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Determination and Linearization of Telescopic Damper Nonlinear Characteristic in Passenger Vehicles Suspension System

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Experiment;
Damping force.

Summary

This paper aims to contribute to defining damping in elements with nonlinear damping in the passenger motor vehicles suspension system and the method of its linearization, suitable for mathematical modeling of the entirepassenger motor vehicle suspension system. The methodology for determining the kinematic characteristics of the passenger motor vehicle suspension system has been developed. In the given case acceleration measurements of vehicle characteristic positions (excitation device, unsprung mass – wheel and sprung mass – body) were made. The velocity and travel of these characteristic positions were obtained by integration of measured accelerations. Accelerations were measured using triaxial accelerometers, whose operation is enabled up to 20 kHz sampling frequency. By using the 1D mathematical model of suspension system and the experimental results, it was possible to determine current values of the damping force, the damping values, in the function ofvelocity and travel of the telescopic damper. Based on the obtained results, it is possible to define the real damping value, and further the damping coefficient of the passenger motor vehicle suspension system damper. The measurements were conducted on two vehicles. The first vehicle is new and had travelled approximately 2000 kmup tothe measurement day, and the second vehicle is older and had travelled approximately 56000 km up to the measurement day. These measurements and the analysis of the results can also be used to determine the exploitation life of the passenger motor vehicle suspension system.

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1. Introduction

From the early stages of road vehicles development, the suspension system has taken an important role in vehicle construction. This role is manifested through acceptance of all active forces in the form of excitation (from the road or from the forces acting on the vehicle body) which should be accepted and amortized in order to providea stable vehicle movement and maintainthe conditions of passengers comfort, and wheel guidance.To achieve this, the suspension system must form an elastic contact between the vehicle's main frame, as the sprung mass, and the axle with wheels, as the unsprung mass.

Active forces acting on the vehicle cause oscillations of the vehicle. These oscillations impair the stability of vehicle movement and comfort of the passengers, as they cause a reduction in the dynamic reaction of the road. This can lead to unwanted consequences for the safety and health of passengers.

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Various types of suspension systems are used today, which increasingly reduces the unwanted consequences of vehicles oscillations. Basic functions of the suspension system can be summarized in several groups according to [1]:

- transferring all forces from vehicle to wheels and from wheels to vehicle with continuous maintenance of contact between the wheel and the road,
- ensuring oscillatory comfort,
- ensuring steering stability,
- maintaining distance from the road and the lowest vehicle point and
- extendingwork life of vehicle elements.

In order to achieve these functions according to [1], the suspension system is a complex set of elements that are interconnected, and which according to [2] can be grouped into four subsystems:

- elastic elements subsystem that transfers vertical reactive forces to vehicle chassis or body and at the same time provides reduction of vertical impact loads,
- damper subsystem that dampens oscillations,
- subsystem for wheel guiding,
- stabilizer subsystem which ensures the necessary lateral stiffness of the suspension system.

An example of the suspension system on a passenger motor vehicle is shown in Figure 1 [3].

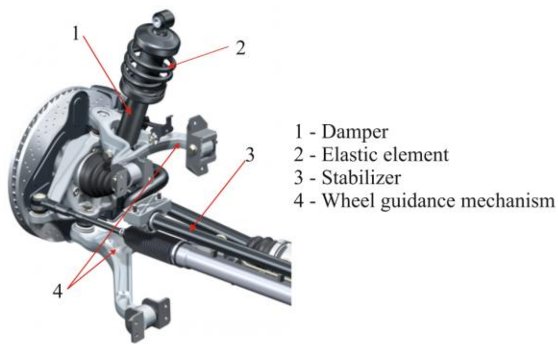


Figure 1.*Subsystems of vehicle suspension system* [3]

The greatest contribution in reducing the consequences of oscillations that the vehicle is subjected to is provided by elastic elements and elements for damping of oscillations. Today, coil springs as elastic elements are most commonly used on passenger vehicles. These springs can be made with linear and nonlinear characteristic [4]. Spring characteristic is achieved by changing number of coils, steps between them and the diameter of individual coils. Springs with linear characteristic are most widely used on passenger vehicles. However, there are also vehicles withnonlinearcharacteristiccoil springs or even with pneumatic and hydropneumatic elastic elements. Dampers, or oscillation damping elements, are now commonly used on passenger vehicles as hydraulic telescopic (monutube and twintube) dampers. They have extremely nonlinear characteristic and today it is becoming a more frequent subject of research [5, 6].

