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DESIGN OF ACTIVE SUSPENSION MECHANISM

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Abstract. *This work deals with the design of a 3-DOF active suspension mechanism that is able to control Vehicle Body Roll, Toe and Camber angles. This type of mechanism is able to improve handling performance and vehicle stability control, when it is compared to a passive suspension. Several mechanism designs were generated and one of them was described in this work. Firstly, the initial design of a product was chosen for all the components of the new suspension system. It includes the layout for the mechanism to be assembled on a vehicle. Then, using the commercial software ANSYS, a structural analysis was developed by applying the finite elements method. The input forces considered in these analyses correspond to the maximum values obtained considering a vehicle executing two different maneuvers. The first one is the fishhook maneuver, which was used to analyze the vehicle rollover tendency. The second one, a breaking test, was applied to analyze the vehicle behavior during hard breaking conditions. The results show that the mechanism stresses caused by the forces imposed on it during these maneuvers are below the allowable stress.*

Keywords: *Suspension Mechanism, Active Suspension System, Finite Element Analysis, Vehicle Dynamics.*

1. INTRODUCTION

Traffic accidents have generated a large number of deaths in Brazil. In 2013, the DPVAT insurance paid 54,880 death benefits and 444,000 invalidity benefits (VIAS SEGURAS, 2014). Among these accidents, 18% of them were caused by car skidding and 3.3% were caused by rollover. As an attempt to attenuate this scenario, stability control systems for passenger vehicles have been employed. The most relevant control measurement applied for stability control systems is the ESP – Electronic Stability Program Stability based on active brake control. Several international studies carried by car manufacturers and safety agencies show that the ESP system can reduce by up to 80% the accidents caused by skidding, which is one of the main causes of serious traffic accidents. In Europe, the use of ESP in all vehicles resulted in an annual reduction of 4,000 fatalities and 100,000 injuries (BOSCH, 2017).

Although the ESP can contribute for vehicle stability control, this system presents some disadvantages, such as the vehicle speed reduction (Eduardo, 2009; Sohn and Park, 2012). In contrast, other means of control, so as active suspension and active steering systems (Deur, et al, 2011) can contribute to vehicle stability control without the loss of velocity unlike ESP. Moreover, many studies have shown that currently there is a tendency to integrate two or more technologies for stability control, such as in (Lu, et al; Rengaraj and Crolla, 2011).

Malvezzi and Hess-Coelho proposed a mechanism that controls the camber and rear steering angles, as well as provides an anti-roll control. This mechanism was designed on the base of a parallel kinematic structure that can contribute to vehicle stability, as shown in (Malvezzi, 2014; Malvezzi and Hess-Coelho, 2015).

This work aimed to design an active suspension mechanism, based on the mechanism presented in (Malvezzi, 2014). Firstly, the initial design was chosen, including all the components of the new suspension, as well as an assembly viability study. Following, a structural analysis was conducted to assure that every part complies with structural demands. Finally, the designs of all mechanism parts were primed.

2. COMPUTACIONAL PROCEDURE

In this work a design for a 3-DOF active suspension mechanism was developed. It is based on the RTC mechanism (a mechanism that can actuate on Vehicle Body Roll, Toe and Camber angles) proposed by Malvezzi (2014). According to the results presented in that work, the vehicle equipped with RTC active mechanism had an improved handling performance and stability control when compared to the original one, which is equipped with a passive suspension.

For the advantages of applying the RTC mechanism to make able, a layout study had to be conducted, so the assembly could be viable in a vehicle. To achieve this goal, the flowchart presented in Fig. 1 was employed. The commercial software ANSYS was used for structural analysis.

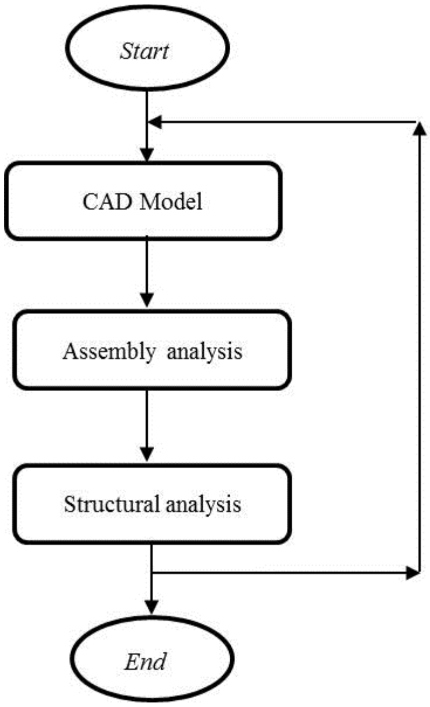

Figure 1. Flowchart for design of the active suspension mechanism

Figure 2-a shows the RTC mechanism proposed by Malvezzi (2014). Figure 2-b presented the first mechanism design generated in this work and Fig. 2-c shows the main parts of the final mechanism design of this work.

