

PAPER • OPEN ACCESS

The behaviour of a vehicle's suspension system on dynamic testing conditions

To cite this article: L Mihon *et al* 2018 *IOP Conf. Ser.: Mater. Sci. Eng.* **294** 012083

View the [article online](#) for updates and enhancements.

Related content

- [Effect of suspension kinematic on 14 DOF vehicle model](#)
T Wongpattananukul and C Chantharasenawong
- [Apparatus for Electrical Measurement of the Resistive Force on a Wheel Rolling on Soft and Uneven Grounds](#)
Kiyoshi Takahashi, Shigeaki Yokoyama, Hisashi Ohta et al.
- [Electrical Measurement of Contact Pressure Distribution of a Wheel Running on Soft Ground](#)
Kiyoshi Takahashi, Susumu Chiyo and Akimasa Wakabayashi

The behaviour of a vehicle's suspension system on dynamic testing conditions

L Mihon, N Lontiş and S Deac

Politehnica University of Timisoara, Mechanical Machines, Equipment and Transportation Department, Mihai Viteazu blvd., no. 1, 300222 Timisoara, Romania

E-mail: liviulmihon@upt.ro

Abstract. The paper presents a car suspension's behaviour on dynamic testing conditions through theoretical and mathematical simulation on specific model, on the single traction wheel, according to the real vehicle and by experiment on the test bench by reproducing the road's geometry and vehicle's speed and measuring the acceleration and damping response of the suspension system on that wheel. There are taking in consideration also the geometry and properties of the tyre-wheel model and physical wheel's properties. The results are important due to the suspension's model properties which allows to extend the theory and applications to the whole vehicle for improving the vehicle's dynamics.

1. Introduction

Road vehicle's movement on a dedicated surface, like concrete and asphalt, could be controlled by a driver in longitudinal and transvers directions as well as around the vertical axis (yaw rotation) within the limits imposed by the physical laws and the surfaces interaction through adherence. The transverse and yaw movement are very well connected each other.

In the vertical direction the movement of the vehicle follows the roadway, including all the bumps (ascending or descending) and declivities, without any action on the part of the driver. For comfort and a safety driving all these irregularities of the road should be minimized through the tires and suspension system of the vehicle.

2. Suspension System of the vehicle

All the external forces that act on a vehicle during his ride, with the exception of gravity and aerodynamic forces, are transmitted through the contact area between tire and road surface (tire patches). Thus, the suspension of a vehicle could be precept as a complex system which generate forces at the tire's contact patches and serve to transmit these to the vehicle: wheels and tires, brakes, suspension arms, hubs and nuts, steering system, springs and shock absorbers [1].

Due to his role of linking the vehicle and the road, the suspension system direct influence the handling dynamics and ride comfort and also affects space utilization, aerodynamics and costs. The suspension alone is not decisive in establishing handling dynamics and ride comfort of the vehicle, but the overall vehicle size, center of gravity height, weight distribution, wheelbase and truck and aerodynamics, at high speeds, influence also the vehicle's behavior.

Starting from this basic concept, where the vehicle's driving consists of moving a body within the atmosphere and the gravity field and in this process the forces (vertical, propulsion and braking, lateral



guiding) are transmitted to the road by tire contact patches, we will take in consideration all these forces for vehicle's dynamics analysis [2].

2.1. Longitudinal and vertical suspension forces

The major longitudinal forces transmitted from the road to the vehicle body are propulsion and braking forces. The primary task of propulsion and braking forces is to control longitudinal dynamics of the vehicles by acceleration and speed. Longitudinal acceleration also gives rise to pitching and vertical motions. Coupling depends on center of gravity position, wheelbase and suspension kinematics. Stabilizing factor are pitch and lift stiffness but also shock absorber damping. For a live axis the driveshaft torque reaction on the body must also be considered.

As a result of vehicle acceleration or braking [3] there are changing of the instantaneous normal forces at the front and rear axles as a dynamic weight transfer reaction. The elastokinematic toe changes due to the direct changes in tire tangential force and indirectly by the changes in wheel travel and the resulting change in side forces, generate side force components which affect vehicle handling. Toe changes on the other hand allow to improve the handling response of the vehicle's braking system as response to the load changes. Figure 1 [4] present the general problem of a vehicle under the loads and movements constraints and degrees of freedom (displacements, accelerations, rotations), with specific notations.

