UDC: 629.33.027.4:004.942
Conference Paper

PERFORMANCE ANALYSIS OF PASSIVE, SEMI-ACTIVE AND ACTIVE-CONTROLLED SUSPENSION SYSTEMS USING MATLAB/SIMULINK

Nexhbedin Ismaili

Department of Mechatronics, Faculty of Applied Sciences, University of Tetova, RNM
Corresponding author e-mail: nexhbedin.ismaili@unite.edu.mk

Abstract

Knowing the fact that the suspension system is the only connecting point between the vehicle and the rarely uniform road, we strongly believe that the suspension system is the most significant part of extraordinary importance in providing comfort and driving safety against specific road conditions and situations. Therefore, the purpose of this research was to study the performance of three different suspension systems: passive, semi-active and active.

To analyze the specific parameters and to compare the performance results of three types of suspension systems, the representative ¼ model of the vehicle, represented as a two degree of freedom system, is modeled and simulated in the MATLAB/Simulink environment. Differential motion equations describing the three systems are subjected to software modeling and simulation, providing information on the behavior of these systems for the same parameters, under the same conditions.

The comparison of the numerical and graphical results between the three systems proved that the active suspension systems, controlled by a controller, offer significantly better performance than the previous systems in terms of response time (by tuning, in order to achieve proper rise time, overshoot, settling time and steady-state error) and adjustment of system suspension as well as reduction to dissipation of vertical oscillations, roll and pitch movements, which was also the aim of this research.

Keywords: Control systems, feedback controller, modeling, simulation, suspension

1. Introduction

Because of the non-flat road surfaces, the suspension systems are a must and at the same time are key elements of the design of any vehicle. The necessity of using these systems stems mainly from the appearance of numerous destabilizing oscillations, which impose the need to increase safety and comfort while traveling. The vehicle's suspension system is the only component that serves as the only contact point between the vehicle and the road. For this reason, the suspension system must provide proper driving control in terms of regular wheel positioning and ride quality in terms of passenger comfort and avoiding fatigue. The fulfillment of these functions is very important to maintain the stability and safety of the vehicle as well as the driver's comfort. If the suspension system allows excessive vertical oscillations of the wheels, the driver may lose control when the vehicle is moving on an irregular surface road. This loss of steering control can result in a vehicle crash and, in the most unwanted cases, in injuries (Knowles, 2007).

Based on the principles of operation and the types of components used to suppress oscillations, there are three main types of suspension systems:

- Passive suspension systems (most common)
- Semi-active suspension, and
- Active suspension.

Several models can be used to investigate the behavior and operation of these systems:

- The quarter vehicle model - only one suspension center on one of the four wheels is explored.
- Half vehicle model – two suspension centers on two wheels are explored, and
- Full vehicle model – all of four suspension centers on each wheel are explored. (Zhou, 2013)

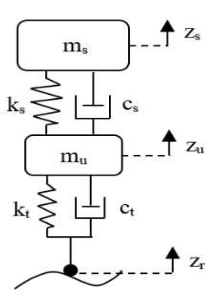
In this research, all three types of suspension systems are modeled and simulated in accordance with the quarter vehicle model. Due to the limited space, only the displacement of the sprung mass and the difference between the sprung and unsprung mass are presented in this paper.

2. Modeling and simulation of passive suspension system

In order to be able to construct representative block diagrams of the model in Simulink, we first need to obtain the differential equations describing the model we have chosen (¼ of the vehicle). The vehicle's suspension system consists of ¼ of the vehicle's body mass, shock absorbers (a spring and a damper) and a wheel.

2.1. Mathematical modeling of the vehicle suspension system

By combining the representative values of the masses *m*, the stiffness coefficients *k* and the coefficients of the quenching components *c*, it is possible to create a model for each type of vehicle. Fig. 2 shows the schematic of the model for ¼ of the vehicle.



m_s – ¼ of the vehicle's mass that relies on the suspension system
m_u – unsprung mass (one-wheel assembly mass)
k_s – stiffness coefficient of the spring
k_t – tire stiffness coefficient
c_s – damper coefficient
c_t – tire damping coefficient
z_s – vertical displacement of the sprung mass
z_u – vertical displacement of the unsprung mass
z_r – road displacement

Figure 1. Model of the passive vehicle suspension

The mathematical model of this passive suspension system is constructed based on the equations of motion, which will be developed by referring to Newton's Second Law for each of the measses.