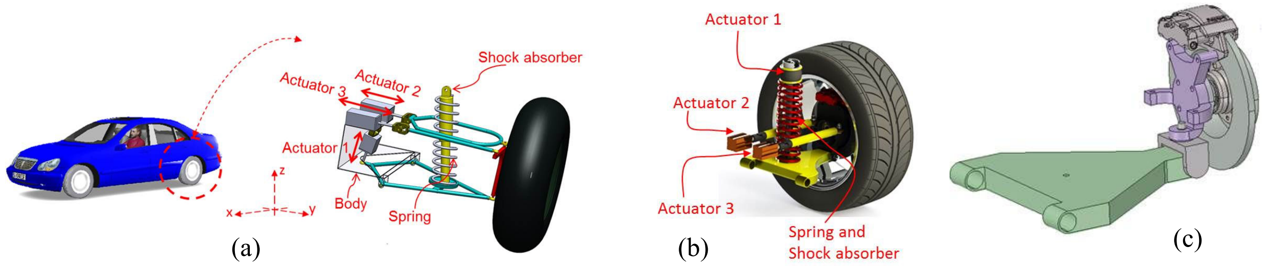

Figure 2. (a) RTC Mechanism; (b) First mechanism design; (c) Final mechanism design

According to Fig. 2-b, the first active motion sets the camber angle, which is achieved by the action simultaneously of the actuators 2 and 3 in the same direction. The second active motion sets the toe angle, obtained by the movement of the actuators 2 and 3 in the opposite direction. The third motion can be either passive or active. In the passive motion, it is assumed that the car does not perform evasive maneuvers or cornering. In this case, the actuator 1 does not exert force on the sprung mass (so there is no energy consumption) and the wheel travel is influenced only by the action of the spring and the shock absorber. On the other hand, the active motion provides anti-roll control by a force exerted by the actuator 1 on the sprung mass (body).

In order to analyze the mechanism mobility the Gruebler-Kutzbach criterion (Tsai, 1999), defined by Eq. (1), is applied.

$$M = \lambda(n - j - 1) + \sum f_f \tag{1}$$

where M is the mobility of the mechanism, the index λ corresponds to the space dimension where the mechanism is supposed to function, n is the number of links in the mechanism (including the fixed base), j is the number of joints in the mechanism, f_f is the number of degrees of freedom of relative motion permitted by joint f.

According to Figure 3, the mechanism has 8 limbs and 9 joints. By applying the Eq. (1), the mobility is equal to 3:

$$M = 6(8 - 9 - 1) + 15 = 3 \tag{2}$$

Figures 3-a and 3-b show the mechanism parts identification used to apply Gruebler-Kutzbach criterion. In Fig. 3-c the active joint R is equivalent to the spring-damper-actuator 1 assembly.

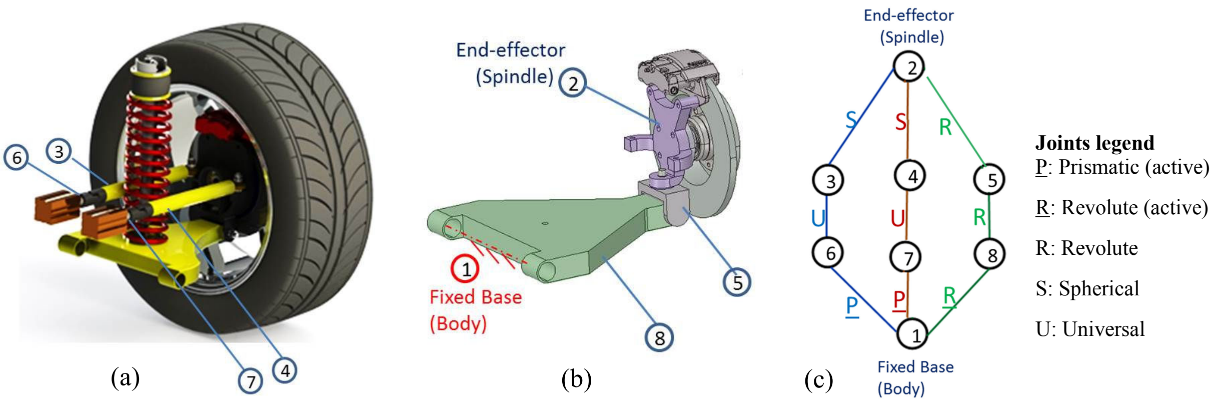

Figure 3. (a) and (b) Mechanism parts identification; (c) Mechanism graph representation

3. PREPROCESSING FOR FINITE ELEMENT ANALYSIS

All the required data for the simulations were extracted from the work of Malvezzi (2014). These data were generated from simulating the vehicle dynamic behavior on the CarSim software. It was employed a C-Class vehicle which 1,274 kg, available in the CarSim data library.

