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INFLUENCE OF COMPLIANCE FOR AN ELASTOKINEMATIC MODEL OF A PROPOSED REAR SUSPENSION

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ABSTRACT−The research is carried out to improve passenger's comfort to increase the vehicles stability in dynamic conditions. The literature available in the automotive engineering considers different topics for studying suspensions. An example represents mechanisms structure and analysis (synthesis, kinematics, and dynamics) under various operating conditions. These aspects have been approached before analytically, numerical. The current paper studies the influence of the lateral force on the contact patch of the wheel and the corresponding variations of vehicle stability parameters, such as camber angle and wheel rear track. The study is performed for a newer innovative rear suspensions mechanism which does not have a wheel track and camber angle variation, relative to the chassis, when the suspension components was considered rigid. A numerical solution is obtained through a virtual model on several commercial codes: MSC Adams, Patran, Nastran. Concerning the analysed parametes, their variation increases as the applied force is increased. Moreover, the largest variation corresponds to the case were elastic bushings and deformable links are considered.

KEY WORDS : Automotive suspension, Elastokinematics, Camber angle, Track width

1. INTRODUCTION

The automotive industry draws up many research problems.The literature in the field is available in best represented in papers reference such as: Zomotor (1987); Lukin *et al*. (1989); Ellis (1989); Bastow and Howard (1993); Reimpell *et al*. (2001); Reimpell and Betzler (2005); Genta and Morello (2009a, 2009b), etc.

A problem that is studied intensively by both researchers and companies represents the improvement of passenger car suspensions structures, increasing passenger comfort and vehicle dynamic.

An important task is to improve the features of the suspension and, hence mechanisms. A brief state of the art regarding the study of the automotive suspension configurations and their kinematic analysis with influence on some vehicle parameters is provided below.

Knapcsyk and Dzierzek (1995) present a vector algebraic method to analyze the displacement and force of the five-rod wheel suspension mechanism of a very generalized and independent rear suspension

Cambiaghi *et al*. (1996) present a method for the force-displacement analysis based on PC program of a five-rod suspension to study rubber bushing compliance effects, to minimize camber angle and track width changes and to result in a toe-in tendency under throttle-off condition, thus improving stability at a rear axle.

A Michelin Optimal Contact Patch (OCP) concept (2001) proposes an innovative automotive rear suspension mechanism in which the transverse arms of the two wheels are jointed to a mobile frame linked in turn on the car body by link rods. This determines an additional degree of mobility of the two wheels in comparison with conventional axles that leads to negative camber at cornering.

Laurent and Sebe (2001) have patented an active camber suspension in order to improve operating safety vehicle and maintain the tires in a position relative to the ground; thus minimize the extremely severe stresses and improve fatigue strength.

Simionescu and Beale (2002) studied the kinematics of a classic rigid body guidance mechanism multilink suspension to determine a minimum variation of wheel track, toe angle and camber angle during bump (jounce) and rebound of the wheel.

Rocca and Russo (2002) present a kinematic analysis algorithm for a multilink suspension taking into account the joint compliance. This proposed analysis could be used in a procedure based on the solution of a typical non-linear least-squares problem to determine elastokinematic parameters starting from experimental data. The comparisons between MSC ADAMS results and the numerical procedure in the

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case of elastokinematic analysis show a good agreement.

Heuze *et al*. (2003) analyze the kinematic influence of the structure (position of the joints of the Michelin optimized contact patch concept (OCP) on several parameters - half-track and camber angle. The two-dimensional suspension mechanism was studied semi-analytically and numerically.

Knapczyk and Maniowski (2006) perform tests and experiment the kinematic and elasto-kinematic characteristics of the five-rod suspension with subframe. The proposed vector-algebraic approach is useful in iterative elastokinematic studies. The influence of the compliant bushings (in the suspension rods and subframe) on the position and orientation of the wheel is confirmed.

Alexandru (2009) proposes an analytical method solved numerically using the DELPHI software to optimize kinematically a rear multi-link mechanism. The method is based on the determination of global coordinates of the joints fixed on car body to generate an imposed axle trajectory.

Alexandru and Alexandru (2010) study the virtual prototype of an automotive front suspension system with an actuators that generate force for a quarter-car model containing a guiding and suspension system of a front wheel. The virtual prototype of the active suspension (a control loop) contains a multi-body mechanical model connected to the dynamic model of the actuator and the controller model.

