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## MODELING, SIMULATION AND SENSITIVE ANALYZING OF THE TERRAIN VEHICLE SUSPENSION SYSTEM

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**ABSTRACT:** Effective method for analyzing of the terrain vehicle suspension performance is presented. The key objective of this approach is to make quick analyses of the vehicle suspension system performance. The vehicle is simplified by mathematical modeling of a quarter of it and equations of motion are solved by using Matlab software. To verify the reliability of the proposed method a comparison with one of the simply commercial software is done. The usefulness of the proposed method on the real terrain vehicle is presented in order to analyze the sensitivity of the several suspension parameters such as spring stiffness coefficient, damping coefficient, tire stiffness coefficient, sprung and un-sprung masses. The sensitivity of the suspension parameters is determined by considering of the vehicle body acceleration, vertical tire force and suspension travel that directly influences on the driving comfort and driving safety.

**KEYWORDS:** Modeling, suspension parameters, driving comfort, driving safety

### INTRODUCTION

The key objective of the terrain vehicles is to ensure the tire contact with the ground surface. Because of emphasized ground roughness this can be ensured by more or less comprehensive suspension system. The suspension system connects physically the vehicle chassis and wheels, transmits all loads from ground to the vehicle, enables the vehicle driving, braking and steering, The suspension system is fixed on the vehicle chassis and consists from the wheels and tires, springs, shock absorbers, few rods and linkages, as well as steering system, Lajqi et al. [1] and Pehan et al. [2]. The driving comfort is directly related to the vertical vehicle body acceleration. The driving safety is depended by the quality of the contact between the tire and ground surface. The suspension travel denotes relative motion between vehicle body and wheels. The designers devote great attention to the suspension systems in order to improve both characteristics of the driving comfort and driving safety.

The vehicle suspension systems are categorized as passive, semi-active and active systems Lajqi et al. [3], Sentil [4]. The passive suspension system includes beside mechanisms also one of the conventional springs and shock absorbers. Springs have linear or nonlinear characteristics, while the shock absorbers exhibit nonlinear relationship between force and relative velocity. Although this suspension system do not fulfill all expectations about comfort and safety are used widely. The semi-active suspension system contains instead passive shock absorber an active one that is automatically controlled by an integrated regulator. Because of this device the suspension system becomes semi-active and it offers some advantages by extreme driving conditions, Ram Mohan Rao et al. [5]. Comparable with the fully active system the semi-active suspension requires less energy, it is cheaper and it is simplest in design. The fully active suspension system beside normal components includes also an actuator, sensors and CPU unit. Sensors measure the acceleration and programmable CPU unit computes and guides the actuator that produces the additional forces when it is desired. After comparison of counted suspension systems the priority is dedicated to the passive one-because of the simplicity and comparable lower weight.

Suspension systems for terrain vehicles enable very extensive wheel movements that require enough space because of collision risk. It is often required a many attempts to get finally an effective suspension design. At first stage the designer should have an effective tool to make quick analyzes of lengths and other geometrical appearance of the suspension components. At second stage the designer must make 2D or 3D-model of the suspension in order to check the possible movements. In the follow steps the strength is checked. It is repeated until the design engineer does get satisfied results. To speed these design procedure it would help the effective methodology that simulates the suspension characteristic in dependence of essentials parameters. One of the good established models for

understanding and explanation of vehicle suspension is so called a quarter vehicle model (QVM), Yu et al. [6]. The QVM is comparatively easy to transform into mathematical model that consist actually from two second order differential equations, which can be solved on analytical or numerical way.

Before the mathematical model, the initial suspension system should be designed, and then it must be transformed into simplified two masses model, Figure 1.

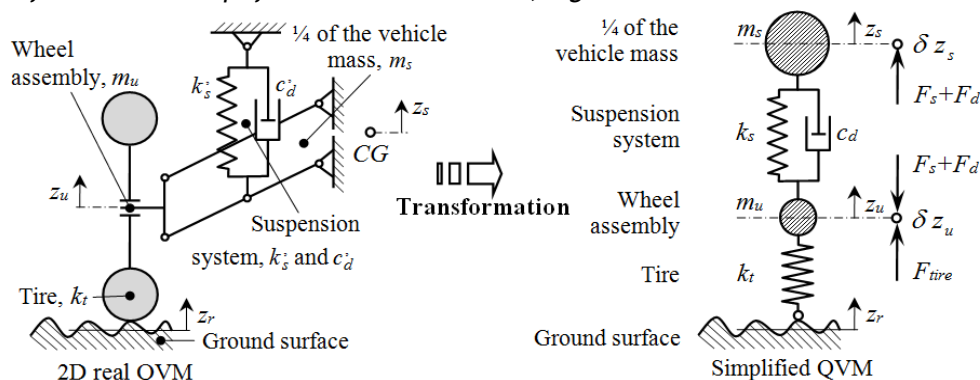


Figure 1. Transformation from 2D real QVM into simplified QVM

Due to simplicity (in special cases it is true) it is assumed that also in the simplified model the spring stiffness coefficient and damping coefficient remain constant. At this stage it is needed powerful computing program for simulation the suspension behavior and its characteristics. It is supposed that considering only vertical ground excitation would give usable results. Eventual longitudinal and lateral excitations that may arise during vehicle movement are neglected.

