OPTIMIZATION STUDY OF A CAR SUSPENSION–STEERING LINKAGE

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Abstract: *The study presents the main steeps and some results of a design, analysis and optimization process of a complex articulated mechanism required for a car suspension and steering. The authors used a top-down approach, starting from the requirements of the assembly and the close connections between the vehicle suspension and steering. Then the computer model was successively refined and new components were added as the design solutions became clearer. More steps were done, starting with a simplified analysis aiming geometrical optimization, continuing with a multibody study of the mechanism, part and assembly manufacturability-oriented CAD-modeling, a finite element analysis of some important parts and finishing with corrections and tunings of the CAD models.*

Key words: *MacPherson suspension linkage, rack-and-pinion steering, multibody analysis, finite element analysis, optimization*

1. INTRODUCTION

Designing a suspension-steering mechanism for an automotive vehicle is a complex task. Starting from the very beginning, a lot of requirement must be fulfilled. The process needs considering the vehicle kinematics and dynamics, continues with aspects regarding the general layout, packaging and compatibility with the other systems of the vehicle and finalizes with fine tuning, manufacturability and durability. The existence of many specialized software programs can be today very helpful for the automotive engineer. Such computer programs, used for part and assembly design, engineering computations and virtual testing, represent valuable tools in the engineering activities, assisting the teams to finalize projects in shorter time, with fewer mistakes and with smaller costs. This paper presents one computer aided approach to model and virtually test a suspension and steering mechanism for cars.

2. PRELIMINARY ANALYSIS OF LINKAGES GEOMETRY

The MacPherson articulated axle (schematized in figure 1 as a Catia-V5 2D-model) consist on a swing arm, connected to the vehicle body by a revolute joint placed in the lower side of the mechanism, and on a strut, connected to the body by elements behaving as an spherical joint.

The strut is made of two parts, having two degrees of freedom one vs. other: a translation along the strut axis, permitting the wheel bump-rebound movement, and a rotation around the same axis, permitting the wheel steering. The function of this cylindrical joint is taken by the suspension damper (shock absorber), reducing thus the number of parts needed to ensure the main suspension functions: cushioning, damping and wheel guiding.

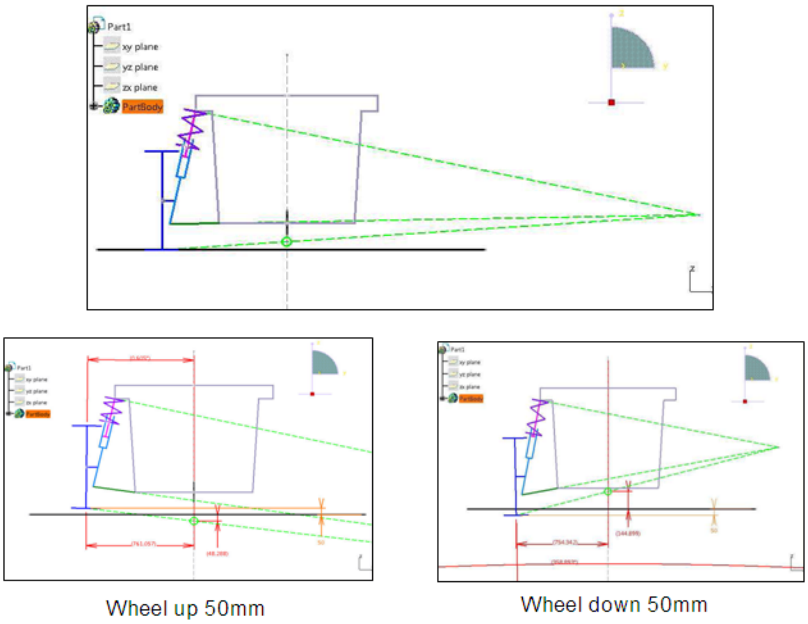

Fig. 1 – Schematic representation and behavior of the MacPherson suspension linkage

The MacPherson linkage is now commonly used on cars and vans because needs less lateral space on the vehicle, leaving enough free space for a transversal engine-transaxle mounting.