Within research of oscillatory characteristics of motor vehicles, vehicle tire is an essential element in the analysis [7]. Although not considered a direct part of the vehicles suspension system, its role is very important when achieving oscillatory comfort and stability of vehicle movement. There are several different oscillatory models of the tire [8]. These models often represent tire as a nonlinear element with characteristic of an elastic element with its stiffness and internal damping of tire rubber material. However, in research of the vehicles oscillatory characteristics, the tire is described as a linear elastic element with disregard to damping [9].

The investigation of vehicles oscillatory characteristics, in various vehicle conditions, is most often carried out by setting one of the vehicle's oscillatory models [10]. Solving these models is done numerically by using one of the numerical modeling softwares, such as Simulink. Input data in models are taken as known constants (masses, stiffness). Characteristics of elastic elements (stiffness) are adopted from already known data obtained from previous research or determined experimentally [11] or numerically [12]. Researchers often take the damping value as a constant value based on some of the recommended data and in this way simplify the nonlinear damping characteristics for further calculation of vehicle oscillatory characteristics [13]. Damping element characteristic can be determined more precisely in one of the following ways:

- by developing a special module for modeling and calculation of hydrodynamic characteristics in telescopic damper [5], where fluid flow coefficients and effective flow sections are required, for which experimental results must be used and

- byapplying experimental approach where the measurement of the nonlinear damper characteristics is performed indirectly in function of the damper piston travel.

The data obtained by second method can be linearized in such a way as to determine the mean value of the damping coefficient from the obtained data for measuring the nonlinear characteristic of the damper. Which is precisely the topic of this paper.

2. Method for determining the nonlinear characteristics of the damper and its linearization

Damping force of the damper is a function of the damper piston velocity and represents an extremely nonlinear function$F_p=f(\dot{z}_k)$.It can be determined using experimental data obtained by measuring the vehicle's kinematics. Model most commonly used for calculating the damping force is the one-dimensional oscillatory model of the vehicle or the ¼ model of vehicle [9], shown in Figure 2.

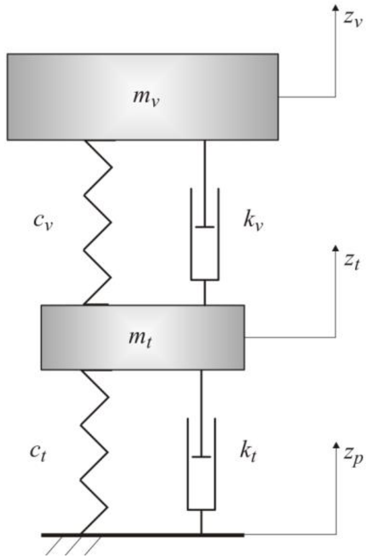


Figure 2. *One dimensional oscillatory vehicle model* [9]

Mathematical model of oscillatory system shown inFigure 2 can be expressed as:

$$m_v\ddot{z}_v+F_p+c_v(z_v-z_t)=0 \tag{1}$$
$$m_t\ddot{z}_t-F_p+c_v(z_t-z_v)+k_t(\dot{z}_t-\dot{z}_p)+c_t(z_t-z_p)=0 \tag{2}$$

Where$F_p$is damping force in telescopic damper.

