

SUSPENSION SYSTEM IN AUTOMOBILE SYSTEM

ANNA SMERINGAIOVA, IMRICH VOJTKO, BARBARA LESKOVA

Technical university of Kosice, Faculty of Manufacturing Technologies, Presov, Slovak Republic

DOI: 10.17973/MMSJ.2016_09_201664

e-mail: anna.smeringaiova@tuke.sk

This paper is aimed on the possibility of modeling, simulation and diagnostics of the suspension system of passenger cars. A simplified dynamic model of the car suspension system is defined. The model transformation into the Matlab – Simulink environment is described for the purpose of suspension system performance analysis. Following are examples of modeling and simulation of vehicle suspension in Autodesk Inventor Professional and control of the technical condition of the vehicle shock absorber using the test equipment MAHA MSD 3000th.

KEYWORDS

car, suspension system, modeling, simulation, dynamic model

1 INTRODUCTION

Comfort and safety of driving in the car is closely related to design concept and setting of the suspension system. There are several possible solutions to flexible connection of body with axle, with different characteristics and ability of regulation. The classical passive suspension is compromise solution between comfort and driving safety. An effort to ensure this two conflicting requirements has required the creation and development of active and semi - active suspension systems for cars. The current trend is to use computer simulation as a tool for testing and verification of the suspension system function in finding an optimal design solution. Equally important is the periodic inspection of the technical state of shock absorbers of cars which have already been in operation.

The article describes the simulation and diagnostic methods used for functional analysis of the passenger car Audi A6 Avant suspension system. Based on the available technical documentation about AUDI A6 Avant was in the graphical system Autodesk Inventor Professional, created a simplified 3D model of the front axle (Fig. 1). In modeling of individual components the main focus was placed on compliance of shape and dimensions of the springing elements of the car: the front wheels, shock absorbers, coil springs, stabiliser and axle arms.

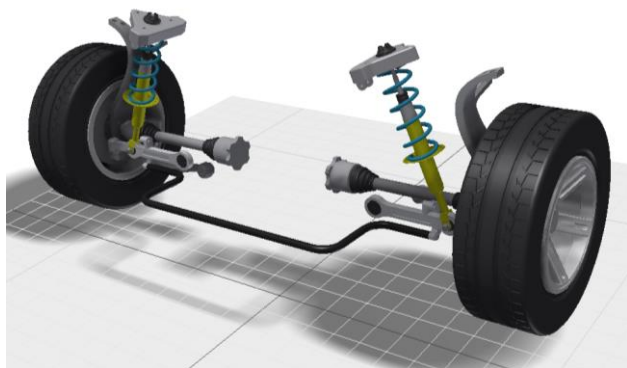


Figure 1. Front axle model created in Autodesk Inventor Professional

Design and basic parameters of the passenger car Audi A6 Avant [Audi 2016] were the basis for creation of the simulation models of car suspension system. The basic monitored

parameter realized functional analysis was sprung mass displacement in the vertical direction.

2 FUNCTIONAL ANALYSIS SHOCK-ABSORBER BY RESONANCE METHOD

State of the vehicle suspension is diagnosed by means of special equipment and devices. At present the most used is the resonance method, which works on two principles: EUSAMA and THETA (Fig. 2). THETA design principle is considered more accurate because when testing it simulates driving conditions on the road, the result is not affected by the weight of the vehicle and the state of the inflated tire.

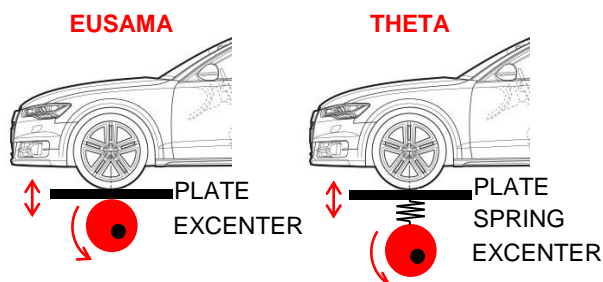


Figure 2. Design principle a) EUSAMA, b) THETA

Control of the shock absorbers was performed on the test stand MAHA MSD 3000 for vehicles with max. axle load 2.5 tons, which is working on the design principle of THETA. Technical parameters of laboratory are listed in [Maha 2016].



