Simulation assessment of the impact of inertia of the vibration plate of a diagnostic suspension tester on results of the EUSAMA test of shock absorbers mounted in a vehicle

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Abstract. As mentioned in the title, the paper presents an assessment of the impact of inertia of the vibration plate of a diagnostic suspension tester on results of the EUSAMA test of the condition of shock absorbers mounted in a vehicle. In the method adopted by EUSAMA, the normal force between the vehicle tyre and the vibration plate is measured during the tests. The percentage ratio of the minimum value of this force to its static value, within the excitation frequency range of 0-25 Hz, is the “EUSAMA indicator”. It is taken as a basis for diagnostic assessment of the condition of the shock absorbers mounted in a vehicle. In the measuring systems of the testers, the force between the tyre and the vibration plate is measured indirectly at the point where the excitation is applied to the plate on which the wheel of the suspension system under test is placed. This means that the force measured includes not only the normal tyre-plate contact force but also the force of inertia of the plate. The authors made a simulation analysis of the impact of the inertia of suspension tester’s vibration plate on the EUSAMA test results. The simulations were carried out both in the frequency and time domain. In the latter case, the EUSAMA test conditions, i.e. the excitation frequency varying in time, were replicated. The results reveal that the vibration plate inertia may considerably distort results of the EUSAMA test. For this inertia to be taken into account, the tester must be enabled to measure the vertical acceleration of the vibration plate.

1. Introduction with a review of the literature dealing with the EUSAMA method

Two groups of methods are used to test the condition of shock absorbers mounted in a vehicle: “free vibration methods” and “forced vibration methods” [1, 4, 8, 9, 11, 13]. In the former group, a record of vehicle body vibration (or, more precisely, the number of half-periods of the vibration) caused by initial test conditions is assessed. In some variants of such an approach, the vertical force exerted by the tyre on the ground is measured. In the other group of methods, the vehicle wheel is forced to vibrate in vertical direction with a frequency of 16-25 Hz and then this frequency is gradually reduced until the vibration fades out to “sweep” the frequency range that covers the natural frequencies of the “sprung mass” and “unsprung mass” supported by spring elements of the suspension system and flexible tyres. Thus, the known fact of high sensitivity of system vibration to damping in the resonance zone is made use of. This group includes the variant proposed by BOGE, a company, where the initial excitation frequency is about 16 Hz and the peak-to-peak values of vertical displacements of the vibration plate are assessed. A newer version of the forced vibration methods is the one introduced by EUSAMA (EUropean Shock Absorber Manufacturers Association) [1, 2, 3, 4, 5, 6, 7, 8, 9, 11, 13, 15,
In this case, the initial excitation frequency is about 25 Hz and the assessment of the shock absorber condition is based on the force exerted by the tyre on the ground (i.e. vibration plate of the tester). An expanded version of the EUSAMA method is the one proposed by the Hunter company, where the phase shift angle between the excitation (vertical motion of the vibration plate) and the force exerted by the tyre on the plate is measured [12, 14].

In this study, the authors deal with the EUSAMA method [1, 5, 7, 8, 9, 11, 13, 15, 16, 17] and the normal tyre-plate contact force is measured. The percentage ratio of the minimum value \( N_{\text{min}} \) of this force measured in the excitation frequency range 0-25 Hz to the static value \( N_{\text{st}} \) of this force is the “EUSAMA indicator” denoted here by \( \text{WE} \).

\[
\text{WE} = \left( \frac{N_{\text{min}}}{N_{\text{st}}} \right) \cdot 100\%
\]

Figure 1 shows a graphic interpretation of determining the WE indicator. The WE value is compared with a table of EUSAMA requirements for an opinion on the shock absorbers’ condition to be given. Its formula is general enough that there is no need to know any reference or limit values for individual vehicle makes and models. Only general criteria of assessment of the shock absorbers’ condition are adopted: “very good” if \( \text{WE} > 60\% \); “good” if \( \text{WE} \) is within 40-60 %; “to be checked on a test stand after removal from the vehicle” if \( \text{WE} \) is within 20-40 %; “to be replaced” if \( \text{WE} < 20\% \).

![Figure 1. Graphic interpretation of determining the EUSAMA WE indicator [7, 9]](image)

In the measuring systems of the testers, the force between the tyre and the vibration plate is measured indirectly at the point where the excitation is applied to the plate on which the wheel of the suspension system under test is placed (see Figure 2). This means that the force measured is not the exact normal tyre-plate contact force but it also includes the force of inertia of the plate.

