ANNALS of the ORADEA UNIVERSITY.
Fascicle of Management and Technological Engineering, Volume X (XX), 2011, NR2

MODELS FOR THE DYNAMIC ANALYSIS OF THE SUSPENSION SYSTEM OF THE VEHICLES' REAR AXLE

Alexandru Cătălin
"Transilvania" University of Braşov, Product Design and Robotics Department,
calex@unitbv.ro

Keywords: motor vehicle, rear axle, suspension system, dynamic analysis.

Abstract: The paper presents a comparative analysis between different half-car models for the suspension system of the rear axle of the motor vehicles, the difference between models consisting of how the equilibrium of the car body is assured. The results of the dynamic analysis are reported to a full-vehicle assembly, having in view to evaluate the roll motion of the car body, which has great influence on the vehicle stability and comfort. The models are analyzed using the multi-body systems (MBS) software ADAMS.

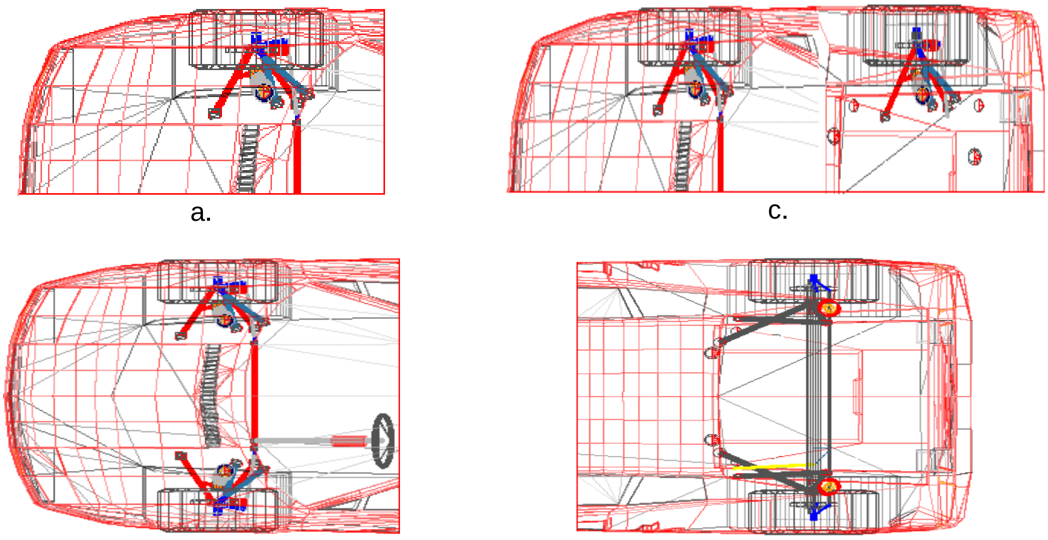
1. INTRODUCTION

The dynamic model of the guiding (suspension and steering) system of the motor vehicles includes the kinematic elements - bodies (car body, axle, wheel carrier, guiding links), as well as the elastic and damping elements (ex. springs, dampers, bushings, antiroll bar, bumper and rebound elements). The increasingly growing demand for more comfortable passenger car imposes new models of the guiding systems, close to real systems on the vehicle [3, 7, 9]. The complexity of the guiding systems of the passenger cars makes it very difficult to handle the error in the case of simultaneous achievement of the entire model (full-vehicle assembly). In these terms, in the preliminary stages, segmented models (suspension assemblies) can be developed & analyzed, for the wheel suspension (quarter-car models) and for the axle suspension (half-car models) [6].

In this paper, for the guiding system of the rear axle, two half-car models are analyzed having in view to evaluate the roll motion of the car body, one of the more important factors for the vehicle stability and comfort. The study is made using the multibody systems environment ADAMS (Automatic Dynamic Analysis of Mechanical Systems) of MSC Software, licensed to the Product Design and Robotics Department from the "Transilvania" University of Braşov. This software package was lanced in commercial versions even in the 1980's but in the last decade a new type of study was defined through it usage: virtual prototyping. This consists mainly in conceiving a detailed model and using it in a virtual experiment, in a similar way with the real case, which permits a large-scale evaluation of the system behavior. The working precision is very high, such that the dynamic behavior of the vehicle may be predicted in early design stages. Another advantage of this kind of simulation consists in the possibility of make easy virtual measurements in any point and area of the system and for any parameter, without going through expensive prototype building and testing.