In the above mentioned model, the equations (1) and (2) represent ordinary linear differential equations of the second order, if damping force in the damper is defined by the equation (3):

$$F_p=\bar{k}_v(\dot{z}_v\text{-}\dot{z}_t)=\bar{k}_v\dot{z}_k \tag{3}$$

By using equation (3) it is also possible to determine damper nonlinear characteristic. It can be presented as a function of travel or velocity of damper piston. Research shows that the damping force is a complex function that depends on travel and velocity of the damper piston [14].

If a mathematical model is simplified, the obtained results can also be simplified in terms of linearization of damping coefficient of the damper. Due to this, it is necessary toproperly define the elastic work (W_{el}) and the damping work (W_p), which can be obtained by identifying dependence of the damping force F_p and travel of the damper piston z_k. According to [15], ratio of damping and elastic work is a relative damping$\psi_r$and can be defined as:

$$\psi_r=\frac{W_p}{W_{el}} \tag{4}$$

If relative damping $\psi_r$is known, according to [15], Lehr'sdamping can be defined as:

$$\xi_r=\frac{\psi_r}{4\pi} \tag{5}$$

Based on expressions (4) and (5), according to [15, 16], damping coefficient $\bar{k}_v$, can be defined as:

$$\bar{k}_v = 2m_v\omega_0\xi_r = 2\sqrt{m_v c_v}\xi_r \qquad (6)$$

3. Experimental measurement and measurement results

3.1. Experimental device

The damping force F_p can be determined experimentally by measuring the kinematic values at characteristic points of the elastic suspension system and using the mathematical model described by the equations (1) and (2). It is possible to determine the damping force F_p if we know the kinematic data (travel, velocity and acceleration) of the vehicle body and wheel, and with the known system excitation. For this purpose, an example of acceleration measurement of excitation device, unsprung mass (m_t) and sprung mass (m_v) of two passenger motor vehicles are shown next. Measurements were made by using accelerometers. First vehicle is new and had travelled approximately 2000 km (Vehicle 1), and second vehicle is older and had travelled approximately 56000 km (vehicle 2) up to the measurement day.

The largest problem in setting up the sensors (accelerometers) for experimental research was the limited space, so when purchasing the device special attention was paid to the quality of devices of relatively small dimensions. Triaxial accelerometers SLAM STICK of the company Midé Technology (USA) have been selected, with the ability of measuring in direction of all three axes of the coordinate system belonging to the device, unsprung and sprung vehicle mass. This device, with its basic characteristics, is shown in Figure 3.

Measurable range of acceleration	± 100 g
Frequency of data acquisition	100 ÷ 20000 Hz
Possible data records	up to $4 \cdot 10^9$
Weight	65 g
Dimensions	76.2 x 29.8 x 15 mm
Temperature work range	- 40 ÷ 80 °C
Type of PC connection	USB
Battery charge lifetime	5.5 ÷ 15.5 h