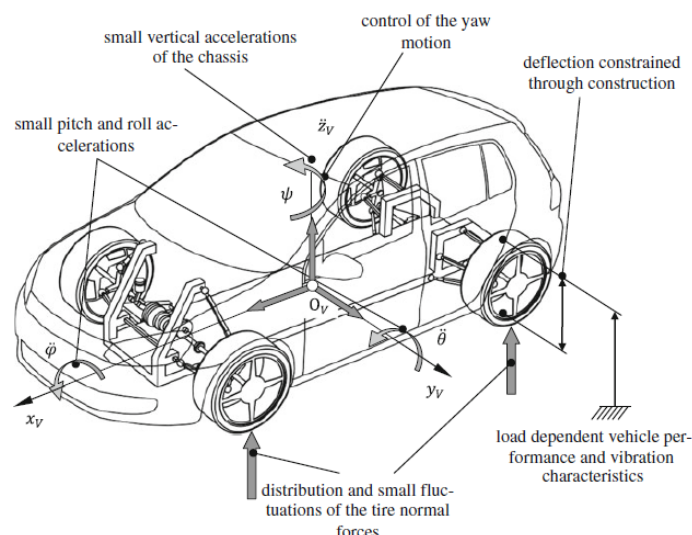


Figure 1. Vehicle model with applied forces

The simulation and analysis of vehicle models is influenced significantly by the type and the temporal behaviour of the applied forces. For a complete study one differentiates between different fundamental types of forces: (i) external and internal forces and torques, (ii) applied and reaction forces and torques and (iii) surface and volume forces. The first category (i) depends on the boundaries of the system in question. The outside forces and torques acting on said system are referred to as external forces. The forces acting within a system are thus referred to as the internal forces. Through a change in the system boundaries, the internal forces become external forces of the new system boundaries. The second category (ii) refers to the source of the forces. Applied forces can be described using the physical force laws with respect to position and velocity. The applied forces in vehicle systems are the forces due to gravity, the spring and damper forces, but also the gas forces in the cylinders of the combustion engine, the air forces and especially the tire forces. The reaction forces are the forces occurring inside the joints. If modeling the wheel and the road surface as rigid bodies, the normal forces from the road to the wheels are reaction forces. In the case of the modeling of the wheel with an elastic tire however, they are considered to be applied forces. The third category (iii)

refers to the distribution of the forces. Surface forces are constrained on a surface, and the volume forces are distributed spatially. The surface forces in vehicles are the wind forces and the forces between the road surface and the tread of the tire. The volume forces in vehicles consist of all the forces that are due to gravity.

Another differentiation results from the temporal characteristic of the forces. One differentiates between forces with deterministic, and those with stochastic temporal characteristics. Deterministic forces are defined explicitly through their time functions. Stochastic forces do not possess a rule based temporal characteristic and need to be described using statistical methods.

For stability and driving comfort of a vehicle there is also important to reduce at maximum the tangential force differences between left and right sides, by appropriate choice of brake system and, for the driven axles, by use of a differential. In this situation the brake and slip control are similarly tuned. The yaw moment caused by unequal tangential forces may also be employed positively. Limited-slip and locking differentials generate a locking effect which tends to produce a correcting yaw moment, tending to bring the car back to a straight path. In order to prevent exaggerated understeer in acceleration, the differential locking effect is limited. In decelerating process, using engine braking, differential locking torque helps to compensate for oversteering moments caused by dynamic weight transfer. To improve the stability of the vehicle using this effect some vehicles use differentials with a higher limited slip factor while under engine braking than in acceleration. So, the vehicle stability control may be realized by means of differentials with controllable locking torque, which permit introduction of a correcting yaw moment.

Modern brake control systems use deliberate, controlled, asymmetric, driver-independent brake intervention to stabilize the vehicle at the handling limit.

Vertical forces, acting at the wheel road surface patch than transmitted and applied between suspension and body, consist mainly of spring and shock absorber (damping) forces. Their task is to support the vehicle's sprung mass on the wheels and impose limits on vehicle lift, roll and pitch motion relatively to the road surface. Through the extent limit permitted by spring travel, the body should be isolated against road surface irregularities and dynamic changes in wheel load minimized.