Additional results to the accelerations, forces and displacements are the diagrams that represent the sensitivity of those responses in dependence to the input geometrical and other parameters. Those diagrams can help the designer to see the influence of chosen parameters on the driving comfort and driving safety. With this in his mind can he faster and reliable make following modifications.

#### MATHEMATICAL MODELING OF THE VEHICLE SUSPENSION SYSTEM

Simplified QVM consists from two masses, tire stiffness coefficient, coil spring stiffness coefficient and shock absorber damping coefficient, all except shock absorber sequential connected, as shown in Figure 1. Lower mass is so called un-sprung mass  $m_u$ , while upper one is so called sprung mass  $m_s$ . The un-sprung mass represents approximately one wheel mass, while sprung mass represents approximately  $\frac{1}{4}$  of the rest total vehicle mass. By the coil lower spring the tire stiffness  $k_t$ , is directly presented. In the reality parallel to the tire stiffness acting also tire damping that is common neglected and it is done in this case also because of its unimportance on final results, Jazar [7]. By the coil spring and the shock absorber schematically inserted between two masses the completely suspension system is represented. The spring stiffness and shock absorber damping coefficients are denoted by  $k_s$  and  $c_d$ . They should be properly transformed. The lower or un-sprung mass is excited by the ground surface through the tire. In the continuation only the simplified QVM is to be treated. For the presented QVM the differential equations of motions are derived by applying the Lagrange's equations:

$$\frac{d}{dt} \left( \frac{\partial E_k}{\partial \dot{z}_u} \right) - \frac{\partial E_k}{\partial z_u} = \frac{\delta W}{\delta z_u}; \quad \frac{d}{dt} \left( \frac{\partial E_k}{\partial \dot{z}_s} \right) - \frac{\partial E_k}{\partial z_s} = \frac{\delta W}{\delta z_s} \quad (1)$$

The kinetic energy  $E_k$ , is determined by the following expression:

$$E_k = \frac{1}{2} \cdot m_u \cdot \dot{z}_u^2 + \frac{1}{2} \cdot m_s \cdot \dot{z}_s^2 \quad (2)$$

The derivatives of the Lagrange's equations can be written as follow:

$$\frac{d}{dt} \left( \frac{\partial E_k}{\partial \dot{z}_u} \right) = m_u \cdot \ddot{z}_u, \quad \frac{\partial E_k}{\partial z_u} = 0; \quad \frac{d}{dt} \left( \frac{\partial E_k}{\partial \dot{z}_s} \right) = m_s \cdot \ddot{z}_s, \quad \frac{\partial E_k}{\partial z_s} = 0 \quad (3)$$

The elementary work  $\delta W$  is given by the following expression:

$$\delta W = (F_{tire} - F_s - F_d) \cdot \delta z_u + (F_s + F_d) \cdot \delta z_s \quad (4)$$

The elementary work during a virtual displacement of un-sprung mass  $\delta z_u$ , and sprung mass  $\delta z_s$ , are expressed, as follows:

$$\frac{\delta W}{\delta z_u} = F_{tire} - F_s - F_d; \quad \frac{\delta W}{\delta z_s} = F_s + F_d \quad (5)$$

Forces that act in the suspension system, such as dynamic tire force  $F_{tire}$ , spring force  $F_s$ , and damping force  $F_d$ , respectively, are determined by the following expressions:

$$F_{tire} = k_t \cdot (z_r - z_u) \quad (6)$$

$$F_s = k_s \cdot (z_u - z_s) \quad (7)$$

$$F_d = c_d \cdot (\dot{z}_u - \dot{z}_s) \quad (8)$$

By few mathematical rearrangements and substituting equations (3), (5), (6), (7) and (8) the Lagrange's equations, equation (1) generate the following differential equations of motion:

$$m_u \cdot \ddot{z}_u = k_t \cdot (z_r - z_u) - k_s \cdot (z_u - z_s) - c_d \cdot (\dot{z}_u - \dot{z}_s) \quad (9)$$

$$m_s \cdot \ddot{z}_s = k_s \cdot (z_u - z_s) + c_d \cdot (\dot{z}_u - \dot{z}_s)$$

where  $\ddot{z}_u$ ,  $\dot{z}_u$  and  $z_u$  denote acceleration, velocity and displacement of the un-sprung mass, while  $\ddot{z}_s$ ,  $\dot{z}_s$  and  $z_s$  denote acceleration, velocity and displacement of the sprung mass.