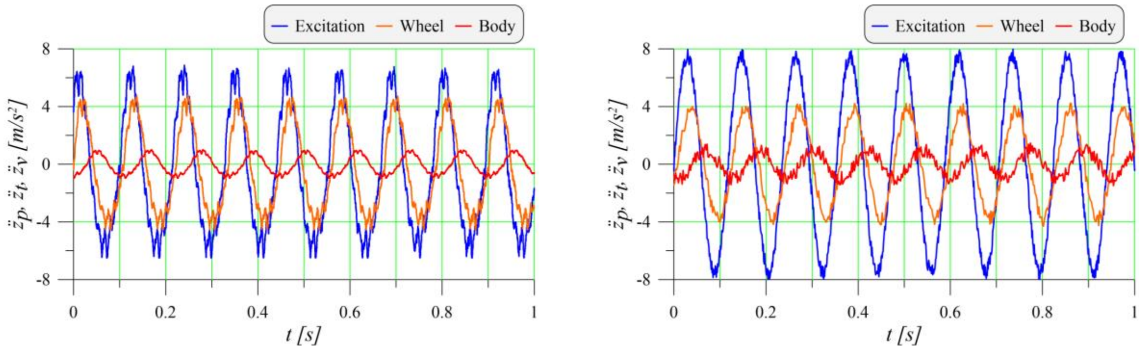
Figure 3. *Triaxial accelerometer*

3.2. Acceleration measurements and measurement results

It was necessary to define the excitation signal for specific measurements. Unevenness on the road is relatively small and unpredictable due to its macro and micro structure. The set obstacle can be clearly defined, but it does not allow obtaining a realistic quality oscillation of the vehicle over a period of time. Therefore, it was decided to generate the excitation by artificial means using a shock absorber testing device at vehicle inspection stations, where the first accelerometer was installed. Installation of the remaining two acceleration measuring devices (accelerometers) was carried out at characteristic positions of the suspension system unsprung (on lower arm position) and sprung mass (on vehicle body).

Accelerometers installed in this manner measured and recorded acceleration of the mentioned characteristic positions under the effect of excitation induced by the shock absorber testing device. Although accelerometer devices can record measurement results with a wide frequency range, while taking into account the fact that frequency of the excitation device is about 50 Hz, the sampling frequency of the measurement of 5000 Hz was selected to carry out the above mentioned test.

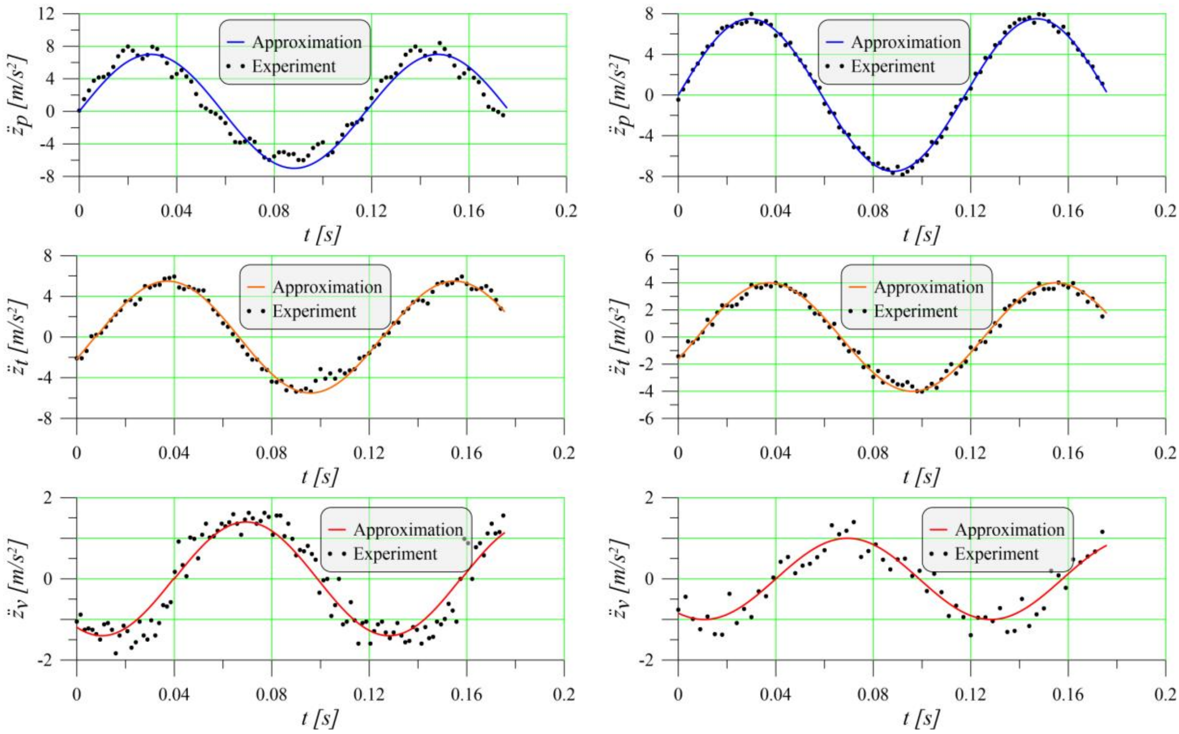
Signal synchronization was performed on all three sensors (accelerometers) with tester for the excitation for three accelerometers set in characteristic positions. The excitation itself was activated for a period of about 17 s, and the results obtained from the excitation device (testing device), lower arm and vehicle body were recorded on the same diagram. An example of recorded acceleration results at marked positions, for a period of 1 s, is shown in following Figures. Figure 4 shows example of recorded acceleration results for two vehicles at characteristic positions.