Tire normal forces, Figure 2, differ from the corresponding spring and shock absorber forces introduced to the body by the inertia of unsprung masses. The spring and shock absorber forces, Figure 3, through their vertical reaction forces from tire and tangential forces also affect the body, as do unsprung masses. These forces results from coupling of the tire contact patch longitudinal and transverse movement in response to vertical suspension motion (antidive, antisquat, sideforce reaction angle). These effects are used to reduce roll and pitch angles.

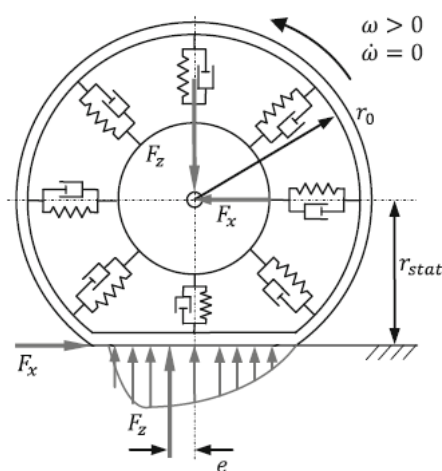


Figure 2. Model of a wheel

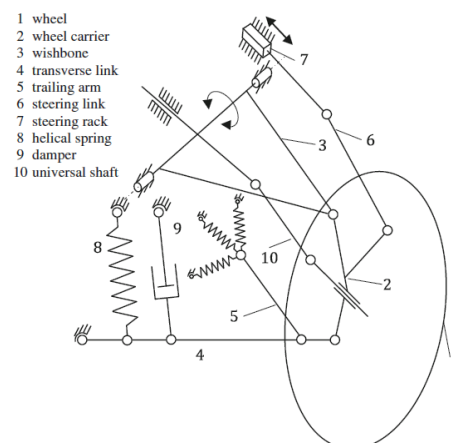


Figure 3. Scheme of an independent suspension

2.2. One Dimensional Quarter Vehicle Model

In order to be able to assess the possibilities and limits of mechanical systems, the use of theoretical mathematical techniques has proven to be of great value. By means of a suitable reduction of the model, the computing time and interpretability of the results can be optimized. Thus it was used a simple equivalent mass-spring system in order to theoretically analyze the vehicle vibrations in vertical direction. The influence of vehicle suspension and damping, wheel mass and tire elasticity on comfort and range of spring deflection can be determined simply by using quarter vehicle models.

The vehicle model hereby consists of a two-body-system, which is linked to a wheel model, Figure 4. This model reduction is acceptable for a passenger car because the coupling masses are usually very small compared to the car body mass. Because the influence on vibration's properties are reduced, for a correct analyze we neglect also the effects of pitch and roll oscillations in the evaluation of comfort.

