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STUDY OF NEW AUTOMOTIVE REAR SUSPENSION MECHANISM

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ABSTRACT. *The paper presents firstly an introduction discussing: the problem's statement emphasizing the actual problem of research of passenger car suspension to improve driving comfort and dynamic of the vehicle movement; aspects regarding the studies of car suspension mechanisms; the paper's aims and objectives showing that a new configuration of suspension mechanism having in structure mobile frame and an actuator offering some important operational advantages is proposed. After this, the innovative concept is presented. The motions of mechanism elements and actuator and kinematic displacements are largely presented. The simulation using modern software (MSC Patran 2007, MD R2 Nastran and MSC ADAMS/View) accentuates the actuality of the results, which are presented and discussed for an application. Final conclusions are given.*

KEYWORDS. *Automotive. Suspension mechanisms. Mobile frame. Actuator. Displacements. Simulations.*

1.1. Problem's statement

The automotive industry draws up a lot of necessary researcher problems. Some of these are published, but another have a secret character. Nevertheless, a known literature in field is disposable in recognized titles: Zomotor (1987); Lukin, Gasparyants and Rodionov (1989); Reimpell, Stoll and Betzler (2001); Reimpell and Betzler (2005); Genta, and Morello in two volumes (2009), etc.

A problem studied intensively by researchers and companies is the improved structure of passenger car suspensions. The task is to improve thus some needed features, for example the driving comfort and dynamic of the vehicle movement.

1.2. On the study of automotive suspension mechanims

Regarding the studies of car suspension mechanisms the literature offers some interesting conclusions.

The variant of double wishbone suspension mechanism is most applied in practice because of its presenting advantages: different possible and simple configurations, simple study of different quantities, etc. (Genta, and Morello, Vol. 1 and 2, 2009; Reimpell, Stoll and Betzler, 2001; Reimpell and Betzler, 2005).

The literature offers largely studies in research of mechanism configuration and determined associated quantities. The present list of references gives a limited numbers of papers in this area. The paper of Simionescu and Beale (2002) starts from a brevetted mechanism solution having rigid and compliant joints (Michelin, Société Michelin, 2001) to study the kinematics using multibody simulation software. Mariotti and Ficarra (2008) studied McPherson suspension mounted on both the front and the rear the road holding of a virtual vehicle using multi-body simulation.

Some Romanian researches in this filed have been developed. Thus, the study of double wishbone front suspension mechanisms and car multi-link rear axle are studied by Dobre, Mateescu and Tica (2010), Dobre, Mateescu, Tica and Mirica (2011) or Mateescu, Dobre and Tica (2010).

1.3. Paper's aims and objectives

Starting from the idea to improve the driving comfort and dynamic of the vehicle movement, the present paper has the following aims:

1. the proposal of innovative suspension mechanism having in structure mobile frame and an actuator offering some important operational advantages;
2. the study (kinematics displacements) of this mechanism and associated actuator.

The following cases are studied:

1. rigid arms and joints;
2. rigid arms and compliant bushings;
3. flexible arm and rigid joints;
4. flexible arm and compliant bushing.

The study is promoted the null lateral force applied in wheel contact patch.

The simulation is largely applied in the present paper, using the following modern software offering some important operational advantages.

Results and their comparison and interpretation are followed by final conclusions that finish the paper.

An important mention: the theoretical results and their interpretation will be followed by physical tests, which are expected to be realized in future.

2. ON THE PROPOSED SUSPENSION MECHANISM CONFIGURATION

One of the suspension mechanisms pioneering the idea of the new solution was brevetted by Michelin (Société Michelin, 2001) (Fig. 1). This has a single mobile frame for both wheels; in this case the position in motion of both wheels is reciprocally influenced.