a) Vehicle 1 b) Vehicle 2

Figure 4. *Recorded acceleration diagrams at excitation and vehicles characteristic positions*

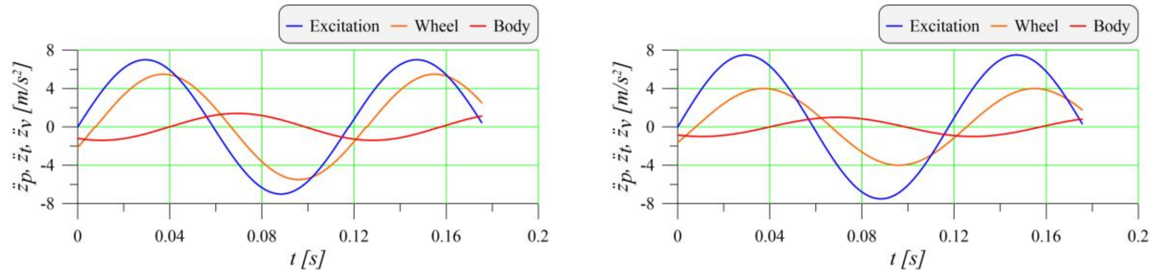
A part of the system excitation was selected to analyze the recorded results for the duration of 0,118 s. Due to high sensitivity of the acceleration sensors and real movement of the vehicle's suspension system, the obtained acceleration results on all three characteristic measurement positions have a relative dissipation compared to the expected results. This is best seen in Figure 5, where the experimental results are represented by dotted lines. Since the excitation signal is of a harmonic character, the acceleration results are harmonic functions as well. The resulting harmonic functions were obtained by approximating the experimental results and shown by solid lines in Figure 5.



a) Vehicle 1 b) Vehicle 2

Figure 5. *Acceleration diagrams at characteristic positions for one period of oscillation*

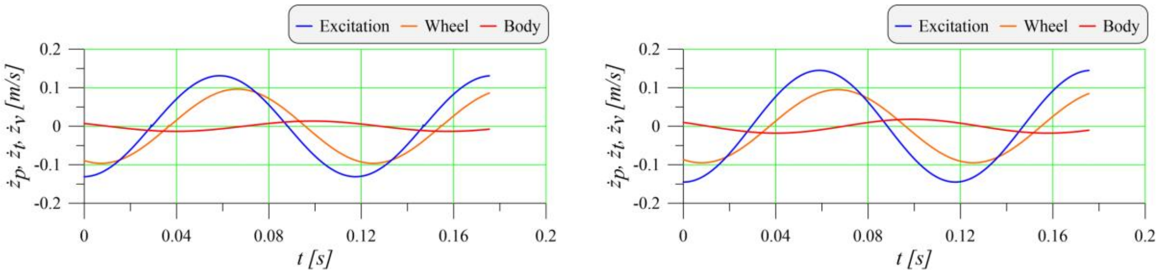
Acceleration results obtained after approximation, as based on the experiment, are shown for one oscillation cycle (of excitation) in Figure 6. Given the harmonic character of acceleration atcharacteristic positions (Figure 6), the prospect of processing these results for further use is much simplified. The original measurement results (dotted lines in Figure 5) are not suitable for further signal processing (measurement errors, high sensor sensitivity, different travel of the suspension system), which can lead to a complete distortion of the signal character presentation.



