ANALYSIS OF EFFECTS OF VEHICLE SUSPENSION SYSTEM STIFFNESS AND DAMPING ON OSCILLATORY COMFORT

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ABSTRACT

Primarily due to road irregularities, vehicle movement is invariably accompanied by oscillations of vehicle masses that cause tiredness and fatigue of passengers on one hand, and additional road and vehicle loads on the other hand. Oscillations of vehicle masses can cause changes in dynamic reactions of the road, which could furthermore decreasethe traction or brake forces, and jeopardize stability of movement. These negative consequences of oscillations are reduced by applying different designs of vehicle suspension systems. The study of complex mechanism oscillations with a large number of degrees of freedoms is generally performed on a simplified model. The selected model should include only the most important parameters and the parameters that are of a less importance should be ignored. By doing this, the analytical processing of the problem is accelerated and the practical verification of the results is simplified. Additionally, costs and time needed for the experimental testing are reduced. In this paper, an analysis of travel of sprung and unsprung vehicle masses was conducted, in dependence on vehicle suspension stiffness and damping, along with the speed of oscillation settling in dependence on vehicle suspension damping.Also, the analysis of vehicle oscillatory behavior on passengers' comfort and stability of movement on the road is presented. The model and results shown in this paper are an adequate basis for the next step, which is designing active suspension system ofthe vehicle.

Key words: Oscillations, Stiffness, Damping, Comfort, Stability

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INTRODUCTION

Elastic elements and dampers are found as the most important elements of vehicle suspension system. The task of elastic elements is to receive impulses from the road and active forces during its movement, and to accumulate energy given within itself. Also, it reduces impact loads on the vehicle. Because elastic elements exist in vehicle suspension system, oscillations appear while vehicle moves on the road, so their damping must be performed with elements for damping or dampers. During the past twenty years of vehicle development, great attention has been paid to introducing semi-active and active suspension system [1, 2, 3,4], whose main goal is improving vehicle dynamic characteristics in the movement process, especially while braking and passing through the curves, but also improving comfort. For development of such systems,the most commonly used models are quarter car suspension system models [1, 2, 3, 4], some of which have included seat and driver models for defining comfort [4].However, during these simulations it is necessary to analyze a complete, very complex vehicle system, which can be done by introducing 3D model of sprung and unsprung masses travel.While using such a model,the largestissue is determining the real parameters of stiffness and damping of individual elements, which can be quite accurately determined by using measurement equipment as shown in [4].

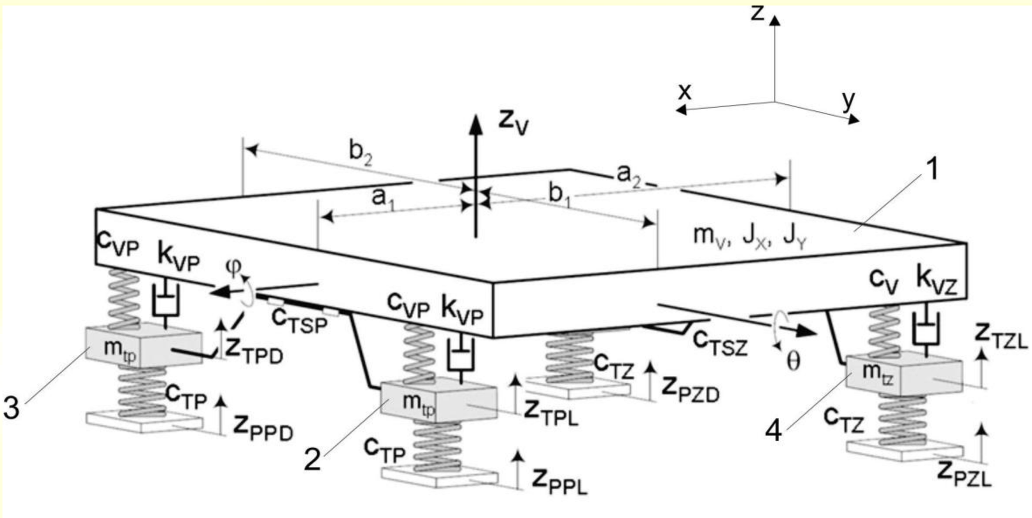
While bearing in mind the aforementioned, the objective of this paper is to establish the real mathematical model to simulate oscillations of sprung and unsprung vehicle masses for different moving conditions of vehicles. In the future,this model should enable the introduction of elements of active suspension which should, along with the experimental verification, enable further development of the suspension system of modern vehicles.

THREE-DIMENSIONAL OSCILLATORY MODEL OF VEHICLE

If a motor vehicle is observed as a real system, then it can be concluded that the analysis of masses travel caused by the movement of vehicle on the road can be a very complex problem, whose solution would take a lot of time. Thus the equivalent oscillatory vehicle models are used for mathematical modelling problems like these. The equivalent oscillatory system is obtained on the basis of principle of transforming real to equivalent oscillating system, which is reflected in:

- equality of the kinetic energy of the real and the equivalent system and
- equality of the potential energy of the real and the equivalent system.