A vehicle steering mechanism consists of two subsystems:

- a steering linkage, (or better, a correlation mechanism), used to relate the swivel angles of all the steerable wheels (figure 2), so that to minimize positioning error of these wheels during cornering;

- an actuation mechanism, needed to transfer the rotation of the driver's steering wheel as a positioning command of the steering linkage.

For cars and vans, the most spread steering mechanism uses a rack and pinion steering box. In this type of steering mechanism, the rack is part of both steering linkage and actuation mechanisms, making this the simplest steering mechanism for wheeled vehicles.

Figure 2 schematizes the main components of such a rack and pinion steering linkage in the case of two-axle vehicle with independent front suspension. The actuation mechanism transmits the rotation (imposed by the driver) of the steering wheel trough the steering shaft to the pinion. The pinion rotation will determine the lateral translation of the rack and the change of the linkage position will force the road wheels to steer.

The project started with a multi-parametric optimization in a vertical-transversal plane of the suspension-linkage 2D-model. The parameters modified during optimization, between the limits permitted by packaging requirements, were: the arm's length and laying angle, the wheel scrub radius, the kingpin angle and the height of the strut-body joining point.

During wheel bouncing displacement, the optimization goals were:

- ensuring sufficient wheel bump-rebound travel;

- ensuring an acceptable ground clearance;

- maintaining minimal change of the wheel camber, so that to permit maximal tire grip;

- maintaining minimal change of the vehicle track width, avoiding thus the "fight" of the axle wheels, that generates useless "consume" of grip, poor handling and supplementary stress for linkage;

- maintaining the axle's roll-center as high as possible, in order to have small body roll, so diminishing its influence when cornering and consequently reducing the change of the wheel camber and track width.

Figure 1 shows graphical representations of the 2D simplified mechanism behavior during vertical wheel bouncing, obtained at the end of the 2D optimization process.

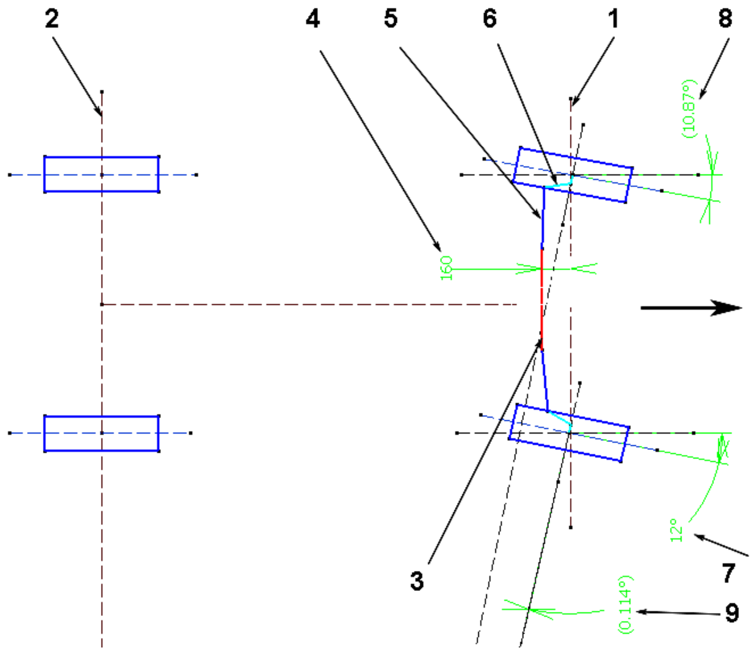

Fig. 2 – Simple steering model used to primarily adopt the design dimensional parameters

A similar geometrical optimization was made for the planar model of the rack and pinion linkage. As shoed in figure 2, this mechanism contains two steering arms 6, two steering rods 5, the rack 3 and the vehicle body as mechanism base.

In the lack of lateral forces on tires, a correct positioning of the steerable wheels (in this case, the ones of the front axle) can be realized only if the projections on the ground surface of the front-wheels centerlines intersect both, in the same point, the similar centerline projection of the rear wheels (position 2 on the figure 2). That is also called the Ackermann condition or the geometrical correct steering condition [2].