The development of the mathematical model is based on three basic processes:

1. Coordinates (x_1, x_2) are defined in their respective direction. Let $x_1 = 0$, $x_2 = 0$ when system is in static equilibrium.
2. We divide the elements of inertia to analyze the forces acting on them:

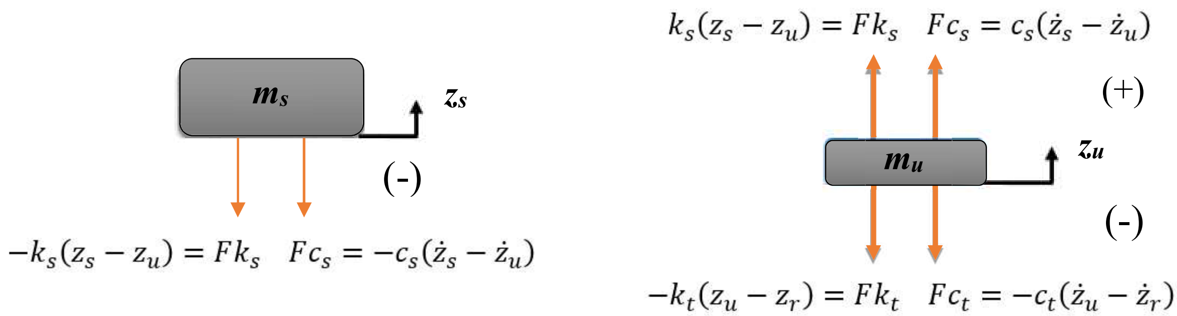


Figure 2. Schematics of the inertias and their respective forces

3 Based on Newton's laws, we formulate the system's equations of motion, which represent second-order differential equations and they represent the mathematical model of the system. The equations are obtained by analyzing the mutual actions of the input and output forces on the inertia components of our system:

- For sprung mass $\boldsymbol{m_s}$ we have:

$$\sum F = m * a \tag{1}$$

$$\sum F = -Fk_s - Fc_s = m_s \ddot{z}_s \tag{2}$$

$$m_s \ddot{z}_s = -k_s(z_s - z_u) - c_s(\dot{z}_s - \dot{z}_u) \tag{3}$$

- For unsprung mass $\boldsymbol{m_u}$ we have:

$$\Sigma F = Fk_s + Fc_s - Fk_t - Fc_t = m_u \ddot{z}_u \tag{4}$$

$$m_u \ddot{z}_u = k_s(z_s - z_u) + c_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - c_t(\dot{z}_u - \dot{z}_r) \tag{5}$$

Equations (3) and (5) are second-order differential equations that describe a passive suspension system. Solving these equations is a complex process that requires a lot of computational effort and precision, which also takes a long time to accomplish. For these reasons, the solution to these equations can be accomplished with the help of MATLAB computing software.

2.2. Modeling and simulation using MATLAB/Simulink

The modeling is performed in the Simulink workspace, from whose library are selected specific blocks that fit the requirements of the system being investigated and interconnection is done between them to create a functional and representative entirety of the system. In other words, Simulink is an environment that enables modeling, simulation and graphical analysis of dynamic systems and their behavior in terms of time and frequency domain (Figure 4).

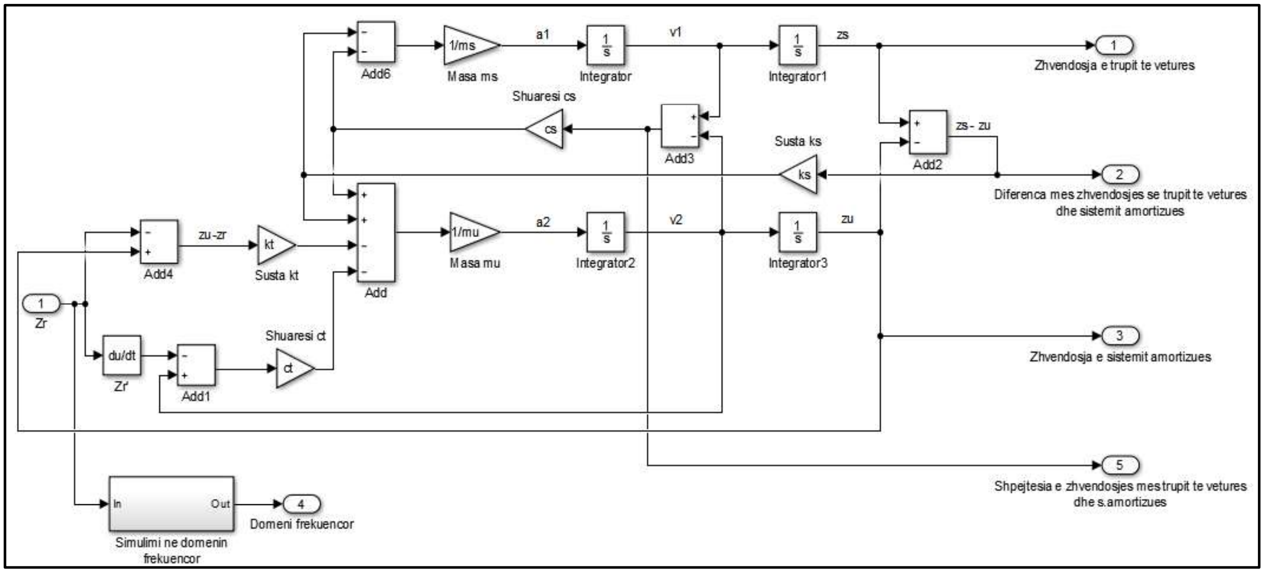


Figure 3. Model of passive ¼ vehicle suspension developed in Simulink environment

The model in Figure 4 is ready to be simulated and to analyze the results, namely the response of the system to the input parameters. The input parameters are given in Table 1.