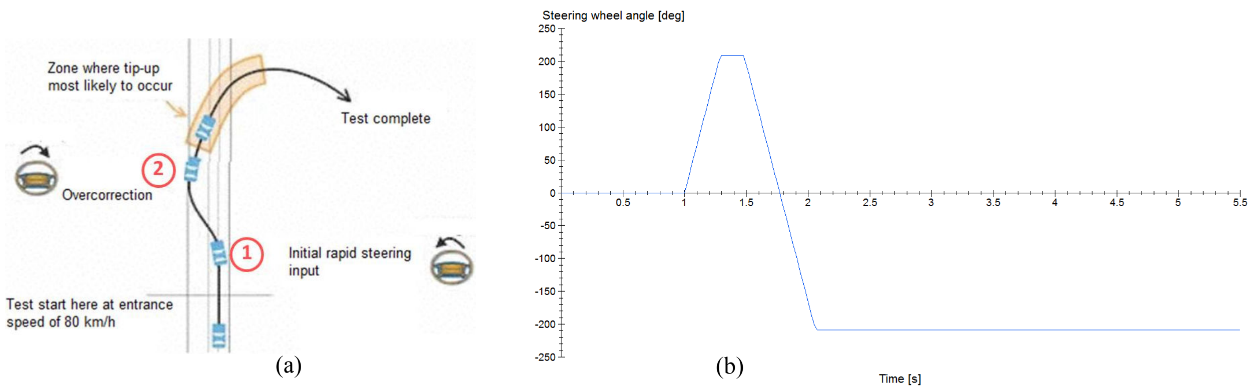

Figure 4. Fishhook Maneuver: (a) Test Procedure; (b) Steering wheel input

The input forces in the finite elements analysis correspond to the maximum values obtained considering a vehicle executing two different maneuvers. The first one is the fishhook maneuver, which is used to analyze the vehicle rollover tendency. This maneuver consists of applying an angular turn to the steering wheel to remove the vehicle from linear movement. After that, a fast correction response is applied and the steering wheel is turned in the opposite direction. The peaks of forces and moments on the rear left wheel were achieved in the second part of the fishhook maneuver (after overcorrection in Fig. 4).

The second maneuver considered the vehicle during hard braking conditions. The weight supported by the rear left suspension system is 2,000N. In addition, it was adopted a coefficient of static friction of 0.8 between the tire and the pavement.

Table 1 shows such values and net configurations employed in the finite element analysis. It is important to note that the tire forces and moments were defined considering the CarSim reference axis (Fig. 2-a), which is different from the ANSYS reference axis (Fig. 5).

Table 1. Inputs data employed in the finite element analysis (rear left wheel).

Tire Forces and Moments (CarSim reference axis)	Braking Test	Fishhook Maneuver	Units
Fx	1,600	-9	N
Fy	0	3,400	N
Fz	2,000	5,015	N
Mx	0	-115	N.m
My	0	0	N.m
Mz	0	53	N.m
Element Types	Hex Dominat	Hex Dominat	-
Midside Nodes	Dropped	Dropped	-
Element size range	1 to 10	1 to 10	mm

The lower control arm is fixed to the chassis by means of bushes. Therefore, the boundary condition used was a cylindrical support with free tangential movement. The upper control arms and their actuators were represented by springs with a rigidity of 2,000N/mm. Only the actuator 1 along with its shock system was represented by a spring's rigidity of 6,500N/mm (Fig. 5).

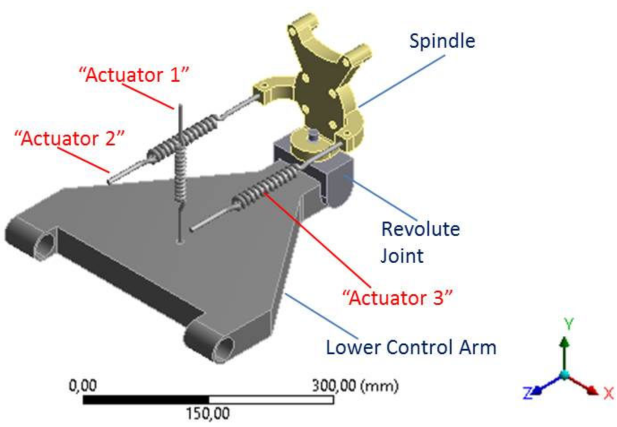

Figure 5. Pre-processing: mechanism CAD model

The parts that do not have any movement between than were bonded into a single component. As a result, the suspension system was composed only by three more important parts: the spindle, the lower control arm and the joint linking the lower control arm and the spindle (Fig. 5). The final geometry is shown in Fig. 6 and 7.

