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TOPOLOGICAL OPTIMIZATION OF THE STRUCTURAL OF A STEERING KNUCKLE FOR THE PUSHROD AUTOMOTIVE SUSPENSION SYSTEM

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Abstract. The steering knuckle is a fundamental part of automobiles, being responsible for the direct integration of the scrolling, steering, braking and suspension systems into the vehicle's chassis. Pushrod type suspension incorporates a double arm system overlapped with rods to activate the damping system, favoring the approximation between the vehicle's center of gravity and the ground, improving the vehicle's performance at high speeds. The knuckle undergoes severe efforts under conditions of use, and catastrophic failures can occur in certain cases. This paper present a static structural analysis of the knuckle under impact loads, maneuver in short radius curves and maneuver in long curves followed by a topological optimization aiming to improve the stress distribution, minimizing the weight of the structural component for critical conditions of use. The topological optimization performed on the knuckle employed a generic CAD model, made of aluminum alloy Al7075-T6, which allowed the mass to be reduced by 35,439%, minimizing the stress gradient by 34,614%, the maximum stress of which was 186.65MPa

Keywords: Knuckle, Topology optimization, finite element analysis, weight optimization.

1 Introduction

The suspension system is a fundamental system for any motor vehicle, acting directly on control, stability and comfort when driving it. The architecture of the suspension system varies according to the need to transfer dynamic loads to the chassis and the desired behavior of the vehicle [1], [2].

Automotive suspension of the pushrod type, although not so common in passenger vehicles, is characterized as a suspension model aimed at sports or competition vehicles where the need for load transfer at high speeds is prioritized through the reduction of a not suspended mass, leading to reduction in the variation of the vehicle's center of gravity (CG) [3].

The knuckle, one of the main components of the pushrod suspension system, connected to the front wheel on the chassis, is designed to maximize vehicle dynamics by improving the transfer of dynamic forces to the damping system and converting the linear motion of the tie rod direction in angular movement of the tires [1], [4], [5]. The knuckle, as well as the other suspension components, is designed to withstand complex and varied mechanical loads arising from the vehicle's behavior on the track, and therefore are subject to fatigue failures [6].

Due to structural requirements, the steering knuckle can be made of cast iron, steel alloys or aluminum alloys [3]. The use of aluminum alloys in the manufacture of knuckle is mainly aimed at improving resistance to efforts and reducing component weight, making it necessary to optimize the geometry using numerical techniques, such as topology optimization [2], [7].

The use of topological tools allows to obtain the ideal geometry of a component, from an initial design, leading to its weight reduction without compromising its performance and the mechanical resistance restrictions [6], [8].

The objective of this article is to present an analysis of the stress distribution and the topological optimization of the knuckle for a suspension system of the pushrod type, subject to loadings of track conditions, seeking to determine an ideal component geometry from the consideration of different alloys aluminum and determining the fatigue life for the structural component.

2 Methodology

2.1 Geometry

The CAD model of the suspension geometry used in the study, Figure 1 (a), is a generic model that considers the connectivity of the steering knuckle with the use of overlapping double arms, activation of the shock absorber via suspended rod and angle of inclination of the wheels of 13.2°. The dimensions of the knuckle, Figure 1 (b), are (i) 574.2 mm, (iii) 154.5 mm, (iii) 285.0 mm and (iv) 75.0 mm having a mass of 9.639 kg. The dimensions were determined based on the analysis of the suspension characteristics and physical restrictions of the component.