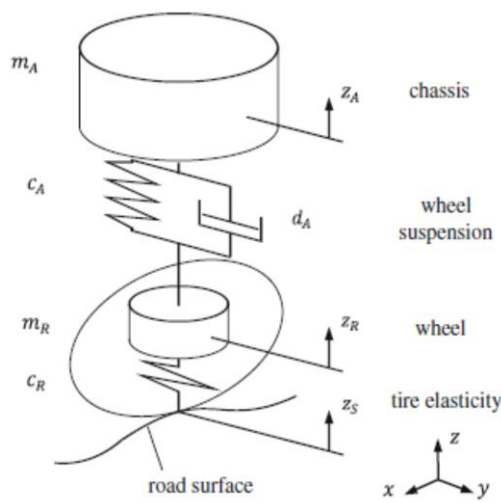


Figure 4. The quarter – vehicle model

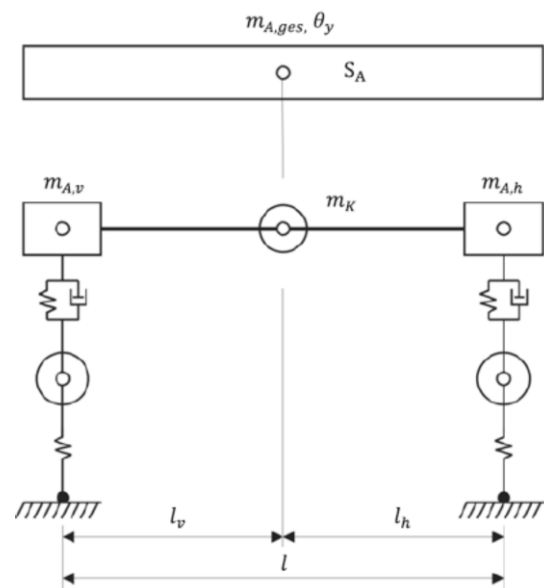


Figure 5. Derivation of the vehicle body mass

In Figure 4 [5] the notations are the following: x , y and z are the current position coordinate (and also Cartesian versors of axis), m_A is the body mass, expressed as ratio from the whole vehicle mass, m_R is the wheel assembly mass and respectively, c_A is the vehicle body spring stiffness, c_R is the wheel spring stiffness and d_A is the vehicle body suspension damper.

The contact to the road is established via the wheel (the patch with the road surface), which is modeled through the spring stiffness c_R . The equations of linear momentum for the system in z -direction, in accordance with Figure 4 notations are:

$$m_A \ddot{z}_A + d_A (\dot{z}_A - \dot{z}_R) + c_A (z_A - z_R) = 0 \quad (1)$$

$$m_R \ddot{z}_R - d_A (\dot{z}_A - \dot{z}_R) - c_A (z_A - z_R) + c_R z_R = c_R z_S \quad (2)$$

Equation 1 refer to the body (chassis) and equation 2 refer to the wheel.

In order to achieve better results from the quarter-vehicle model, the pro rata chassis mass can be determined by breaking down the half chassis mass onto three mass points $m_{A,v}$ (front axle), $m_{A,h}$ (rear axle) and m_K (coupling mass).

From the conditions for the preservation of the chassis mass $m_{A,ges}$, of the center of mass S_A and the mass moment of inertia $\theta_y = m_A i_y^2$, where i_y is the radius of gyration about y -axis

$$m_{A,y} + m_{A,h} + m_K = m_{A,ges} \quad (3)$$

$$m_{A,v}l_v - m_{A,h}l_h = 0 \quad (4)$$

$$m_{A,v}l_v^2 + m_{A,h}l_h^2 = m_A i_y^2 \quad (5)$$

one obtains three equivalent masses

$$m_{A,v} = m_{A,ges} \frac{i_y^2}{l_v l}, \quad m_{A,h} = m_{A,ges} \frac{i_y^2}{l_h l}, \quad m_K = m_{A,ges} \left(1 - \frac{i_y^2}{l_h l_v}\right) \quad (6)$$

For the wheel based, proportional body mass m_A of the quarter-vehicle model, $m_A = \frac{1}{2} m_{A,v}$ (front wheel) respectively $m_A = \frac{1}{2} m_{A,h}$ (rear wheel) is used.

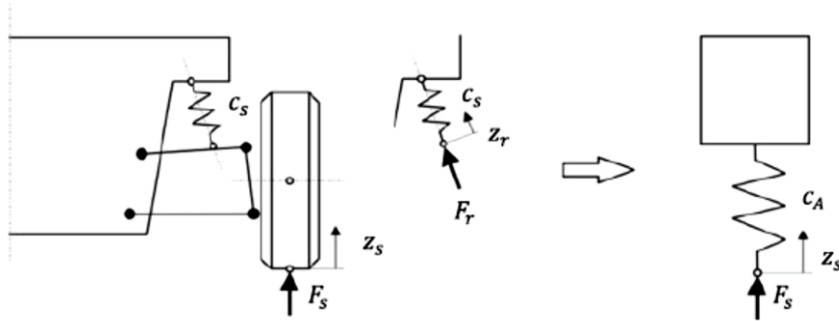


Figure 6. Kinematic transmission of the wheel suspension forces

Furthermore it is necessary to consider the kinematic transmission of the wheel suspension forces as shown in Figure 6. The kinematic transmission k results from the vertical deflection z_r of the wheel suspension as