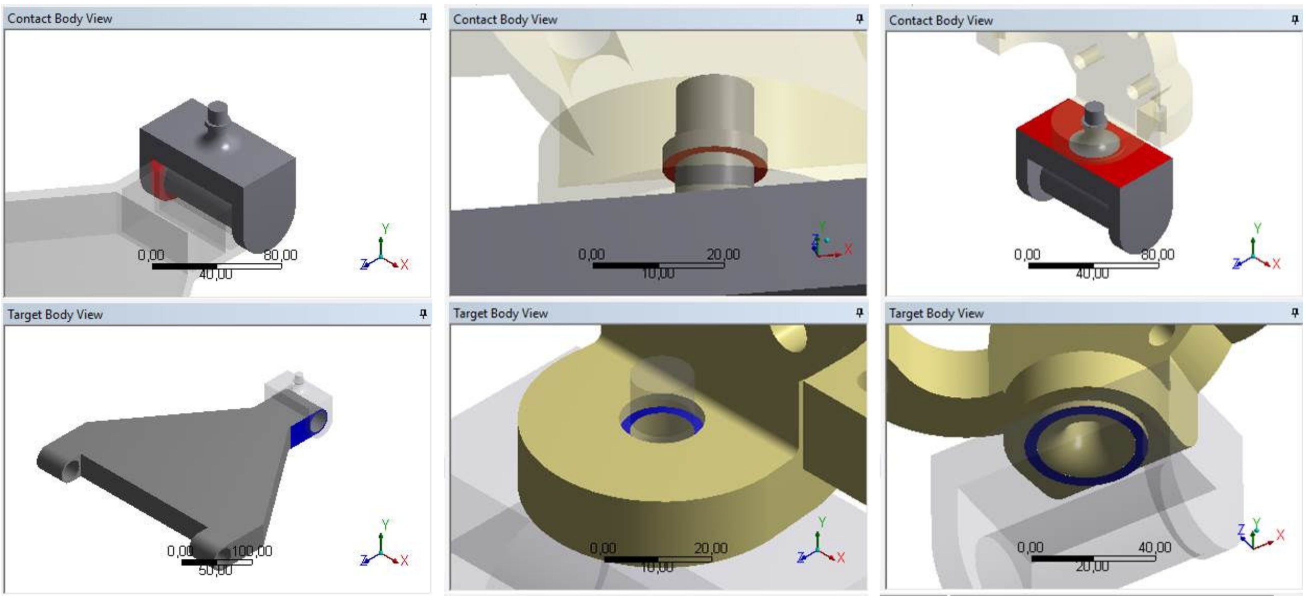

Figure 6. Frictionless contact

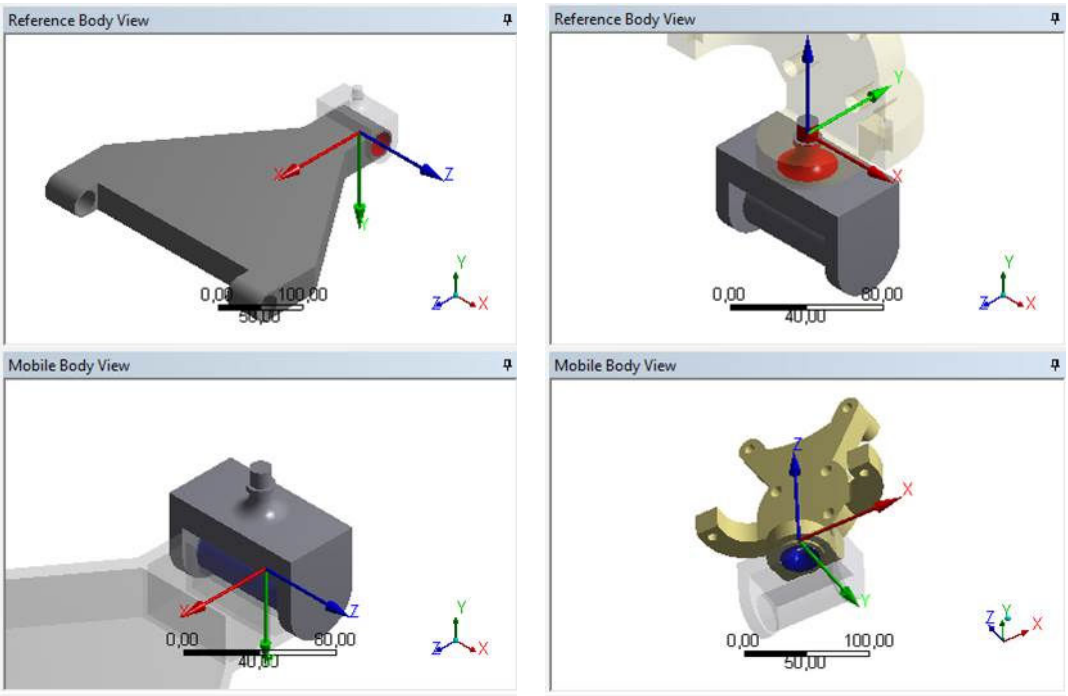

Figure 7. Spherical joint

4. SIMULATIONS AND RESULTS

As the results from the simulations were non-linear, the 'large deflection' command was enabled. As a result, the Ansys software executes a series of interactions adding portions of forces (steps) until the convergence is reached.

Figures 8 and 9 show such interactions for one simulation loop.

