyre was described with using the “point contact” model\([8,9,10,11]\). Therefore, the road roughness wavelength that was taken into account had to be bounded from below for unnatural overestimation of the damping forces in the tyre model to be avoided. Here, this difficulty was overcome by adopting a “fixed footprint tyre model” to simulate the smoothing properties of vehicle tyres \([8,10,11,12]\) by averaging the road profile height over the tyre-road contact patch with filtering the road roughness spectra. In this specific case, a filter with the following absolute value of its transmittance was used:

\[
\left| H_{op}(\Omega) \right| = \left| \frac{\sin(\Omega \cdot l_{op})}{\Omega \cdot l_{op}} \right| \tag{3}
\]

where: \( l_{op} \) is a half of the static length of the tyre-road contact patch.

In formal terms, the said filtration that simulated the smoothing properties of vehicle tyres and led to the use of the PSD function \( S_d(\Omega) \) of road irregularities was expressed by equation (4):

\[
S_{df}(\Omega) = \left| H_{op}(\Omega) \right|^2 \cdot S_d(\Omega) \tag{4}
\]

The applying of \( S_{df}(\Omega) \) as an input made it possible to use the “point tyre-road contact” model \([8,10,11,12]\) and, in consequence, to generate an appropriate realization of the input. Finally, for the needs of accomplishing the objective of this work, a time history of the input displacement was used that corresponded to a test drive on a 102.3 m long section of an average road (of class “C”) with a vehicle speed of \( V=90 \text{ km/h} \), for which \( 2 \cdot l_{op} = 0.185 \text{ m} \).

4. Sliding friction in suspension systems of present-day passenger cars

To confirm the presence of considerable sliding friction forces in suspension systems of present-day passenger cars, a decision was made to carry out preliminary exploration tests and to measure the resistance forces of this type in 13 vehicles (figure 1).

![Figure 1. Friction resistance forces in suspension systems of present-day passenger cars](image)

The cars chosen were in good or very good technical condition and were taken from several different market segments. They also differed from each other in date of manufacture, mileage, and suspension
system design. The friction resistance forces were analysed on the grounds of elasticity characteristics of individual suspension systems.

Based on an analysis of results of more than 500 measurements, the following was ascertained:

- The friction in individual suspension systems depended on their instantaneous deflection; for systems with leaf springs, it also depended on the actual spring operation range.
- The tyre characteristics related to energy dissipation did not considerably affect the evaluation of the friction forces in the suspension systems, even when a simplified test method (based on measurements of the normal tyre-road reaction force) was used.
- The strongest friction forces (about 200 N – figure 1) occurred in the front suspension system with guiding columns (car VII) and in the rear leaf spring suspension systems (cars III and XIII).
- The lowest values of the friction resistance forces (below 50 N – figure 1) were recorded for rear suspension systems with trailing arms (cars V, VII, and VIII).

5. Simulation model

In pursuance of the objective of this work, only numerical calculations were done, with employing a verified version of the largely nonlinear “quarter-car” model \([3,13,14]\) whose structure was adapted to the special simulation tests (figure 2). The analysis covered the vertical movements of the front left wheel and of the vehicle body part situated above the said wheel, induced by the kinematic excitation that took place during rectilinear vehicle motion on an uneven road with a constant speed and, for comparison, during a typical EUSAMA test. As the test specimen, a pick-up passenger car with an independent front suspension system was chosen.