$$k(z_s) = \frac{dz_r}{dz_s} = \frac{z_r}{z_s} \quad (7)$$

From the principle of virtual work, the relation between the force F_s at the tire-road contact point (in the contact patch) and the elastic force F_r can be derived as follows:

$$F_s \delta z_s = F_r \delta z_r = F_r \frac{dz_r}{dz_s} \delta z_s \quad (8)$$

then

$$F_s = \frac{dz_r}{dz_s} F_r = k F_r \quad (9)$$

Based on equations (7) to (9) one calculate the spring stiffness related to vertical displacement z_s of the tire contact point (in the contact patch) as follows:

$$c_s = \frac{dF_s}{dz_s} = \frac{d(kF_r)}{dz_s} = \frac{dk}{dz_s} F_r + k \frac{dF_r}{dz_s} = \frac{dk}{dz_s} F_r + k \frac{dF_r}{dz_r} \frac{dz_r}{dz_s} \quad (10)$$

Because $\frac{dF_r}{dz_r} = c_s$ and $\frac{dz_r}{dz_s} = k$ one obtain

$$c_A = \frac{dk}{dz_s} F_r + k^2 c_s \quad (11)$$

2.3. Tire model

The modeling of the tire forces requires special attention and care especially when along with the stationary behaviour the instationary behaviour has to be covered. There are three types of tire models such as mathematical models, physical models and a combination of the two. For the further processing one can either be accomplished with the help of an approximation through an algebraic function (Magic Formula Tire Model [6, 7]) or through an interpolation. This art of modeling is for the simulation of driving maneuvers mostly sufficient as the excitation frequencies in these cases are well below the Eigen frequencies of the belts. Problems can however occur, when a large number of influencing variables have to be considered, as in this case a considerable characteristic curve needs to be saved and evaluated. And it is practically impossible, to change individual parameters, without recreating the entire characteristic curves. That is why, for investigation and modeling of the comfort and vibration analysis are often use these models only to estimate the deformation of the tire.

From the point of view of a reasonable computation time, such models are still not used in the vehicle dynamics simulation in spite of the massive increase in computation power available today. In stationary models only the rim will be considered to have mass and inertia, and the belt will not be modeled as a separate body. These models are similar to the characteristic curve models mostly used in the simulation of stationary driving maneuvers. In order to extend the application area of these models, very often a first order time element (PT_1) is used to represent the delayed building up of the tangential forces.

However if excitations in the region of the belts eigenfrequency, between 30 Hz and 50 Hz, were to occur, such as through the use of elastic components in the wheel suspension or through the pulsation of the brake pressure during an ABS controlled braking procedure, and these have an effect on the characteristics of the vehicle behaviour, then the eigendynamics of the belt have to be considered as well. Thus the belt will be modeled as a rigid ring with mass. This will be applied at low speeds (below 10 m/s) highly instationary transport processes in the contact surface between the tire and the road (the contact patch), that require a special attention.

A kinematic model for the wheel-road contact [5] will respect the following: (i) the contact geometry tire/road are described through a representative mechanism and can be calculated using simple expressions, (ii) simple tire models, in which the longitudinal forces and the lateral forces can be calculated, without considering the eigendynamics of the belts, as a function of the longitudinal slip and the lateral slip, can access these variables, (iii) for more complex tire models, that need to represent the instationary processes, the patch can be discretized. The required kinematic values are available for these models also.

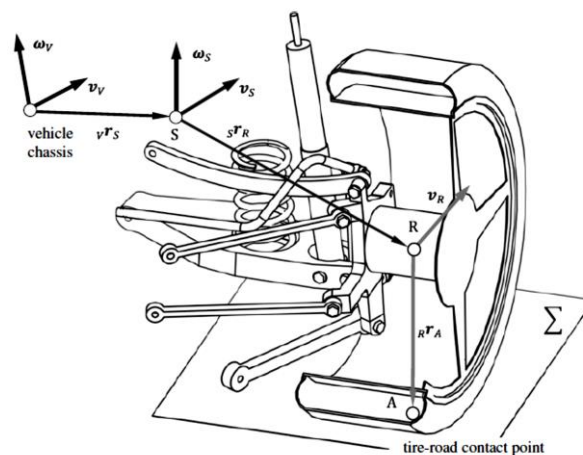


Figure 7. Calculation of the position and velocity of the wheel carrier from the vehicle chassis

It shall be assumed that the position and orientation of the wheel in space is known. For this one measures the position and velocity of the wheel from the vehicle chassis, see Figure 7. The position of the wheel carrier can be described through the wheel center point R and through the introduced wheel carrier fixed coordinate system according to Figure 3 and 4 notations. These values are known through the vehicle kinematics.