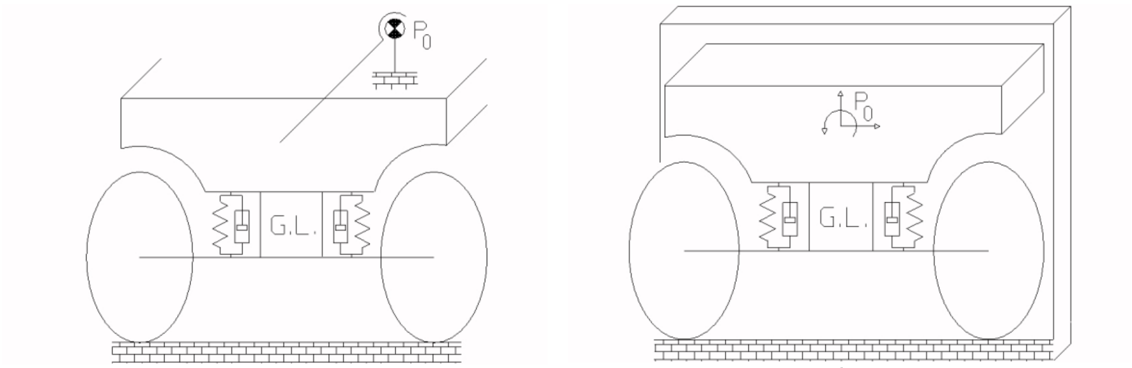
2. THE DYNAMIC MODEL OF THE REAR AXLE SUSPENSION SYSTEM

The suspension assembly takes into consideration the guiding - suspension system of the front or rear wheel / axle. Such models are schematically presented in figure 1: a - quarter car model (independent wheel suspension); b, c and d - half car model (independent axle suspension - b, dependent axle suspension - d, respectively front and rear wheel independent suspension - c).



b. d.

Figure 1. Suspension assemblies (quarter-car and half-car models)



a. b.

Figure 2. Half-car models to evaluate the roll motion (G.L. - guiding linkage)

The quarter-car model (fig. 1, a) is usually used to evaluate the vertical motion of the car body, the body equilibrium being ensured with a translational joint between the car body and ground, in the median plane of the vehicle. The half-car models are used to study the vertical motion as well as the roll motion (fig. 1, b, d) and the pitch motion (fig. 1, c) of the car body. To evaluate the roll motion, the equilibrium can be made using a spherical joint on ground (fig. 2, a) in the median - longitudinal plane (the location of the joint can be obtained with the double conjugate points theory [5]), respectively with a planar joint (fig. 2, b) in the vertical - transversal plane of the vehicle. The half-car model with spherical joint takes into consideration the entire mass of the car body, the joint being placed in the conjugate point on the front axle. The second half-car model, with planar joint, includes only a part of the car body mass, according to the distribution of the vehicle mass on the front and rear axle.

To model the suspension assemblies, a virtual prototyping platform, which integrates MBS, CAD and FEA software solutions [1], is used. The virtual prototyping provides the ability to work within CAD environment, having in view to model mechanical systems, to easily transfer geometry between CAD and MBS software, respectively to transfer loads from MBS to FEA, and to bring component flexibility from FEA back into MBS [8, 10]. The main component of the virtual prototyping platform is the mechanical systems analysis and simulation software (MBS). The steps to create a virtual model with MBS software mirror the same steps to build a physical prototype: build - modeling parts, constrain the parts, create forces acting on the parts; test - measure characteristics, perform simulation, review animation, review numeric results as plots; validate - import, superimpose test data on plots; refine - add friction, define flexible bodies, implement force functions, define controls; parameterize - add parametrics, define design variables; optimize - perform design sensitivity studies, perform design of experiments, perform optimization studies. The dynamic simulation consists mainly in conceiving a detailed model and using it in a virtual experiment, in a similar way with the real case. This improves design quality, reduces the number of costly physical prototypes and the product development time [4].