In numerical solution of the vehicle oscillating masses, one-dimensional, two-dimensional and three-dimensional oscillating models of the vehicle can be used, depending on how much of the process is to be explored. In order to analyze the influence of some of the suspension system parameters on oscillatory behavior of unsprung masses and sprung mass of the vehicle respectively, it is best to use the three-dimensional equivalent vehicle model.A three-dimensional oscillatory vehicle model with five masses (one sprung and four unsprung) is shown in the Figure 1.



1 - Sprung mass, 2-Front left unsprung mass,
3-Front right unsprung mass, 4-Rear left unsprung mass

Figure 1. Three-dimensional oscillatory vehicle modelwith five masses

From Figure 1., it can be seen that the given model has seven degrees of freedom,travel of sprung mass in the direction of z-axis, travel of unsprung masses in the direction of z-axis, and angular movements of vehicle around x and y axis. In order to define all the travels and angles of rotation, it is necessary to solve a system of seven equations of oscillation which will be given below.

The equation of sprung mass (1) oscillation in z-axis direction:

$$m_V \frac{d^2 z_V}{dt^2} + k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPL}}{dt} + b_1 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) + k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPD}}{dt} - b_2 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) +$$
$$+ k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZD}}{dt} - b_2 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZL}}{dt} + b_1 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + \quad (1)$$
$$+ c_{VP}(z_V - z_{TPL} + b_1\varphi - a_1\theta) + c_{VP}(z_V - z_{TPD} - b_2\varphi - a_1\theta) +$$
$$+ c_{VZ}(z_V - z_{TZD} - b_2\varphi + a_2\theta) + c_{VZ}(z_V - z_{TZL} + b_1\varphi + a_2\theta) = 0$$

The equation of sprung mass (1) oscillation around x-axis:

$$J_x \frac{d^2\varphi}{dt^2} + b_1 k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPL}}{dt} + b_1 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) - b_2 k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPD}}{dt} - b_2 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) -$$
$$- b_2 k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZD}}{dt} - b_2 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + b_1 k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZL}}{dt} + b_1 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + \quad (2)$$
$$+ b_1 c_{VP}(z_V - z_{TPL} + b_1\varphi - a_1\theta) - b_2 c_{VP}(z_V - z_{TPD} - b_2\varphi - a_1\theta) -$$
$$- b_2 c_{VZ}(z_V - z_{TZD} - b_2\varphi + a_2\theta) + b_1 c_{VZ}(z_V - z_{TZL} + b_1\varphi + a_2\theta) + c_{TS}\varphi = 0$$

The equation of sprung mass (1) oscillation around y-axis:

$$J_y \frac{d^2\theta}{dt^2} - a_1 k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPL}}{dt} + b_1 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) - a_1 k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPD}}{dt} - b_2 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) +$$
$$+ a_2 k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZD}}{dt} - b_2 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + a_2 k_{VZ}\left(\frac{dz_V}{dt} - \frac{dz_{TZL}}{dt} + b_1 \frac{d\varphi}{dt} + a_2 \frac{d\theta}{dt}\right) + \quad (3)$$
$$- a_1 c_{VP}(z_V - z_{TPL} + b_1\varphi - a_1\theta) - a_1 c_{VP}(z_V - z_{TPD} - b_2\varphi - a_1\theta) +$$
$$+ a_2 c_{VZ}(z_V - z_{TZD} - b_2\varphi + a_2\theta) + a_2 c_{VZ}(z_V - z_{TZL} + b_1\varphi + a_2\theta) = 0$$

The equation of front left unsprung mass (2) oscillation in z-axis direction:

$$m_{TP} \frac{d^2 z_{TPL}}{dt^2} - k_{VP}\left(\frac{dz_V}{dt} - \frac{dz_{TPL}}{dt} + b_1 \frac{d\varphi}{dt} - a_1 \frac{d\theta}{dt}\right) - c_{VP}(z_V - z_{TPL} + b_1\varphi - a_1\theta) - \quad (4)$$
$$- c_{TS} \frac{\varphi}{b_1 + b_2} + c_{TP}(z_{TPL} - z_{PPL}) = 0$$

The equation of front right unsprung mass (3) oscillation in z-axis direction:

$$m_{TP}\frac{d^2z_{TPD}}{dt^2}-k_{VP}\left(\frac{dz_V}{dt}-\frac{dz_{TPD}}{dt}-b_2\frac{d\varphi}{dt}-a_1\frac{d\theta}{dt}\right)-c_{VP}(z_V-z_{TPD}-b_2\varphi-a_1\theta)+ \\ +c_{TS}\frac{\varphi}{b_1+b_2}+c_{TP}(z_{TPD}-z_{PPD})=0 \qquad (5)$$

The equation of rear left unsprung mass (4) oscillation in z-axis direction:

$$m_{TZ}\frac{d^2z_{TZL}}{dt^2}-k_{VZ}\left(\frac{dz_V}{dt}-\frac{dz_{TZL}}{dt}+b_1\frac{d\varphi}{dt}+a_2\frac{d\theta}{dt}\right)- \\ -c_{VZ}(z_V-z_{TZL}+b_1\varphi+a_2\theta)+c_{TZ}(z_{TZL}-z_{PZL})=0 \qquad (6)$$