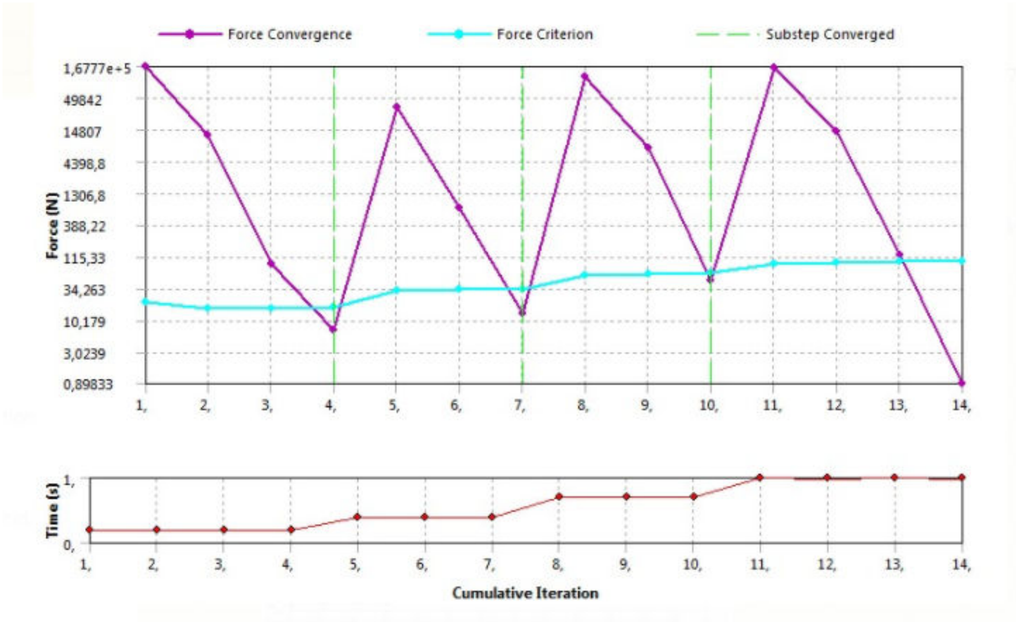

Figure 8. Iteration process for braking test

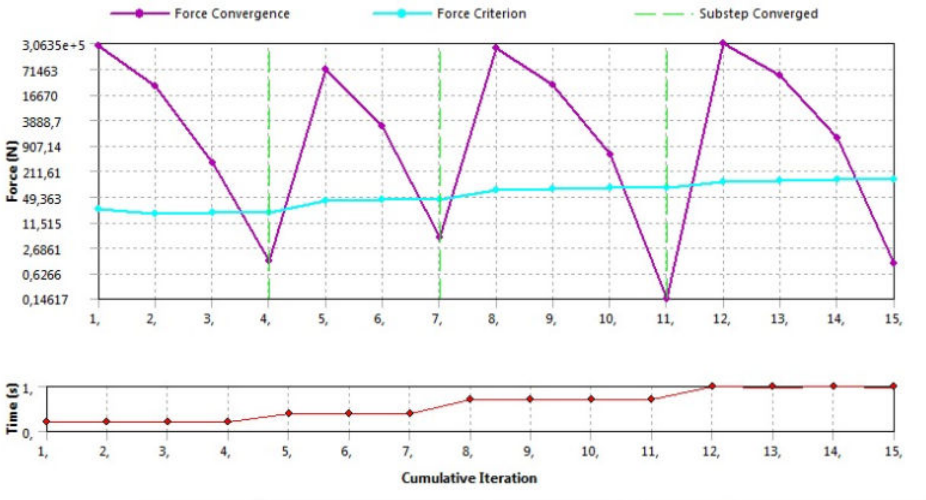

Figure 9. Iteration process for fishhook maneuver

4.1 Simulation Loops

By the end of the simulation 21 loops were done for each maneuver. The suspension's geometry was modified in each one these loops in order to minimize stress concentration regions. Some of these modifications were, for instance, the addition of fillets to eliminate sharp edges, ribs to increase torsion rigidity and minimize over-dimensioned components. Fig. 10, 11 and 12 portrait the most relevant modification loops (loops 4, 7 and 11).

In the loop 4 it was noticed a geometrical singularity, meaning that for both simulations with different loads there was a stress concentration point. It was solved by adding a thicker fillet in that point (Fig. 10).

In loop 7 it was noticed that the stresses for both braking and fishhook maneuvers decreased and are located in different regions of the suspension. During the former, the highest stress is located in the interior part of the lower control arm, close to the connection point between the chassis and the control arm. It was solved by adding ribs and then closing the control arm's profile (Fig. 11).