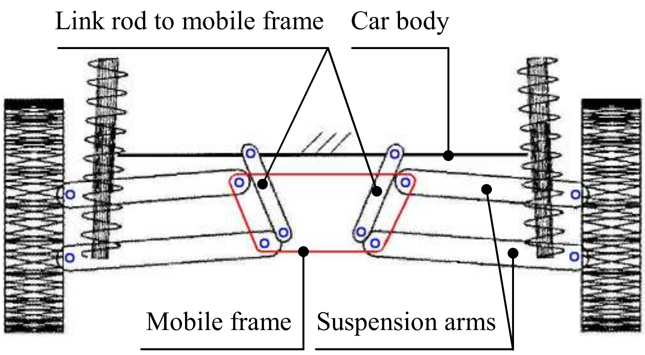


Fig. 1. Michelin OCP suspension concept (see Société Michelin, 2001) to optimize the contact patch (mobile frame represented in red)

The proposed mechanism configuration is given for the right wheel in Fig. 2 (the similar mechanism for the left wheel is symmetric). As difference to the above mentioned Michelin mechanism brevetted in 2001, the proposed configuration has the following main feature: while the Michelin mechanism has a single mobile frame for both wheels, the proposed mechanism have a mobile frame attached to each wheel; thus, two advantages are ensured:

1. both the wheels are not reciprocally influenced;
2. the possibility to control the displacement of the mobile frame using an actuator.

The following features for each mobile frame are mentioned below (Fig. 2):

1. mobile frame motion is controlled by an actuator 1 (jointed chassis 0 and the mobile frame 2) avoiding thus the variation of the track at the corresponding wheel;
2. the arms 3 and 6 and, respectively, 7 and 8 have the same length, to ensure to following feature;
3. the motion of each mobile frame is plan-parallel determining thus the second advantage: the absence of the variation of the camber angle at corresponding wheel.

The associated actuator determines the following advantages:

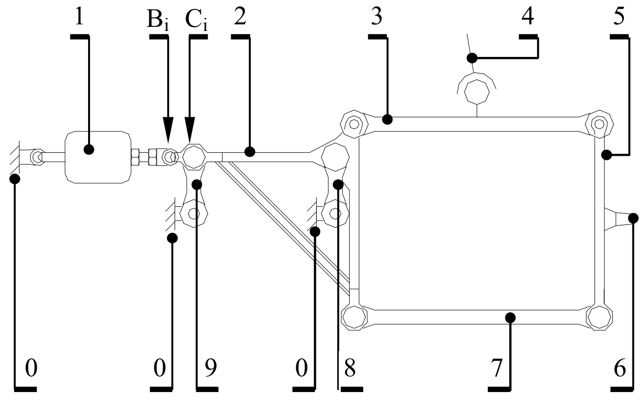
1. the compensation of the variation of the wheel track by the displacement of the joints E and I to exterior;
2. by a closed loop control the variation of the track due to deformations of flexible arms and compliant bushings could be compensated (case non-studied in the present paper).

In the some studying cases (2 and 4 from sub-chapter 1.3), the compliant bushings of horizontal arms are E, G, H, I.

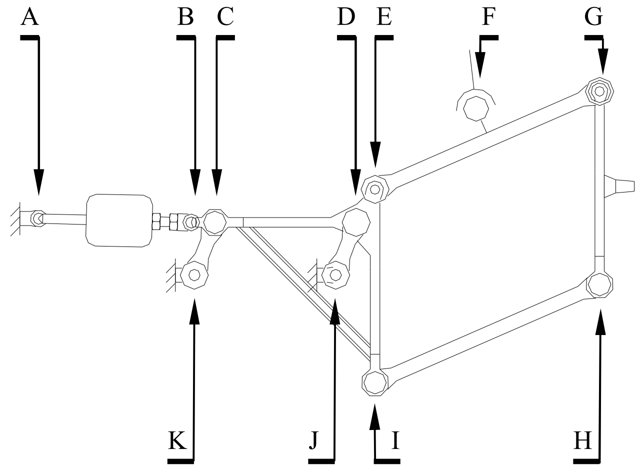
3. MOTIONS OF MECHANISM ELEMENTS AND ACTUATOR

In the initial static equilibrium position, the arms (equal as length) 3 and 7 have a horizontal position.

To maintain the right wheel carrier 5 in the same position (to preserve the same wheel track), the actuator 1 transmits a displacement to the mobile frame 2 (Fig. 2, a and b); thus, the joints E and I of the mobile frame are moved to exterior (Fig. 2, b and Fig. 3). In order to write the equations of the actuator displacement, a kinematics scheme is drawn in Fig. 3 by retaking some notations from Fig. 2 for a good understanding



a)



b)

Fig. 2. Proposed configuration of the automotive suspension mechanism for the right wheel

a) Mechanism in the initial position. b) Mechanism in the maximum bump travel position.