The equation of rear right unsprung mass (5) oscillation in z-axis direction:

$$m_{TZ}\frac{d^2z_{TZD}}{dt^2}-k_{VZ}\left(\frac{dz_V}{dt}-\frac{dz_{TZD}}{dt}-b_2\frac{d\varphi}{dt}-a_2\frac{d\theta}{dt}\right)- \\ -c_{VZ}(z_V-z_{TZD}-b_2\varphi+a_2\theta)+c_{TZ}(z_{TZD}-z_{PZD})=0 \qquad (7)$$

From equations (1) – (7), as well as from Figure 1., it is shown that the influence of tire damping is neglected, and that front and rear stabilizers are shown on the vehicle with its'stiffnesslabelledrespectively as $c_{TSP}$and c_{TSZ}. By observing equations (2), (3) and (4),it can be seen that only the influence of one torsionalstabilizer was shown, and in this case that is the front one, with its stiffness labelled as c_{TS}.

The oscillating equations (1) – (7) will be solved by generating an individual simulation model in Simulink, MatLAB software.

THE IMPACT OF THE SUSPENSION STIFFNESS ON THE TRAVEL OF SPRUNG AND UNSPRUNG MASSES

The most commonly used elastic element in the suspension system is springs. The elastic elements are aimed at accepting active forces that arise during the process of the vehicle movement (inertial force, centrifugal force, aerodynamic force, load redistribution on the rise, impulses and forces due to shock loads of the road, etc.) and get transferred to the frame or body, and provide its reduction.

In order to examine the stiffness of suspension system on the vertical travel of the sprung and unsprung masses of the vehicle, various values of vehicle suspension system stiffness will be analyzed. The analysis will be carried out for the vehicle speed v = 20 km/h, whereby the damping of suspension will be kept constant. Road profile to be used for the analysis is defined inthe Figure 2.

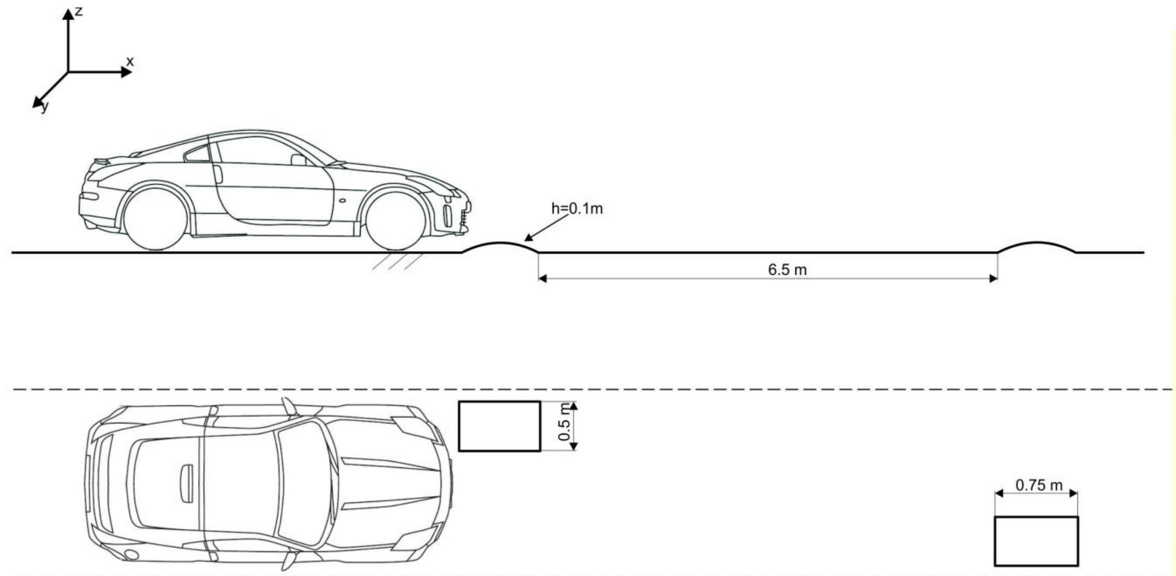


Figure2. Road profile to be used for the analysis

The Table 1. presents the characteristics of vehicle required for the numerical simulation of oscillatory model of the vehicle.

Characteristics of vehicle listed in Table 1 for mass and centre of gravity were adopted according to the literature [5], and data for radius of inertia i_x and i_y around x and y axes from the literature [7]. Characteristics of suspension were adopted according to empirical research.