Esfahani *et al*. (2010) study kinematic aspects of suspension variable camber mechanism using Visual Nastran. The model is represented by a double wishbone front suspension mechanism for which a hydraulic camber angle mechanism is introduced to improve vehicle stability and reduce the rubber abrasion.

Dobre *et al*. (2010) approach the modeling of own proposal models using the Autodesk Inventor. The analysis of operation of double wishbone front suspension mechanisms gives their influence on the guiding of front wheels is appreciated through the variations of the track half-width and the camber angle depending on the wheel vertical displacement.

Dobre *et al*. (2011) present research starting from the analysis of an existing variant (Ford and Volkswagen cars) of multi-link rear axles leading to a multi-link rear axle structure proposed by authors. The models have incompatibilities resulting from different causes (inaccurate measurement, used types of joints - rigid or semi-elastic, supplemental arm, etc.).

Tica *et al*. (2011) proposed an innovative rear suspension mechanism having in structure a mobile frame and an actuator at each rear wheel mobile frame. The paper studied the variations of camber angle and wheel rear track depending on the wheel travel, taking into account the deformations of joint and arms.

From this state of the art the following main conclusions could be noted:

- the structure of automotive suspension mechanisms and their kinematical effects on different parameters influencing the comfort of the passengers and the dynamic of the vehicle remain an actual focused research (Bastow and Howard, 1993; Reimpell *et al*., 2001; Société Michelin, 2001; Alexandru and Alexandru, 2010; Esfahani *et al*., 2010, etc.);
- the use of the concept of active camber angle by the use of an actuator in the structure of mechanism is of actual main interest to influence the vehicle dynamic stability; some examples are given by Laurent and Sebe (2001), Esfahani *et al*. (2010) and newest Tica *et al*. (2011);
- the presence of the compliant bushings and flexible arms in the study of vehicle suspension mechanism is another important trend in the future research (Knapczyk and Dzierzek, 1995; Cambiaghi *et al*., 1996; Rocca and Russo, 2002; Knapczyk and Maniowski, 2006, etc.).

The present paper continues the approaching of similar own aims mentioned before. The aim of the paper is paying attention on the influence of the lateral force in the contact patch of the wheel on the variations of vehicle stability parameters (camber angle and wheel rear track). The study is approached for an innovative proposed mechanism of rear suspension having two main features:

(1) it has in structure two parallelogram mechanisms for each wheel, when other similar structures use quadrilateral mechanisms;
(2) the use of two parallelogram mechanisms is dictated by the existence of an important component: an actuator which corrects the movement of the wheel to obtain the desired variations of the rear wheel track.

The study is approached for a rigid mechanism (rigid components) and mechanisms having deformable components (compliant bushings and flexible arms). Two important characteristics are proposed in connection with these structures:

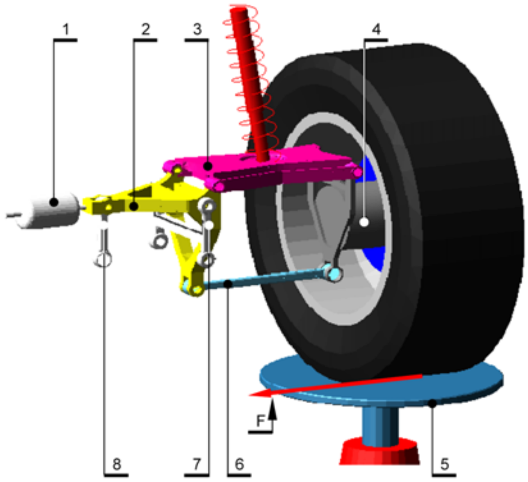
- the camber angle is constant in the case of the rigid mechanism and variable at the existence of deformable components;
- the compensation of the rear wheel track variation determined trough actuator at rigid mechanism, this leads to an active track suspension mechanism with an operating advantage: elimination of the lateral forces on the wheel caused by this variation.

The influence of these features highlights important conclusions for practice.

2. ON PROPOSED SUSPENSION MECHANISM

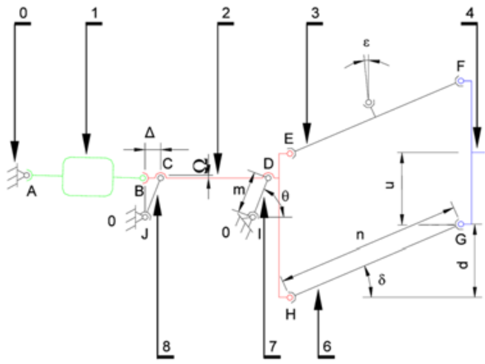
A 3D view of the proposed mechanism is given in Figure 1 for the right wheel, it being similar for the other wheel. The kinematic scheme of this mechanism is drawn in Figure 2.