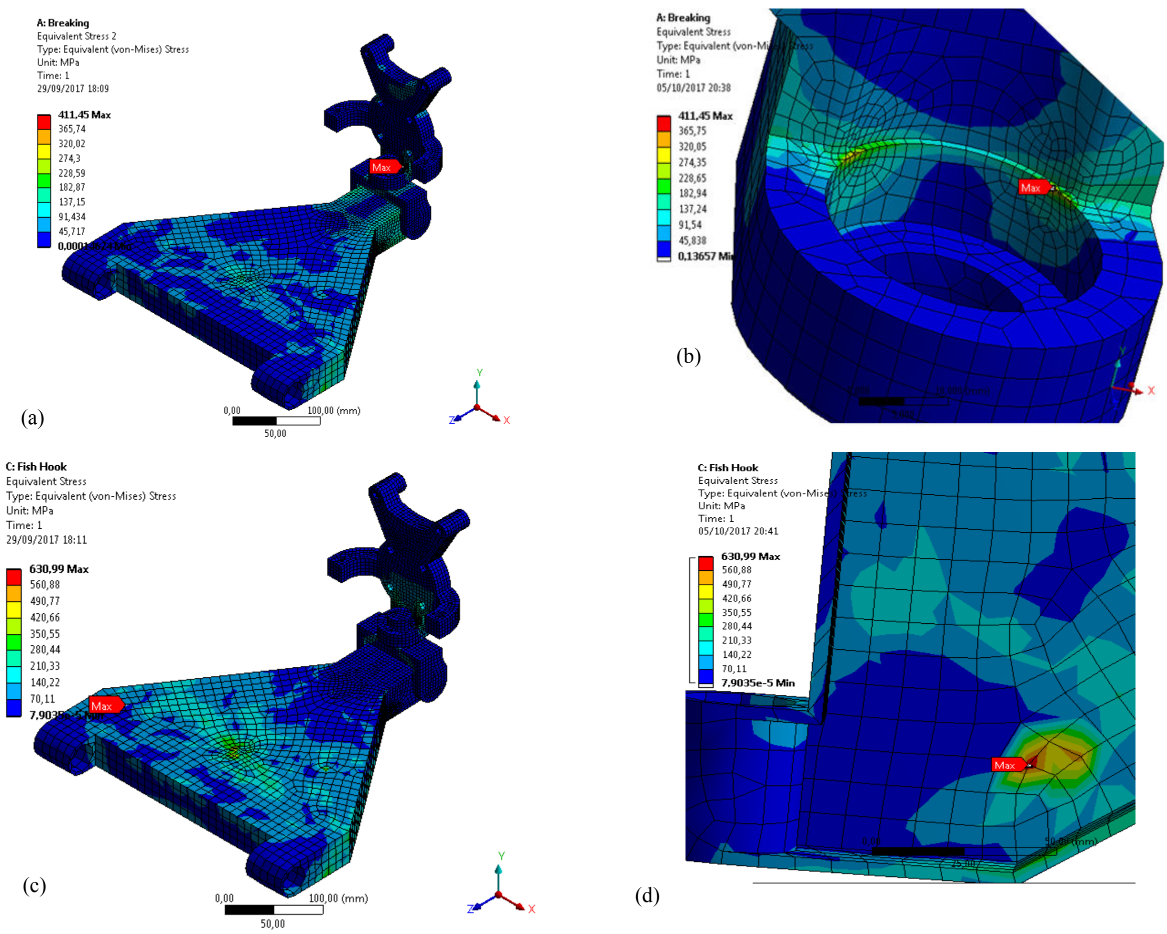

Figure 10. Loop 4 - Results for: (a) and (b) braking test; (c) and (d) fishhook maneuver

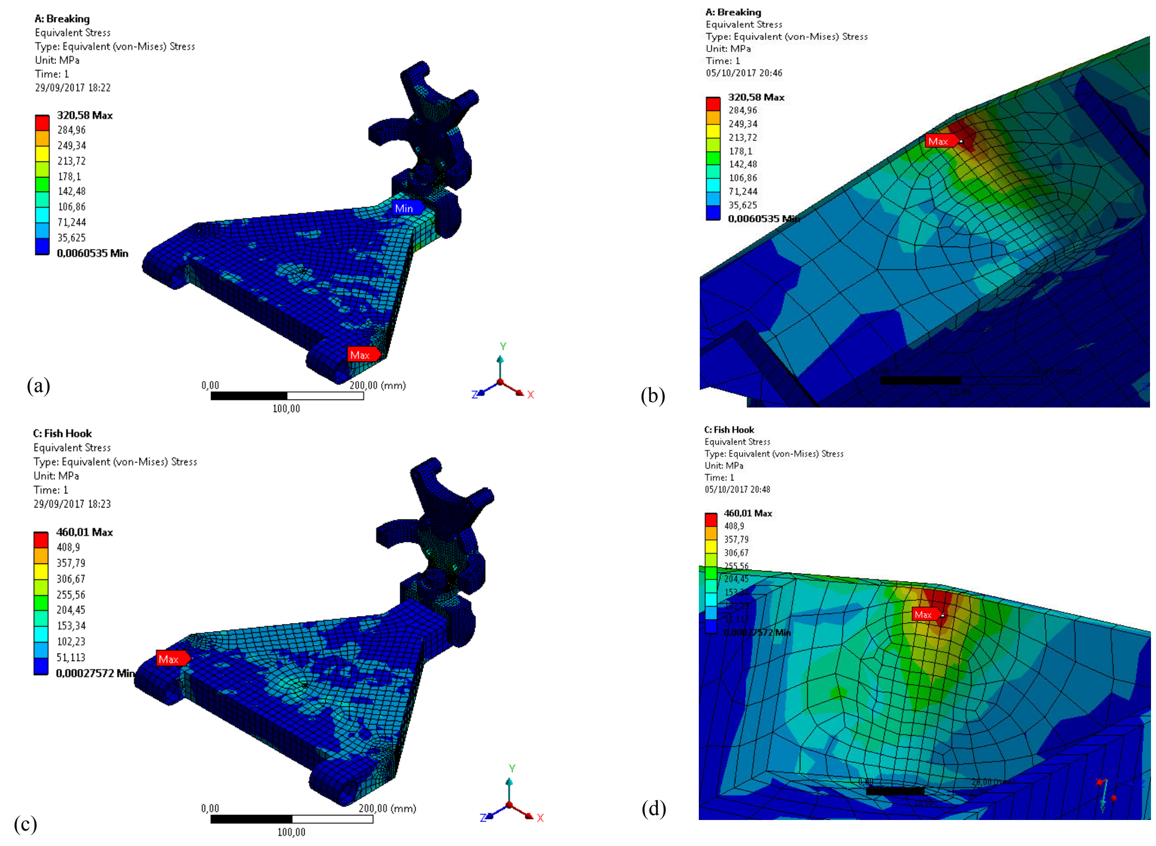

Figure 11. Loop 7 - Results for: (a) and (b) braking test; (c) and (d) fishhook maneuver

It was noted in loop 11 that for both maneuvers the stresses decreased and again they changed place. For braking, the maximum stress was concentrated in the crimping region of the pin in the articulation of the control arm, whereas in the fishhook the fillet of 3 mm was not sufficient to reduce the stress, since it only changed from one side to the other side of the spindle (Fig. 12). The obtained results in the loop 11 are not realistic, because after mesh-refinement in subsequent loops the equivalent stress increased to 650 MPa.