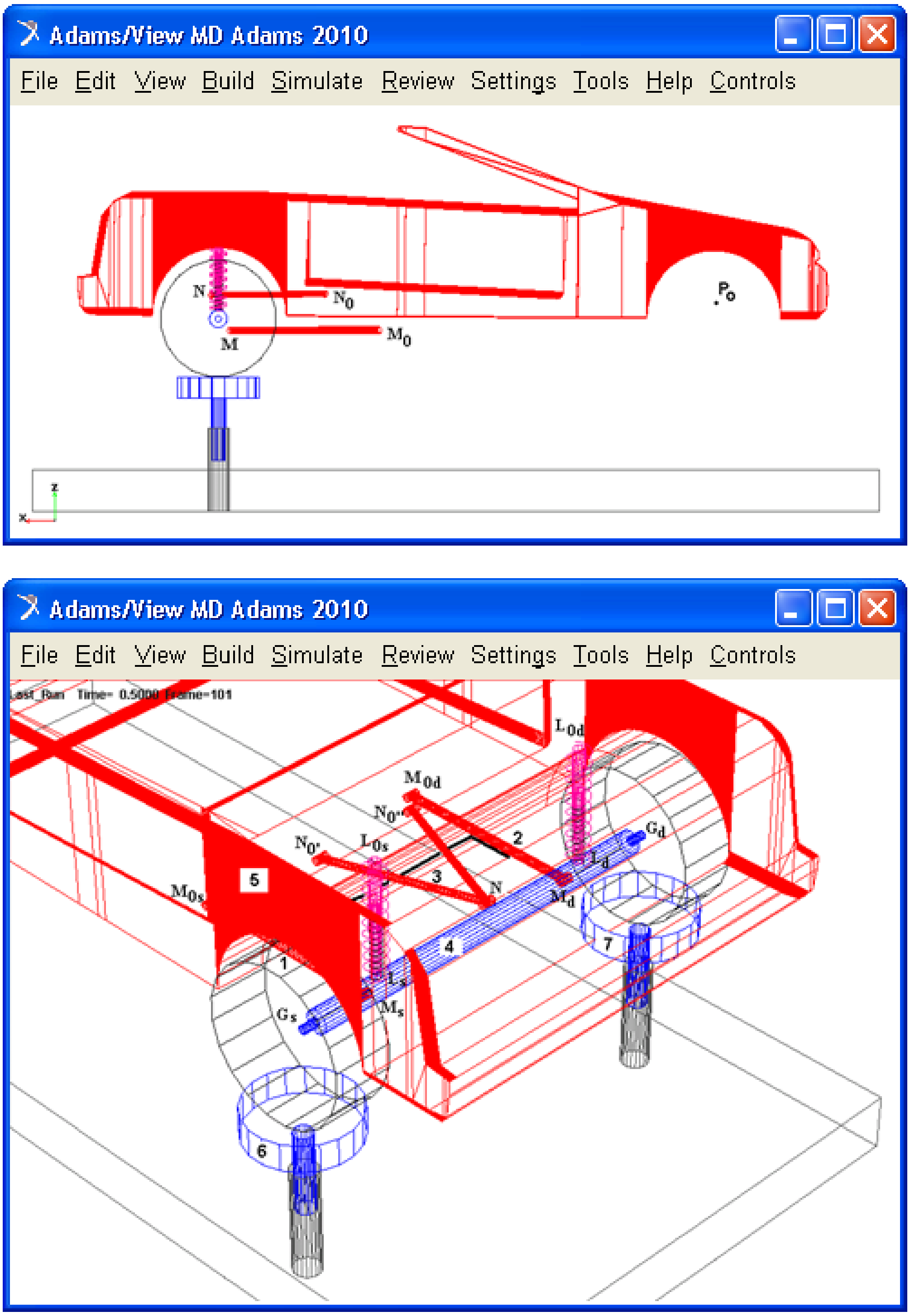


Figure 3. The MBS dynamic model of the rear axle suspension system

Using the MBS environment ADAMS of MSC Software, the half-car model that corresponds to a passenger car with independent suspension of the rear axle suspension was developed (fig. 3). The dynamic model of the guiding - suspension system of the car axle contains the kinematic elements - rigid parts (car body, axle, guiding links) as well as the elastic and damping elements (springs, dampers, bushings, bumper and rebound elements, antiroll bar, tires).