Figure 3. Position of the wheel at the plate

The inspection procedure is as follows:

- Wheels of the front / rear axle shall be placed on the plate (Fig. 3). Vibration is transmitted from the driving unit (motor and excenter) through these plates, which are connected to the spring.
- After starting the test, first is measured the left wheel and then the right wheel.
- Engine is running continuously during measurements. Using the computer, rotational speed is precisely controlled and is changing according to measurement mode.
- Amplitude of vibrations and damping of wheels is drawn on the monitor. After completing the tests was evaluated damping rate (Fig. 4).

State of the shock absorbers is assessed on the basis of found dimensionless unit Theta (damping rate D). "Theta" is a dimensionless number, the conversion coefficient of the suspension spring between the plate and excenter. Where optimal values are in the range from 0.13 to 0.3 (green), the application limit from 0.1 to 0.13 (yellow) and when the dampers takes inconvenient values below 0.1 (red). In this case the greater is damping rate, the greater is comfort of dumping. In Fig. 5 is part of the protocol with the results of the measurement of the front axle.



Figure 4. The course of the measured values in the software EUROSYSTEM

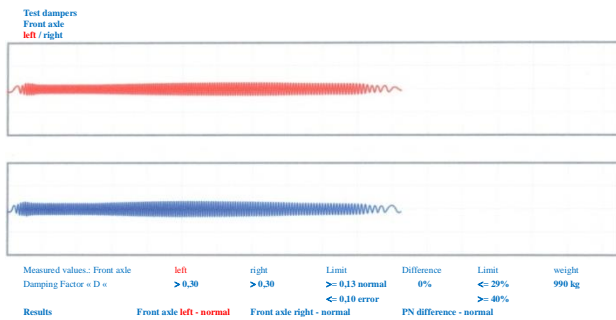


Figure 5. Protocol from the measurement of the shock absorbers test

3 DYNAMIC MODEL OF CAR SUSPENSION

Dynamic model of car damping has six degrees of freedom. By solution we will consider the oscillation of the simple mechanical system provided that the oscillation of the left and right wheels are same and the vehicle is symmetrical to the longitudinal axis.

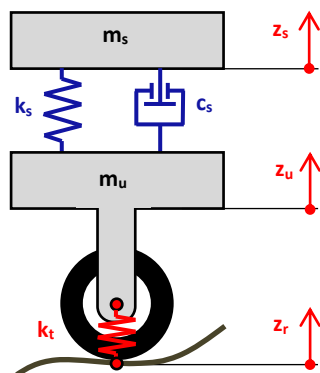


Figure 6. Mathematical model of Passive Suspensions system

A quarter car model is a simplified description of the real car with a weight of one quarter of the car weight and preservation of dynamic and kinematic characteristics of real car. It is used for virtual testing of damping vehicle. It consists of the following springing elements: wheel, damper and spring. With this type of model is taken into account a quarter of the unsprung mass weight (axle, wheel, wheel suspension) and a quarter of the sprung mass weight (body, suspension). The model simulates the vertical motion of the car on the road. In Fig. 6 is a model of passive suspension. Figure describes the displacement z_s , z_u , z_r of individual car parts.