Lather, on the 3D model, the optimization process was resumed, considering supplementary the lateral slip angles affecting tires under medium to hard cornering maneuvers.

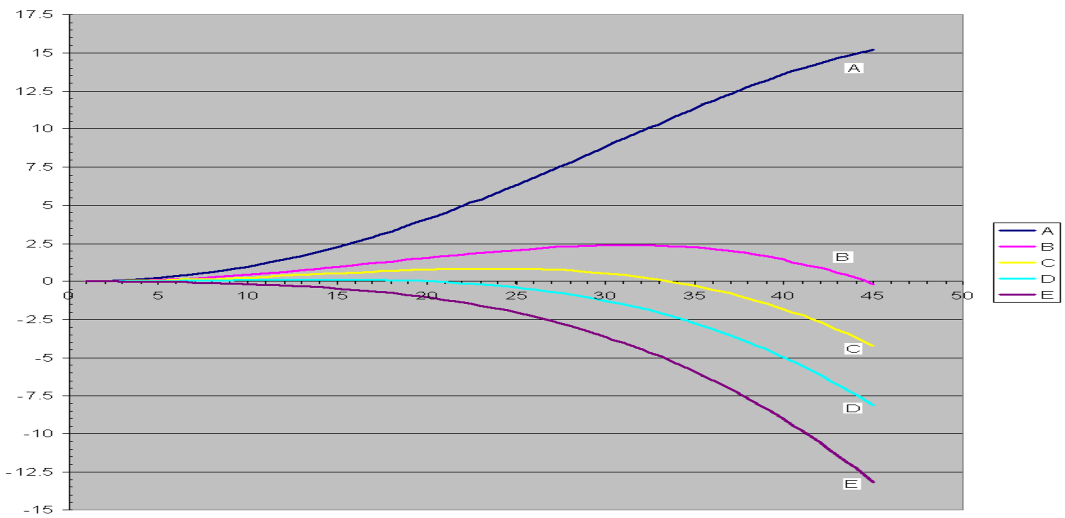

Fig. 3 – Exemplification of the error-angle variations as function of the inner-wheel's steering-angle for different rack positions

The fulfillment of that condition was the starting point for the multi-parametric optimization of the steering linkage. Considering as independent variable the inner-wheel's steering-angle 7, the aim of optimization ([1], [2]) was to reduce the error angle 9, i.e. the difference between the Ackermann-angle and the actual steering-angle of the outer-wheel (the first is the wanted angle, while the second is the actual linkage angle – position 8 in figure 2). The parameters changed during optimization (figure 2) were: the length of the rack 3; the position 4 of the rack with respect to the front-axle 1; the length of the steering arm 6; the position angle of the steering arm with respect to the corresponding-wheel's centerline.

The way the error-angle is changing versus the linkage dimensions and inner-wheel's steering-angle it is presented in figure 3. To appreciate the quality of a certain linkage layout (defined by a set of parameters considered at a time), the error angle was weighted in function of inner-wheel's steering-angle: biggest weighting values for small to mean values of the steering angles, which correspond to vehicle high speeds.

The layout noted with C in figure 3 was found to be the best. The procedure was repeated three times with different geometrical parameters and had as result the optimal layout for a planar model of the linkage. The process was also used to calculate the maximal steering angles that can be realized without the locking danger for the correlation mechanism.

3. MULTIBODY MODELING

Using the main dimensions obtained by the 2D geometrical optimization, the next step was to realize a draft Catia-V5 3D-model. The main parts of the suspension and steering systems were modeled, adding also the disk brake and the components of the wheel (hub, rim and tire), figure 4. The parts were then virtually assembled so that to respect packaging requirements and to avoid geometrical interferences.