3. Modeling and Experimental results

Starting from the technical equipment that consist the core of the Vehicle Dynamics Laboratory of the Politehnica University Timisoara and also the MSC ADAMS package software for modelation and evaluation of the experiments involved in vehicle dynamics were obtained interesting satisfactory results of the proposed models.

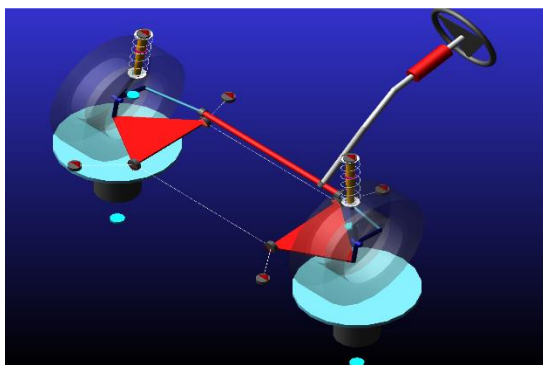


Figure 8. MSC ADAMS Car model for the suspension analysis

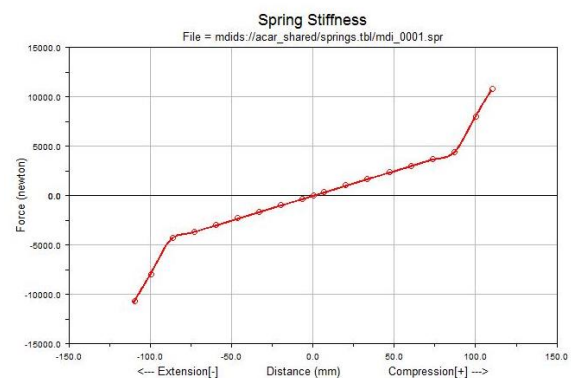


Figure 9. MSC ADAMS quarter-car model for the force loading through measurement

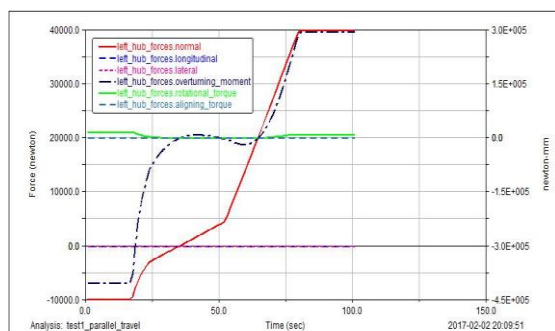


Figure 10. Simulation results for the wheel load and through measurement

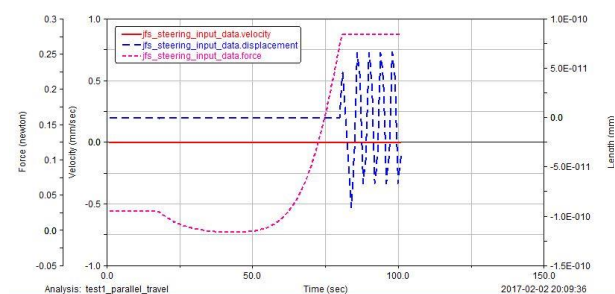


Figure 11. Simulation results for the suspension travel and through measurement

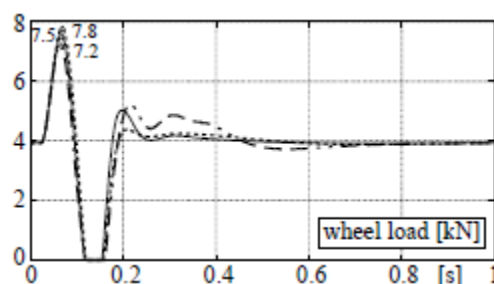


Figure 12. Compared simulation with experiment for the wheel load

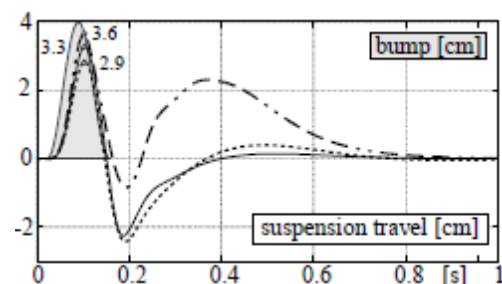


Figure 13. Compared simulation with experiment for the suspension travel

Thus, there were used the following equipment and software: manufactured suspension test bench, electric actuator for applied forces on wheel-spring-damper assembly, electronic equipment for electric actuator control, NI USB-6251 data acquisition and control device, LabView 2015 software [8], MSC Adams 2017 software [9], ASM R1K-L10 linear motion transducer, SACHS 1755017000 – 11 – 05 – 00, 19 – 04 – 00 and 19 – 09 – 01 acceleration transducers.

For the model and test bench was chosen a conventional MacPherson suspension assembly reproduced in Figure 8 after the modelation in MSC ADAMS Car Software. For the complete analysis in the model was included also the steering assembly that is missing on the suspension test bench.

For the quarter car model and through experiment (based on the manufactured test bench) there were used the following important data: sprung mass – 320 kg, spring diameter – 14 mm, spring coil diameter – 128 mm, shear modulus of steel – 85 kN/mm², upper arm length – 270 mm, lower arm length – 280 mm, tire 195/65 R15, wheel mass – 42 kg, wheel spring stiffness – 225000 N/m, damper value for the suspension test rig – 2750 Ns/m. Figure 9 expose the measured values for the loading force that was used in MSC ADAMS simulation.