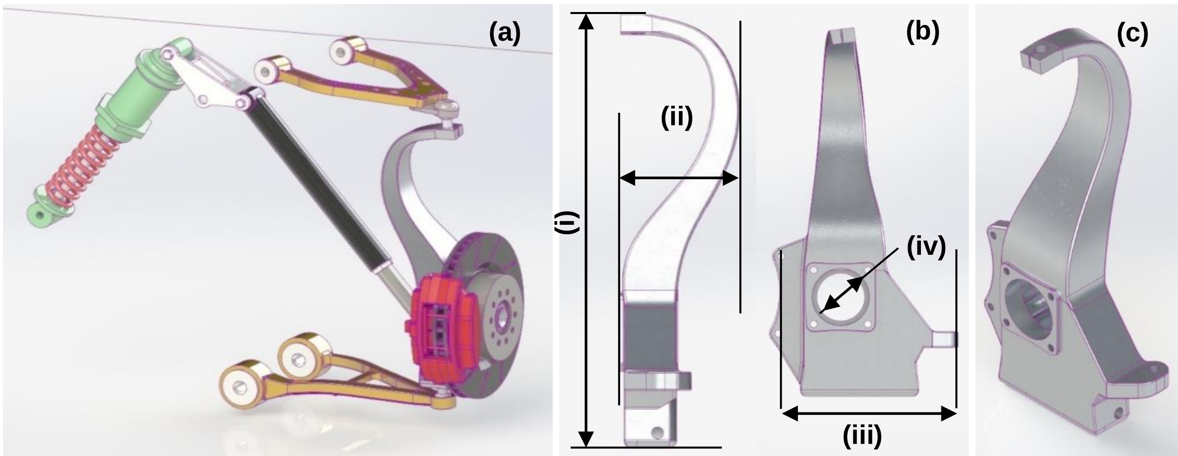

Figure 1: (a) Anterior view for the pushrod suspension system, (b) dimensional restrictions and (c) isometric view of the knuckle

2.2 Material

The selection of material for the analysis considered the need for low component density, high resistance, and cost. The materials used were aluminum alloys Al2011-T3 (I), Al7075-T6 (II) and Al6061-T6 (III). The main mechanical properties of the alloys to be used in the research are listed in Table 1.

Table 1 - Main mechanical properties of the materials used.

Properties	I	II	III
Density (Kg / m³)	2830	2810	2700
Young's modulus (GPa)	70.3	71,7	68,9
Poisson's ratio	0.33	0.33	0.33
Breaking stress (MPa)	373.0	572,0	310,0
Shear modulus (GPa)	26.0	26.9	26.0
Yield strength (MPa)	296.0	503.0	276.0
Buckling module (GPa)	69.6	70.3	67.5

2.3 Loads and Bindings

The loading conditions and linkages for steering knuckle were set based on the work of Ghungarde [7] Anderson [9] and Carello [10]. The load distribution, Table 2, was determined based on the conditions of:

- Total vehicle mass (M) of 1500 kg;
- Mass distribution between the front and rear axles in the 50:50 condition.
- Vehicle with constant speed of 30 ms^{-1};
- Adoption of gravity acceleration with a value of 9.81ms^{-2};
- Realization of curves always in the same direction;

• Load transfer between the steering knuckle and overlapping arms (DW) is null;

Table 2: Uploads and links.

Load application conditions	Force (N)		
	Axis X	Axis Y	Axis Z
Braking (F1)	5518.12	-5518.12	0.00
Wheel hub strength (F2)	3678.75	11036.25	11036.25
Vehicle weight (F3)	0.00	-5518.12	0.00
Damper reaction (F4)	-3678.75	-5518.13	-11036.25
Impact force (F5)	6989.65	19375.00	0.00
Steering link force (F6)	55.18	0.00	55.18
Long radius curve (R = 44 m) (F7)	1559.66	0.00	1559.66
Short radius curve (R = 15 m) (F8)	2287.5	0.00	2287.5

The points of application of the load, Figure 2, they tend to act only at the interface between the steering knuckle, being less the damping characteristics generated by fasteners and bushings.