From the equations (2) and (3) it can be seen that the moments of inertia J_X and $J_Y$occur, which are determined by the recommendations from the literature [8]:

$$J_Y = m_V \cdot i_Y^2 \tag{8}$$

$$J_X = m_V \cdot i_X^2 \tag{9}$$

Table 1 – **Vehicle characteristics**

Name	**Symbol**	**Value**
Sprung mass	m_V	1000 kg
Unsprungmasses	m_{tp}	35 kg
	m_{tz}	25 kg
Suspension damping	k_{VP}, k_{VZ}	1500 Ns/m
Tire stiffness	c_{TP}, c_{TZ}	145000 N/m
Torsion stabilizer stiffness	c_{TS}	25000 Ns/m
Coordinates of centre of gravity in longitudinal plane	a_1	0.933 m
	a_2	1.557 m
Coordinates of centre of gravity in transversal plane	b_1	0.706 m
	b_2	0.706 m
Sprung mass radius of inertia around x-axis	i_X	0.64 m
Sprung mass radius of inertia around y-axis	i_Y	1.13 m

After the inclusion of the necessary data the values of moments of inertia are as follows:

$$J_Y = 1277\,\text{kgm}^2$$

$$J_X = 409{,}6\,\text{kgm}^2$$

While conducting the travel analysis of sprung and unsprung masses, unsprung masses will adopt the initial value in the z axis direction of 0.3 m and 0.8 m of the sprung mass.

Based on the familiar data about the vehicle given in the Table 1, the familiar configuration of the road and initial velocity, by using own mathematical model which is numerically defined in the program Simulink, Matlab, the obtained results are shown in Figures 3-8.

The Figure3 presents the vertical travel of unsprung masses in the direction of the z-axis, depending on variety of stiffness of suspension system and impulse brought to the system in the form of bumps on the road, as defined in the Figure 2. Three different values were taken for the stiffness of suspension system in order to present the effect of stiffness on vertical oscillations of unsprung masses of the vehicle. The values of the suspension system stiffness for individual cases of the simulation for the front axis (c_{VP}) and the rear axis (c_{VZ}) of the vehicle are viewed as the same and are labelled as c_V.

It can be seen that with the increase of vehicle suspension system stiffness, the oscillation amplitude of unsprung masses is reduced because the elastic elements in the vehicle suspension system are in charge of receiving the energy which is transferred to the wheels onto them, and therefore if the elastic element is more stiff it will be less compressed and will receive less energy from the impulse force of the bump that wheel passes over it.

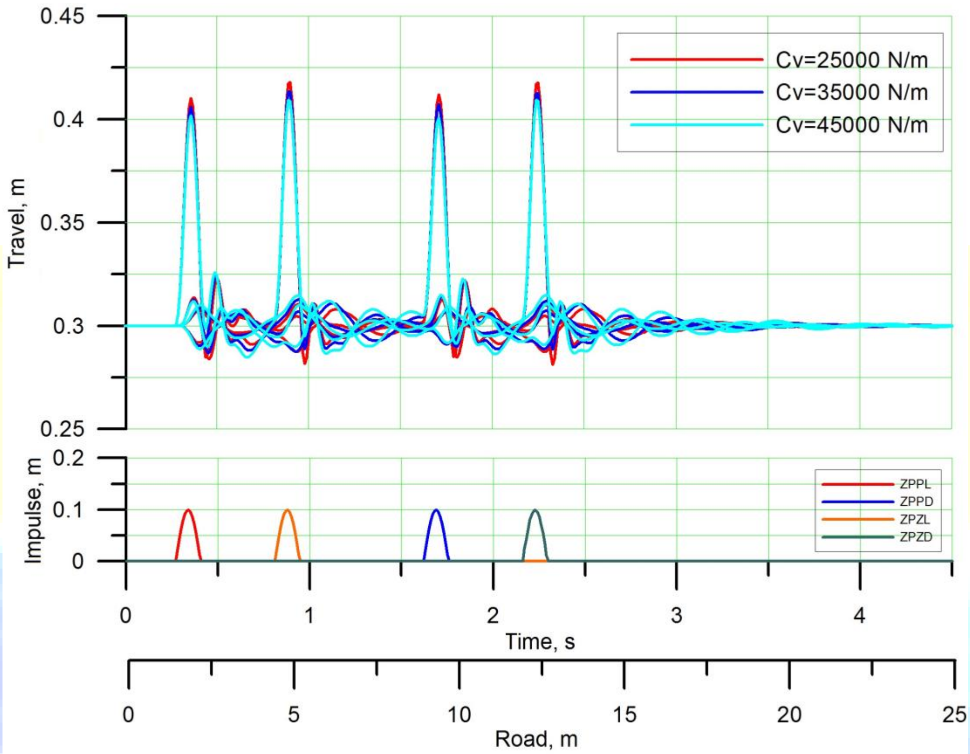


Figure3.Travel of the unsprung masses in the direction of the z-axis in the longitudinal plane of the vehicle

In order to better observe the response of the individual wheels on the impulse of the road when the wheel passes over it, the figures 4-7 present four diagrams of response to the road impulse of each individual wheel.

The Figure 3, as well asthe figures 4-7, show that the increase of vehicle suspension stiffness reduces the amplitude of the oscillations of the unsprung masses (wheels). Also, it can be seen that by increasing the vehicle suspension stiffness, settling of the vibrations lasts longer, i.e. the oscillations settle down slower.