For the guidance of the rear axle, relative to the car body, a spatial linkage with three bars is used; this achieves the axle guidance through driving three axle points. The guiding links are connected to the car body and to the rear axle through bushing elements (compliant joints). The graphics of the parts was made using the solid library from ADAMS and boo-lean operations (adding, subtracting and intersection). To model the geometry of the car body, which is more complex, a specialized CAD software (SolidWorks) was used, the solid model being transferred in ADAMS as Parasolid format file. To import the geometry of the car body, ADAMS/Exchange reads the CAD file and converts the geometry into a set of ADAMS/View geometric elements.

The elastic and damping elements of the suspension system represents forces acting between two parts (car body and axle) over a distance and along a particular direction. The suspension spring is modeled as a double active (tension – compression) elastic element of translational nature. The inputs for modeling are the global coordinates of points in which the springs are connected to the adjacent bodies, the natural spring length, and the force vs. deflection characteristic. The suspension contains also bound-stop and rebound-stop that acts between the car body and axle. The internal forces of elastic bumpers have transitory character, so that these elements were modeled as translational springs with unilateral rigidity, using one-sided impact forces. The anti-roll bar, which represents a flexible bar fitted transversely to the suspension, is disposed between the lower arms of the guiding axle linkage.

Having in view to evaluate the dynamic behavior of the body (mainly the roll motion of the body) the half-car model is analyzed in passing over bumps regime, by simulating an experimental testing machine (fig. 4). The wheels are anchored on the actuators that have translational motion relative to ground. The vertical motions of the actuators (Z_{Ks}, Z_{Kd}) are controlled using motion generators, which are kinematic restrictions (constraints) used to simulate the road profile (disturbances). For this simulation, the right wheel (actuator) is fixed, while the left wheel passes over a sinusoidal bump with the amplitude of 50 mm, in 0.01 seconds (fig. 5).

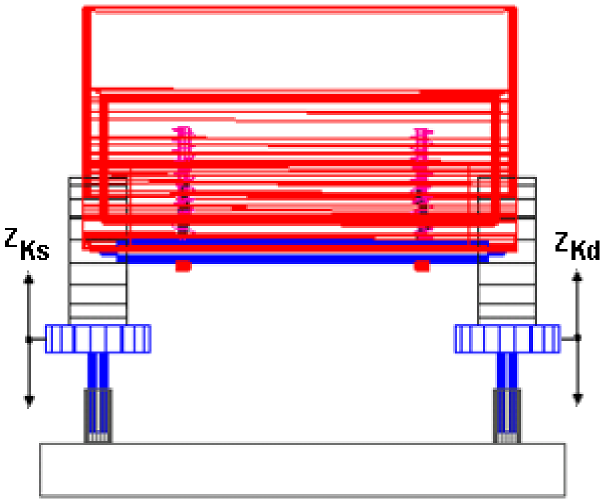


Figure 4. The virtual simulator

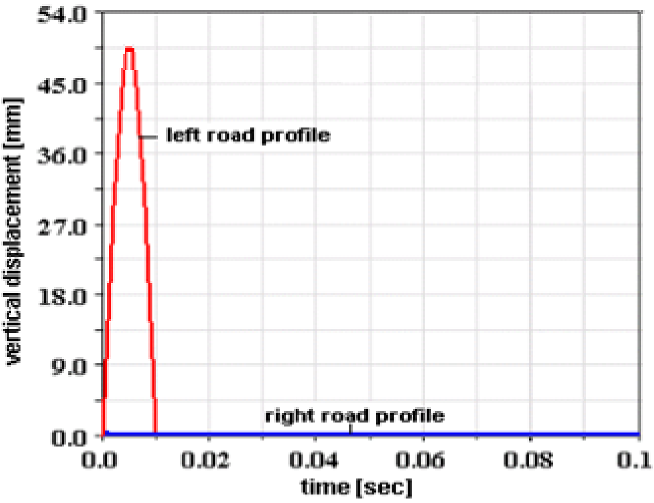


Figure 5. The road profile for simulation

The mathematic function used to model the motion generator is: IF(expression_1: expression_2, expression_3, expression_4), where "expression_1" is the evaluated function, and "expression_2, _3, _4" represents the function value for "expression_1"<0, =0, respectively >0. Therefore, for the left road profile the vertical displacement is: IF(time-0.01: 50*sin(pi*time/0.01), 50*sin(pi*time/0.01),0).