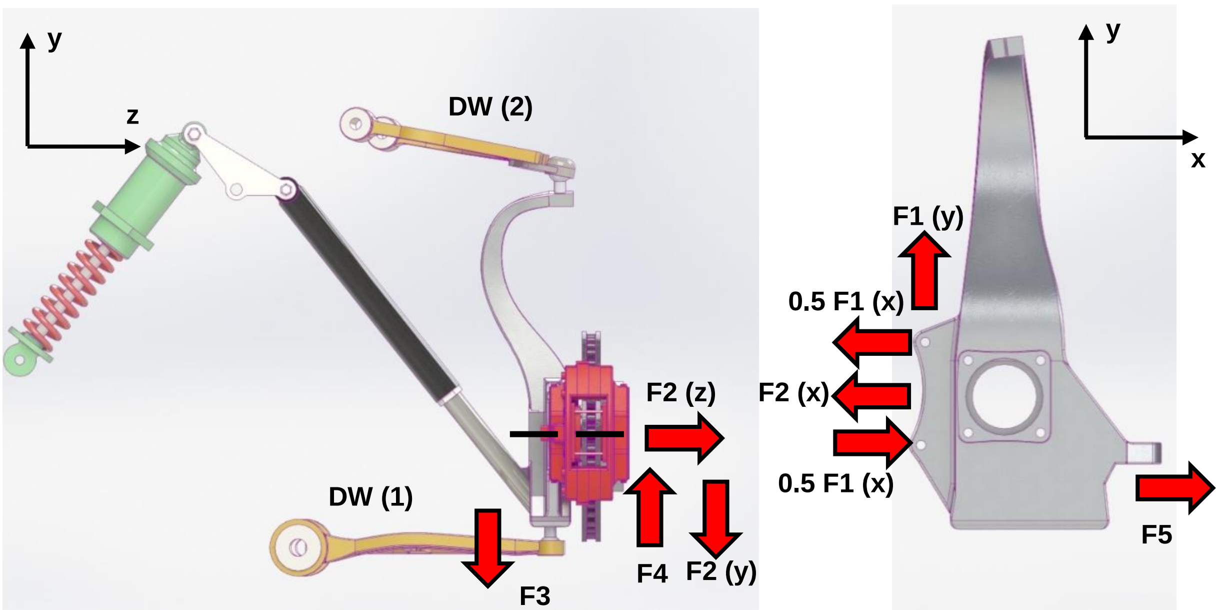

Figure 2. Application of loads and links in the knuckle steering.

Considerations on the behavior of the steering knuckle are based on usage, and listed below:

Condition I - Impact: Assessing the behavior of the steering knuckle when subjected to impact conditions involves determining the maximum displacement of the knuckle and the behavior of the stress field in the related components of the suspension system [4]. The loading for the impact situation corresponds to the application of loads F2, F3, F4 and F5, where it was admitted that the driver did not apply the brakes and consequently reduced the vehicle's speed during the moment of impact.

Condition II - Impact with brake: Corresponds major mechanical stress to the steering knuckle. [11] In this condition it was admitted that the driver applied the brakes during the moment of impact is considered a load with loads F1, F2, F3, F4 and F5.

Condition III - Short radius curve: The maneuver in a short radius curve involves the application of loads F2, F3, F4, F6 and F8. The load transfer in the curves, Equation (1), depends on the vehicle mass (m), gravity acceleration (g), distance between the CG and the front wheel (a2), distance between the CG and the ground (hcg) , vehicle acceleration (a) and the radius of the curve (R). For the determination of efforts, a vehicle was considered at a constant speed of 30 ms^{-1}, making curves with a radius of 15 m [12].

Condition IV - Long radius curve: Maneuver in long curves involves loads F2, F3, F4, F6 and F7. For load transfer in the curves, Equation (1), a vehicle was considered at a constant speed of 30 ms^{-1}, making curves with a radius of 44 m [12].

$$F2 = \frac{M(g.a_2 + h_{cg}.a)}{4.R} \quad (1)$$

2.4 Topological analysis

Topology optimization is a numerical method for generating, from an initial geometry, modifications that aim to maximize structural stiffness by minimizing the volume of the structure under conditions of restrictions and loads [13], [14]. The structural topology optimization process is based on the Solid Isotropic Material (SIMP) method to assign to each mesh element a stiffness matrix that, for each interaction (i) performed, analyzes the pseudo-density value (ηi) under conditions of the volume restrictions established in order to determine the minimum energy of statistical deformation at a given load [14].