![Nonlinear “quarter-car” model with dry friction in the suspension system:](image)

Figure 2. Nonlinear “quarter-car” model with dry friction in the suspension system:

- a) general structure; b) system of forces;
- \(F_{s1}\) – elasticity force in the suspension system; \(F_{s2}\) – elasticity force in the tyre; \(F_{t1}\) – viscous damping force in the suspension system; \(F_{t2}\) – damping force in the tyre; \(F_{ts}\) – sliding friction force in the suspension system; \(m_1\) – sprung mass; \(m_{1g}\) – weight of the sprung parts; \(m_2\) – unsprung mass; \(m_{2g}\) – weight of the unsprung parts; \(q\) – input displacement; \(z_1\) – coordinate of the centre of the sprung mass; \(z_2\) – coordinate of the centre of the unsprung mass; \(\zeta_1\) – vertical axis of the system attached to the sprung mass; \(\zeta_2\) – vertical axis of the system attached to the unsprung mass.
The “quarter-car” model with two degrees of freedom, used for the analyses, made it possible to take into account actual elasticity characteristics of the suspension system and tyres, sliding friction in the system, and tyre separation from the road (“bouncing”). The smoothing tyre properties were also represented in the model. The description of these properties exactly corresponded to the results of measurements carried out on a test rig (table 2) at the Automotive Industry Institute (PIMOT).

The equations of motion for the non-linear “quarter-car” model were derived in accordance with the principle of dynamic force analysis:

\[
\begin{align*}
\ddot{z}_1 &= \frac{F_{s1} + F_{ts1} + F_t}{m_1} - g \\
\ddot{z}_2 &= \frac{-F_{s1} - F_{ts1} - F_t + F_{s2} + F_{ts2}}{m_2} - g
\end{align*}
\]

where: \(\ddot{z}_1\) is acceleration of the sprung mass, \(\ddot{z}_2\) is acceleration of the unsprung mass, and \(g\) is acceleration of gravity.

The elasticity force in the suspension system was approximated by two fifth-degree polynomials:

\[
F_{s1} = \begin{cases} 
A_1s \cdot u_1^5 + B_1s \cdot u_1^4 + C_1s \cdot u_1^3 + D_1s \cdot u_1^2 + E_1s \cdot u_1, & \text{for } u_1 \leq u_{1gr} \\
A_2s \cdot u_1^5 + B_2s \cdot u_1^4 + C_2s \cdot u_1^3 + D_2s \cdot u_1^2 + E_2s \cdot u_1 + F_2s, & \text{for } u_1 > u_{1gr}
\end{cases}
\]

where: \(u_1\) is suspension deflection, as expressed in (7), \(A_1s-\) \(E_1s\) and \(A_2s-\) \(F_2s\) are coefficients of the polynomials that describe the elasticity force in the suspension system at small and big deflections, and \(u_{1gr}\) is threshold between the ranges of applicability of the different representations of the elasticity force in the suspension system:

\[
u_1 = z_2 - z_1 + \omega_{m_0}
\]

where: \(\omega_{m_0}\) is difference between coordinates \(z_1\) and \(z_2\) at which \(F_{s1} = 0\).

The sliding friction force in the suspension system was represented as follows:

\[
F_{ts1} = \begin{cases} 
Ats_1 \cdot \text{sgn} \dot{u}_1, & \text{for } |\dot{u}_1| > \dot{u}_{1gr} \\
Ats_1 \cdot \frac{\dot{u}_1}{\dot{u}_{1gr}}, & \text{for } |\dot{u}_1| \leq \dot{u}_{1gr}
\end{cases}
\]

where: \(Ats_1\) is amplitude of the sliding friction force in the suspension system, \(\dot{u}_1\) is suspension deflection rate, defined by (9), and \(\dot{u}_{1gr}\) is threshold of the suspension deflection rate:

\[
\dot{u}_1 = \ddot{z}_2 - \ddot{z}_1
\]

The viscous damping force in the suspension system was described by equation (10):

\[
F_t = c_1 \cdot \dot{u}_1
\]

where: \(c_1\) is coefficient of viscous damping in the shock absorber, defined as follows:

\[
c_1 = \gamma \cdot c_{1kr}
\]

where: \(\gamma\) is relative damping coefficient, and \(c_{1kr}\) is critical damping coefficient.