The differential equations of motion in a matrix form are expressed, as follow:

$$\begin{bmatrix} m_u & 0 \\ 0 & m_s \end{bmatrix} \cdot \begin{Bmatrix} \ddot{z}_u \\ \ddot{z}_s \end{Bmatrix} + \begin{bmatrix} c_d & -c_d \\ -c_d & c_d \end{bmatrix} \cdot \begin{Bmatrix} \dot{z}_u \\ \dot{z}_s \end{Bmatrix} + \begin{bmatrix} k_s + k_t & -k_s \\ -k_s & k_s \end{bmatrix} \cdot \begin{Bmatrix} z_u \\ z_s \end{Bmatrix} = \begin{Bmatrix} k_t \cdot z_r \\ 0 \end{Bmatrix} \quad (10)$$

Equation (9) represents a system of the second order non-homogeneous linear differential equations of the motion, whereas equation (10) represents general matrix form of the linear multi-degree of freedom vibration system, Happian-Smith [8].

### CONCEIVING THE PROGRAM FOR ANALYSIS OF THE VEHICLE SUSPENSION

In order to solve the differential equations of motion for general case of ground excitation it is more than necessary to have efficient program that speed the numerical calculations. To build the proper program the differential equations of motion should be transformed into the suitable form. Equation (9) is transformed into the state variable equations to simplify the second order differential equation into the first order differential equations. It is supposed that the state space variables are given by the following expressions:

$$\begin{aligned} x(1) &= z_u, x(2) = z_s \\ x(3) &= \dot{z}_u, x(4) = \dot{z}_s \\ dx(1) &= x(3), dx(2) = x(4) \\ dx(3) &= \ddot{z}_u, dx(4) = \ddot{z}_s \end{aligned} \quad (11)$$

By substituting equation (11) into equation (9), the space state equations are written, as follow:

$$\begin{aligned} dx(1) &= x(3) \\ dx(2) &= x(4) \\ dx(3) &= -\frac{k_s + k_t}{m_u} \cdot x(1) + \frac{k_s}{m_u} \cdot x(2) - \frac{c_d}{m_u} \cdot x(3) + \frac{c_d}{m_u} \cdot x(4) + \frac{k_t}{m_u} \cdot z_r \\ dx(4) &= \frac{k_s}{m_s} \cdot x(1) - \frac{k_s}{m_s} \cdot x(2) + \frac{c_d}{m_s} \cdot x(3) - \frac{c_d}{m_s} \cdot x(4) \end{aligned} \quad (12)$$

Equation (12), present general form of the state space equation of the QVM, while in matrix form can be written, as follow:

$$\dot{x}(t) = A \cdot x(t) + B \cdot u(t), \quad (13)$$

where  $x(t)$  is termed the state vector and it contains four state variables. Matrix A denotes the state matrix, while matrix B is the input vector of the ground excitation  $u(t)$ . They are derived by the following expressions:

$$\begin{aligned} x(t) &= \{x(1) \ x(2) \ x(3) \ x(4)\}^T \\ \dot{x}(t) &= \{dx(1) \ dx(2) \ dx(3) \ dx(4)\}^T \\ A &= \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_s + k_t}{m_u} & \frac{k_s}{m_u} & -\frac{c_d}{m_u} & \frac{c_d}{m_u} \\ \frac{k_s}{m_s} & -\frac{k_s}{m_s} & \frac{c_d}{m_s} & -\frac{c_d}{m_s} \end{bmatrix}, \quad B = \begin{bmatrix} 0 \\ 0 \\ \frac{k_t}{m_u} \\ 0 \end{bmatrix}, \quad u(t) = \begin{Bmatrix} 0 \\ 0 \\ z_r \\ 0 \end{Bmatrix} \end{aligned} \quad (14)$$

Program for solving equation of motion is performed in Matlab environment. The flowchart diagram is presented in Figure 2. The Program starts by reading the suspension parameters, ground excitation, number of input and output variables, initial conditions and number of iteration (time). After it the Program continues to solve differential equations of motion by applying Runge-Kutta method. The Program is stopped when iteration condition is fulfilled and results could be presented by diagrams and numerical values.

The output results in dependence of time are accelerations, velocities and displacements of sprung and un-sprung masses. The Program also computes vertical tire forces between ground surface

and tire and suspension travel. The output results are depended from ground excitation. The ground excitation represents the irregularity of the road profile, called road bumps, Figure 3.

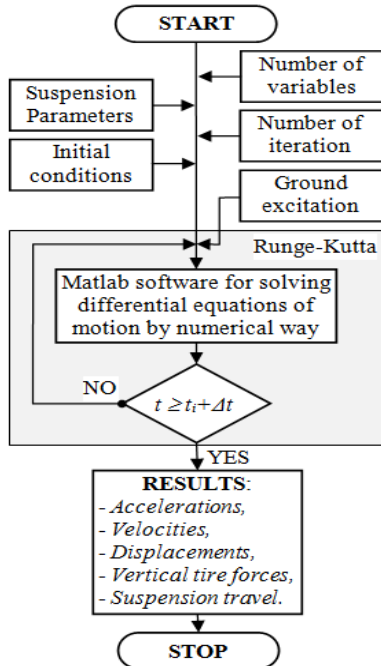


Figure 2. Flowchart diagram of the program for solving the differential equations of the motion

The ground excitation, shown in Figure 3, represents smooth obstacles on flat road and is approximated by a smooth function like cosine. Expression for simulation of ground excitation  $z_{ri}(t)$ , is written, as follow:

$$z_{ri}(t) = \begin{cases} \frac{1}{2} H_1 [1 - \cos(b_i \pi \cdot t)] & \text{if } t_1 \leq t \leq t_2 \\ \frac{1}{2} H_2 [1 - \cos(b_i \pi \cdot t)] & \text{if } t_3 \leq t \leq t_4 \\ 0 & \text{else} \end{cases} \quad (15)$$

where,  $H_1, H_2$  are amplitudes of the first and second bumps.