Elements: 0 – chassis; 1 – actuator; 2 – mobile frame; 3 – upper arm; 4 – damper rod; 5 – wheel carrier. 6 – bearing hub; 7 – lower arm; 8 – outer arm; 9 - inner arm;

Joints in the bump position b between: A –0 and 1; B –1 and 2; C – 2 and 8; D – 2 and 7; E – 2 and 3; F – dumper and 3; - G – 3 and 4; H – 4 and 6; I – 2 and 6; J – 7 and chassis; K – 0 and 8. Joints B_I and C_i in initial position a.

plane motion of the mobile frame 2 is produced. Thus, the joint I is shifted displaced in low and right and the arms 8 and 9 (with equal lengths) are rotated in their joints I and, respectively, K. As a result:

a) the joint H is displaced relative to I:
   - on vertical direction - with the distance d;
   - on horizontal (transversal) direction - with the distance Δ, the same between the joints C_i and C;
b) the joint C is shifted displaced on vertical direction relative to initial position C_i with the distance Ω; the

displacement of center of the wheel the wheel centre relative to ground (non-figured here), which is noted with u.

c) At the bump displacement of the right wheel, the parallel.

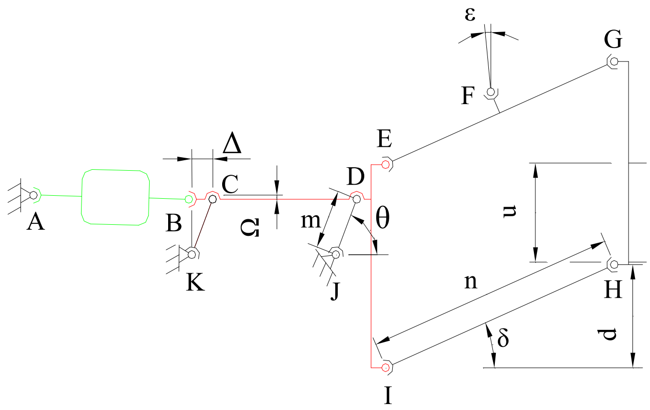


Fig. 3. Kinematics scheme of the automotive suspension mechanism for the right wheel in a displacement position

The notations of the joints are identical with the ones mentioned in Fig. 2, b appearing as new only the notation C_i of the initial position (non-displaced mechanism) of C

Color green: actuator 1. Color red: mobile frame.

4. KINEMATICS DISPLACEMENTS

The equations of some displacements are:

$$d = n \sin \delta ; \tag{1}$$

$$\Delta = n \left(1 - \cos \delta\right); \tag{2}$$

$$\Delta = m \cos \theta . \tag{3}$$

Equalizing (2) and (3), the following expression of the angle θ outcomes:

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

The other displacement Ω is:

$$\Omega = m \left(1 - \sin \theta\right). \tag{5}$$

The displacement u of the wheel centre (bump) is:

$$u = d + \Omega . \tag{6}$$

Replacing (1), (4) and (5) in (6), the following expression results:

$$u = n \sin \delta + + m \left[1 - \sin\left(arc \cos \frac{n \left(1 - \cos \delta\right)}{m}\right)\right]. \tag{7}$$

From this, the following expression of the angle δ is deducted:

$$\delta = 2 \cdot arc \tan\left(\frac{\sqrt{4 \cdot n^2 \cdot m^2 - u^4 + 4 u^3 m + 4 u^2 m^2}}{4 \cdot n^2 + u^2 - 2 m u} + + \frac{2 n u - 2 m n}{4 \cdot n^2 + u^2 - 2 m u}\right). \tag{8}$$

The absolute displacement of the joint C_i is the length of the chord $C_i C$:

$$C_i C = \sqrt{\Delta^2 + \Omega^2} . \tag{9}$$

Supposing a length AC_i between two joints of the actuator, the final length that will be studied depending on the bump is:

$$AB = \sqrt{\left(AB_i + \Delta\right)^2 + \Omega^2} . \tag{9}$$

4. APPLICATION

The length AC was used by MSC ADAMS/View software in order to determine the variation the track and camber angle depending on a given bump.

Entry data are give below.

Some lengths are generally usual for passenger car practice:

- $DJ = CK = 60$ mm;
- $EG = HI = 260$ mm.