The elasticity force in the pneumatic tyre was approximated by a third-degree polynomial:

\[
F_{s2} = As_2 \cdot u_2^3 + Bs_2 \cdot u_2^2 + Cs_2 \cdot u_2
\]

where: \(As_2-\) \(Cs_2\) are coefficients of the polynomial that describes the elasticity force in the tyre and \(u_2\), represented by (13), is radial deflection of the tyre.
$$u_2 = \begin{cases} q - z_2 + R, & \text{for } q - z_2 + R \geq 0 \\ 0, & \text{for } q - z_2 + R < 0 \end{cases}$$

(13)

where: \(R\) is free tyre radius.

The damping force in the tyre model was a linear function of the rate of radial deflection of the tyre, as it has been expressed in (14):

$$F_{t_2} = c_2 \cdot \dot{u}_2$$

(14)

where: \(c_2\) is damping coefficient, and \(\dot{u}_2\) (according to (15)) is rate of radial deflection of the tyre:

$$\dot{u}_2 = \begin{cases} q - \dot{z}_2, & \text{for } q - z_2 + R \geq 0 \\ 0, & \text{for } q - z_2 + R < 0 \end{cases}$$

(15)

The solutions of the equations of motion (5) were obtained in the time domain with using approximation techniques, by numerical integration. The results presented herein have been obtained with using an authorial program built in the Matlab Simulink environment [3].

6. Calculation data: parameters of the model and of the test conditions

The calculation data introduced into the model corresponded to a front quarter of the Isuzu D-max motor vehicle. For the \(c_{kr}\) parameter, a value of 9 062 N\(\cdot\)s/m was adopted, which was determined within works [12, 13]; the relative damping coefficient \(\gamma\) was treated in the calculations as a variable. Every time, 26 values of this parameter, varying from 0 to 0.5 in steps of \(\Delta\gamma = 0.02\), were taken for the calculations.

The length of the tyre-road contact patch in static conditions was \(2l_{op} = 0.185\) m.

The numerical experiment related to the EUSAMA test was carried out with the nominal and zero values of the sliding friction resistance force in the suspension system (52 single tests in total). Apart from that, 26 simulations of vehicle motion in real conditions with a speed of 90 km/h on an average road (of class “C” according to ISO 8608 [2]) were carried out. All the model parameters and basic information about the test conditions have been brought together in table 2 and figure 3.

![Figure 3](image-url)

**Figure 3.** Time histories of the of the input displacement in the third (core) phase of a typical EUSAMA test and during a test drive on an average road (of class “C”) with a speed of 90 km/h
Table 2. Summary of model parameters and test conditions