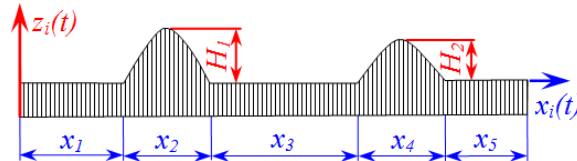


Figure 3. Cross section of the road profile represented by double cosine road bumps

Several authors, like Rill [9], Bolandhemmat [10], and Sam et al. [11] have using similar function for ground excitation, as it is described here. Therefore, road bumps introduced by cosine function equation (15), could be considered as one of the complex road inputs.

**THE PROGRAM VERIFICATION**

To get the comparable value of just presented computer Program, the result comparison by simply commercial Working Model Software [12], is made. Both are run. Some input data are given in Table 1. These characteristics are taken from real terrain vehicle, which is in the develop stage yet, Lajqi [13]. Graphic model of the vehicle suspension system is made into Working Model (WM) environment, Figure 4, left side. The data correspond to one explained in Table 1. The WM environment mimics the “Suspension Test Rig”. Because of the WM limitations the ground excitation is assumed to be pure sine in nature. The ground excitation is theoretical simulated by installation of an hydraulic actuator which produces vertical motion corresponds to the sine function:  $z_r(t)=0.1 \cdot \sin(2t)$ .

Table 1. Parameters for a quarter terrain vehicle suspension, Lajqi (2012)

Suspension parameters	Unit	Symbol	Value
Spring stiffness coefficient	$[Nm^{-1}]$	$k_s$	55227
Damping coefficient	$[Nsm^{-1}]$	$c_d$	5230
Tire stiffness coefficient	$[Nm^{-1}]$	$k_t$	201441
Sprung mass	$[kg]$	$m_s$	295
Un-sprung mass	$[kg]$	$m_u$	39

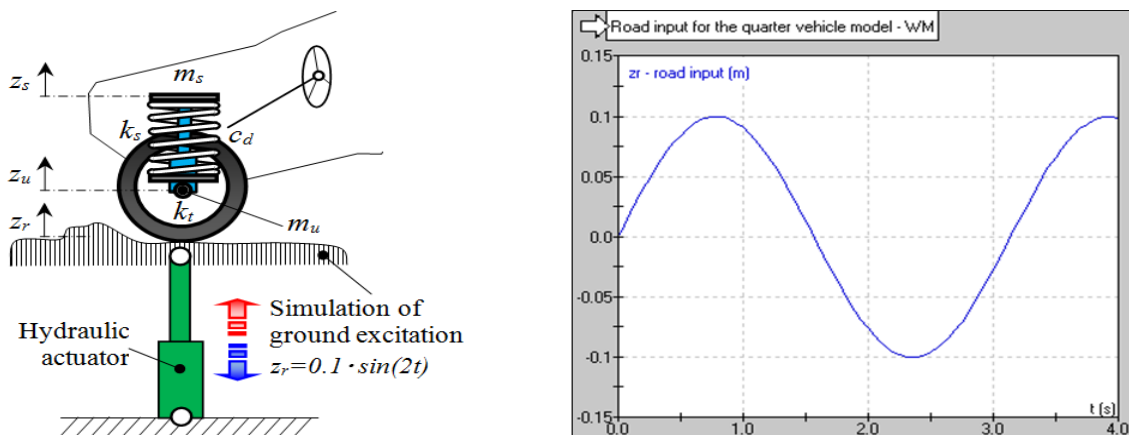


Figure 4. “Suspension Test Rig” and simulation of the ground excitation in a function of time performed by the Working Model

In the Figure 5 it is shown the vertical displacements of sprung and un-sprung masses as a function of time, obtained by WM and MATLAB platforms, respectively. It can be conclude that the results ensured by Program coincide very well with the ones ensured by simple commercial and much more comprehensive WM.

Figure 6, shows the results in form of vertical velocities of the sprung and un-sprung masses as a function of time, performed by WM and MATLAB, respectively. It is obvious that the Program generates satisfactory precise and reliable results as they are compared by WM results.