The dynamic model of a quarter car in Fig. 6 is described by the equations:

$$m_s \ddot{z}_s + c(\dot{z}_s - \dot{z}_u) + k(z_s - z_u) = 0 \quad (1)$$

$$m_u \ddot{z}_u - c(\dot{z}_s - \dot{z}_u) - k(z_s - z_u) + k_t(z_u - z_r) = 0 \quad (2)$$

Expression of acceleration of displacement of the sprung and unsprung masses from the equations (1) and (2) :

$$\ddot{z}_s = -\frac{c}{m_s}(\dot{z}_s - \dot{z}_u) - \frac{k}{m_s}(z_s - z_u) \quad (3)$$

$$\ddot{z}_u = \frac{c}{m_u}(\dot{z}_s - \dot{z}_u) + \frac{k}{m_u}(z_s - z_u) - \frac{k_t}{m_u}(z_u - z_r) \quad (4)$$

where:

m_s - the sprung mass

m_u - the unsprung mass

c - the damper coefficient of suspension

k - the spring coefficients of suspension

k_t - suspension coefficient of tires

z_s - the displacements of the sprung mass

\dot{z}_s - the speed of the sprung mass movement,

\ddot{z}_s - acceleration of the sprung mass movement

z_u - the displacements of the unsprung mass

\dot{z}_u - the speed of the unsprung mass movement

\ddot{z}_u - acceleration of the unsprung mass movement

z_r - the terrain input disturbance.

4 SIMULATION OF THE SUSPENSION SYSTEM IN AUTODESK INVENTOR PROFESSIONAL

Functional simulation of the test vehicle suspension system was conducted in dynamic simulation environment in Autodesk Inventor Professional 2015. The simulation model in Fig. 7 is created on the design principle EUSAMA. Simulation was used for a quarter car model. An extension (sprung mass) and the excentric plate (surface unevenness) has been added to the modified front axle model in Fig. 1.

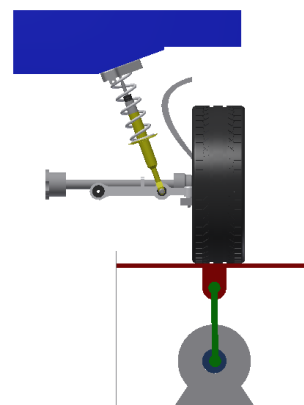


Figure 7. The simulation model of the suspension system

The condition to start the dynamic simulation with relevant results was correct definition of the mechanical system movable parts movement (turning of the excenter, the vertical sliding movement of the plate and entire front axle, shock absorber movement in the direction of its axis) with respect to fixed basis (frame of the excenter). There were also determined properties of the spring/damper - rigidity, damping and selected type of "power joint". Selected was a joint by spring springs. In terms of simplification of the simulation value of spring constant was defined by the sum of the stiffness of the

tire and the spring stiffness. Same procedure has been done also in case of the damping constant. Tab. 1 [Leskova 2016] shows the values of the input parameters.

Table 1. Input values for simulation in Autodesk Inventor Professional

k [N/mm]		c [N.s/mm]		k _p [N/mm]	
38		2		300	
c _{pneu} [N.s/mm]		P [W]		f [Hz]	
1		2200		5	
		F [N]		4512,6	

The rotational part of the excenter has been loaded with torque in order to carry the movement of the plate to simulate the travel path inequalities respectively the movement of the plates of the test equipment described above. Size of the torque M was calculated from equation (5).

$$M = \frac{P \cdot 60}{2\pi f}, \quad (5)$$

where the values of power P and frequency f are based on the technical specifications of the test device MSD 3000. Defined load of the quarter-car model corresponds to the gravitational force of one quarter of the car sprung weight.

Simulation results: Car wheel crossing the obstacle was the object of simulation. By simulation under various conditions has been investigated the reaction of the spring/damper system and the progress of spring deformation (Fig. 8) and force acting on the spring. The simulation results were compared with the results obtained by manual calculation mentioned in [Leskova 2016].