Table 1. Suspension parameters used in the simulation

Parameter name	Parameter notation	Parameter value	Unit
Sprung mass	(m_s)	450	[Kg]
Unsprung mass	(m_u)	50	[Kg]
Spring stiffness	(k_s)	5800	[N/m]
Damping coefficient of the sprung mass	(c_s)	300	[Ns/m]
Tire stiffness	(k_t)	140000	[N/m]
Damping coefficient of unsprung mass	(c_t)	1500	[Ns/m]
Simulation time	*(t)*	20	[s]

In the following, we will present the simulation results according to the input parameters shown in Table 1.

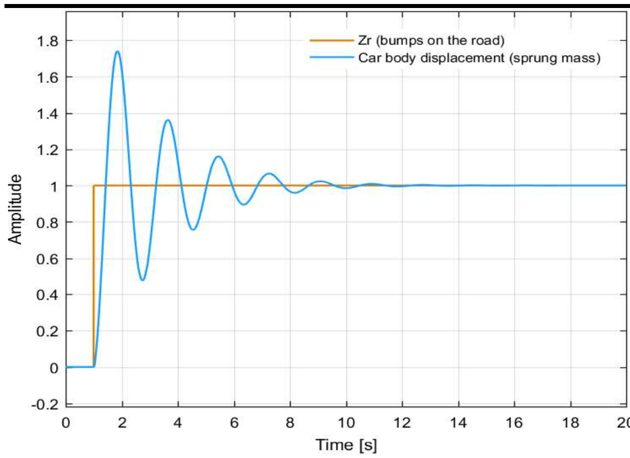


Figure 4. Sprung mass displacement through time

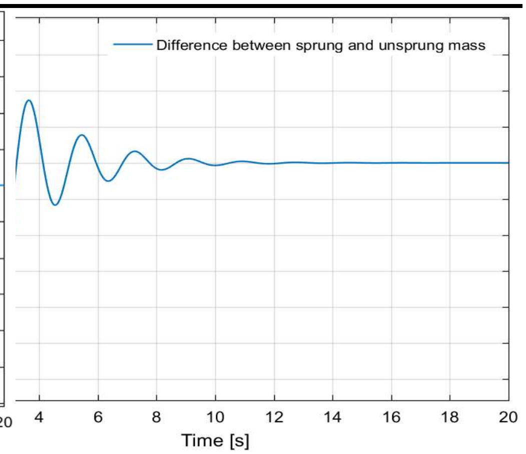


Figure 5. The difference between sprg. & unsprg. mass displacement

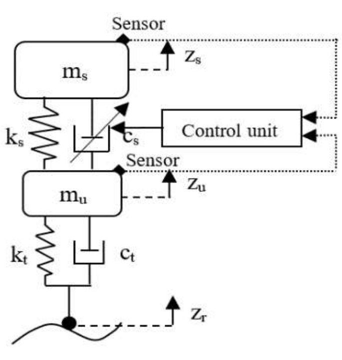
3. Modeling and simulation of the semi-active suspension system

Unlike passive suspension systems, semi-active systems in addition to the usual components such as springs and damper also contain regulators and sensors that form the feedback link. These systems manage to influence and even control the system by modifying the damper state for a very fast time. The regulator, through the information received by the sensors, generates a certain voltage to the damper, causing it to change and adapt to irregularities on the road.

Semi-active suspension systems are characterized by interesting features, such as:

- Almost negligible energy demand,
- More security than passive suspension,
- Low cost and weight, and
- Significant impact on the vehicle (Savaresi, Poussot-Vassal, Spelta, Sename, & Dugard, 2010).

3.1. Mathematical model of the semi-active suspension system



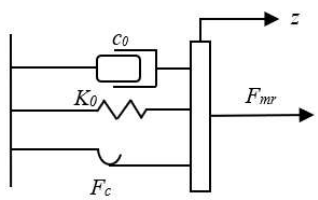
C_s – variable coefficient of the damper
Control unit – an electronic device that changes the damper's coefficient
Sensors - Signal providers on the position and speed of the vehicle components.

Figure 6. Model of the semi-active suspension system

The damper used in this research is of a magnetorheological type, filled with a magnetorheological fluid, which is controlled through a magnetic field. The fluid consists of micrometrical iron particles, which are influenced by the magnetic field and become lined up in parallel directions, creating high resistance of the damper with very little power source (Wikipedia, 2019).

There are several models to represent the magnetorheological damper, including:

- Bingham model (used in this research)
- LuGre
- Dahl and
- Bouc-Wen



Fmr – the controlling force of the model
Fc – frictional force
C_0 – damping constant
K_0 – stiffness of an elastic component
F_0 – offset force
z(t) – sprung mass displacement
z'(t) – velocity of sprung mass displacement

Figure 7. Bingham's model and its descriptive equation.

$$F_{mr} = \frac{2F_C \tan^{-1}(d \cdot \dot{z})}{\pi} + c_0\dot{z} + K_0 z + F_0 \quad (6)$$

Differential equations of the mathematical model:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - c_s(\dot{z}_s - \dot{z}_u) - F_{mr} \quad (7)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + c_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - c_t(\dot{z}_u - \dot{z}_r) + F_{mr} \quad (8)$$