(1) The two parallelogram mechanisms are the joints: J, C, D and I, and, respectively, E, F, G, H.
(2) The actuator 1 moves the mobile frame 2 through the joint B.



Elements: 1 – actuator; 2 – mobile frame; 3 – upper arm; 4 – wheel carrier; 6 – lower arm; 7 – outer arm; 8 – inner arm

Figure 1. Right suspension mechanism 3D model.



Elements: 0 – car body; 1 – actuator; 2 – mobile frame; 3 – upper arm; 4 – wheel carrier; 6 – lower arm; 7 – outer arm; 8 – inner arm.
Joints: A – between 0 and 1; B – between 1 and 2; C – between 2 and 8; D – between 2 and 7; E – 2 and 3; F – between 3 and 4; G – between 4 and 6; H – between 2 and 6; I – between 7 and 0; J – between 8 and 0.

Figure 2. Kinematic scheme at the right wheel of the proposed suspension mechanism in the maximum bump travel position.

(3) In a plan representation the mobile frame is connected to the outer and inner arms 7 and 8 by the joint D and C. Also this mobile frame is connected to the lower and upper arms 3 and 6 by the joint E and H.

The element 5 is a mobile plateau that makes the bump and rebound on vertical of the wheel.

An observation: the damper is not considered in the mechanism structure, because the wheel oscillations are not studied.

The suspension mechanism is loaded with the action of the lateral force F acting in the contact patch of the wheel.

(4) The link 7 have a four joint quadrilateral structure and connect the mobile frame to the car body :
a. by two joints on the lower part (I joint) at the same vertical and lateral position;
b. by two joints on the upper part (E joint) at the same vertical and lateral position;

In order to produce a suspension system stiffer is possible that the two joint link 6 and link 8 can be substitute with a four joint quadrilateral structure, like the link 3 and 7. This action increase the elements number and the complexity of the proposed structure.

The actuator compensates:

(1) the variation of the wheel track by the displacement to exterior (to right on Figure 2) of the joints E and H;
(2) the changes of the camber angle and wheel track if the compliant bushing and flexible arms are used.

3. KINEMATIC ASPECTS

The expressions have been deducted by Tica *et al*. (2011). Further only the expressions used in the paper studies will be retaken. The actuator distance between the external joints of the actuator is:

$$AB = \sqrt{(AB_i + \Delta)^2 + \Omega^2} \tag{1}$$

in which: B_i is the initial position of the joint B figured in Figure 2 in the bump displacement of the right wheel (given as entry data in an application, so that the arm length AB_i is known); Δ and Ω – displacements shown in Figure 2. The actuator distance (1) will be expressed in function of the bump u of the wheel using the following expressions:

• the lateral displacement Δ:

$$\Delta = m\cos\theta \tag{2}$$

• the vertical displacement Ω:

$$\Omega = m(1-\sin\theta) \tag{3}$$

where the angle θ can be expressed in function of the angle δ:

$$\theta = arc\cos\frac{n(1-\cos\delta)}{m} \tag{4}$$

• the angle δ can be expressed in function of the bump u of the wheel and the arms length m and n:

$$\delta = 2\cdot \mathrm{arc\,tan}\left(\frac{\sqrt{4\cdot n^2\cdot m^2 - u^4 + 4u^3 m + 4u^2 m^2}}{4\cdot n^2 + u^2 - 2mu} + \frac{2nu - 2mn}{4\cdot n^2 + u^2 - 2mu}\right) \tag{5}$$

The actuator displacement AB is presented in the Figure 10. It depends on the bump or rebound travel u of the wheel and the arms length m and n; it will be used to control the wheel track variation depending on the bump u of the wheel according to the analysis procedure discussed below.

4. ANALYSIS METHODOLOGY

Procedure of analysis consists in the following steps.

(1) The geometry of the entire system including the

suspension mechanism and the wheel was created in Autodesk Inventor Professional 2008.