Conditions of the simulation: Loading force: 4512,6 N
Frequency: 5 Hz

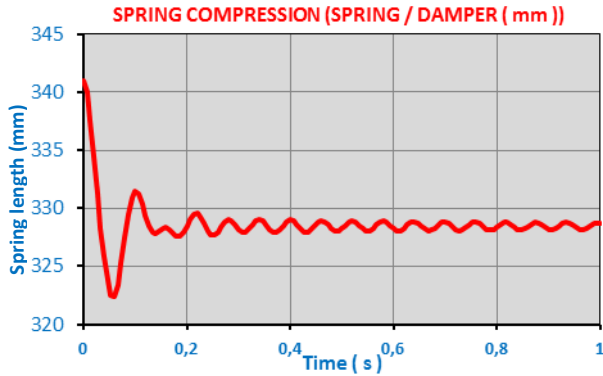


Figure 8. The course of the measured values in Autodesk Inventor

Spring has been compressed from the total length of 341 mm to length 328,8 mm. In computing part it has been selected compression of 130 mm what corresponds approximately to the compression in this simulation.

5 DYNAMIC SIMULATION OF THE SUSPENSION SYSTEM IN THE ENVIRONMENT OF MATLAB - SIMULINK

In Fig. 9 is described a simplified half-car model that includes independent suspension of front and rear axles. The model allows movement in two degrees of freedom: the bounce (z) and the pitch of the car (θ). Demo version of the vehicle suspension may be used for investigation of driving characteristics.

Front and rear wheel suspensions consists of suspension system - damper and spring. A more detailed model could even include tire model and nonlinear damper. In the model, two

degrees of freedom are represented by four situations: vertical displacement (z), vertical velocity (Z), angular displacement (θ) and angular velocity (θ̇). Complete model would be described by six degrees of freedom.

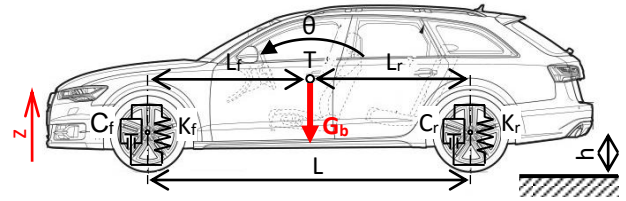


Figure 9. A free-body diagram of the half-car model

MATHEMATICAL MODEL ANALYSIS

The front suspension influences the bounce (i.e. vertical degree of freedom) according to Equation (6).

$$F_{front} = 2K_f (L_f \theta - z) + 2C_f (L_f \dot{\theta} - \dot{z}) \quad (6)$$

Equation (7) determines pitch angle at the front axle:

F_{front} , F_{rear} - upward force on body from front/rear suspension, K_f , K_r - front and rear suspension spring constant, C_f , C_r - front and rear suspension damping rate, L_f , L_r - horizontal distance from gravity center to front/rear suspension,

θ , $\dot{\theta}$ - pitch (rotational) angle and its rate of change, z , \dot{z} - bounce (vertical) distance and its rate of change.

$$M_{front} = -L_f \cdot F_{front} \quad (7)$$

M_{front} - pitch moment due to the front axle.

Equation (8), (9) contains expressions for the rear suspension.

$$F_{rear} = -2K_r (L_r \theta + z) - 2C_r (L_r \dot{\theta} + \dot{z}) \quad (8)$$

$$M_{rear} = -L_r \cdot F_{rear} \quad (9)$$

M_{rear} - pitch moment due to rear suspension.

The forces and moments result in body motion according to Newton's Second Law.

$$m_b \ddot{z} = F_{front} + F_{rear} - m_b g \quad (10)$$

$$I_{yy} \ddot{\theta} = M_{front} + M_{rear} + M_y \quad (11)$$

m_b - suspended body mass,

\ddot{z} - acceleration bounce (vertical) distance

$\ddot{\theta}$ - angular acceleration of pitch

M_y - pitch moment induced by vehicle acceleration,

I_{yy} - body moment of inertia about gravity center.

Input values shown in Table 2 are calculated based on available data on the tested car. Because testing in Simulink was focused on driving comfort (selected soft suspension) calculated spring rate of the front axle has been modified from 38 N/m to 28 N/m and the damping of the front axle has been modified to 3,000 Ns/m and rear axle to 2,500 Ns/m.

The dimensions of the vehicle (Fig. 9) and mass values (Tab. 2) in [Leskova 2016], [Audi 2016] were used for the calculation.