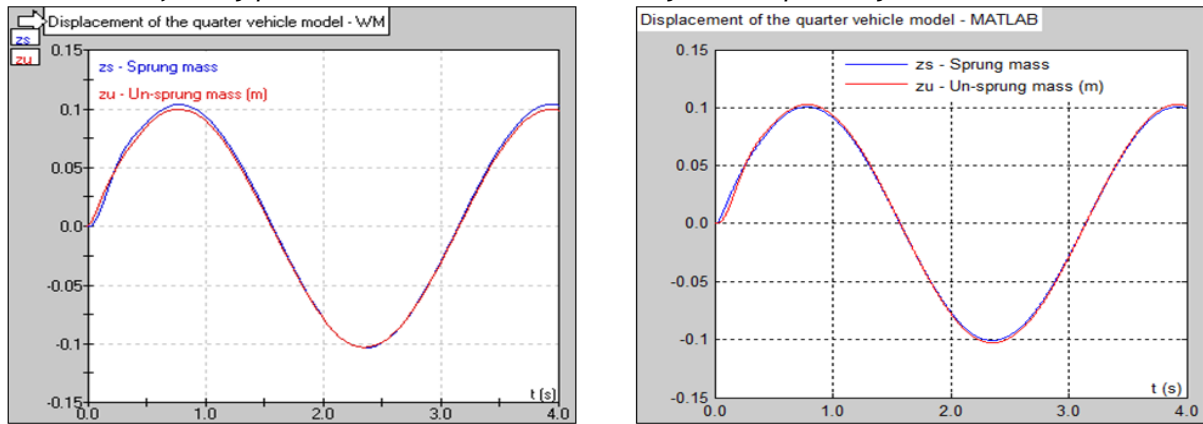


Figure 5. Calculation of the vertical displacements of the sprung and un-sprung masses, performed by WM and MATLAB platforms

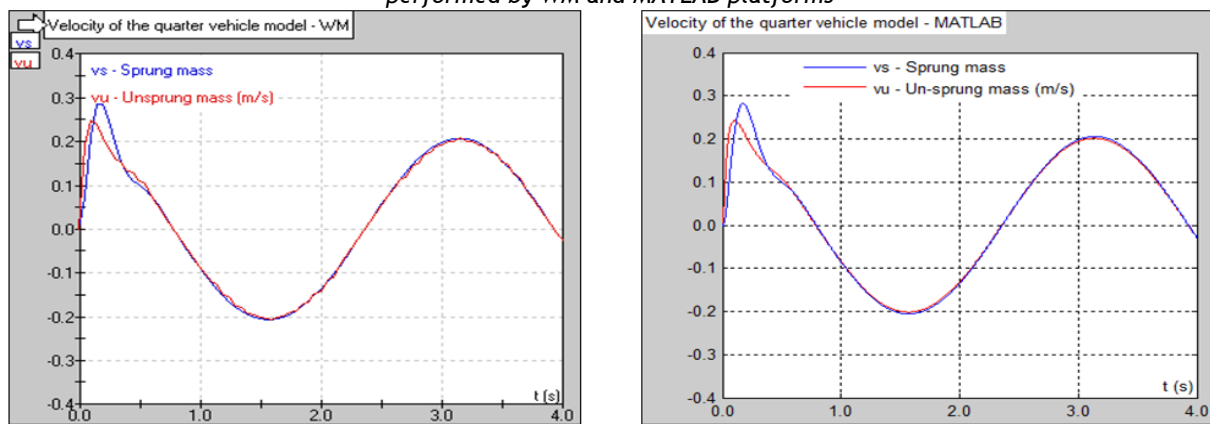


Figure 6. Vertical velocities for sprung and un-sprung masses, performed by WM and MATLAB platforms

Consequently it can be concluded that the developed Program is suitable for use in further analysis of the vehicle suspension behavior.

**USING THE PROGRAM FOR THE TERRAIN VEHICLE SUSPENSION DESIGN**

Before the comprehensive suspension calculations the initial terrain vehicle design is prepared. In order to improve it few numerical attempts is expected. Flowchart diagram is shown in Figure 7 where it is explained in details how the Program calculates accelerations, forces and displacements in dependence of time and caused by ground excitation (Figure 8). The vehicle is run over the road bumps by three different speeds.