Thus, from the experiment and simulation analysis we can compare and calibrate the values for the load forces that act on the tire – wheel – damper assembly of a quarter-car model. Based on the results and taking in consideration the equations and input data we will be able to obtain the comfort behaviour of the suspension assembly of the current classic MacPherson suspension system through variation of different values of tire pressure, tire model/manufacturer, tire damping, spring stiffness, absorber damper, etc.

Figures 10 and 11 present the simulation results made in MSC ADAMS starting from the measured elements (forces, displacements, accelerations) and time dependent working process, and in Figures 12 and 13 there are presented the compared values measured through data acquisition chain versus modeled values for the wheel load and suspension travel of the assembly on a bump-speed rolling sequence.

4. Conclusions

The studies developed for these models was first conceived and tested like a theoretical system and then simulated on specific environment. The results was compared with experimental data, obtained on a specialized test rig through data acquisition values and the results are very favourable, proving the good approximations and initial values conditions for the simulation and experiment. The models for the tires and suspension components could be modified due to the simulation and the time and costs of testing experiments could be preserved and the testing conditions could be varied on demands.

References

- [1] Blundell M and Harty D 2004 *The Multibody System Approach to Vehicle Dynamics*, Elsevier Butterworth-Heinemann Publications
- [2] Kiencke U and Nielsen L 2000 *Automotive Control Systems*, Springer
- [3] Genta G 1997 *Motor Vehicle Dynamics. Modeling and Simulation*, World Scientific Publishing, Singapore
- [4] Schramm D, Hiller M and Bardini R 2014 *Vehicle Dynamics. Modeling and Simulation*, Springer
- [5] Rill G 1012 *Road Vehicle Dynamics. Fundamentals and Modeling*, CRC Press
- [6] Pacejka H B 2006 *Tyre and vehicle dynamics. 2nd (ed.)*, Butterworth-Heinemann, Oxford
- [7] Pacejka H B and Besselink I J M 1993 *Magic formula tyre model with transient properties*, Proceedings of the Berlin Tyre Colloquium, Vehicle System Dynamics Supplement 27, 145–155, Swets&Zeitlinger, Lisse, Berlin
- [8] ***Labview/National Instruments – 2015 User's manual
- [9] ***MSC ADAMS Car – 2017 User's manual