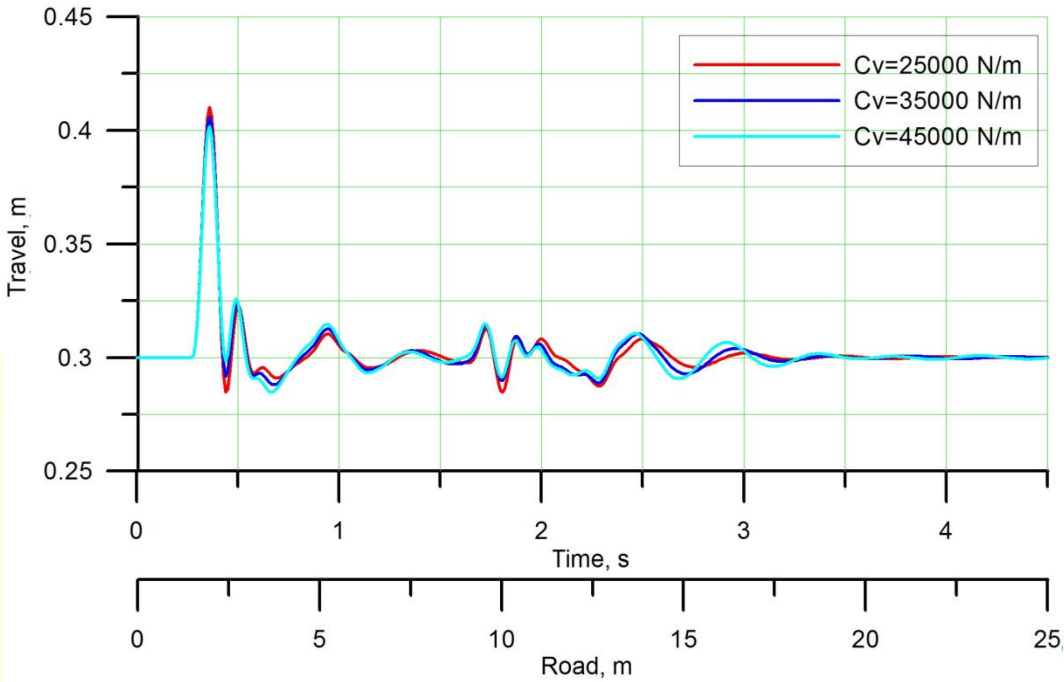


Figure 4. The response of the front left wheel (unsprung mass) to its impulse

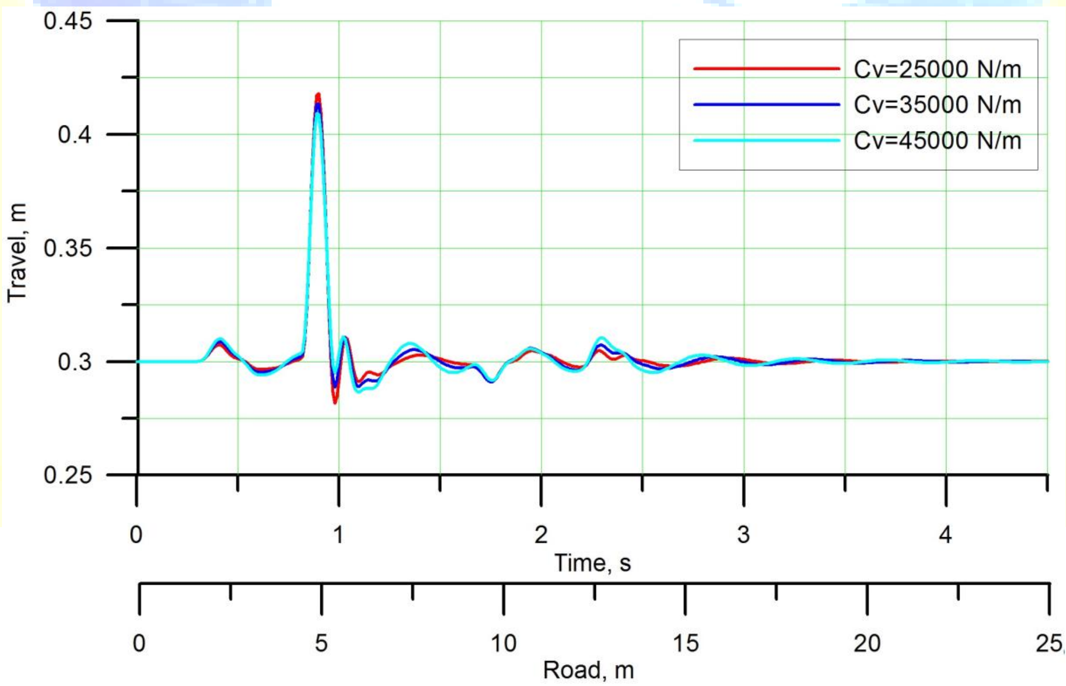


Figure 5. The response of the rear left wheel (unsprung mass) to its impulse

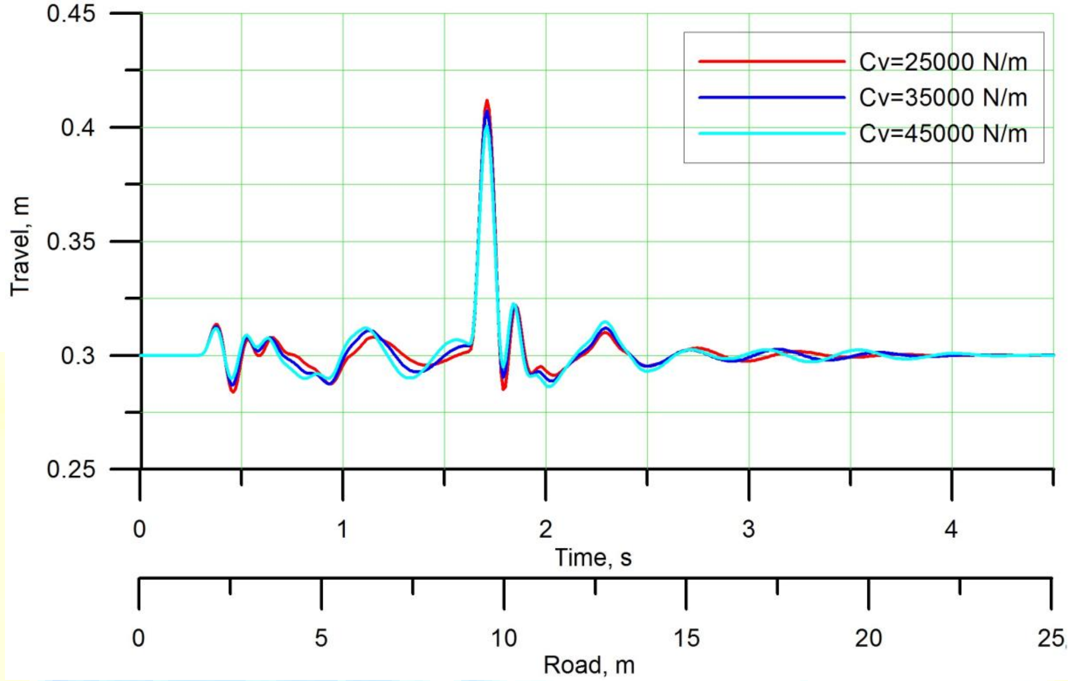


Figure 6. The response of the front right wheel (unsprung mass) to its impulse

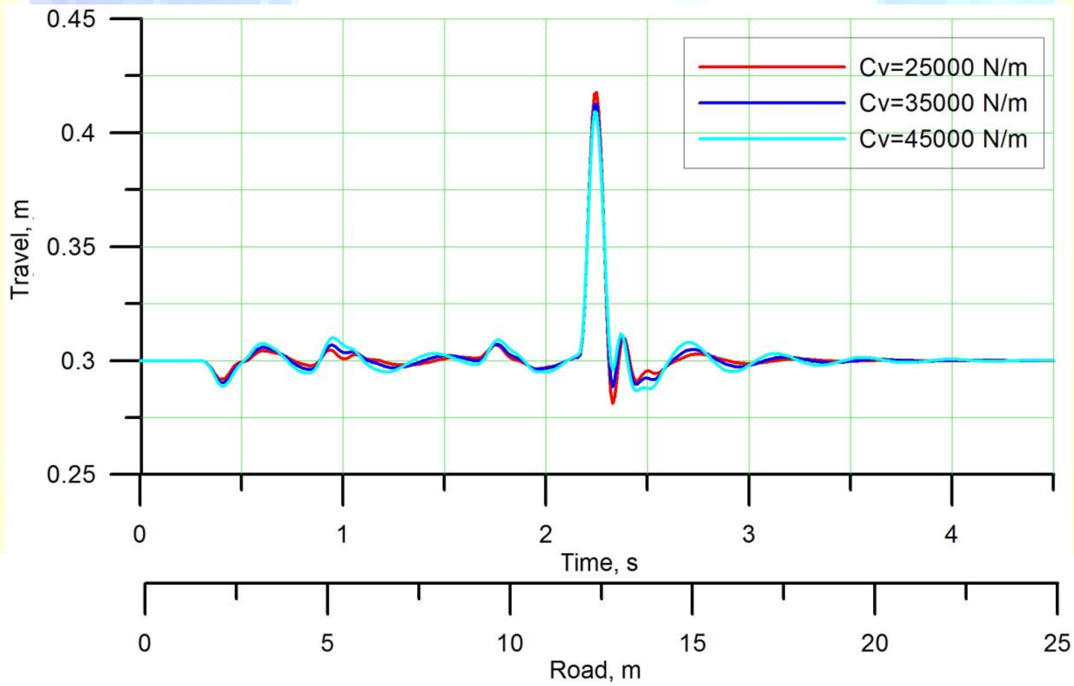


Figure 7. The response to the right rear wheel (unsprung mass) to its impulse

Finally, the Figure 8. shows the impact of the suspension system stiffness on the travel of the sprung mass of the vehicle, whereby it can be concluded that with the increase in the stiffness of vehicle suspension system, the shocks are increased onto the sprung mass, i.e.thesprung mass oscillates with the increasing amplitude the higher the stiffness of suspension system gets.