The maximization of structural stiffness is based on minimizing the conformity of the structure (uc) from the consideration of multi-loads and links in the structure. The reduction in compliance (uc) allows the structure to do less work for the applied mechanical stresses [15]. The variables compliance (uc), pseudo-density (ηi), proportion of the preserved volume (α) and the Pre-optimization volume (V2) can be correlated according to Equation (2). The pseudo-density values (ηi) alternate between 0, material removal, and 1, material addition, so that these values influence the elastic tensor behavior of each mesh element so that i values close to 0 represent material to be removed [15].

$$\begin{cases} u_{c=\min F(\eta_i)} \\ s.t. \int \eta_i d\theta < \alpha V_2 \end{cases} \quad (2)$$

3 Results and Discussions

3.1 Structural analysis

Based on the geometry adopted for the steering knuckle and the boundary conditions and material emphasized in the methodology, there were static structural analysis in computer application ANSYS through the Static Structural module. The structural behavior of the steering knuckle on the displacement (mm) Maximum stress (MPa) are shown in Tables 3 and 4, respectively

Table 3: Maximum displacement (mm) of the steering knuckle for boundary conditions and connections.

Material	Maximum displacement (mm)			
	Condition I	Condition II	Condition III	Condition IV
Al2011-T3	0.460	0.738	0.299	0.286
Al7075-T6	0.450	0.724	0.293	0.280
Al6061-T6	0.469	0.753	0.306	0.292

Table 4: Maximum stress (MPa) of the steering knuckle for boundary conditions and bonding.

Material	Maximum stress (MPa)			
	Condition I	Condition II	Condition III	Condition IV
Al2011-T3	122.21	285.38	128.84	127.61
Al7075-T6	122.21	285.46	128.84	127.61
Al6061-T6	122.21	285.46	128.84	127.61

The computational results presented in Tables 3 and 4 were used to determine the critical behavior of stress and strain to which the steering knuckle. Condition II, which involves impact with braking, promotes a maximum deformation of 0.738mm with a maximum stress of 285.46 MPa. The zones of occurrences of the maximum (σ_{max}) and minimum (σ_{max}) and maximum (δ_{max}) and minimum (δ_{min}) displacement for the geometry are shown in Figure 3.

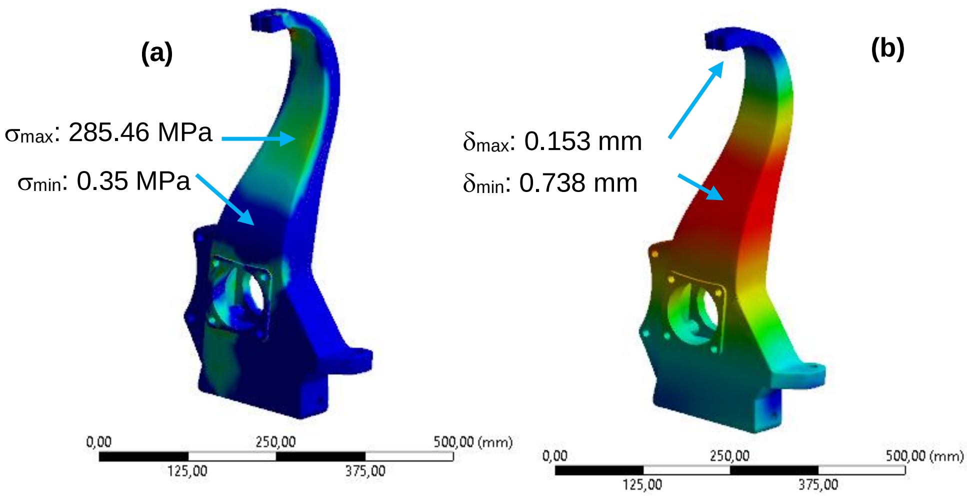

Figure 3. Areas of maximum stress (MPa) (a) and maximum displacement (mm) (b) to the geometry of the steering knuckle.