3.2. Modeling and simulation using MATLAB/Simulink

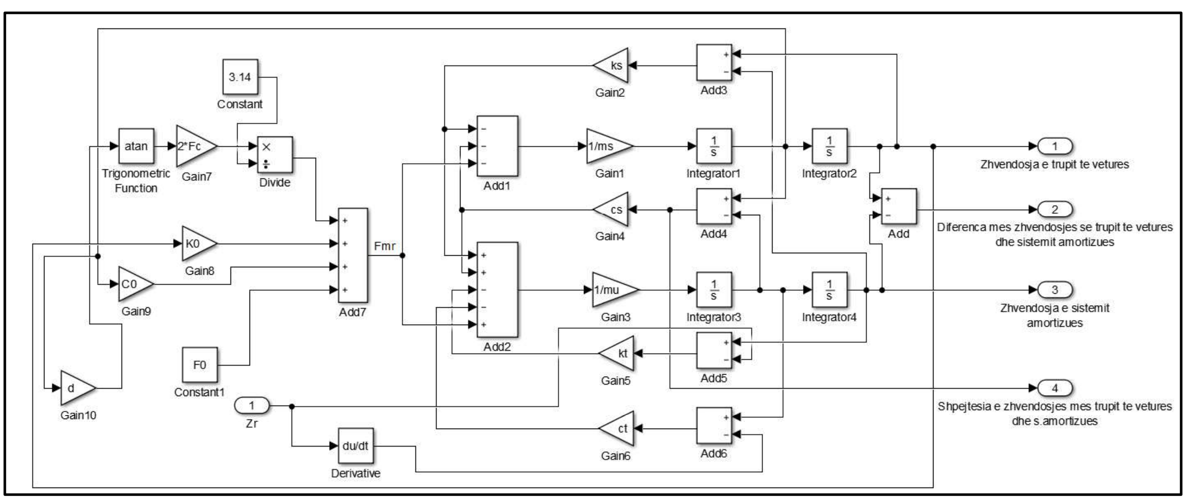
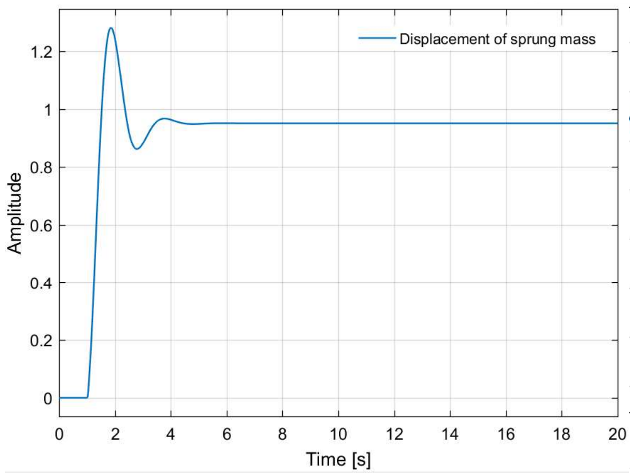


Figure 8. Simulink model of the semi-active suspension system

Table 2. Simulation input parameters for Bingham model (Eshkabilov., 2016)

Parameter name	Parameter notation	Parameter value	Unit
Damping coefficient of the Bingham model	(c_0)	650	[Ns/m]
Offset force	(f_0)	0	[N]
Frictional force	(Fc)	210	[N]
Stiffness of an elastic component	(K_0)	300	[N/m]
Form factor	(d)	[5, 10, 20, 30, 60]	

After performing the simulation in Simulink, the following results are obtained:



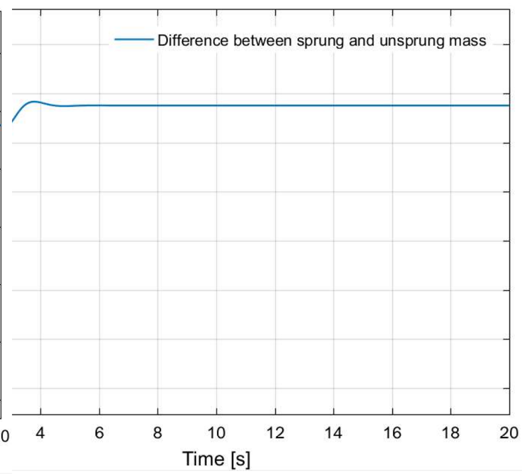


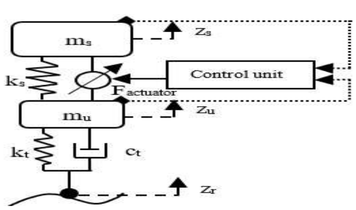
Figure 9. Sprung mass displacement through time

Figure 10. The difference between sprung and unsprung mass

4. Modeling and simulation of active suspension system

Unlike passive and semi-active systems, active suspension systems contain actuator, which generates sufficient force to lift the car body to non-flat surfaces of the road. It is currently the most advanced type of suspension system and offers exceptional stability and maneuverability. It is also a system that requires more energy and is characterized by heavy weight and high financial cost.