2. Objective
The work was undertaken to assess by simulation the influence of inertia of the vibration plate of a diagnostic suspension tester on results of the EUSAMA test.

3. Description of the linear “quarter-car” model where the inertia of the vibration plate of a diagnostic suspension tester is additionally taken into account
Figure 2 shows a linear “quarter-car” model placed on a diagnostic suspension tester. It consists of “sprung mass” \( m_1 \), “unsprung mass” \( m_2 \), and mass \( m_3 \) representing the vibration plate. Coefficients \( k_1 \) and \( k_3 \) represent the suspension stiffness and the tyre radial stiffness, respectively; \( c_1 \) and \( c_2 \) are suspension system damping and tyre radial damping coefficients, respectively. Symbol \( \zeta \) represents the time-dependent kinematic excitation from the vibration plate, measured along axis \( O\zeta \). The vertical displacements of individual model masses from their static equilibrium positions for the vibration
plate being at rest are denoted by \( z_1 \) and \( z_2 \), respectively. The vehicle (and its model) does not move in the horizontal plane; hence, the velocity of this movement is \( v=0 \). The dynamic component \( F_{dz} \) of the vertical tyre-plate contact force and the dynamic component \( F_{md} \) of the input force applied to the vibration plate and measured by tester’s instrumentation have also been shown.

\[ F_{dz} \] – dynamic component of the vertical tyre-plate contact force, actually applied to the tyre

\[ F_{md} \] – dynamic component of the input force applied to the vibration plate

**Figure 2.** Linear “quarter-car” model placed on a diagnostic suspension tester

The tyre-plate contact force \( F_{op} \) is the sum of static load \( N_{st} \) and dynamic component of the vertical tyre-plate contact force \( F_{dz} \).

\[ F_{op} = N_{st} + F_{dz} \]  \hspace{1cm} (2)

where:

\[ N_{st} = (m_1 + m_2) \cdot g \]  \hspace{1cm} (3)

and \( g \) – acceleration of gravity. Component \( m_3 \cdot g \) is reduced during the reset of the measuring system.

\( F_{dz} \) is the sum of dynamic elasticity force \( F_{ds} \) (measured in relation to the static equilibrium state, i.e. for radial tyre deformations in relation to the static deformation) and viscous damping force \( F_{two} \) in the tyre (see Figure 2).

\[ F_{dz} = F_{ds} + F_{two} \]  \hspace{1cm} (4)

The tester instrumentation measures force \( F_{opm} \), which consists of total (i.e. static load inclusive) tyre-plate contact force \( F_{op} \) and plate inertia force \( F_{bp} \) with a minus sign. The plate inertia force is:

\[ F_{bp} = -m_3 \cdot d^2 \zeta(t)/dt^2 \]  \hspace{1cm} (5)

\[ F_{opm} = F_{op} - F_{bp} = F_{op} + m_3 \cdot d^2 \zeta(t)/dt^2 = F_{dz} + N_{st} + m_3 \cdot d^2 \zeta(t)/dt^2 = F_{md} + N_{st} \]  \hspace{1cm} (6)

where:

\[ F_{md} = F_{dz} + m_3 \cdot d^2 \zeta(t)/dt^2 \]  \hspace{1cm} (7)

is a result of measurement of the dynamic component (i.e. exclusive of static load \( N_{st} \)) of the force in the tyre \( F_{dz} \), i.e. a result of measurement of the force that excites vibration of the system under test. The plate inertia force \( F_{bp} \) distorts the result of measurement of force \( F_{dz} \) (i.e. its value) and, in consequence, the value of force \( F_{op} \). It also introduces phase shift of \( F_{md} \) in relation to \( F_{dz} \).

The equations of motion for the linear model (Figure 2) have been derived from the principles of dynamic force analysis. Their matrix form has been shown in relation (8).
where the symbols of the matrices of inertia $M$, viscous damping $C$, stiffness $K$, excitation influences transmitted by the damping in, and the radial stiffness of, the tyre ($C_\zeta$ and $K_\zeta$, respectively) have been indicated. The vectors of generalized coordinates (displacements), velocities, and accelerations have been denoted by $\mathbf{q}, \dot{\mathbf{q}}, \ddot{\mathbf{q}}$, respectively. This concise notation has been adopted in relation (9).