Table 2. Input values for the simulation in MATLAB software

L_f [m]	L_r [m]	m_b [kg]	I_{yy} [kgm ²]
1,29	1,62	1636	2500
K_f [N/m]	K_r [N/m]	C_f [Ns/m]	C_r [Ns/m]
28 000	21 000	3 000	2 500

THE MODEL OF THE SUSPENSION

The suspension model shown in Figure 10 has two inputs, and both input blocks are blue on the model diagram. The first input is the road height. A step input here corresponds to the vehicle driving over a road surface with a step change in height. The second input is a horizontal force acting through the center of the wheels that results from braking or acceleration maneuvers. This input appears only as a moment about the pitch axis because the longitudinal body motion is not modelled. [Humusoft 2016]

Simulation output graphs are displayed through an oscilloscope and signal oscilloscope. In the scheme of suspension subsystem (Fig. 11) are defined equations (6) to (11), by the amplifier and the summation block. For each subsystem (front / rear wheel suspension) it is possible to enter values of K, C, L separately, that means the front subsystem K_f, C_f, L_f a pre zadný and rear subsystem K_r, C_r, L_r .

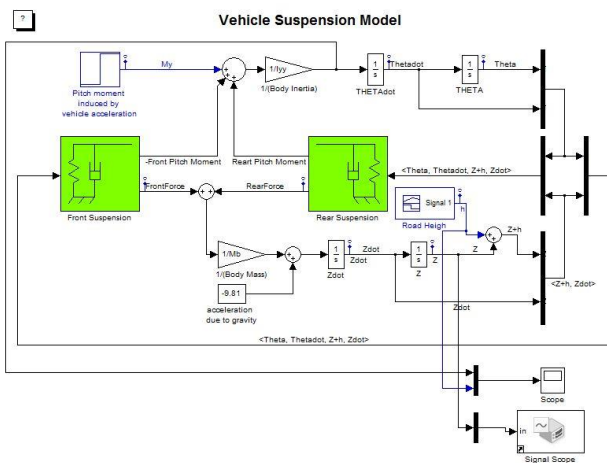


Figure 10. Top-level diagram of the suspension model

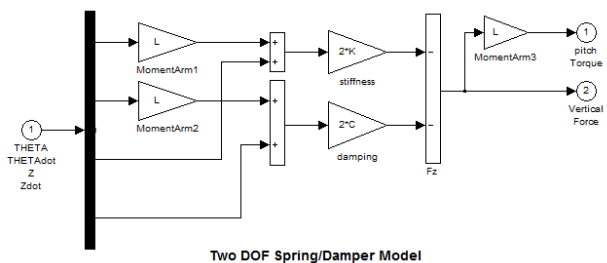


Figure 11. The Spring/Damper model used in Front Suspension and Rear Suspension subsystems

The values in Table 2 were entered into the model environment as follows: View menu - Model Explorer - Model Workspace. After entering required input values has been running the simulation. The objective of this simulation was to approximate the behavior of the half-car model in the following cases:

CASE A

Simulation of obstacles on the road and the subsequent reaction are described by graphs shown in Fig. 12 and Fig. 13. Waveform: In the third second of the car starts to slow down in the sixth-second encounters the obstacle with height of 5 cm and after two seconds drive out of the obstacle.

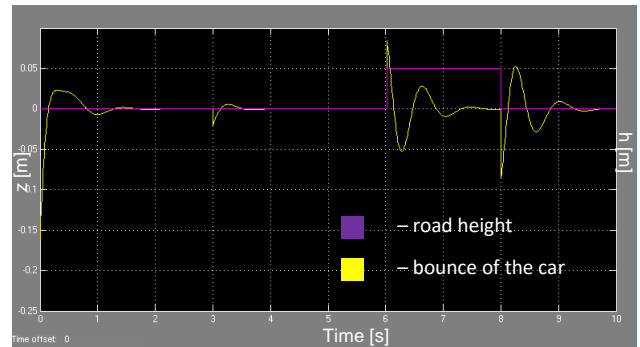


Figure 12. Graph of the course of the vertical reaction of the vehicle to slow down and its reaction to collision with the obstacle