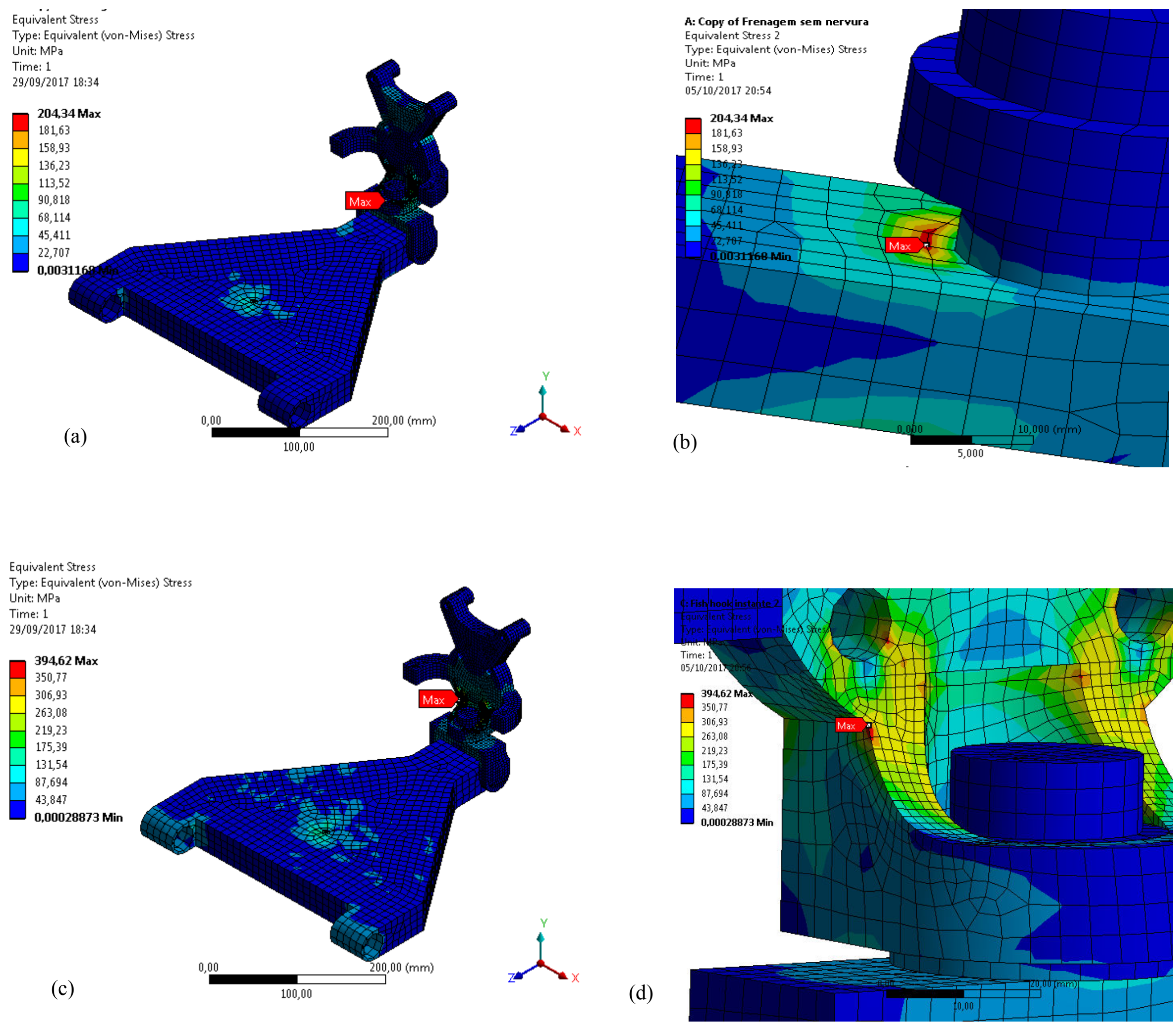

Figure 12. Loop 11: Results for: (a) and (b) braking test; (c) and (d) fishhook maneuver

4.2 Final Results

During each simulation, it was possible to observe loop the reduction and change of position of the maximum Von-Misses Stress. However, the initial target for the project was to obtain equivalent stress lower than 400 MPa. To achieve this target, it was necessary to decrease the diameter of the pin, thus, the fixing nut. This allowed for remodeling the spindle and removing the cut that allowed the mechanical assembly of the suspension. This change resulted in a uniform surface reducing the stress to 353.17 MPa in the fishhook maneuver.

At the critical instant of the fishhook maneuver (position 2 in Fig. 4-a), the vehicle tends to roll out of the turn, with the body approaching the left rear tire, compressing the spring, trailing the upper control arm and causing a buildup of stress in the fillet of the spindle, as shown in figure 13-b.