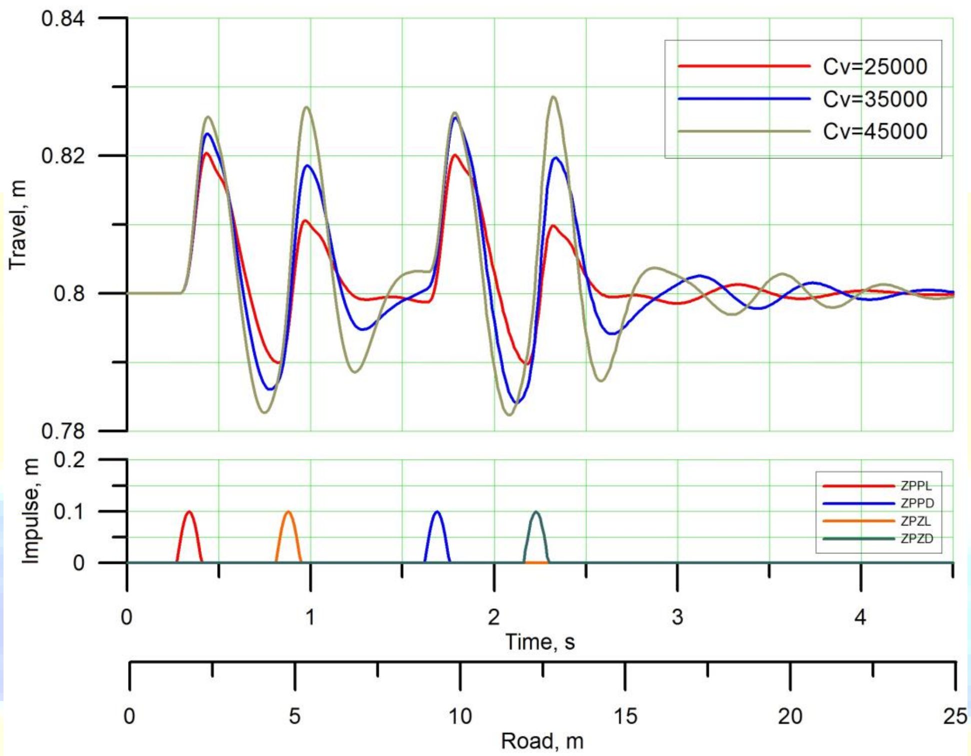


Figure 8. Travel of the sprung mass, depending on the stiffness of suspension system

Based on the conclusions of the results of the simulation, it can be said that with the increase of the vehicle suspension system stiffness, the amplitude of sprung mass travel will increase, which additionally means that the oscillatory comfort of the passengers decreases, but a better holding of the car on the road on which the vehicle is moving is achieved. Based on this, it can be concluded that when choosing the stiffness of suspension in the construction of a vehicle, the exploitation conditions in which the vehicle shall be moving should always be predicted, and on the basis of those conditions, it should be decidedupon the type of suspension (more or less stiff).

THE IMPACT OF DAMPING OF THE SUSPENSION SYSTEM ON SETTLING THE OSCILLATIONS OF SPRUNG MASS

Energy of the movement that is brought by road impulse force into the vehicle suspension system is partly used on compression of that element, and the other part of the energy must somehow be spent or transferred to another part that is attached to the unsprung mass through the suspension system. The remaining part is transferred to the sprung mass, i.e., the vehicle body via dampers (shock absorbers), which is responsible for damping, i.e. settling the oscillations that occur when the vehicle drives over bumps or obstacles. Only the impact of damping of the vehicle suspension system on settling the oscillations of sprung mass of the vehicle is presented here, without any analysis of the impact on the movement of unsprung masses (wheels).Figure 9. presents the influence of vehicle suspension system damping on sprung mass oscillations settling.

By analyzing Figure 9., it can be concluded that by increasing the damping of suspension system the oscillations of the sprung mass of the vehicle, which result from the movement of vehicle across the bumps or obstacles, are settling more quickly the greater the damping of vehicle suspension system is.

The characteristics of the vehicle that were taken during the calculations are the same as in Table 1, except that the coefficient of stiffness of elastic suspension system is at a value of 25000 N/m and the damping coefficient of suspension varies.

As can be seen from the conducted analysis, by increasing damping ratio of suspension system, a larger travel of sprung mass of the vehicle occurs, which deteriorates the passengers' comfort, but also achieves better hold of the car on the road which then increases vehicle stability. As for the coefficient of stiffness, the selection of appropriate values of damping ratio of elastic suspension system must be made as a compromise between passengers' comfort while driving, and safety of the vehicle on the road.