a) Vehicle 1 b) Vehicle 2

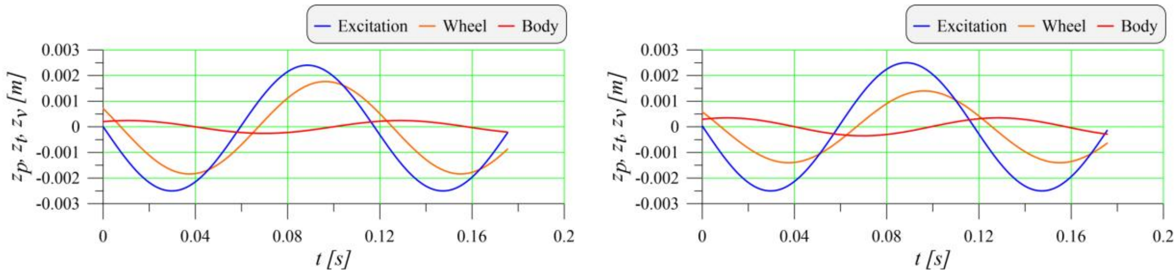
Figure 6.*Acceleration of excitation device, vehicle wheel and body oscillation diagrams*

After integration of the approximated acceleration functions shown in Figure 6, the results for velocity and travel of the excitation signal, wheel and body are obtained and shown in Figures 7 and 8.



a) Vehicle 1 b) Vehicle 2

Figure 7. *Velocity of excitation device, vehicle wheel and body oscillation diagrams*

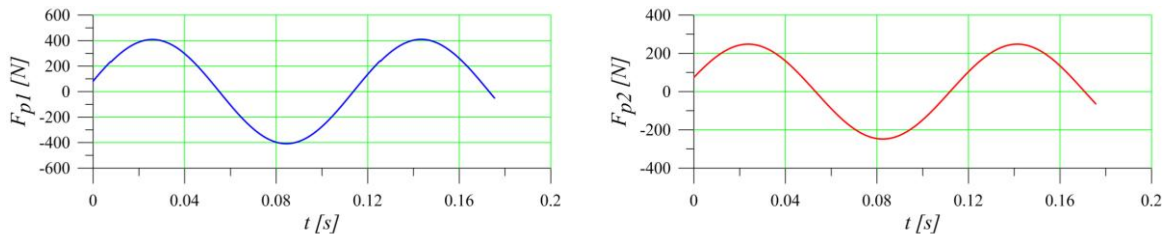


a) Vehicle 1 b) Vehicle 2

Figure 8. *Travel of excitation device, vehicle wheel and body oscillation diagrams*