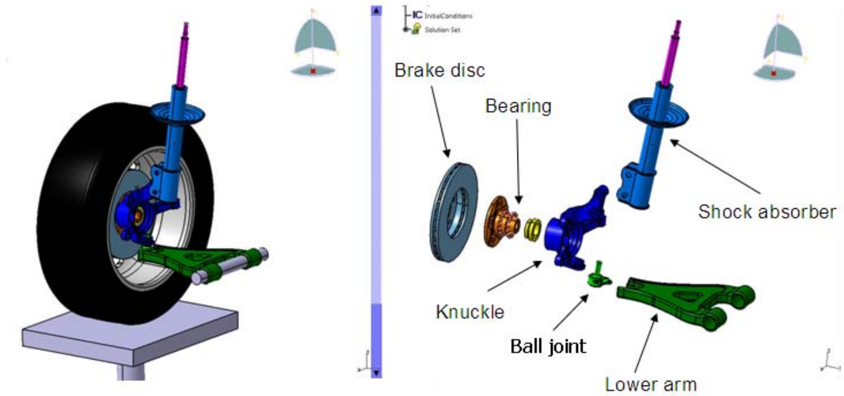

Fig. 4 – CAD (right) and MBS (left) models of the suspension

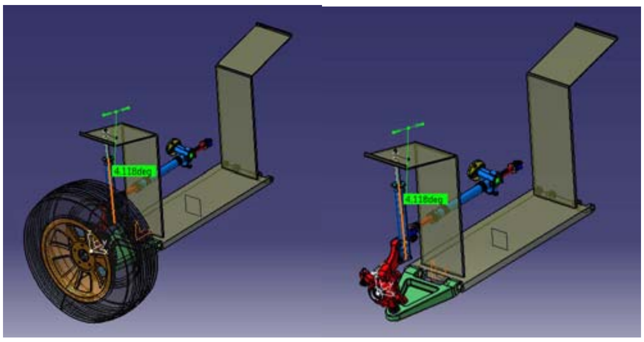

Fig. 5 –CAD model of the suspension-steering, considering the caster angle

Together with the wheel camber and the kingpin positioning angles, the other laying angle of the kingpin-axis, the caster-angle, was supplementary introduced into the CAD model (figure 5) and considered as design parameters for a new optimization stage [1], [4], [6], [7].

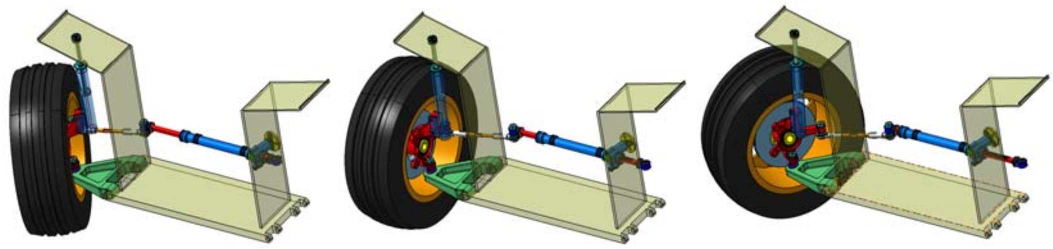

Fig. 6 – The multibody model subjected to the analysis
left – extreme left-steering, middle – straight-line travel, right – extreme right-steering

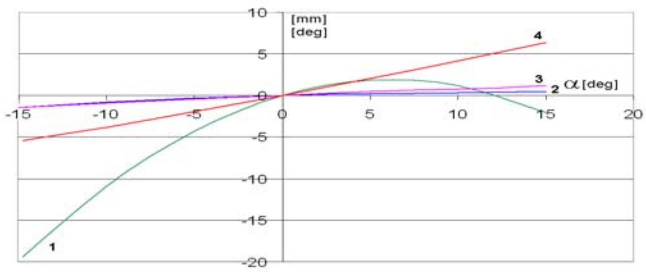

Fig. 7 – Multibody analysis results of axle kinematic-behavior on jounce-rebound (straight-line motion): independent parameter – the lower-arm swing-angle;
1 – track-width change; 2 – camber-angle change;
3 – steering angle change (bump-induced steering); 4 – wheel lift, divided by 10

Starting from the three-dimensional Catia model and using LMS Virtual Lab Motion software, a new model was realized. This multibody model (presented in figure 6, in the case of steering with no vertical-motion for the wheel) permitted to study the kinematics and dynamics of the steering-suspension mechanism. Final simulation results are presented in figure 7.