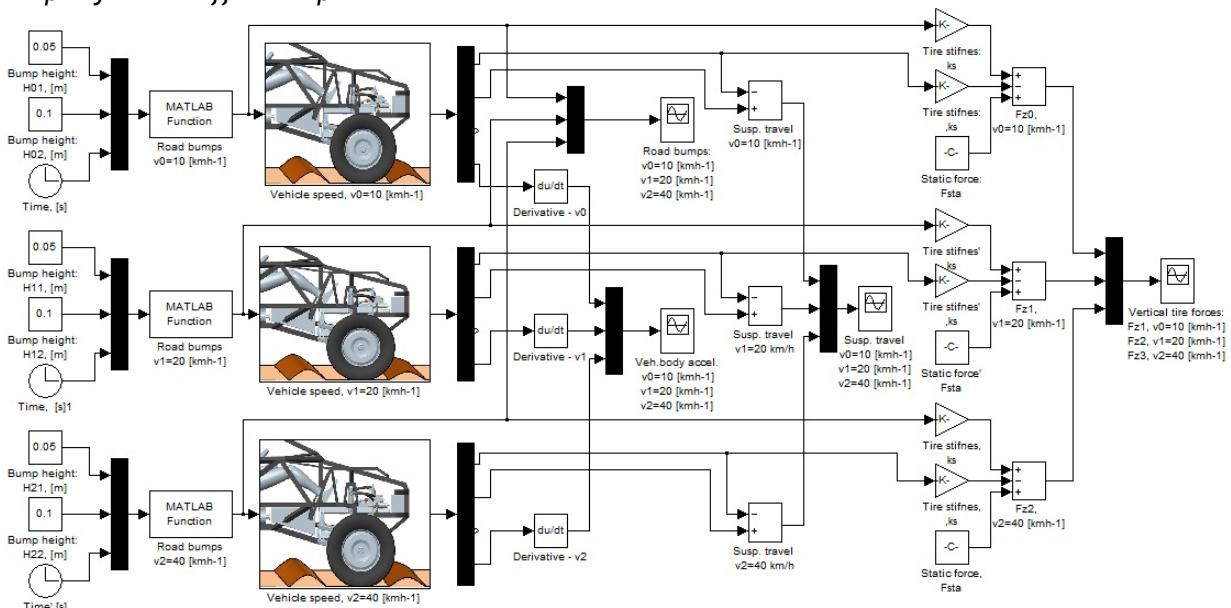


Figure 7. Flowchart diagram for calculation of accelerations, forces and displacement obtained by three different vehicle speeds

Table 2. Data for calculation of the road bumps when vehicle moves by speeds 10, 20 and 40 [kmh<sup>-1</sup>]

Road data			Cosine road bumps, $H_1=0.05$ [m], $H_2=0.1$ [m]		
j	$x_j$ , [m]	$L_j$ , [m]	For $v_0=10$ [kmh <sup>-1</sup> ],	For $v_1=20$ [kmh <sup>-1</sup> ],	For $v_2=40$ [kmh <sup>-1</sup> ],
			$t_j=L_j/v_0$ , [s]	$t_j=L_j/v_1$ , [s]	$t_j=L_j/v_2$ , [s]
1	11.11	11.11	4.00	2.00	1.00
2	1.39	12.50	4.50	2.25	1.125
3	11.11	23.61	8.50	4.25	2.125
4	1.39	25.00	9.00	4.50	2.25
5	2.78	27.78	10.00	5.00	2.50

To describe the vehicle behavior during it runs over road bumps by different speed the ground excitation, equation (22) is written as  $z_{ri}(t)$ . Index  $i$  is to be 0, 1 and 2. When it is  $i=0$  the vehicle speed is  $v_0=10$  [kmh<sup>-1</sup>] and cosine parameters is  $b_0=4$ . When it is  $i=1$  the vehicle speed is  $v_1=20$  [kmh<sup>-1</sup>] and cosine parameters is  $b_1=8$ . And finally, when it is  $i=2$  then the vehicle speed is  $v_2=40$  [kmh<sup>-1</sup>] and cosine parameters is  $b_2=16$ .

Driving comfort and safety are closely related to accelerations and displacements of the vehicle tire and vehicle body, Mastinu et al. [14] and Popp et al. [15]. The driving comfort is determined by the vehicle body acceleration (Figure 9), where the greater values are not desired. The driving safety is related to the vertical tire forces acting between ground surface and the tire (Figure 10). If the tire forces oscillates too much then the contact between the tire and ground surface is weak. Consequential the vehicle maneuverability and steering is in question. The suspension travel is defined as relative displacement between sprung and un-sprung masses. Both, the vertical tire forces and suspension travel are strictly related to the active safety, Mastinu et al. [14].

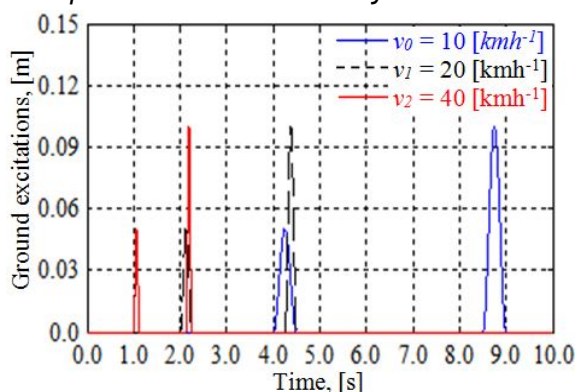


Figure 8. Double cosine road bumps in a function of time for different vehicle speeds

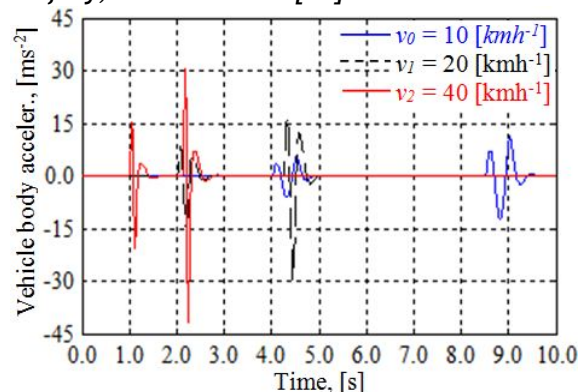


Figure 9. Vertical vehicle body acceleration in a function of time for different vehicle speeds