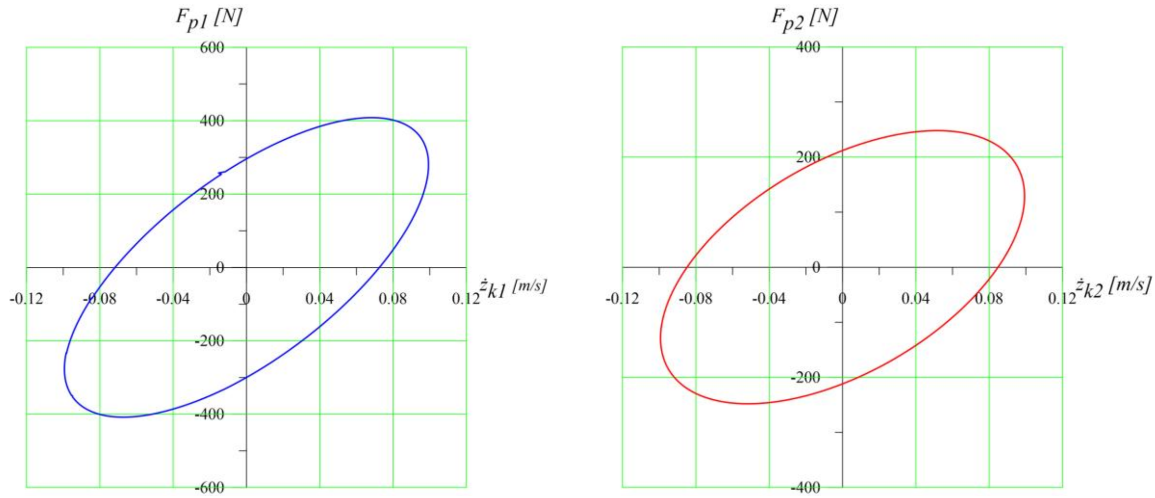
It is possible to determine damping force (F_p) of the suspension system based on recorded acceleration data for the excitation device, vehicle wheel and body, and their approximation, and further the integration of the obtained results for velocity and travel of the indicated positions for both vehicles. The damping force, as it was already mentioned, can be determined using the equations of oscillations of one-dimensional oscillatory model of the vehicle (1) and (2), as defined in Figure 2. As input data to equations (1) and (2) we used measured values of ¼ of body masses $m_{v1}=350\ kg$, $m_{v2}=250\ kg$,wheel masses$m_{t1}=25\ kg$, $m_{t2}=20\ kg$ coil spring stiffnesses$c_{v1}=4.5\cdot10^4\ N/m$, $c_{v2}=3.5\cdot10^4\ N/m$, and tire stiffnesses$c_{t1}=2.5\cdot10^5\ N/m$, $c_{t2}=1.75\cdot10^5\ N/m$.

Damping force $F_{p1}$and F_{p2}diagrams, obtained using equations (1) or (2), are shown in Figure 9. If the results of the damper velocity $\dot{z}_k=\dot{z}_v-\dot{z}_t$and the results of damping force of the damper are known, as shown in the diagrams in Figure 9, one can define their mutual dependence in the form of an ellipse, easily recognizable in literature that treats this field, which is best seen in Figure 10.



a) Vehicle 1 b) Vehicle 2

Figure9. *Damping force in oscillation time domain diagram*



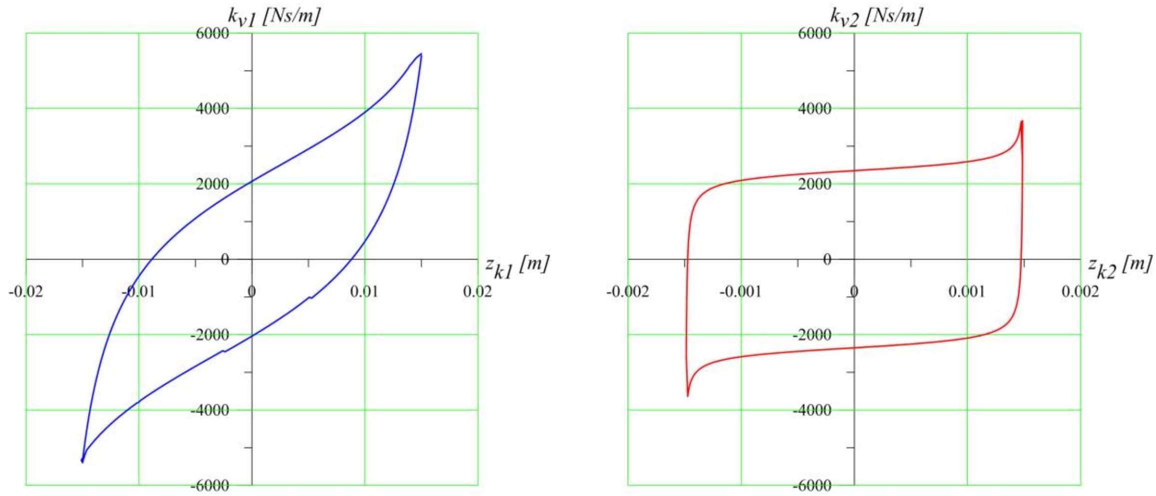
a) Vehicle 1 b) Vehicle 2

Figure 10. *Damping force as a function of damper piston velocity diagram*

Current values of damping coefficient k_v can be calculated using the expression:

$$k_v = \frac{F_p}{|\dot{z}_k|} \tag{7}$$

so finally dependance of k_v which defines damping character in function of damper piston travel, $z_k = z_v - z_t$, can be shown, as in Figure 11, for both vehicles.