4. CAD MODELING

After the narrow examination of the mechanism behavior by multibody simulation, the parts design was revised, having in mind functional and manufacturing requirements. Consequently, the CAD models included the shapes of the preformed part and some intermediate manufacturing stages [3], figure 8. All the parts were virtually assembled to form firstly subassemblies (figure 9) and then the complex mechanism of steering and suspension. The packaging of all the parts and the mounting-dismounting possibilities were also being verified.

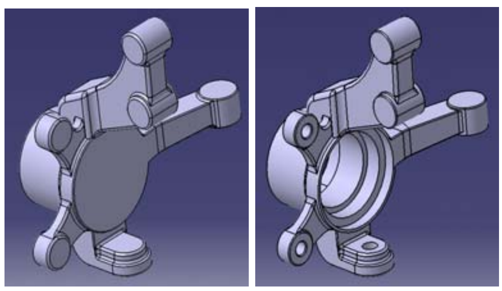

Fig. 8 – Manufacturing-oriented CAD modeling (knuckle)

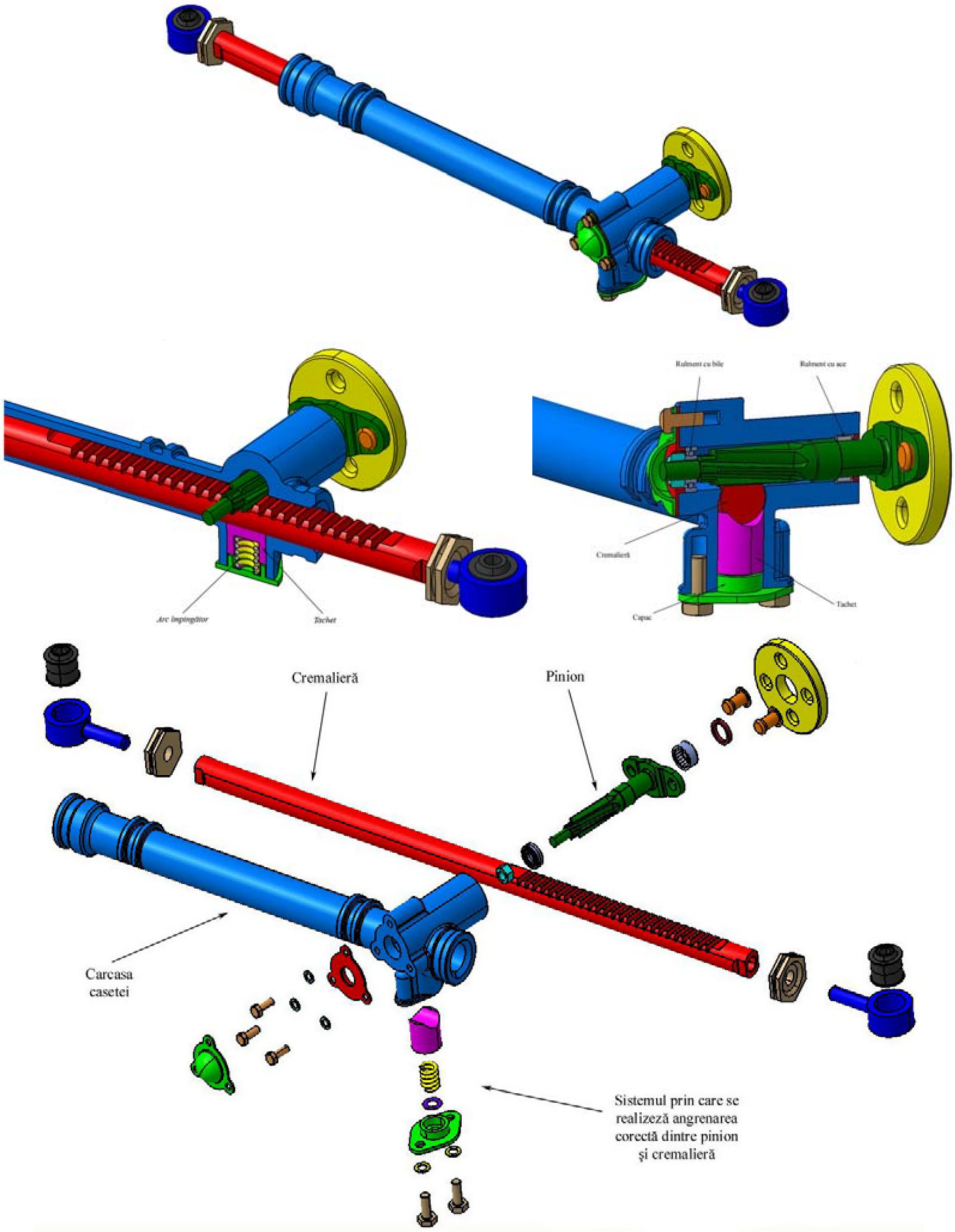

Fig. 9 – Subassembly CAD model
included into the designed mechanism – rack and pinion steering box

5. FINITE ELEMENT ANALYSIS

To verify if the parts were well designed and dimensioned, a study of the stresses and deformations was necessary. The finite element analysis (FEA) represent today the method used most often to verify if a part resists under the loads or if is stiff enough to fulfill its function. The steps involved by this method [3] and applied for the main parts are presented further for the most complex stressed part of the designed mechanism – the steering knuckle, figures 10-12.