In these terms, the above-described model contains 7 mobile parts (see figure 3): car body - 5, rear axle - 4 (including the wheel rims, which are rigidly connected to the axle), upper guiding link - 3, left and right lower guiding links - 1/2, left and right driving actuators - 6/7. The total number of degrees of freedom (DOF) represents the difference between the number of allowed part motions (n - the number of moving parts) and the number of geometric and kinematic constraints (Σr), through the Gruebler count, $DOF=6\cdot n-\Sigma r$, as follows:

- generalized coordinates for 7 moving parts: $7\times 6=42$,
- geometric constraints:
  - translational joints (actuators - ground): $2\times 5=10$,
  - spherical / planar joint (car body - ground): $1\times 3=3$,
- kinematic constraints (vertical displacements of the actuators): $2\times 1=2$.

Therefore, the dynamic model has DOF=42-(13+2)=27 degrees of freedom (independent generalized coordinates). As was mentioned, the connections of the guiding links to the car body and axle are made through bushings, which are compliant joints with six elastic restricted degrees of freedom (so, these are not geometric constraints, but force generating elements).

3. RESULTS AND CONCLUSIONS

To compare the behavior of the half-car models, they were comparatively analyzed and simulated in relation to the full-vehicle assembly. The full-vehicle prototype (fig. 6), in detail described in [2], takes into consideration the guiding - suspension system of the rear axle as well as the front wheels. For the independent suspension of the front wheels, four-bar mechanisms (double-wishbone suspension) are used. The lower and upper control arms are connected to the wheel carrier with spherical joints. Two bushings connect the lower / upper arm to the car body. The steering mechanism contains a rack and pinion gear that translates the rotary motion of the steering wheel into the linear motion of the rack. Universal joints connect the three steering shafts.

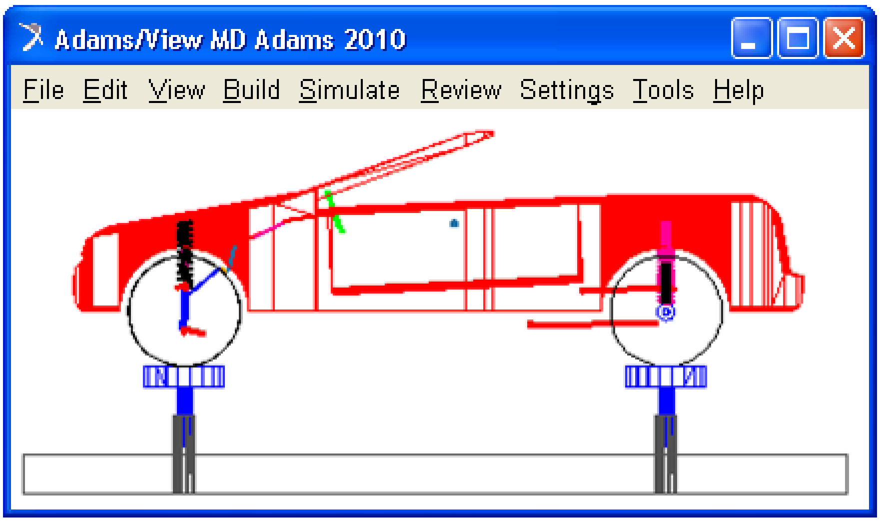
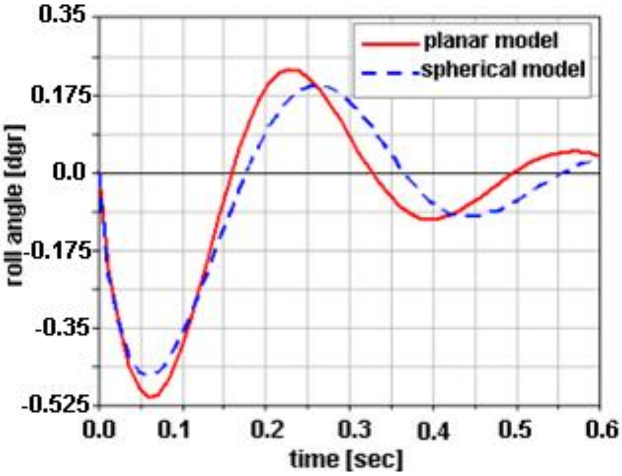
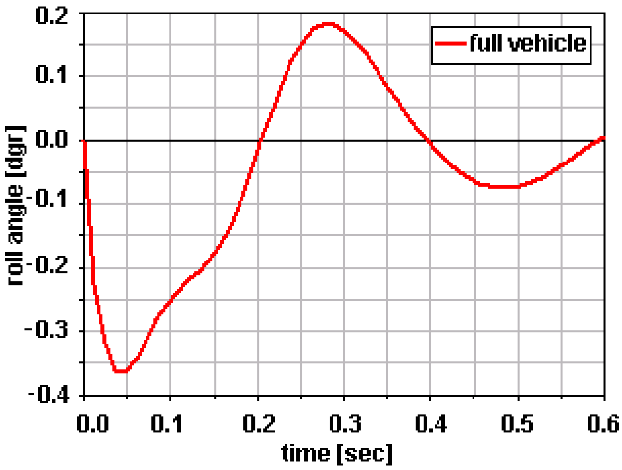