(2) The geometry of the mobile frame 2 and upper 3 and lower 6 arms were imported in MSC Patran 2007 as STEP files. The mesh elements Tria 4 have been used. A multi point constrain (MPC) was used to constrain the nodes of the interior cylinder of bushing with respect to the node from the joint center to avoid the application of the loads (forces, moments) in a single node. Figure 3 illustrates two constraint states in MPC images of the mobile frame, for two different types of joints.

(3) Patran as preprocessor gives a command to MD R2 Nastran that generates the MNF files for flexible elements (mobile frame 2 and upper 3 and lower 6 arms).

(4) The STEP files for rigid components (others the ones deformable) and the MNF files for flexible elements are imported in ADAMS MD R2/View and jointed with rigid or compliant bushings. Then the motions of mechanism components are simulated considering the variable distance AB of the actuator depending on the bump or rebound (applied by a vertical oscillator, Figure 1) to the wheel.

(5) Finally the variations of the two interesting quantities (camber angle and wheel track) are established by ADAMS post processing depending on the bump and rebound of the wheel for several cases of study.

The study cases are:

(1) rigid joints and arms (of reference);
(2) compliant bushings and rigid arms;
(3) rigid joints and flexible arms;
(4) compliant bushings and flexible arms.

The following data are chosen for the simulation process:

- the force F in the wheel contact patch (Figure 1) is 0 N (case a), 2,000 N (case b), and 4,000 N (case c);
- the translational bushing stiffness is 5,000 N/mm (conventional practice value and constant during the simulation process);
- the material characteristics of steel for arms: Young's modulus $E = 2.1 \cdot 10^5$ MPa; Poisson coefficient $\nu = 0.3$; density $\rho = 7{,}800$ kg/m³.

The actuator displacement determined using the

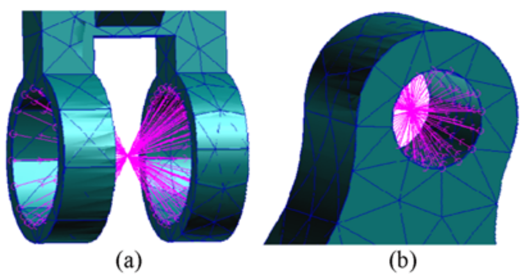


Figure 3. Images MPC of the joints H (a) and E (b) from Figure 2.

equations presented above was used to keep the wheel track constant in case 1 is also transmited to the mobile frame in the other cases.

5. RESULTS

The results given in diagrams show the variations of the camber angle and wheel track depending on the wheel travel on vertical direction. These diagrams are representative for the study cases mentioned above. The positive sense of the ordinate axis corresponds to the vertical motion of wheel when the suspension spring is compressed; the positive sense of the abscissa axis corresponds to the motion of the wheel to exterior in the sense of growth of the camber angle.

Figure 4 and 5 shows the variations of the parameters for the study case 2 comparing with the case 1. The variation of the camber angle (Figure 4) is the following:

- the values of the angle increase proportionally with the

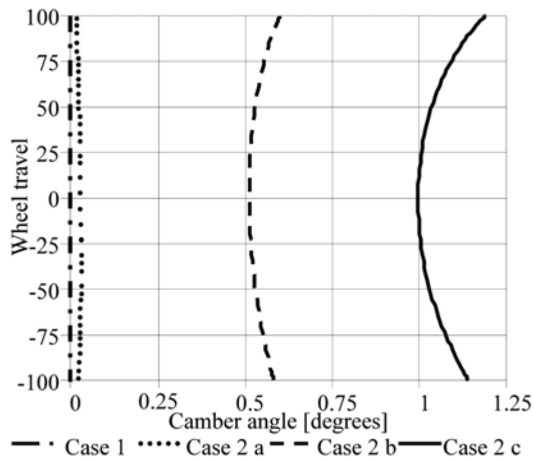


Figure 4. Camber angle variation in function of the wheel travel.

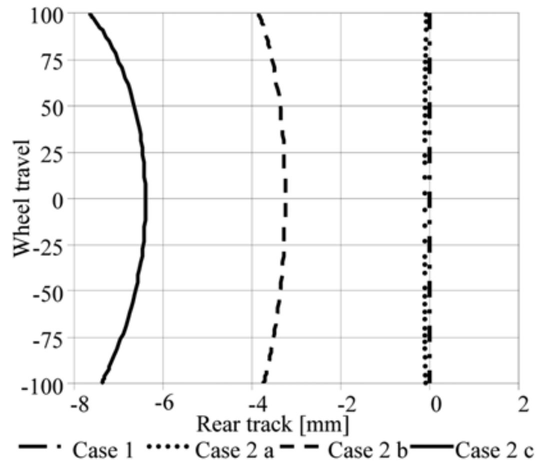


Figure 5. Rear track variation in function of the wheel travel.

lateral force;
- the values are marked with the wheel travel (at bump an rebound);
- the influence of the compliant bushing seems to be reduced at the absence of the lateral force on contact page.