For equation (9), the Laplace transform was formulated, with zero initial conditions. After transformations, equation (10) was obtained, where the domain $s=r+i\omega$ has a real part $r$ and an imaginary part $\omega$, while $i^2=-1$ ($\omega$ is the radian frequency [rad/s]):

$$
\mathbf{q}(s) = H_q(s) \cdot \zeta(s)
$$

The operational transmittances (transfer functions) for displacements $H_q(s)$, velocities, and accelerations have been represented in the form of equations (11), (12), and (13), respectively.

$$
H_q(s) = \begin{bmatrix} H_{q_1}(s) \\ H_{q_2}(s) \end{bmatrix} = \begin{bmatrix} \dot{\mathbf{q}}(s) \\ \ddot{\mathbf{q}}(s) \end{bmatrix} = \begin{bmatrix} \mathbf{q}(s) \\ \zeta(s) \end{bmatrix} = s \cdot H_q(s)
$$

$$
H_{\dot{q}}(s) = \begin{bmatrix} H_{\dot{q}_1}(s) \\ H_{\dot{q}_2}(s) \end{bmatrix} = \begin{bmatrix} \dot{\mathbf{q}}(s) \\ \ddot{\mathbf{q}}(s) \end{bmatrix} = s^2 \cdot H_q(s)
$$

The Laplace transform of the dynamic component of the vertical tyre-plate contact force is expressed by formula (14):

$$
F_{dz}(s) = c_2 \cdot [\zeta(s) - \dot{\zeta}(s)] + k_2 \cdot [\zeta(s) - z_2(s)]
$$

Based on equations (10)-(14), after transformations, a final concise form (15) of the transmittance of the dynamic vertical tyre-plate contact force was obtained (bearing in mind that $q_1=z_1$, $q_2=z_2$):

$$
H_{Fdz}(s) = \frac{F_{dz}(s)}{\zeta(s)} = (c_2 \cdot s + k_2) \cdot \left[1 - H_{q_2}(s)\right]
$$

The difference between the dynamic component $F_{imd}$ of the force measured in the tester and the dynamic vertical tyre-plate contact force $F_{dz}$ arises from the tester plate inertia (the last term in (16)).

$$
H_{Fimd}(s) = \frac{F_{Fimd}(s)}{\zeta(s)} = \frac{F_{dz}(s)}{\zeta(s)} + m_3 \cdot \ddot{\zeta}(s) = \frac{F_{dz}(s) + m_3 \cdot s^2 \cdot \zeta(s)}{\zeta(s)}
$$

$$
= (c_2 \cdot s + k_2) \cdot \left[1 - H_{q_2}(s)\right] + m_3 \cdot s^2 = H_{Fdz}(s) + m_3 \cdot s^2
$$
The values of spectral transmittances \( H_{Fdz}(i \cdot \omega) \) and \( H_{Fdzm}(i \cdot \omega) \), where \( \omega \) is the radian frequency of kinematic excitation \( \zeta \), are defined for \( s=i \cdot \omega \). Their absolute values are equal to the ratios of force amplitudes \( F_{dz} \) and \( F_{dzm} \) to the amplitude of excitation \( \zeta \); the arguments are defined by phase shifts.

A comparison of the absolute values of transmittances \( H_{Fdz}(i \cdot \omega) \) and \( H_{Fdzm}(i \cdot \omega) \) (for an identical excitation) will enable indirect assessment of the impact of vibration plate mass \( m_3 \) on differences between forces \( F_{dz} \) and \( F_{dzm} \), i.e. the declared and actually measured value of the diagnostic parameter.

4. Data taken for the calculations. Parameters of the model and of the test conditions

The example model parameters were adopted for two cases, i.e. vehicle No 1 (“small”) and vehicle No 2 (“medium size”) described in [12], with the tester data (\( m_3 \)) being also taken into account:

- for vehicle No 1, \( m_1=124 \) kg, \( m_2=16 \) kg, \( m_3=14.5 \) kg, \( k_1=11 \) 000 N/m, \( k_2=130 \) 000 N/m;
- for vehicle No 2, \( m_1=266 \) kg, \( m_2=34 \) kg, \( m_3=14.5 \) kg, \( k_1=17 \) 700 N/m, \( k_2=150 \) 000 N/m.

The viscous damping coefficient of the tyre was \( c_2=0 \); the \( c_1 \) value was changed in the calculations.