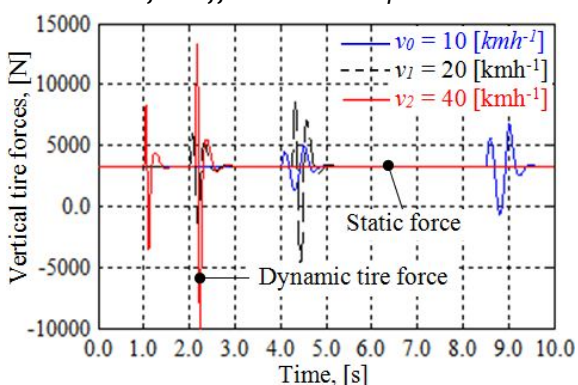


Figure 10. Vertical tire forces in a function of time for different vehicle speeds

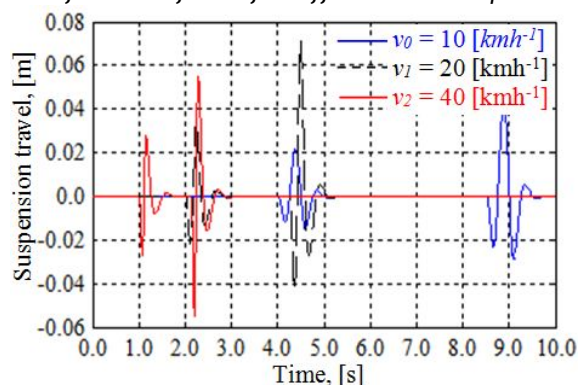


Figure 11. Suspension travel in a function of time for different vehicle speeds

Figure 8 shows a ground excitation derived from cosine function with amplitude  $H_1=0.05$  [m] and  $H_2=0.1$  [m] for three different vehicle speeds. Figure 9 shows vehicle body accelerations in dependence of time for different vehicle speeds: 10, 20 and 40 [kmh<sup>-1</sup>], respectively. Figure 10 shows vertical tire forces in dependence of time for different vehicle speeds.

The vertical tire forces are composed from static and dynamic ones. That is given by the following expression:

$$F_z(t) = F_{sta} + F_{tire} = g \cdot (m_s + m_u) + k_t \cdot [z_i(t) - z_u(t)] \quad (16)$$

where,  $F_z(t)$  denotes vertical tire forces,  $F_{sta}$  is static force and  $F_{tire}$  is termed dynamic tire force.

Figure 11 presents suspension travel in dependence of time for different vehicle speeds. The suspension travel is determined by the following expression:

$$s_{trav}(t) = z_s(t) - z_u(t), \quad (17)$$

where,  $s_{trav}(t)$  denotes suspension travel (Figure 11).

To analyze the suspensions parameters that contribute in the vehicle comfort, vehicle safety and suspensions travel it is necessary to varying these parameters. The lower and upper limit of the suspension parameters are given in Table 3. Further analyses are made for cases where the vehicle moves over the double cosine road bumps with speed 10 [kmh<sup>-1</sup>].

Figures 12 to 14 show the influence of the suspension parameters sensitivity. The vehicle body acceleration, vertical tire forces and suspension travel is presented for all parameter limits.

Table 3. Reference, lower and upper limits values of the suspension parameters

Suspension parameters	Unit	Reference data		Lower and upper limits	
		Symbol	Values	Symbol	Values
Spring stiffness coefficient	[Nm <sup>-1</sup> ]	$k_s$	55227	$k_{sr}$	35000 ... 120000
Damping coefficient	[Nsm <sup>-1</sup> ]	$c_d$	5230	$c_{dr}$	1000 ... 8000
Tire stiffness coefficient	[Nm <sup>-1</sup> ]	$k_t$	201441	$k_{tr}$	120000 ... 320000
Sprung mass	[kg]	$m_s$	295	$m_{sr}$	200 ... 350
Un-sprung mass	[kg]	$m_u$	39	$m_{ur}$	30 ... 59

**DISCUSSION OF THE RESULTS**

To evaluate the terrain vehicle suspension the three vehicle speeds are considered: 10, 20 and 40 [kmh<sup>-1</sup>]. Figure 8 shows the ground excitation in dependence of time. The presented speeds are chosen to get the distribution where the medium value is twice as much as the lowest one and the medium is only the half of the upper one at once. This helps to better understand the values of the vehicle body acceleration, vertical tire force and suspension travel.

When vehicle moves with speed 20 and 40 [kmh<sup>-1</sup>] (Figure 9 and Figure 10) it can be observed that the vehicle body acceleration and dynamic tire forces in the first peak is increased for 2.36 and 4.36 times, on the second peak for 1.55 and 3.4 times, while the third peak is increase for 10% more compared when vehicle moves over the roads bumps by speed 10 [kmh<sup>-1</sup>].