4.1. Mathematical model of active suspension system



$F_{actuator}$ – the force generated by the regulator to the actuator (to increase or decrease the vehicle's body position).

Figure 11. Model of the active suspension system

The differential equations that describe the system and form the mathematical model are:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - c_s(\dot{z}_s - \dot{z}_u) + Fa \quad (9)$$
$$m_u\ddot{z}_u = k_s(z_s - z_u) + c_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - c_t(\dot{z}_u - \dot{z}_r) - Fa \quad (10)$$

4.2. Modeling and simulation using MATLAB/Simulink

Differential equations (9) and (10) have been transformed into an organised group of blocks from the Simulink environment.

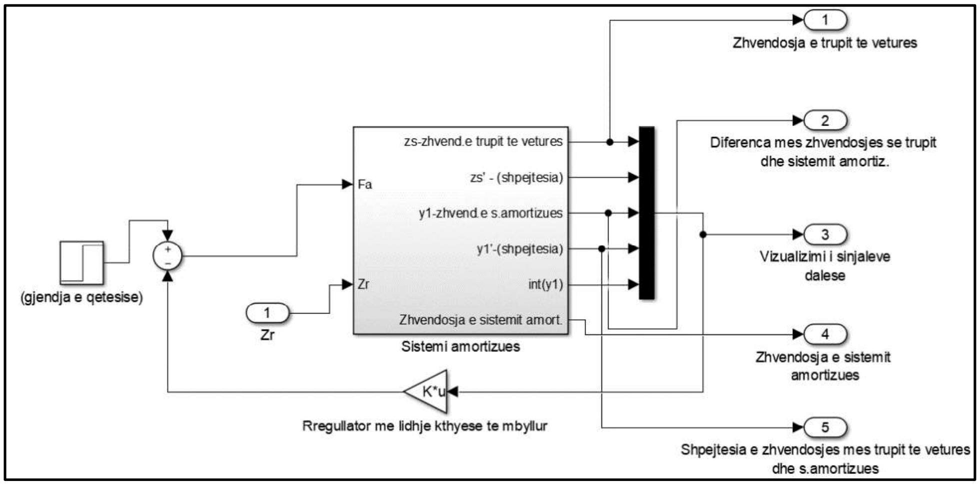


Figure 12. Simulink model of the active suspension system

The following figure shows the contents of the vehicle's suspension block.

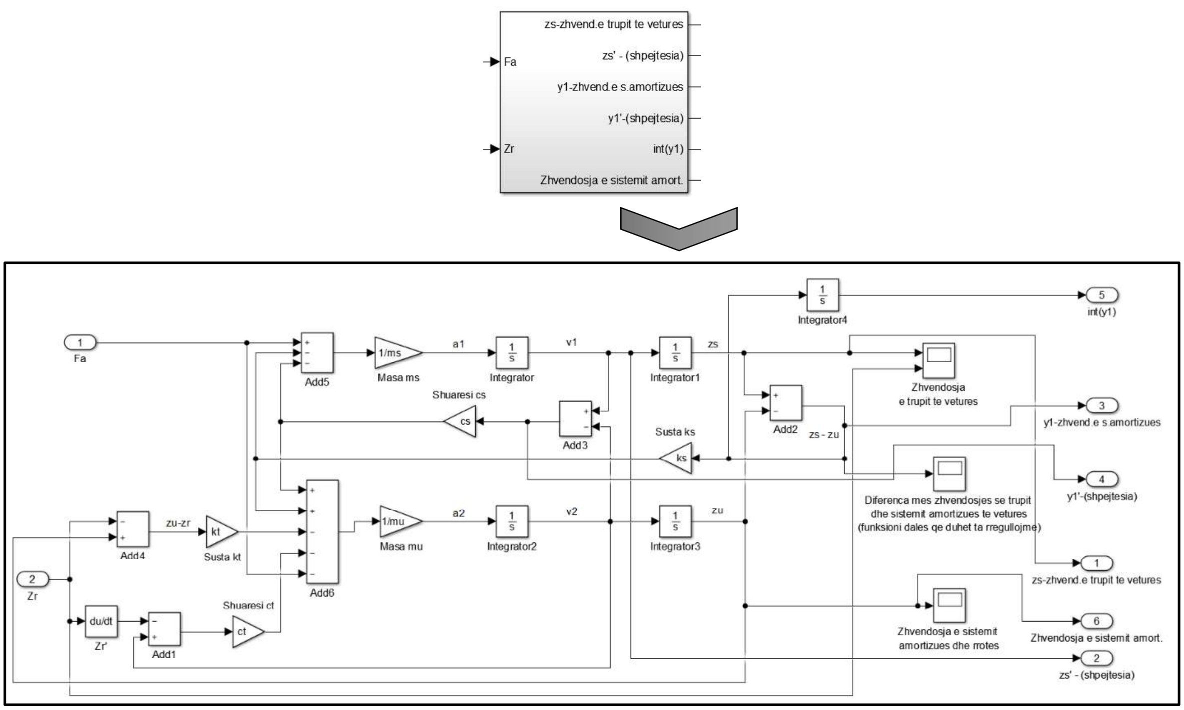


Figure 13. Subsystem content of the active suspension model

After the simulation, the following results were achieved:

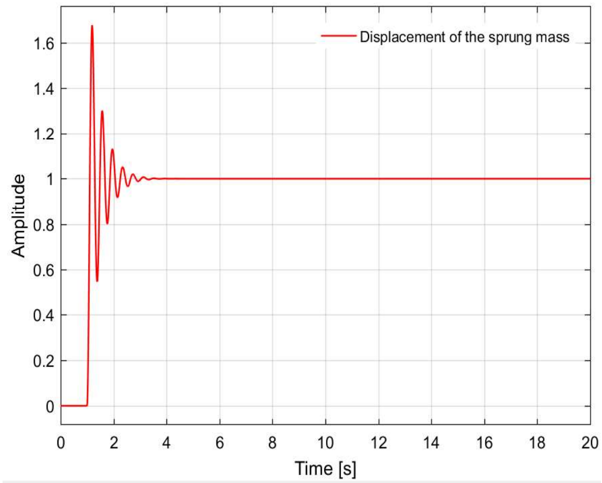


Figure 14. Displacement of sprung mass through time

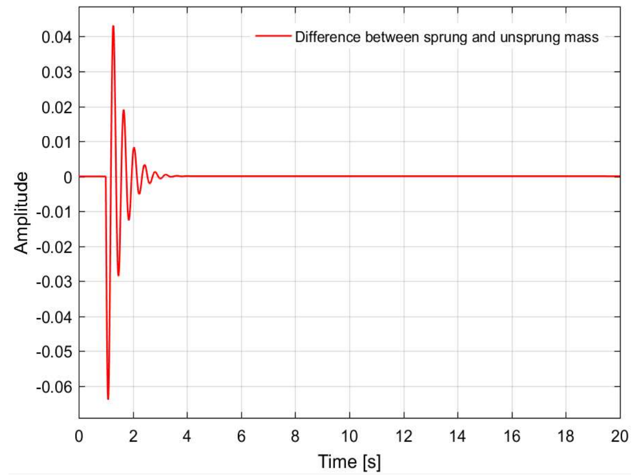


Figure 15. The difference between sprung and unsprung mass

5. Performance comparison between passive, semi-active and active suspension systems

In order to determine which system offers the most favorable performance with regard to the speed of quenching oscillations and improved ride quality, the following simulation of the three systems for the same conditions is presented, namely for a Step and Band-Limited White Noise types of input. The content of the subsystems is already known from Figures 4, 9 and 14.

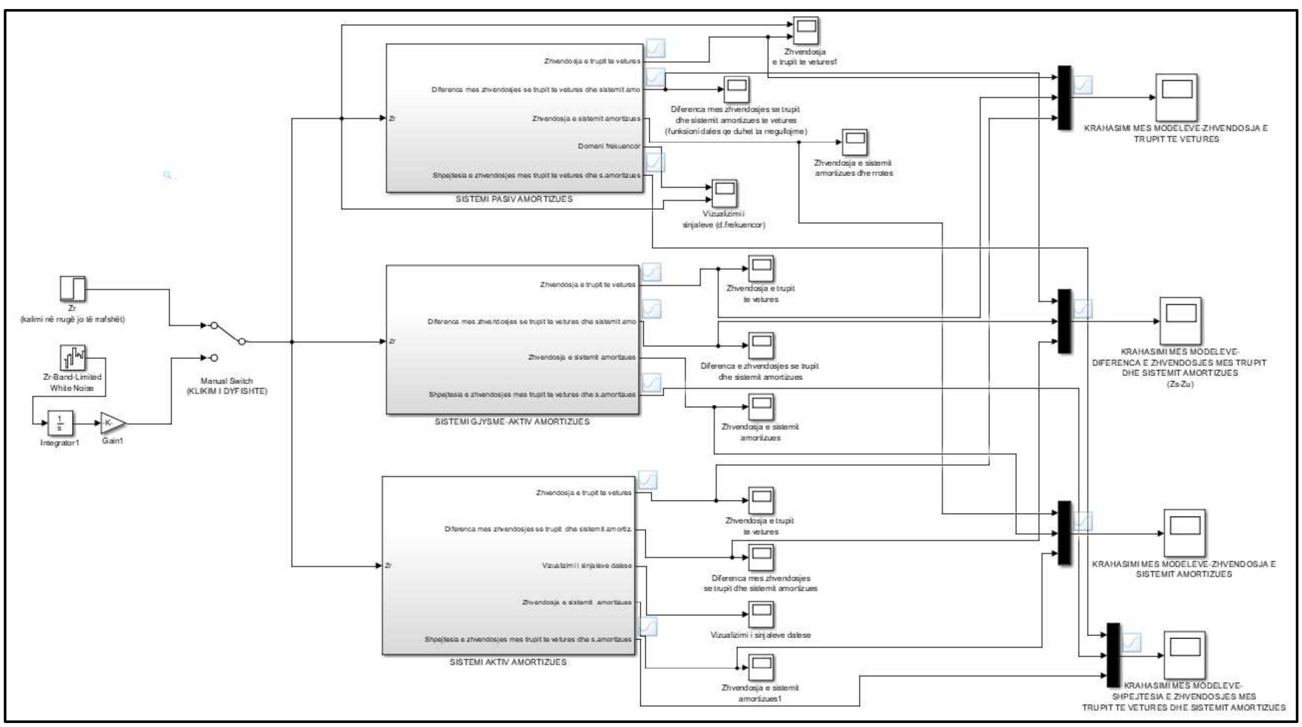


Figure 16. A comparative model of three suspension systems - developed in the Simulink environment