The 1st and 2nd natural frequency of the undamped system were \( f_{01}=1.439 \) Hz and \( f_{02}=14.947 \) Hz for vehicle No 1 and \( f_{01}=1.227 \) Hz and \( f_{02}=11.186 \) Hz for vehicle No 2, respectively. The “critical damping coefficient” \( c_{1kr} \) value was defined, but for a system with two degrees of freedom (2DOF), it is:

\[
c_{1kr} = 2 \cdot m_1 \cdot \omega_{01} (17)
\]

Its physical sense is different from that for a 1DOF system: it does not mean that the motion excited by initial conditions is aperiodic. Nevertheless, it is an imaginary reference value and offers a possibility of using relative values. The relative damping coefficient \( \gamma \) [-] is calculated as the ratio of the current damping coefficient value \( c_1 \) to the critical damping coefficient value \( c_{1kr} \)

\[
\gamma = \frac{c_1}{c_{1kr}} = \frac{c_1}{2 \cdot m_1 \cdot \omega_{01}} (18)
\]

where: \( c_{1kr} \) [N·s/m] – critical damping coefficient;
\( \omega_{01} = 2 \cdot \pi \cdot f_{01} \) – the first (lower) natural radian frequency of the undamped system [rad/s];
\( f_{01} \) – the first (lower) natural Hertz frequency of the undamped system [Hz].

For the example data as above, \( c_{1kr}=2424.0 \) N·s/m for vehicle No 1 and \( c_{1kr}=4101.3 \) N·s/m for for vehicle No 2. The \( \gamma \) values were taken from the range of 0-0.6, especially from the range of 0-0.4, which is applicable to motor vehicle suspension systems.

The properties of the system under consideration were analysed in the 0-25 Hz frequency band, consistent with the suspension system testing conditions according to the EUSAMA method.

5. Results of calculations in the frequency domain

For the given two example sets of model parameter values (for vehicles Nos 1 and 2), calculations were carried out in the frequency domain. Figure 3 shows absolute values of the transmittances of the dynamic components of the vertical tyre-plate contact force \( F_{dz} \) and the measured force \( F_{dzm} \) in relation to the kinematic excitation from the ground, i.e. to the vertical displacement of the tester’s vibration plate, for frequencies in the range of 0-25 Hz and for three selected values of the relative damping coefficient \( \gamma \) of the suspension system. It should be noted that \( F_{dz} \) and \( F_{dzm} \) do not include the static load \( N_{st}=(m_1+m_2) \cdot g \). Component \( m_3 \cdot g \) is reduced during the reset of the measuring system. The value of the second Hertz frequency \( f_{02} \) of the undamped system has also been marked in Figures 3a and 3b. Big qualitative and quantitative differences can be seen in the absolute values of transmittances of forces \( F_{dz} \) and \( F_{dzm} \), especially for the frequencies close to, and higher than, the \( f_{02} \) value (14.94 Hz for the “small” vehicle and 11.18 Hz for the “medium size” vehicle). Meanwhile, it is close to the \( f_{02} \) value where a minimum usually occurs in the vertical tyre-plate contact force and this minimum is used to calculate the WE indicator (1). For such excitation frequencies, the absolute values of the transmittance of the measured force are lower than those determined for the tyre-plate.
contact force. Hence, the dynamic component \( F_{dzm} \) of the measured force is lower than the dynamic component of the tyre-plate contact force, which results in a higher minimum value of force \( N = F_{opm} \), i.e. \( N_{\text{min}} = F_{opm\text{ min}} \) (see also equations (1)-(7)). This may also be explained another way: the modulation by the dynamic component \( F_{dzm} \) (compared with \( F_{dz} \)) is shallower than the modulation by the dynamic component \( F_{dz} \).

![Figure 3](image)

**Figure 3.** Absolute values (AHFDZ and AHFDZM) of the transmittances of the dynamic components of the tyre-plate contact force and of the measured force (\( F_{dz} \) and \( F_{dzm} \), respectively, exclusive of static load \( N_{s} = (m_1 + m_2) \cdot g \)) relative to the kinematic excitation, i.e. to the vertical displacement of tester’s vibration plate: \( f \) – frequency; \( f_02 \) – second natural Hertz frequency of the undamped system; 

\( \gamma \) (Gamt) – relative damping coefficient of the suspension system; 

a) graph plotted for vehicle No 1 (“small”); b) graph plotted for vehicle No 2 (“medium size”)
6. Results of time domain calculations reflecting the EUSAMA test conditions