Regarding the variations of the wheel track (Figure 5) the following conclusions could be drawn:
- the values decrease proportionally with the lateral force;
- the decrease of the wheel track accentuates with the entire vertical motion on the wheel;
- the conclusion mentioned for the camber angle remains: the influence of the compliant bushings is insignificant in relation to the case 1 (of reference) in the absence of lateral force.

The Figure 6 shows the camber angle variations which are similar to the case 2:
- the values is proportional with the lateral force;
- the increase of values is continuous for on the entire vertical motion of the wheel;
- the influence of the flexible arms is more important to the reference case 1, in comparison of the deformability previous case (Figure 4) in absence of lateral force on the contact patch.

The Figure 7 shows the wheel track variations which are similar to the case 2:
- the decrease is proportional with the lateral force;
- the decrease is accentuated with the entire vertical travel of the wheel;
- the influence of the deformability of arms is insignificant in relation to the reference case 1 in the absence of lateral forces.

The last case is plotted in Figure 6. The camber angle (Figure 6 (a)):
- increases with increased force lateral;
- increases with the bump and rebound courses;
- has a greater variation in relation to the reference case 1

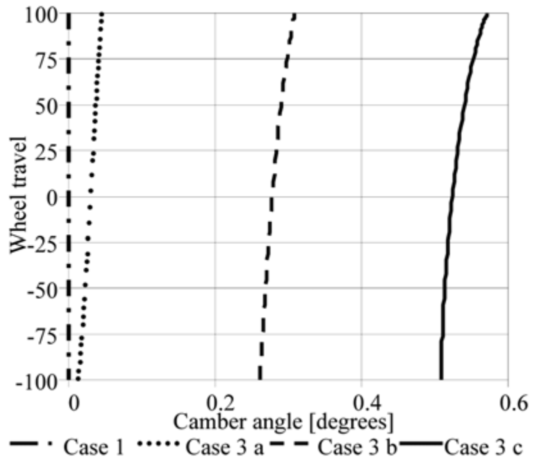


Figure 6. Camber angle variation in function of the wheel travel.

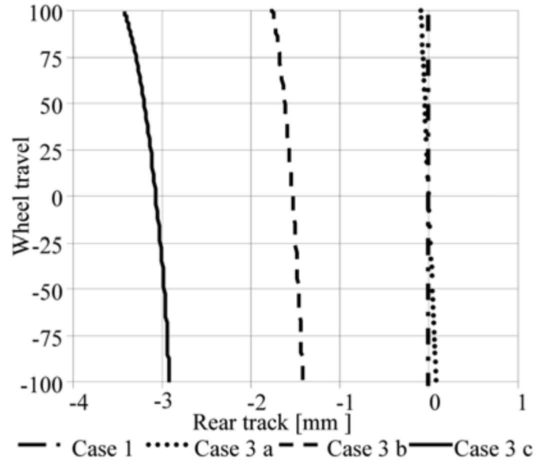


Figure 7. Rear track variation in function of the wheel travel.

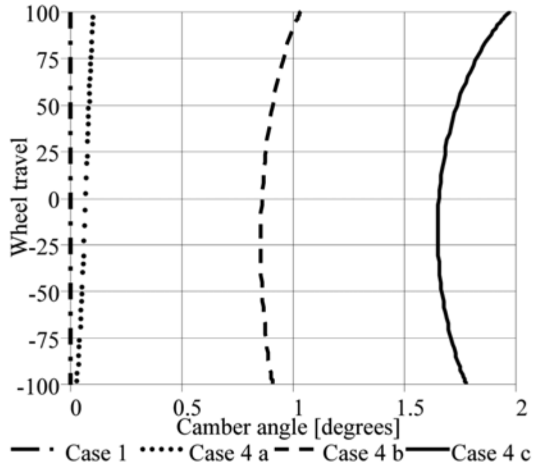


Figure 8. Rear track variation in function of the wheel travel.

in the absence of lateral force.