In the case of compliant bushings used in the studying cases 2 and 4 mentioned in the sub-chapter 1.3:

- the linear stiffness of each bushing is adopted is 5,000 N/mm;
- the rotational stiffness and dumping and, respectively, translation dumping are considered negligible.

The material for arms is steel having the following data:

- Young's modulus, $E = 2.0 \cdot 10^5 \ N / mm^2$;
- Poisson's coefficient, $\upsilon = 0.3$;
- density, $\rho = 7.8 \ kg / dm^3$.

The following limits:

- for bump, $u_{\max} = 100 \ mm$;
- for rebound, $u_{\min} = -100 \ mm$.

The used procedure consists in the following steps.

1. The geometry was created by Autodesk Inventor Professional 2008.
2. The geometry were imported in MSC Patran 2007 as STEP files to generate the format MNF in MD R2 Nastran where a FEA study is performed.
3. The MSC Patran 2007 analyzed the FEA of the arms 3 and 7 and the mobile frame 2 using the disposable hard resources. The mesh elements Tria 4 have been used. For the arm 7, the number of mesh elements was 15,348 and the number of node was 3,861. 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 (Fig. 4). Patran gives a command to MD R2 Nastran to study the FEA process.
4. The resulted MNF file in MD R2 Nastran is imported in MSC ADAMS/View and is jointed with the rigid or compliant bushings of other elements. A bump or rebound is applied to the wheel center by means of a

vertical oscillator (fig. 5). The simulation is started, thus follow-on a lot of results.

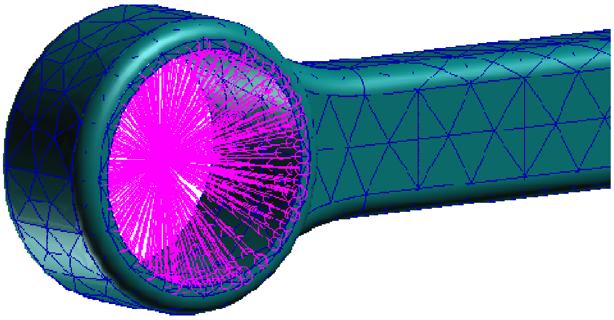


Fig. 4. Example of meshing and multi point constrain for an analyzed arm 7

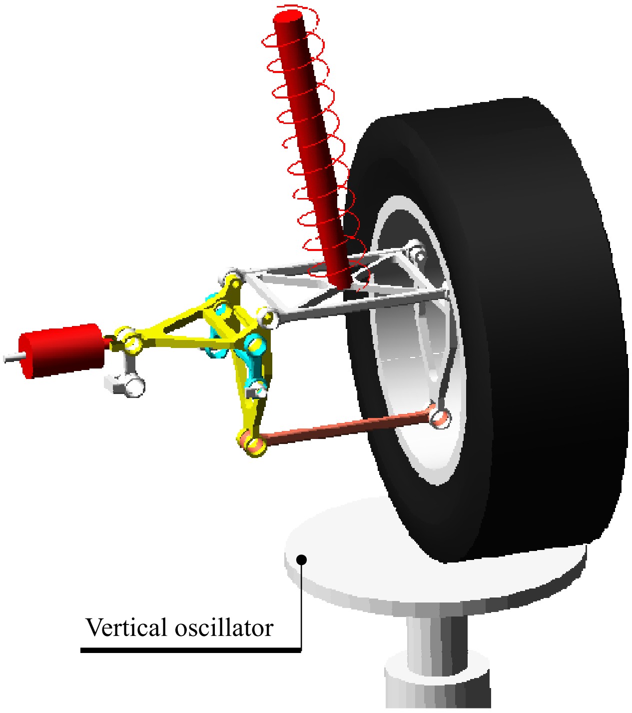


Fig. 5. 3D model created in MSC ADAMS/View

5. RESULTS, INTERPRETATIONS

Results. The simulation of the application on MSC ADAMS/View offers a lot of results of different parameters of elements or markers: positions, velocities, accelerations, kinetic energy, etc.

From this lot of results, the paper present only two quantities: camber angle and rear track; these were obtained using specific commands. The representation of these quantities is given in Fig. 6 and 7 for all cases mentioned in the sub-chapter 1.3 only for the absence of lateral forces on the wheel.