For the given model parameter values, calculations were also made in the time domain. The vertical excitation exerted by the tester’s vibration plate had the form of a sinusoid with an amplitude of \( A = 0.003 \) m and with a frequency changed during the test as shown in Figure 4a. When analysing the simulation test results, the vertical tyre-plate contact force \( F_{op} \) vs frequency curves were plotted and compared with the corresponding curves characterizing the \( F_{opm} \) force, which is normally measured by diagnostic suspension testers (Figure 4b). From the EUSAMA test point of view, it was sufficient to analyse only the lower envelope of these curves (Figure 5). Such a test results presentation method is actually adopted in some machines used to test shock absorbers without dismounting them from the vehicle (e.g. Hofmann Contactest 1000). The results of numerical experiments carried out for vehicle No 2 (“medium size”) confirm the findings based on the calculations carried out in the frequency domain. The quantitative and qualitative differences between the force vs frequency curves plotted for the tyre-plate contact force and the force measured by diagnostic testers are particularly conspicuous at the frequencies of the range 10-25 Hz. The peak-to-peak \( F_{opm} \) values are here markedly lower than those of force \( F_{op} \). At the frequency of about 19 Hz, the amplitude of changes in the simulated \( F_{opm} \) curve is even equal to a small fraction of that of the tyre-plate contact force. For many present-day passenger cars, this is the excitation frequency value at which the EUSAMA indicator is determined.

At low levels of viscous damping in the suspension system of vehicle No 2 (“medium size”), the minimum values of forces \( F_{op} \) and \( F_{opm} \), i.e. \( F_{op \text{ min}} \) and \( F_{opm \text{ min}} \), respectively, and the corresponding excitation frequencies are close to each other (see Figures 5a and 5b). With a growth in the damping, some divergences appear, but they should be deemed moderate (Figures 5c and 5d). In the cases discussed here, a real diagnostic tester would overestimate the minimum force \( F_{opm \text{ min}} \) by somewhat more than 100 N, with obtaining this result at a frequency lower by about 1.5 Hz. These findings may be confirmed by Figure 6a, where EUSAMA indicator (WE) values determined from the values of forces \( F_{op} \) and \( F_{opm} \) have been shown. In the latter case, the WE values obtained for high viscous damping in the suspension system were higher by about 5 percentage points.

![Figure 4](image-url)
It can be seen in Figure 4b that the force of inertia of the tester’s vibration plate may considerably distort the result of a shock absorber test, especially in the vehicles with high values of the upper resonance frequency. At the next stage of simulation tests, calculations were carried out for such a vehicle, herein referred to as vehicle No 1 (“small”). At high levels of damping in the suspension system ($\gamma > 0.3$), the final test result was overstated by about 10 percentage points (Figure 6b).

**Figure 5.** Lower envelopes of the curves representing the vertical tyre-plate contact force ($F_{\text{op}}$) and the force measured by testers ($F_{\text{opm}}$) as functions of the excitation frequency, plotted for various values of the relative damping coefficient of the suspension system: a) $\gamma = 0.1$; b) $\gamma = 0.2$; c) $\gamma = 0.3$; d) $\gamma = 0.4$
Figure 6. EUSAMA indicator values as a function of the relative damping coefficient of the suspension system, obtained from the vertical tyre-plate contact force ($F_{op}$) and the force measured by testers ($F_{opm}$): a) for vehicle No 2 (“medium size”); b) for vehicle No 1 (“small”)

7. Conclusion

The calculation results presented herein show the distorting impact of the force of inertia of the diagnostic tester’s vibration plate on the final results of the EUSAMA test. The relative deviations of the EUSAMA indicator values measured by the tester from the correct values reached even 18 % (see Figure 7). The scale of this problem may be even greater in the case of modern vehicles with relatively stiff suspension systems and high radial stiffness of their tyres. For such vehicles, the EUSAMA indicator is determined at higher frequencies, even of about 20 Hz.

Figure 7. Relative differences between the EUSAMA indicator values obtained from the force normally measured by diagnostic shock absorber testers ($F_{opm}$) and the vertical tyre-plate contact force ($F_{op}$) for various levels of the relative damping coefficient of the suspension system

In authors’ opinion, the simulation research started within this work should be continued, but with using nonlinear models, where the sliding friction in the suspension system should also be represented. The numerical calculations carried out should be then verified by appropriate experimental tests of a real motor vehicle on a typical EUSAMA shock absorber tester.
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