a) Vehicle 1 b) Vehicle 2

Figure 11. *Damping coefficient as a function of damper piston travel diagram*

Figure 11 defines nonlinear characteristic of vehicle ocillations damper.For the mathematical description of the function in Figure 11, $k_v = f(z_k)$, one of the known mathematical functions needs to be introduced, which would best approximate the character of the damping shown in Figure 11.

For the majority of the vehicle oscillatory characteristics investigation, it is sufficient to know the damping coefficient $\bar{k}_v$obtainedbylinearizationofthe function in Figure 11. Using experimental results for the damping work (W_p) and the elastic work (W_{el}), by means of the expression (6) for a complete damper (for both vehicles) the mean damping coefficients of the tested vehicles were calculated, and they are $\bar{k}_{v1}=2197$ *Ns/m* i $\bar{k}_{v2}=1360$ *Ns/m*.

The presented method enables simple and quick determination of the mean value of nonlinear characteristic of damping element in the suspension system. This way, during the exploitation life of the vehicle, i.e. various number of kilometers traveled, the condition of telescopic damper can easily be tracked and the periodical maintenance of vehicle in terms of the suspension system can be defined.

4. Conclusion

Proper performance of vehicle suspension system is one of key elements in safe travel of vehicles along the road, as well as passenger comfort. During exploitation of the vehicle, as well as the suspension system itself, characteristics of these elements deteriorate. Here, attention is paid primarily to elastic elements (springs) and dampers, for which is necessary to define their characteristics. In case of springs, above mentioned characteristic is expressed by stiffness, which is generally of linear value and can easily be determined. On the other hand, in case of a damper, this characteristic is expressed by damping coefficient, which is extremely non-linear and requires a more complex approach using experimental and mathematical (numerical) methods.

In this paper, using triaxialaccelerometers, acceleration values were experimentally determined directly on characteristic positions of vehicle suspension system (device for excitation, lower arm and vehicle body). These results enabled obtaining information on the oscillatory character, and by using the 1D mathematical oscillation model, the basis for defining damping force, damping values, and piston velocity and travel of the telescopic damper were obtained.

By taking into consideration recommendations from the literature and the results obtained through experimental and mathematical modeling, this paper presents methods of using approximate expressions to define the damping itself in the function of telescopic damper travel, which once again confirmed the nonlinear character of the oscillations damper. At the same time, a demonstration of mean damping coefficient value, which is mainly used for simpler modeling of the suspension system, is also given. This methodology can be very successful in determining the state of dampers during its exploitation life.

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Nomenclature

Label	Unit of measurement	Description
c_t	*N/m*	Tire stiffness
c_v	*N/m*	Vehicle spring stiffness
F_p	*N*	Damping force
k_t	*Ns/m*	Tire damping
k_v	*Ns/m*	Damper damping coefficient
$\bar{k}_v$	*Ns/m*	Damper mean value of damping
m_t	*kg*	Wheel mass

m_v	*kg*	Part of the vehicles mass per wheel
t	*s*	Time
W_{el}	*Nm*	Elastic work per cycle
W_p	*Nm*	Damping work per cycle
z_k	*m*	Damper piston travel
z_p	*m*	Travel of excitation device
z_t	*m*	Travel of unsprung mass (wheel)
z_v	*m*	Travel of sprung mass (body)
ω_0	*1/s*	Natural frequency of vehicle oscillation
ξ_r	-	Lehr's damping
ψ_r	-	Relative damping