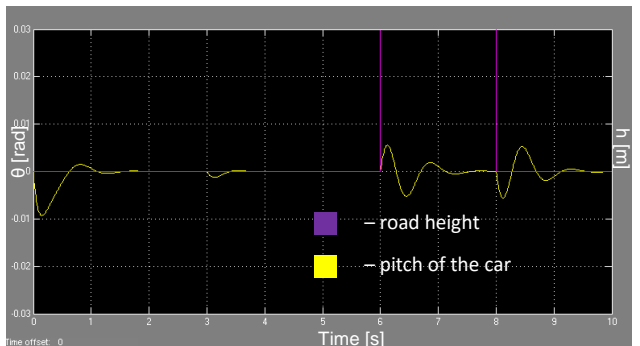


Figure 13. Graph of the course of the pitch of the vehicle - slowdown and collision with obstacle

CASE B



Figure 14. Graph of the waveform of the vertical reaction of the vehicle to acceleration and collision with the pothole

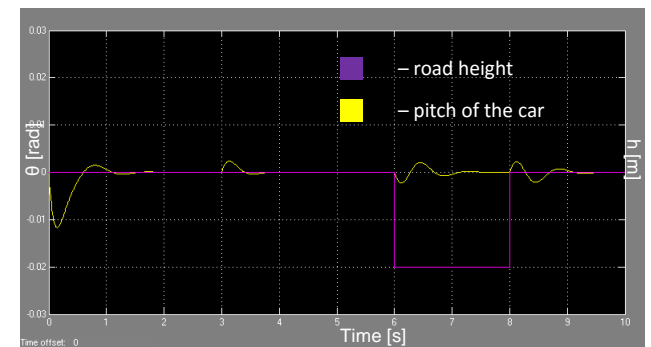


Figure 15. Graph of the waveform of the car pitch - acceleration and collision with the pothole

Simulation of acceleration of the car and unexpected entrance into the pothole on the road are describing graphs shown in Fig. 14 and Fig. 15. Waveform: in the third second the car is accelerating, in the sixth second it goes into the pothole with a depth about 2 cm and after two seconds it is driving out of the pothole. The graph of Fig. 16 shows the waveforms of the monitored variables (bounce, pitch of the car, power, road unevenness and torque).

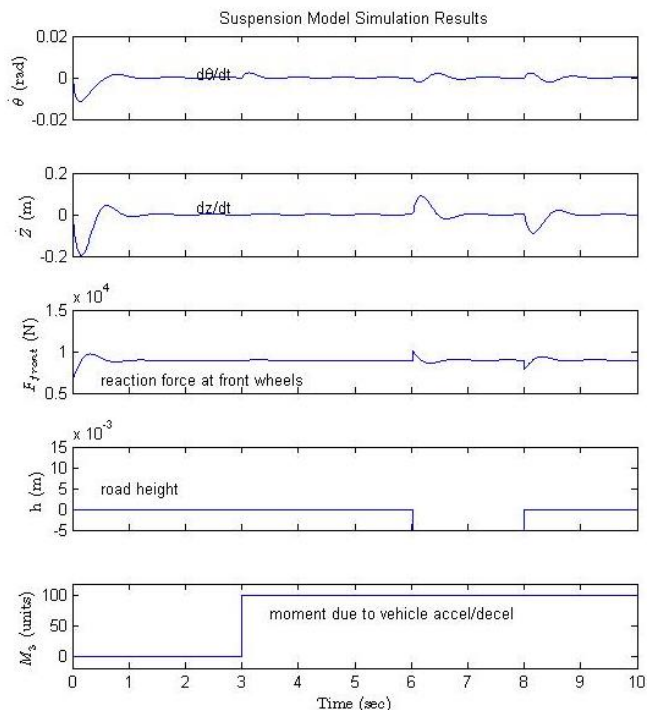


Figure 16. Simulation results

6 CONCLUSION

In the development of the suspension system is the current trend of the development of new suspension system for vehicles, such as active and semi-active systems. Active and semi-active systems are electronically controlled while active systems have the ability to adapt to the constantly changing driving conditions on the road. Modeling and simulation in the virtual environment are the basic tools for the development and design of the cars.