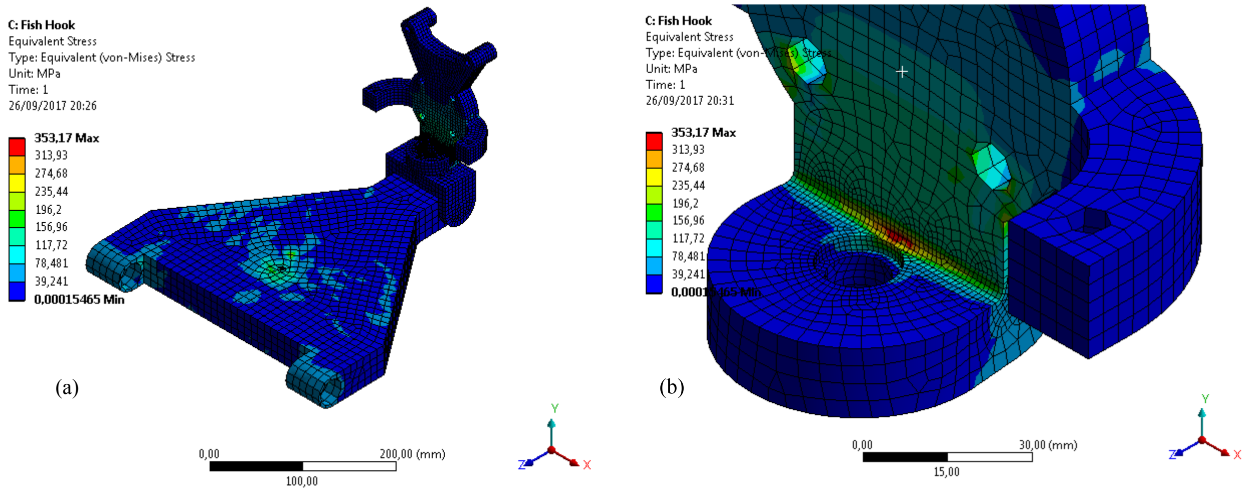

Figure 13. Final results of fishhook maneuver

After several loops for the braking maneuver, by changing the geometry, the point of stress concentration did not change, remaining in the pin-set region of the pin with the joint. It was possible to reduce the maximum stress to 412.57 MPa by increasing thickness of the joint. In the braking situation, with the proposed geometry, the forces and moments are transferred from the tire-pavement contact to the lower control arm. It causes a bending stress in the pin, according to Fig. 14-b.

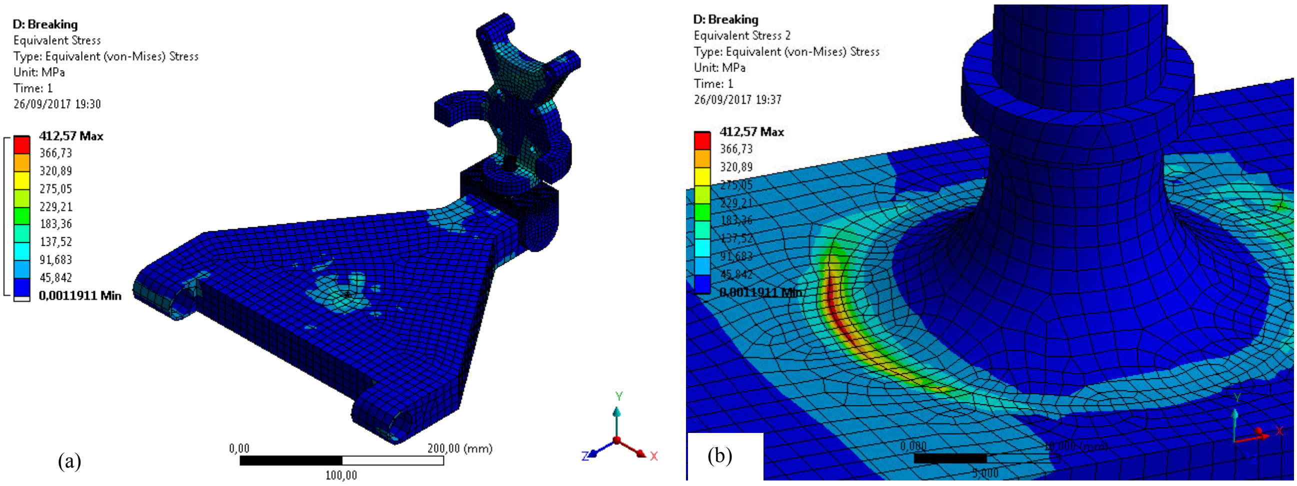

Figure 14. Final results of braking test

5. CONCLUSIONS

This paper describes the design of a 3-DOF active suspension mechanism that is able to control the Toe, Camber and Body Roll angles simultaneously. Firstly, the study of the layout for the mechanism assembly on a vehicle was carried out. Second, using the commercial software ANSYS, a structural analysis was developed by applying the finite elements method. In this analysis the forces and moments in the tire-road contact area were obtained making use of a C-Class vehicle running in two different situations: in the fishhook maneuver and under hard braking condition. Finally, the obtained results were analyzed. They revealed that the maximum equivalent stress is 2.7 times lower than the ultimate stress of high tensile steel. In order to build a mechanism prototype to be assembled on a vehicle, in the future woks a topology optimization, a fatigue and an impact analyses will be conducted. This vehicle will be used only to evaluate the performance of this mechanism at a proving ground.

6. ACKNOWLEDGEMENTS

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