Figure 11 shows that the suspension travel gets it highest values at speed 20 [kmh<sup>-1</sup>] comparable with the one at speed 40 [kmh<sup>-1</sup>], while at 10 [kmh<sup>-1</sup>] the suspension travel is much lower.

When the terrain vehicle runs over the road bumps at lower speeds (bellow 10 [kmh<sup>-1</sup>]) the driving comfort and the driving safety are acceptable. At higher speeds arise higher acceleration and dynamic tire forces, up to 4 g, what could cause the significant decreasing of the tire contact with the ground surface.

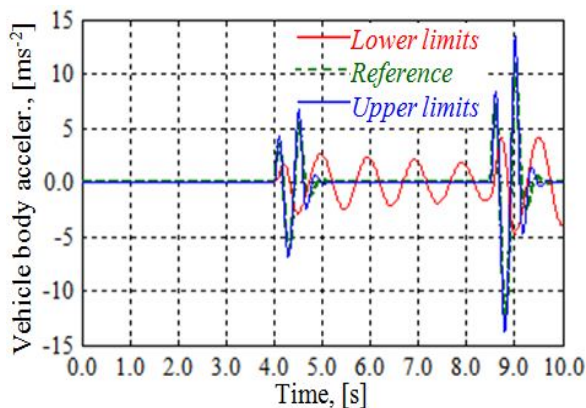


Figure 12. Vertical vehicle body acceleration in a function of time by varying the suspension parameters

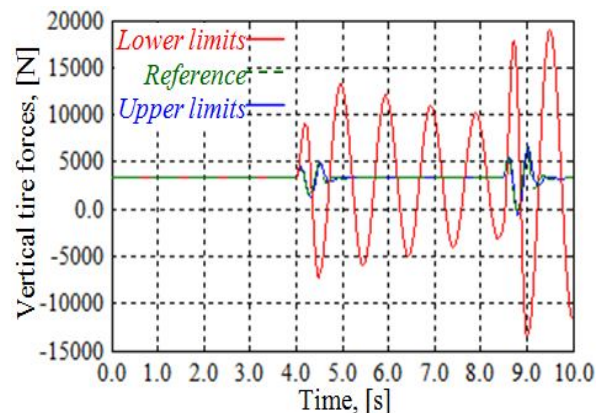


Figure 13. Vertical tire forces in a function of time by varying the suspension parameters

From Figures 12 to 14 it can be concluded that the lower values of parameters provide better driving comfort but in this case it is not possible to isolate the vibration caused by road bumps from the vehicle body. The vehicle body undammed vibrates long time after excitation.

Consequently the dynamic tire forces are greater up to 3 times compare by the reference ones. At lower limits of the parameters immediately after the road bumps the suspension travels stronger and it doesn't become quitter enough on the flat terrain. This situation is to be arising in case when the shock absorber becomes weak. In this situation contact with the ground is reduced and the driving safety especially on uneven or bumpy roads is limited. The reference

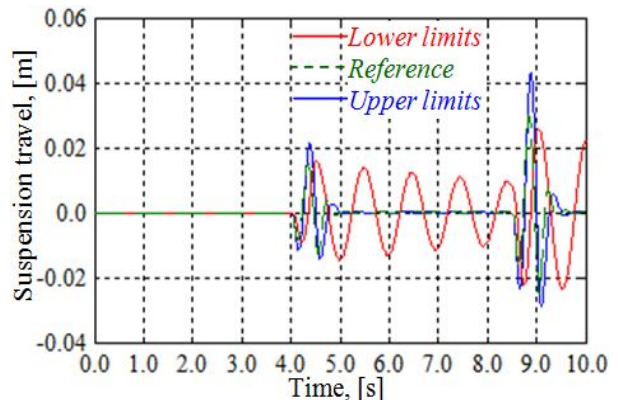


Figure 14. Suspension travel in a function of time by varying the suspension parameters

and upper values of the suspension parameters provide the better performances by sufficient isolation of vibration that is caused by ground excitation and they lead much closer to optimal suspension design.

### CONCLUSION AND RECOMMENDATIONS

A methodology for building the effective methodology for evaluation of the vehicle suspension performances has been presented. The simplified quarter vehicles model is analyzed. To verify the proposed numerical solving of differential equations of motion in Matlab environment the basic calculation of two-mass system are performed in Working Model. Real road bumps is modeled by few cosine bumps and flat lines that is proved to be effective. Analysis of the suspension parameters is performed by estimations of the driving comfort, the driving safety and the suspension travel. Results are presented in diagrams. Each diagram is obtained by varying lower, reference and upper limits of the essential suspension parameter. The higher values of spring stiffness and damping coefficients are not desire, while smaller values can improve the comfort, considerably. However, soft spring and weak damping result in extensive vibrations from equilibrium position and vehicle becomes unstable.

The obtained results confirm that, the reference values of the suspension parameters are close to optimum design values. This means that passive suspension system gives sufficient driving comfort, driving safety and low suspension travel.

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