After the simulation for the time of 20s, for the *Step* input, the following results were obtained:

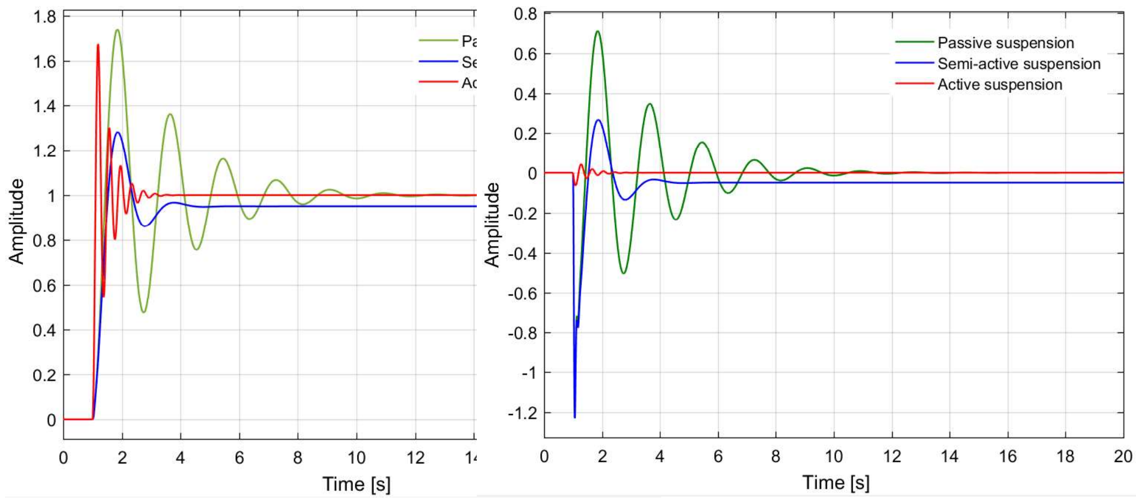
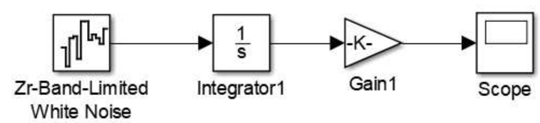


Figure 17. Sprung mass displacement

Simulation under the conditions where the input function is a signal that mimics the uneven road profile (Band-Limited White Noise):



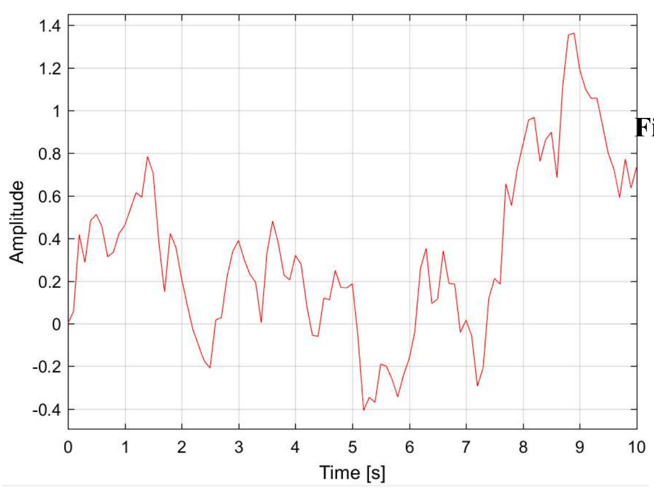


Figure 18. The difference between sprung and unsprung mass displacement

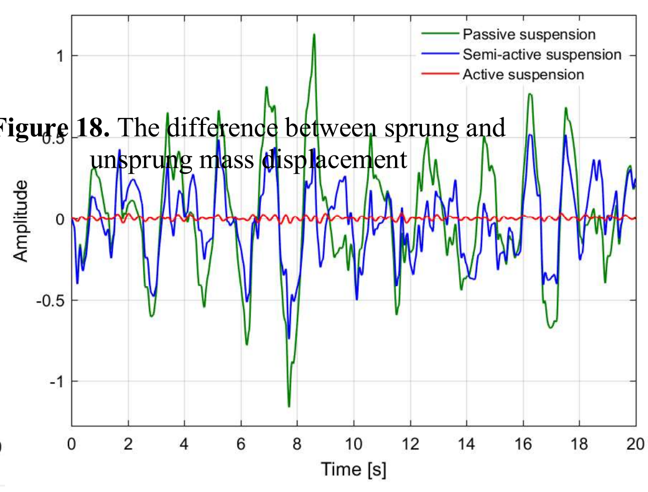


Figure 19. Model of the road simulation and its graph

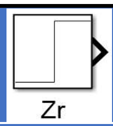
Figure 20. The difference: sprung-unsprung mass displacement

In addition, analogous to the current flow, the simulation results for the displacement difference between the car body and the suspension system is presented.