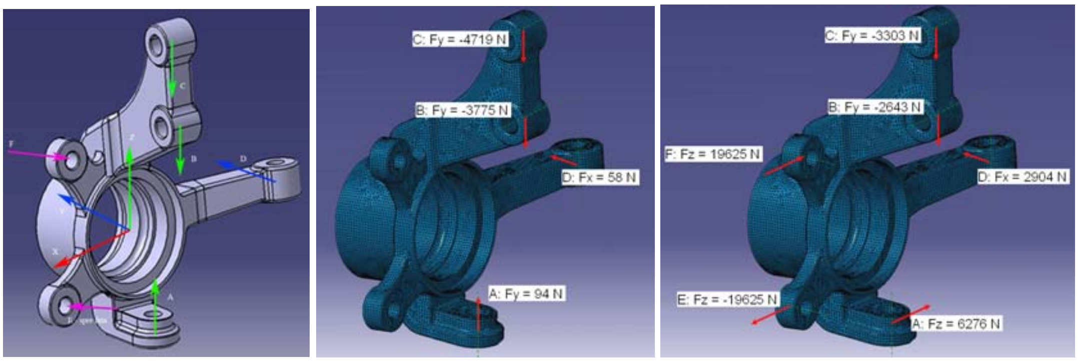

Fig. 10 – Knuckle's coordinate system and forces orientation

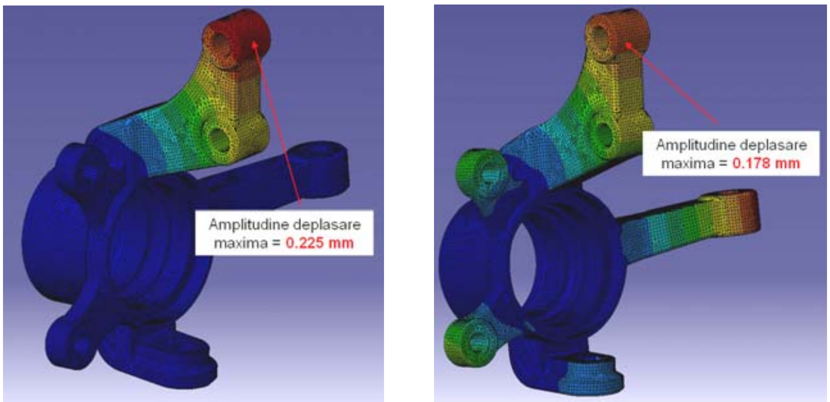

Fig. 11 – Knuckle deformation: left – bump passing, right - braking

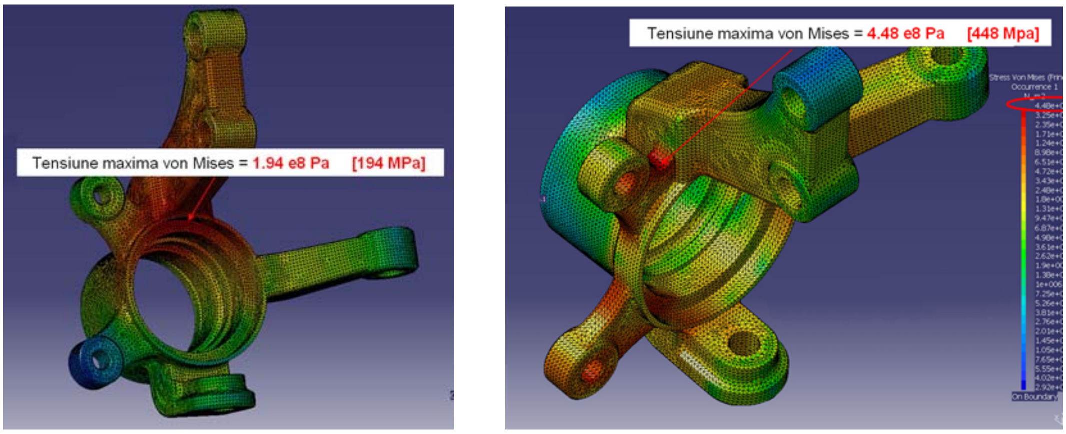

Fig. 12 – Knuckle stress: left – bump passing, right - braking

The material adopted for the knuckle was a forged allied steel. Firstly, a coordinate system was chosen, accordingly with the ISO standard. For the study, the knuckle was considered supported by the wheel bearings and the interactions with the other parts were replaced by forces, figure 10, left. Two very important load cases were analyzed [1], [5], [8]: passage over road unevenness (singular bump) and hard braking on a quality road. The orientations of the forces for these cases are presented in figure 10, middle and right, respectively. The magnitudes of the forces were obtained through the multibody simulation. Comparing with the bump passing, the braking case presents smaller vertical forces, but new big forces act due to the braking torque.

The CAD model of the knuckle was imported into LMS Virtual Lab software where it was divided in finite elements, figures 10-12. This mesh, containing approximately 314000 elements and 63000 de nodes, was refined so that it was obtained a good discretization quality of 98%.

The deformations and stresses resulted, after the FE model solving, are shown in figures 11 and 12 respectively. The figures' left-side corresponds to the bump passing and the right-side, to the braking.