| Description | Notation | Unit   | Value       |
|-------------|----------|--------|-------------|
| Sprung mass | \(m_1\)  | kg     | 578         |
| Unsprung mass | \(m_2\) | kg     | 69.5        |
| A1s1        | \(N/m^3\) |        | 5.526E+10   |
| B1s1        | \(N/m^4\) |        | -9.942E+09  |
| C1s1        | \(N/m^3\) |        | 6.844E+08   |
| D1s1        | \(N/m^2\) |        | -2.246E+07  |
| E1s1        | \(N/m\)   |        | 3.933E+05   |
| A2s1        | \(N/m^3\) |        | 1.344E+09   |
| B2s1        | \(N/m^4\) |        | -6.493E+08  |
| C2s1        | \(N/m^3\) |        | 1.224E+08   |
| D2s1        | \(N/m^2\) |        | -1.123E+07  |
| E2s1        | \(N/m\)   |        | 5.398E+05   |
| F2s1        | \(N\)     |        | -6.427E+03  |
| Threshold between different suspension elasticity descriptions | \(u_{1gr}\) | m | 0.054 |
| Difference between coordinates \(z_1\) and \(z_2\) at which \(F_{s1} = 0\) | \(\Delta m_0\) | m | 0.432 |
| Critical damping coefficient | \(c_{1kr}\) | \(N/s/m\) | 9.062 |
| Minimum value of the relative damping coefficient | \(\gamma_{\min}\) | - | 0 |
| Maximum value of the relative damping coefficient | \(\gamma_{\max}\) | - | 0.5 |
| Amplitude of the sliding friction force in the suspension system | \(A_{ts1}\) | N | 158 |
| Threshold of the suspension deflection rate (friction model) | \(u_{1gr}\) | m/s | 0.005 |
| Coefficients of the polynomial that describes the elasticity force in the tyre | \(A_{2s2}\) | \(N/m^3\) | 2.680E+04 |
| | \(B_{2s2}\) | \(N/m^2\) | 1.640E+06 |
| | \(C_{2s2}\) | \(N/m\) | 1.357E+05 |
| Free tyre radius | \(R\) | m | 0.382 |
| Coefficient of damping in the tyre | \(c_2\) | \(N/s/m\) | 150 |
| Length of the tyre-road contact patch in static conditions | \(2 \cdot l_{op}\) | m | 0.185 |
| Minimum road roughness wavelength value | \(L_{\min}\) | m | 0.1 |
| Maximum road roughness wavelength value | \(L_{\max}\) | m | 100.0 |
| Roughness index of the average road (of class “C”) | \(S_0(\Omega_0)\) | \(m^3/\text{rad}\) | 0.000016 |
| Reference angular frequency of the road profile | \(\Omega_0\) | l/m | 1.0 |
| Waviness index of the average road (of class “C”) | \(w\) | - | 2.0 |
| Minimum Hertz (radian) frequency of the analysed vibrations in the road test | \(f_{\min}\) | Hz | 0 |
| | \(\Omega_{\min}\) | (rad/s) | (0) |
| Maximum Hertz (radian) frequency of the analysed vibrations in the road test | \(f_{\max}\) | Hz | 80 |
| | \(\Omega_{\max}\) | (rad/s) | (503) |

7. Calculation results
During the simulation of the EUSAMA test for the Isuzu D-max vehicle in its nominal condition \((\gamma = 0.31)\), the minimum and maximum suspension deflection rates were about \(-0.2\) m/s and about \(+0.2\) m/s, respectively (figure 4). When the viscous damping was then reduced, nonlinear growth was observed in the absolute values of these extremums, with conspicuous symmetry of the \(\dot{u}_{1\min}\) and \(\dot{u}_{1\max}\) curves in relation to the zero level. For \(\gamma = 0.1\) (deemed as corresponding to the maximum wear of shock absorbers), the amplitude of changes in the suspension deflection rates did not exceed \(0.5\) m/s; only for the system with no viscous damping, this amplitude grew to about \(1.4\) m/s.

When the EUSAMA test simulation results were compared with those obtained for the vehicle moving on an uneven road, good qualitative consistency was noticed between the curves representing the extreme suspension deflection rates (figure 4). For the test drive on an average road (of class “C”) with a speed of 90 km/h, however, the peak-to-valley values of this quantity were always markedly
higher (from about 0.6 m/s at strong damping to about 1.4 m/s at $\gamma = 0$). In this case, the suspension deflection rate for the vehicle in its nominal condition varied within limits of $\pm 0.5$ m/s; when the viscous damping was removed from the system, this rate varied between $-2.1$ m/s and $+1.95$ m/s.

Much bigger differences in the functioning of the suspension system during the EUSAMA test and the test drive on an average road ("C") with a speed of 90 km/h could be seen between the curves representing the extreme viscous damping forces (figure 5). In the diagnostic test simulations, the curves showing $|F_{t_{\text{1min}}}|$ and $F_{t_{\text{1max}}}$ as functions of $\gamma$ were markedly degressive. The distinct saturation began at as low a relative damping coefficient as $\gamma = 0.1$. In such conditions, the amplitude of force $F_{t_{1}}$ was about 400 N, as against a value higher by only about 100 N at nominal vehicle specifications ($\gamma = 0.31$). Thus, the sliding friction resistance forces made at least 30% of the viscous damping force in the cases under interest and considerably affected the process of energy dissipation in the system, distorting the final result of the EUSAMA test even by more than 20 percentage points (figure 6).