The formation of gradients of high stress gradients along the structure ends up inducing fatigue states, leading the part to premature failure. The fatigue analysis was generated using the Goodman relation, to quantify the interaction of medium and alternating stresses and to determine the fatigue life of the mechanical component. Table 5 shows the fatigue life in loading and unloading cycles for loading and bonding conditions.

Table 4: Fatigue life in cycles for the steering knuckle for boundary and boundary conditions.

Material	Condition I	Condition II	Condition III	Condition IV
Al2011-T3	6.77e+4	3.4e+4	1e+8	1e+8
Al7075-T6	7.77e+4	3.6e+4	1e+8	1e+8
Al6061-T6	6.01e+4	2.9e+4	1e+8	1e+8

The computational results present in Table 5 show that condition II has the lowest values for life in fatigue in the structure. The Al6061-T6 alloy has the lowest fatigue life of the entire structure, while the Al7075-T6 alloy has, for this condition, the longest fatigue life in cycles for the structure. The condition used for topological optimization of the component will be condition II, considered a critical case, due to its maximum displacement (mm) and maximum stress (MPa) and minimum fatigue life values.

3.2 Topological optimization

The objective of the optimization process of the topology of the steering knuckle is to know the ideal geometry with minimum mass distribution that allows to reduce the variation between the maximum and minimum stresses along the geometry. It is expected that, by reducing the variation in stresses, the displacement and mass of the component will be reduced to values lower than the originals and that the fatigue life of the component will increase. The evolution of the ideal geometry layout for load conditions, restrictions and boundary conditions is illustrated in Figure 4.

The behavior of the maximum stress (MPa), maximum displacement (mm) and fatigue life for the topologically optimized geometry is shown in Table 5. Figure 5 illustrates the maximum (σ_{max}) and minimum (σ_{max}) stress zones. and the maximum (δ_{max}) and minimum (δ_{min}) displacement for the optimized steering knuckle geometry.

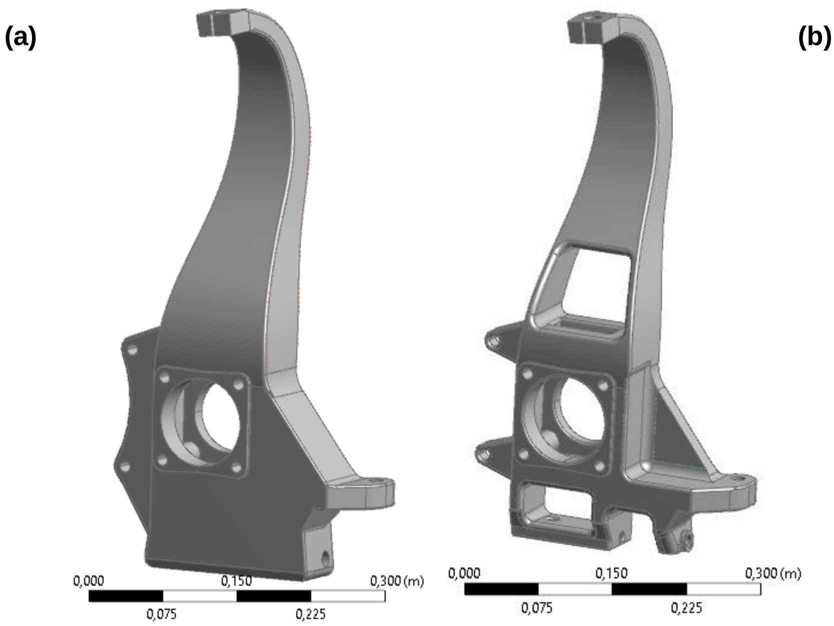

Figure 4. Original geometry (a) and optimized geometry (b) of the steering knuckle.