Some interpretations regarding the analysis and interpretations of the resulted data are given below.

1. If all elements (arms and bushings) are rigid (case 1), the suspension does not present variations of the camber angle and wheel rear track. This is a major advantage of driving comfort and dynamic of the vehicle movement.
2. In all other cases 2…4, the variations of the camber angle and wheel rear track are very low (negligible). Thus the proposed solution of mechanism including mobile frames and actuators improved the mentioned features: driving comfort and dynamic of the vehicle movement
3. Another observations: increasing the number of flexible arms and compliant bushings (passing from case 1 to case 4), an amplification of variations of the analyzed quantities could be noted, , but these remain negligible.
4. The obtained values determined by the innovative configuration of mechanism having mobiles frames and associated actuator are very different with respect to the values existent to practice mechanism of rear wheel.

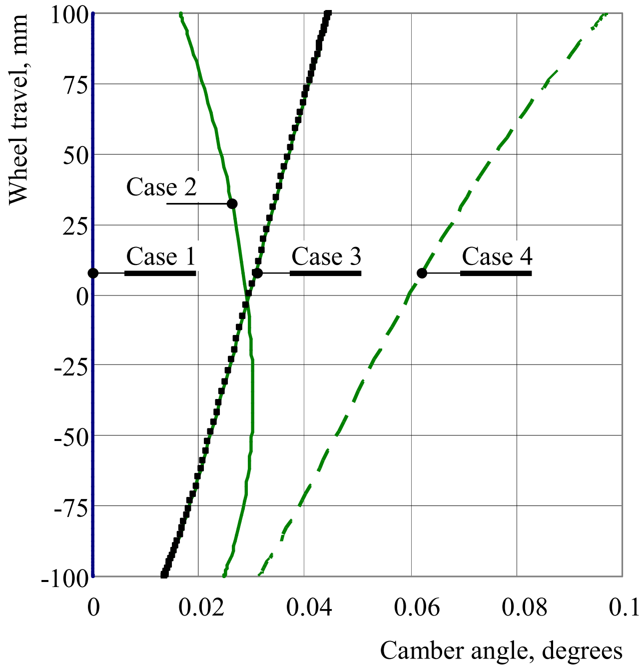


Fig. 6. Camber angle depending on the wheel travel

6. FINAL CONCLUSIONS

The conclusions are given punctually below.

1. The proposed mechanism has an innovative configuration with respect to the similar types (Michelin OCP suspension concept) by the existence of a mobile frame and actuator at each rear wheel.
2. The 3D representation made in Autodesk Inventor Professional 2008 offer the following studies by modern simulations software: MSC Patran 2007, MD R2 Nastran and MSC ADAMS/View.
3. This configuration adds new improved features on the driving comfort and dynamic of the vehicle movement with respect to other existent cars.

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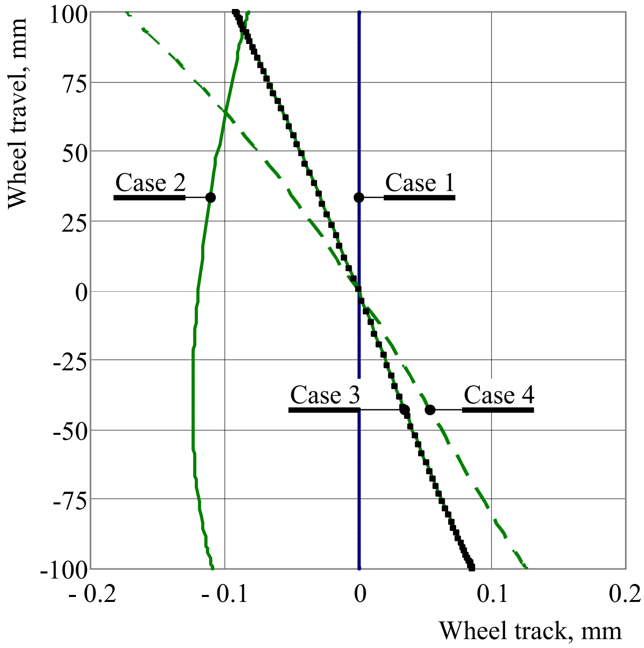

Fig. 6. Wheel rear track depending on the wheel travel

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