In the test drives on an average road (of class “C”) with a speed of 90 km/h, the peak-to-valley values of force $F_{t_{1}}$ marked increased (linearly, in rough terms) with growing relative suspension damping (figure 5). For $\gamma = 0.1$, the amplitude of force $F_{t_{1}}$ was about 750 N; for the nominal vehicle specifications, this value was $F_{t_{\text{1max}}} \approx 1 465$ N. In this situation, the sliding friction resistance forces already made as little as about 10% of the viscous damping force in the suspension system and did not significantly influence the course of dissipation of the energy of vertical vibrations of the vehicle.

Similar conclusions could be drawn from a comparison between the characteristic curves of the minimum vertical tyre-road contact forces, obtained in two test series (figure 7). The shape of the curve representing $N_{\text{min}}$ as a function of relative suspension damping, obtained from 26 simulations of the EUSAMA test, was markedly degressive. The minimum values of the said force begun to decrease to a considerable degree only when the value of the relative damping coefficient $\gamma$ was reduced in successive tests from 0.1 to 0. Outside of this range, the dependence of $N_{\text{min}}$ on the relative damping in the suspension system was weak. At $\gamma = 0$, the minimum value of the force exerted by the wheel on the tester’s vibration plate was about 2 000 N, while it already exceeded 4 600 N at $\gamma = 0.1$. On the other hand, the result obtained for the vehicle in its nominal condition was only slightly higher (5 300 N).

Figure 4. Comparison of extreme suspension deflection rates in the EUSAMA test and on an average road (“C”) at 90 km/h for various relative damping coefficient values

Figure 5. Comparison of extreme viscous damping forces in the suspension system in the EUSAMA test and on an average road (“C”) at 90 km/h for various relative damping coefficient values as against the sliding damping forces
The curve under consideration differed not only in quantitative but also in qualitative terms from the one plotted for the vehicle driven on an average road (“C”) with a speed of 90 km/h (figure 7). Based on the latter curve, it was found that even the tyre separation from the road happened to occur when there was no viscous damping in the suspension system. The highest value of the minimum vertical tyre-road contact force ($N_{\text{min}} \approx 3900$ N) was obtained for $\gamma=0.24$. For the vehicle in its nominal condition, however, the value of this force was 3675 N. The biggest differences (even of about 2000 N) between the curves compared with each other were observed at very low and very high levels of the viscous damping in the suspension system.

8. Conclusions
The use of a time history of the normal force between the tyre and the vibration plate of a diagnostic shock absorber tester should be considered a good point of the EUSAMA method, as this parameter is very strongly associated with the safety of motor vehicle motion. A problem arises, however, from the fact that the suspension deflection rates are rather moderate in this case, even of the order of a small fraction of those likely to occur on a typical road of the average class (figure 4). In such a situation, the forces generated by the shock absorber under test are not as strong as they can be when the vehicle moves in real conditions (figure 5). Thus, the sliding friction resistance forces that develop during the EUSAMA test can make even more than 30% of the viscous damping force (figure 5), erroneously raising the final test result by over 20 percentage points (figure 6) or even more [3,4,5]. On the other hand, the sliding friction should not be expected to have a considerable impact on the process of dissipation of the energy of vertical vibrations of a vehicle with nominal parameters when moving with a speed of 90 km/h on an average road. In such conditions, the resistance forces of this type may already make as little as about 10% of the viscous damping force in the suspension system (figure 5). The discrepancies between the vertical vibrations of a motor car examined on an EUSAMA suspension tester and the vibrations that occur in real drive conditions are also indicated by the differences in the minimum vertical forces in the tyre-road contact area. In the tests reported here, these differences even reached a level of 2000 N (figure 7).

In author’s opinion, the research results might even be more interesting if passenger cars smaller and, consequently, lighter than the Isuzu D-Max were subjected to the analysis. Therefore, similar research
works should be carried out for other vehicles, possibly with using models and, afterwards, experimental methods of more sophisticated type.

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