5.1. Discussion of the results

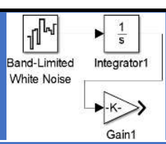
We should emphasize that our goal was to achieve the smallest possible difference between sprung and unsprung mass displacements (also seen in the graphs in Figures 18, 19) and at the same time to achieve the fastest time of stabilizingthe oscillations, achieved results are shown in Tables 3 and 4.

Table 3. The difference between sprung and unsprung mass displacement – Step function

	**For the input function: Step (crossing over a bump on the road)**					
Type of suspension / Parameter	**Passive suspension**	**Semi-active suspension**	**Active suspension**	**Improvement passive›semi-active (%)**	**Improvement semi-active ›active (%)**	**Improvement passive› active (%)**
The difference between sprung and unsprung mass displacement	0.71cm	0.27 cm	0.04 cm	**61.97%**	**85.18%**	**94.36%**
Stabilising time	14.5s	4.3s	3.4s	**70.34%**	**20.93%**	**76.55%**

- Semi-active suspension system is shown to be more suitable for suppressing the oscillations against passive system by **61.97%**,
- Active suspension system is shown to be more suitable against the semi-active system by **85.18%,** while against passive suspension system by **94.36%**,
- Regarding the stabilization time, the semi-active system is faster against the passive system by **70.34%**,
- Whereas, the active suspension system is faster in eliminating oscillations against the semi-active system by **20.93%,** while against the passive system by **76.55%**.

Table 4. The difference between sprung and unsprung mass displacement - Brand-Limited White Noise Signal

	**For the input function: Band-Limited White Noise Signal (imitation of the uneven road profile)**					
Type of suspension / Parameter	**Passive suspension**	**Semi-active suspension**	**Active suspension**	**Improvement passive›semi-active (%)**	**Improvement semi-active ›active (%)**	**Improvement passive› active (%)**
The difference between sprung and unsprung mass displacement	0.32 cm	0.14 cm	0.02 cm	**56.25%**	**85.71%**	**93.75%**

It can be concluded from Table 4 that the Brand-Limited White Noise function as the input function in the suspension systems, has shown results and improvements between the systems in similar percentages as the previous input function (Step function).

6. Conclusion

From the research conducted in the field of suspension systems, we can conclude that:

- Active suspensions are the most suitable systems for application thanks to their excellent ability to quench the oscillations in a record time.
- Following them are semi-active systems, as a very suitable form of suspensions due to their good performance and lower price than active suspension systems.
- Whereas, the passive suspension system is defeated by both of the above-mentioned systems, but the same is still widely used in ordinary cars, due to the average performance requirements and affordable price.

References

[1]. Banks, J. (1999). Discrete event simulation. Marietta: Prentice-Hall. pp. 10-25.

[2]. Knowles, D. (2007). Automotive Suspension & Steering Systems. Clifton Park New York: Thompson Delmar Learning, 4th Edition, ISBN 1401896820, pp. 5-40.

[3]. Savaresi, S., Poussot-Vassal, C., Spelta, C., Sename, O., & Dugard, L. (2010). Semi-Active Suspension Control Design for Vehicles. Oxford: Elsevier Ltd, ISBN: 978-0-08-096678-6, pp.15-39.

[4]. Shpetim Lajqi, Tanislav Pehan. *Designs and Optimizations of Active and Semi-Active Non-linear Suspension Systems for a Terrain Vehicle*. Slovenia: Strojniški vestnik - Journal of Mechanical Engineering 58(2012)12, pp. 732-743, DOI:10.5545/sv-jme.2012.776.

[5]. Zhou, Qi.(2013). *Research and Simulation on New Active Suspension Control System*. *Theses and Dissertations.* Pennsylvania: Lehigh University. Paper 1700.

[6]. Choudhury Farhan Soud, Dr. Sarkar Rashid M. A. *An approach on performance comparison between automotive passive suspension and active suspension system (pid controller) using Matlab/Simulink*. Bangladesh: Journal of Theoretical and Applied Information Technology.

[7]. Andronic Florin, Manolache-Rusu Ioan Cozmin, Patuleanu Liliana. *Pasive suspension modeling using matlab, quarter car model, imput signal step type*. Romania: TEHNOMUS.

[8]. P. Sathishkumar, J. Jancirani, Dennie John, S. Manikandan. *Mathematical modeling and simulation of a quarter car vehicle suspension.* India. Anna University.

[9]. Parasuram Harihara, Dara W. Childs. *Solving Problems in Dynamics and Vibrations using MATLAB*. USA. Dept. of Mechanical Engineering Texas A&M University.

[10]. Eshkabilov Sulaymon L. *Modeling and Simulation of Non-Linear and Hysteresis Behavior of Magneto-Rheological Dampers in the Example of Quarter-Car Model*. Engineering Mathematics. Vol. 1, No. 1, 2016, pp. 19-38. DOI: 10.11648/j.engmath.20160101.12.