Figure 6. The full-vehicle virtual prototype



a.



b.

Figure 7. The variation of the roll angle

Analyzing the behavior of the half-car models in study (fig. 7, a), relative to full-vehicle assembly (fig. 7, b), we can conclude that the dynamic response of the half-car model with spherical joint is close to full-vehicle behavior. In addition, this model allows all important motions of the car body (vertical displacement, roll and pitch oscillations), while the half-car model with planar joints cannot consider the pitch rotation. Therefore, the half-car model with spherical joint is more useful to study the dynamic behavior of the vehicle (before the conception of the full-vehicle assembly).

ACKNOWLEDGEMENT

This work was supported by CNCSIS-UEFISCDI, project number PNII-IDEI 607/2008.

References

[1] Alexandru, C. Testing the Guiding - Suspension System of the Motor Vehicles in Virtual Environment. International Review of Mechanical Engineering, Vol. 3 (5), 2009, p. 521-528.
[2] Alexandru, P., Vişa, I., Talabă, D., Alexandru, C., Antonya, C. Static & Dynamic Analysis of the Wheels Guiding Mechanisms (in Romanian), Ed. Lux Libris, Braşov, 2005.
[3] Ambrosio, J., Verissimo, P. Sensitivity of a Vehicle Ride to the Suspension Bushing Characteristics, Journal of Mechanical Science and Technology, Vol. 23 (4), 2009, p. 1075-1082.
[4] Bernard, A. Virtual Engineering: Methods and Tools. Proceedings of the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture, Vol. 219 (5), 2005, p. 413-421.
[5] Campbell, C. New Directions in Suspension Design: Making the Fast Car Faster, Bentley, 1991.
[6] Chernykh, V.V. Multicriterion Optimization of the Kinematics of the Spatial Mechanisms of Passenger Car Wheel Suspensions, Journal of Machinery Manufacture and Reliability, Vol. 36 (5), 2007, p. 419-424.
[7] Fischer, E. Standard Multi-Body System Software in the Vehicle Development Process. Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-Body Dynamics, Vol. 221 (1), 2007, p. 13-20.
[8] Haug, E. J., Choi, K. K., Kuhl, J. G., Vargo, J. D. Virtual Prototyping Simulation for Design of Mechanical Systems. Journal of Mechanical Design, Vol. 117 (63), 1995, p. 63-70.
[9] Hegazy, S., Rahnejat, H., Hussain, K. Multi-Body Dynamics in Full-Vehicle Handling Analysis. Journal of Multibody Dynamics, Vol. 23 (1), 1999, p. 19-31.
[10] Höhne, G. et al.. Extended Virtual Prototyping. Springer, 2007.