The maximal deformations and stresses, also indicated in the figures, show high but normal levels, meaning that this complex part was well designed. The maximal von Mises stresses for the two case studies were 194 MPa (bump) at the damper-support end and 448 MPa (braking) at the caliper's upper fixing element. Comparing these stress values to 950 MPa, the tensile strength of the chosen material (41MoC11 alloy steel), result the safety coefficient values: 8 for bump passing and 2.5 for hard braking. Also, the places where these maximal efforts appear represent indication for the areas that may be subjected to a further shape optimization.

6. CONCLUSIONS

The paper showed a possible way to design, study and optimize a complex mechanism as a car suspension-steering linkage (MacPherson strut and rack and pinion). The authors used a top-down approach, starting from the requirements of the assembly and successively refining and adding components as the design solutions became clearer. The main mechanism dimensions were obtained using very simple 2D models, which permit to judge and interpret easier the mechanism behavior. After this geometrical and kinematic optimization, more other steps were done, as the multibody study of the mechanism, part and assembly CAD modeling aiming manufacturability, finite element analysis of some important parts and, finaly, corrections and tunings of the CAD models.

7. ACKNOLEDGEMTS

This work, made partly as a graduate projects, beneficiates of the LMS subsidiary in Brasov's support, consisting in study themes, knowledge, suggestions and hardware and software resources for computation. It was intended to be a new step in the co-operation between Transilvania University of Brasov and LMS Romania.

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