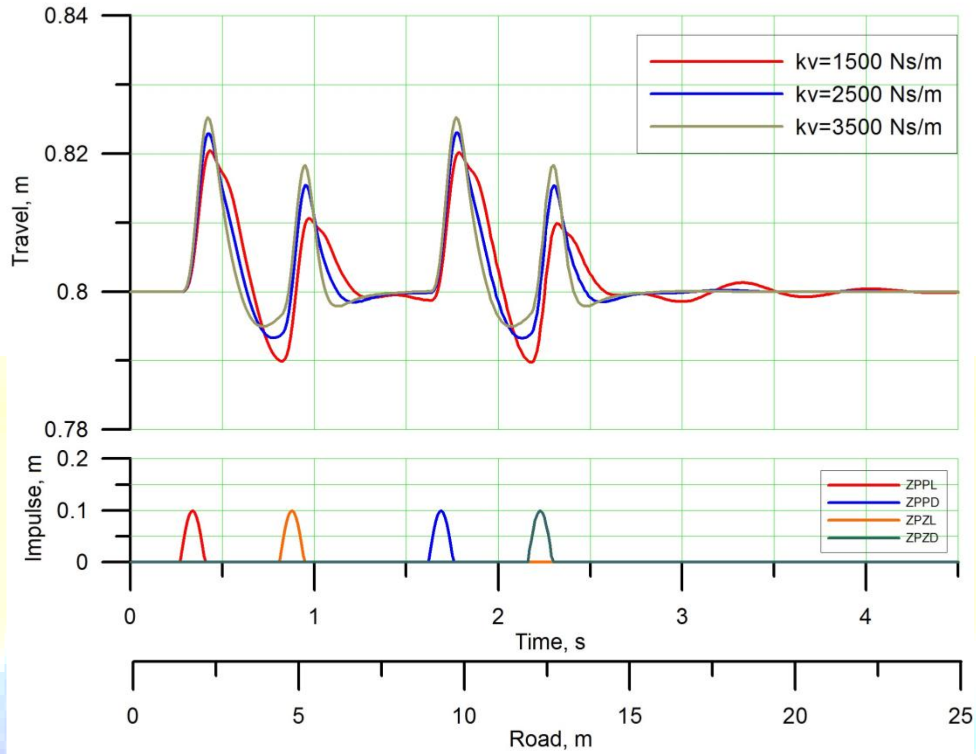


Figure 9. The impact of damping of vehicle suspension system on settling of oscillations of the vehicle sprung mass

CONCLUSION

In this paper, the influence of the stiffness and damping of elastic suspension system on the oscillatory comfort of the vehicle and its movement on the road were presented. A three-dimensional oscillatory model of vehicle with five masses and seven degrees of freedom was used for the analysis, which was solved numerically in the program Simulink, Matlab.

For different values ofstiffness of the vehicle suspension system, it was shown that by increasing stiffness, unsprung masses (wheels) oscillate with a smaller amplitude. This results in greater vertical travel of the vehicle's sprung mass. By increasing the damping ratio of suspension system, oscillations of the sprung mass of the vehicle settle more quickly. In terms of passengers' comfort and behavior of the vehicle on the road, the stiffer suspension (with greater stiffness) means a reduced passengers' comfort, but better behavior of the vehicle on the road, while the softer suspension (with less stiffness) means a more comfortable ride for passengers, but worse behavior of vehicle on the road. It is therefore necessary, when selecting the parameters of suspension, to always know the exploitation conditions in which the vehicle will be moving.

A mathematical model that was used for the analysis of parameters of passive suspension system and the obtained results, provide a further basis formodellingthe active suspension system and analysis of the parameters of that system.

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Nomenclature:

a_1, a_2	- coordinates of gravity centre in vehicle longitudinal plane [m],
b_1, b_2	- coordinates of gravity centre in vehicle transverse plane,
c_{TS}	- torsion stabilizer stiffness [N/m],
c_{TP}	- front unsprung mass (tire) stiffness [N/m],
c_{TZ}	- rear unsprung mass (tire) stiffness[N/m],
c_{VP}	- front suspension system stiffness [N/m],
c_{VZ}	- rear suspension system stiffness [N/m],
i_x	- radius of inertia around x-axis [m],
i_y	- radius of inertia around y-axis [m],
Jx	- moment of inertia of sprung mass around x-axis [kg m^2],
Jy	- moment of inertia of sprung mass around y-axis [kg m^2],
k_{VP}	- front suspension system damping [Ns/m],
k_{VZ}	- front suspension system damping [Ns/m],
m_{TP}	- mass of front unsprung mass [kg],
m_{TZ}	- mass of rear unsprung mass [kg],
m_V	- mass of sprung mass [kg],
z_{PPL}	- impulse from the road on front left wheel [m],
z_{PPD}	- impulse from the road on front right wheel [m],
z_{PZL}	- impulse from the road on rear left wheel [m],
z_{PZD}	- impulse from the road on rear right wheel [m],
z_{TPL}	- travel of front left unsprung mass in z-axis direction [m],
z_{TPD}	- travel of front right unsprung mass in z-axis direction [m],
z_{TZL}	- travel of rear left unsprung mass in z-axis direction [m],
z_{TZD}	-travel of rear right unsprung mass in z-axis direction [m],
z_V	- travel of sprung mass in z-axis direction [m],
θ	- angle of rotation of sprung mass around y-axis [°],
φ	- angle of rotation of sprung mass around x-axis [°].