In the paper, by an example of the particular vehicle, are suggested the possibilities of modeling and simulation of suspension in an environment of two selected softwares. Functional analysis of the selected vehicle suspension system was implemented in Matlab Simulink software. There was used the modified demo version of the half-car model suspension simulation (design principle THETA), available in the application block Simulink. Through the simulation was monitored the vertical movement of the car pitch angle depending on the road surface. In Autodesk Inventor Professional was used the quarter model of the front axle supplemented by excentric platform which simulates road unevenness. Waveforms of damping vibrations and acting forces were recorded graphically according to the load. Disadvantage of this model is that it pursues exclusively vertical movement - the bounce of the wheel and describes the less precise design principle EUSAMA.

Simulation in this environment is associated with engineering design of the front axle 3D model. It provides demonstrative simulation of the behavior of the suspension system under load. Simulation model of the car suspension in Simulink environment is the transformation of mathematical model. The possibilities for functional analysis of passive and active suspension and for simulation of control of the active suspension with varying driving conditions are open. On the same design principle (THETA), also operates test device MAHA MSD 3000, on which has been accomplished real vehicle shock absorbers control.

ACKNOWLEDGMENT

This research was supported by the Ministry of Education of the Slovak republic, through grants VEGA 1/0614/15 and KEGA 013TUKE-4/2014.

REFERENCES

- [Audi 2016] Audi A6 Avant '05 Self-Study Programme 344 Germany. Available at: <http://maben.homeip.net/static/auto/audi/SSP%20344%20Audi%20A6%20Avant%202005.pdf> [online].
- [Dubravcik 2011] Dubravcik, M. and Kovac, M. Testing in Automotive Manufacturing. Kosice: TUKE, 2011. ISBN 978-80-553082-8-9. (in Slovak)
- [Humusoft 2016] Matlab & Simulink [online]. Humusoft, 2016. [cit. 2016-04-20]. Available at: <http://www.humusoft.cz/matlab/details/>
- [Maha 2016] Maha [online]. 2016. [cit. 2016-04-20]. Available at: <http://www.maha.de/shock-absorber-tester-msd-3000.htm>
- [Monkova 2015] Monkova, K. Kinematic Analysis of Mechanisms Part 2. Praha: RISE Association, 2015. ISBN 978-80-87670-17-0. (in Slovak)
- [Murcinko 2013] Murcinko, J. and Murcinkova, Z. Implementation of intelligent elements in vibration diagnostics of CNC machines. In: Applied Mechanics and Materials. Vol. 308 (2013), p. 87-93. - ISSN 1660-9336.
- [Krenicky 2010] Krenicky, T. The Monitoring of Technical Systems Operation Using Virtual Instrumentation. In: Strojarsstvo extra. č. 5 (2010), s. 25/1-25/2. - ISSN 1335-2938. (in Slovak)
- [Leskova 2016] Leskova, B. Functional Analysis of the Passenger Cars Suspension System. Diploma work. 2016 (in Slovak)
- [Pavlenko 2015] Pavlenko, S. and Mascenik, J. Design of Machine Parts and Mechanisms. Kosice: TU, 2015. ISBN 978-80-553-2534-7. (in Slovak)

CONTACTS:

Ing. Anna Smeringaiova, PhD.
 doc., Ing. Imrich Vojtko, PhD
 Ing. Barbara Leskova
 Technical University of Kosice
 Faculty of Manufacturing Technologies
 Bayerova 1, 08001 Presov, Slovak Republic
 Tel. : +421 556 026 312 , +421 556 026 364
 e-mail: anna.smeringaiova@tuke.sk, imrich.vojtko@tuke.sk, barbara.leskova@student.tuke.sk