Table 5: Maximum values for stress (MPa), displacement (mm) and fatigue life for topologically optimized geometry of the steering knuckle

Material	Maximum displacement (mm)	Maximum stress (MPa)	Fatigue life
Al2011-T3	0.548	183.65	4.94e+4
Al7075-T6	0.511	183.65	5.16e+4
Al6061-T6	0.573	183.66	4.22e+4

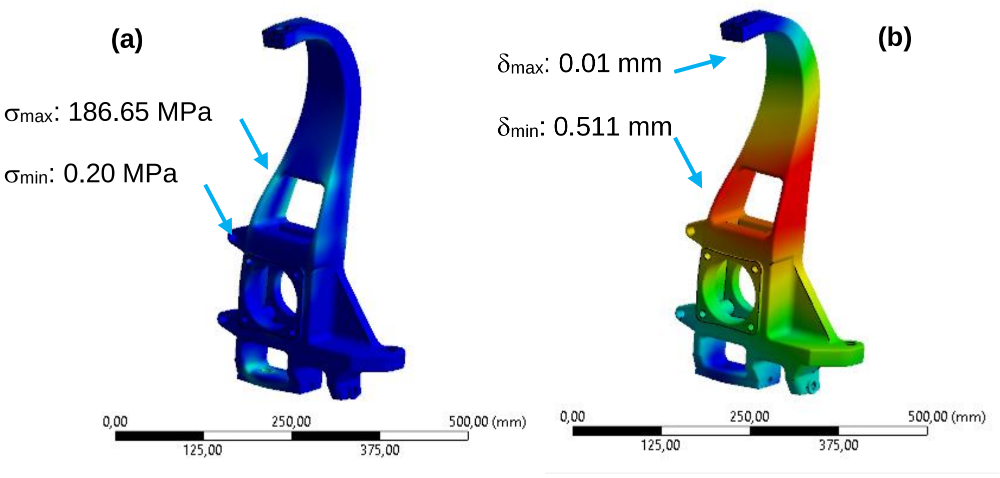

Figure 5. Areas of maximum stress (MPa) (a) and maximum displacement (mm) (b) for the optimized geometry of the steering knuckle

The material selection for the optimized steering knuckle was based on the behaviors of maximum stress (MPa), maximum displacement (mm), fatigue life and the mass (kg) of the optimized geometry. The material selected was the aluminum alloy Al7075-T6 due to having, among the aluminum alloys used, the lowest deflection values and the longest fatigue life for use.

The simulation results indicate a reduction of 29,420% in the maximum displacement in the optimized knuckle, producing maximum deformations of 0.511 mm for a maximum stress of 186.65 MPa developed on the side wall of the knuckle, close to the brake assembly, as shown in Figure 5. Although the maximum stress on the optimized joint is concentrated in a specific area, no excessive variations were observed in the local safety factor, suffering a variation from 2.65 to 2.54, which is 4.15% of the original steering knuckle. The mass of the optimized knuckle is 3.4 kg less, implying a reduction of 35.439%. The comparison of the results is shown in Table 6.

Table 6: Comparison of results

Properties	Knuckle		Δ (%)
	Initial	Optimized	
Mass (kg)	9.639	6.223	-35.439
Maximum displacement (mm)	0.724	0.511	-29.420
Maximum stress (MPa)	285.46	186.65	-34.614
Fatigue life (cicles)	3.6e+4	5.16e+4	+43.33

4 Conclusions

The steering Knuckle as part of the suspension mechanism is generally oversized due to the efforts to which it is subjected. In this study, the articulation joint is designed to reduce its mass (kg) while maintaining its operating conditions and capacity to withstand the critical efforts present in its use. The optimization of the structure aims to improve the efficiency of the joint, making it light without compromising its life of use and its initial performance. The use of topological optimization as a design tool allowed for substantial mass reductions, reaching a reduction of 35,439%.

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