A comparative analysis of variation of studying parameters for different structure cases mentioned in Figure 4 and 5 (compliant bushings and rigid arms) and Figure 6 and 7 (rigid joints and flexible arms) is of interest. Thus, the analysis for the variations of the camber angle (Figure 4, Figure 6, Figure 8) and the wheel track (Figure 5, Figure 7, Figure 9) offers the following specific conclusions:

The deformability of compliant bushings has a great influence on studied parameters as those flexible arms, because the values and the shape of variations of the studied parameters are relatively (Figure 4 in relation to Figure 5) more important at the same parameter.

If there are the presence of deformable bushings and arms, their effect on the studied parameter is about cumulative of the singular cases having a single deformable element type.

The force and displacement provided by the actuator is

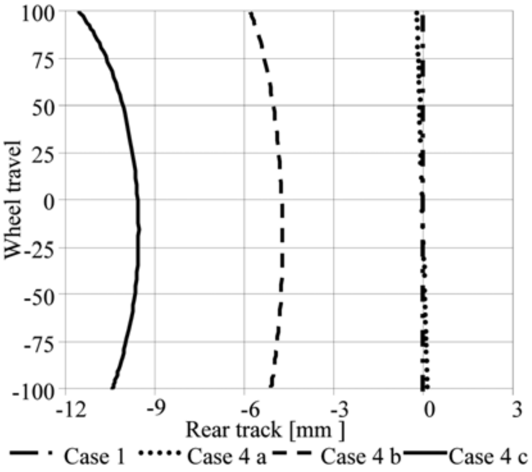


Figure 9. Rear track variation in function of the wheel travel.

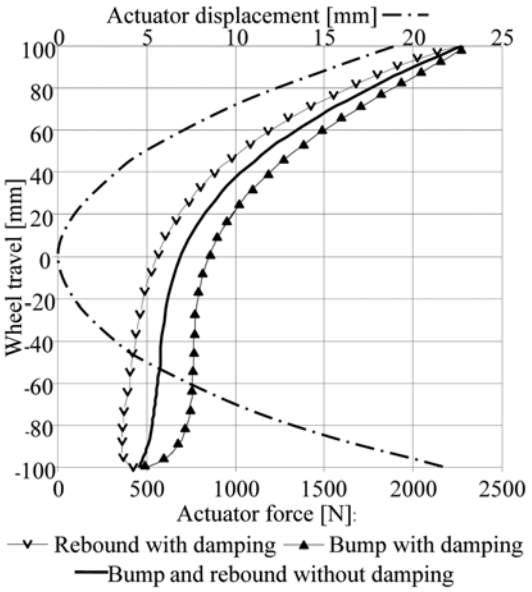


Figure 10. Force and displacement provided by the actuator in function of the wheel travel.

represented on the Figure 10 in function of the vertical position of the wheel.

When the damping characteristics was considered, the resulted force provided by the actuator have different value for the bump and the rebound wheel displacement.

6. CONCLUSION

The following conclusions could be enounced.

(1) The use of innovative automotive suspension mechanism brings constructive possibilities (actuator, deformability of bushings and arms) to influence the stability parameters taken in study (camber angle and wheel track).

(2) The actuator is an important constructive component integrated in the mechanism structure. Its main task is to make as active track suspension mechanism by the compensation of the variation of the rear wheel track.

(3) The camber angle is constant in the case of the rigid mechanism and variable at the existence of deformable components.

(4) The variations of the camber angle and wheel track are influenced by the deformability of components, but also by the operations factors (the lateral force from the wheel contact patch was considered as this factor in the paper).

(5) In the absence of the actuator, the wheel track parameter is influenced not only by the deformability of some components, but also by the constructive structure of mechanism.

(6) The tack variation of the ordinary suspension like McPherson or double wishbone is usually in the range of 20–30 mm (Reimpell 2001), without external lateral force acting on the contact patch. The kinematic advantage can be clearly seen since the proposed configuration have no track variation when the component flexibility is not considetated.

(7) The arm flexibility and bushing compliance has a semnificative influence on the suspension elastokinematics. When the lateral force is acting on the contact patch, the influence of the compliant bushings studied in the case 2 at the analyzed parameter is more important that the deformability of the flexible arms studied in the case 3.

(8) The variation of the camber angle and track width increase at the growing of the lateral force when the arm and bushing flexibility were considered.

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