Optimization of steering knuckle geometry of an off-road vehicle applying DoE

Felipe Rauber, Sérgio Junichi Idehara

Abstract— Improvement in the product quality along with an increase in performance and reduction of design costs are constant themes in the automotive industry. In this context, several techniques have been developed for analysis and optimization in the design of mechanical components. One of the most commonly applied techniques is the finite elements method (FEM). The aim of this study was to use the FEM to analyze the steering knuckle of an SAE Baja off-road vehicle and then apply an optimization tool in order to provide a lighter component. The stress level and safety factors associated with the fatigue life of the component were analyzed and a parametric optimization was employed. The variables used for the optimization were chosen by the design of experiments (DoE) method. Based on the results, a new geometry was generated, where it was possible to obtain a reduction of 12.7% in the mass while retaining a safety factor of 3.5 for the infinite fatigue life.

Index Terms— fatigue, finite element method, optimization, steering knuckle.

1 Introduction

THE continuous growth of the automotive market and the need to further reduce the design time and increase the quality of the products offered means that numerical optimization during vehicle design is becoming increasingly important ([1], [2], [3]). The optimization results contribute to improving attributes such as the reliability, performance, safety and cost ([4]). These variables in the design of a vehicle are essential for obtaining a product which is competitive in the market. For example, [5] applied an optimization process in vehicle cabin design to improve the sound pressure level as a criterion of passenger comfort. In their study, the DoE tool was used to identify the components that have the greatest influence on the acoustic pressure. The optimization resulted in a panel thickness that reduced the noise inside the cabin.

As commonly occurs in the automotive industry, vehicle components have a complex geometry, which complicates the calculations in an analytical analysis. However, the use of computational tools can help to solve this problem, for instance, the finite element method (FEM) can be used to analyze complex structures, as discussed by [6]. The authors simplified the model of a vehicle in the body-in-white (BIW) stage and compared the results with experimental data. The authors then carried out a validation of a simplified geometry based on the torsional and flexural rigidity of the structure.

An automotive application of a relatively novel technique is presented by [7]. The authors describe a methodology based on topological optimization to achieve a reduction in the chassis weight. In this case, the authors exemplify the process obtaining the ideal layout of the trusses and the thickness distribution of the structure. In another study, a multi-objective optimization process was applied by [8] to improve the shock absorption capacity in vehicular structures. The authors used a genetic algorithm and the neural network optimization method. The design defines the optimal values for the aluminum tube dimensions based on a finite elements model. Design optimization has thus become an indispensable tool for engineers to deal with complex systems.

In general, the optimization of a mechanical component is based on the analysis of the parameters that have the greatest influence on the structural and functional behavior, in order to change fewer design parameters to maximize the improvement of the final product. In this regard, [9] proposed optimization applying a genetic algorithm method to a bus structure, subject to a torsional excitation and material weight load. The optimal solution provides the beam thickness that minimizes the objective function. In the paper, the authors performed sensitivity analysis to determine the main parameters to be optimized. It is essential to identify these main parameters in order to better understand the analysis and the results obtained in the process of numerical model optimization. Thus, in engineering tasks different techniques of sensitivity analysis are employed during optimization processes. For instance, [10] discusses the relationships and simplifications with regard to the sensitivity matrix for the topological optimization of flexible multibody structures. The authors noted the great importance of parameter sensitivity in optimization.

The steering knuckle of a vehicle is a structural component that generally has a complex geometry and connects several vehicle systems. Therefore, it is an important component in terms of motor vehicle safety. It can be developed using optimization tools, as demonstrated in [11], where the authors used the topological optimization approach in the design of the axle knuckle of a Formula SAE vehicle. [12] applied a multi-objective optimization approach to the manufacturing process of a steering knuckle, based on a finite elements model, to reduce defects resulting from the fabrication process.

In this study, the steering knuckle of an SAE Baja off-road vehicle (Fig. 1) is analyzed. The numerical model for the component must represent the interaction of the three main systems responsible for controlling the movements of the vehicle: suspension, steering and brake systems. In addition, the steering knuckle is connected to the wheel hub through a bearing. This is responsible for supporting the axle, which enables the rotation of the wheels. Therefore, because this is a critical component in terms of structural strength, safety and vehicle maneuverability ([13]), the verification and

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optimization of its design during the vehicle development is of great importance.

Fig. 1. SAE Baja off-road vehicle.

The main vehicle subsystems connected to the steering knuckle are:

- the suspension system, which is responsible for filtering the excitation of the wheel in contact with the ground. In general, suspension systems are divided into two types: dependent and independent ([14]). In this study, the vehicle was designed with independent suspension of the double-A type (Fig. 2). According to [15], the advantages of an independent suspension system are compact dimensions, easily handling, lower weight and no interference with the movement between the wheels. Also, kinematic adjustment of the convergence angle can be performed while the vehicle moves forward. The suspension is coupled to the steering knuckle through ball joints. Thus, the loads of the suspended mass are transmitted to the steering knuckle through these mounting points;
- the steering system, the function of which is to act on the front wheels in response to the driver's commands, in order to provide directional control of the vehicle ([14]). The most common type of construction is rack and pinion, since this system is simpler and cheaper. This type of construction is applied to the vehicle considered in this study, and it is connected to the steering knuckle by the steering arm;
- the brake system, which is a mechanism for modulating or stopping the movement of rotating parts or the vehicle itself. Basically, the brake system is a combination of components that should slow down the vehicle speed progressively, make it stop, and if it is stopped, keep it stationary. The most common types of brakes used in the automotive industry for passenger cars are drum brakes and disc brakes. In the case of disc brakes, which are used in this design, the brake calipers are mounted on the steering knuckle. When the driver presses the brake pedal, the calipers press the brake disc, which is fixed to the wheel, and the frictional force generated by the brake reduces the speed of the vehicle. The brake force is supported by the steering knuckle through a set of screws.

For the design of this component the finite element software used was Ansys Workbench.

Fig. 2. Double-A suspension of SAE Baja vehicle.

2 Methodology

Firstly, a parametric geometry was developed in computer-aided design (CAD) software. To solve the optimization problem, the design of experiments (DoE) method was used ([16]) to determine the main parameters. The constrained minimization function was written as

$$\mathrm{f}(\mathrm{p}_1, \mathrm{p}_2, \mathrm{p}_3, \mathrm{p}_4, \mathrm{p}_5) = \min\left\{\int_{\mathrm{R}} \mathrm{g}(\mathrm{x}, \mathrm{y}, \mathrm{z}).\,\mathrm{dm}\,, \mathrm{p}_\mathrm{i}\right\}, \quad \text{for } \sigma_{\mathrm{vm}}(\mathrm{x}, \mathrm{y}, \mathrm{z}) \leq \sigma_{\max} \tag{1}$$

where $\int_R g(x,y,z).dm$ is the mass with dimensional variables x, y and z and constrained by maximum von Mises stress (σ_{max}). The mass and volume data are provided by the CAD-FEM software and are used in the optimization algorithm. The original design is updated in the CAD model and subsequently a new FEM simulation is carried out. This process is repeated until an optimal solution is reached or the maximum number of interactions ends.

2.1 Design of experiments (DoE)

Design of experiments (DoE) is the name given to the techniques used to guide the choice of experiments so that they are performed efficiently ([17]). As noted by [18], there is a wide range of algorithms or DoE methods available for engineering. They all have one feature in common: to find, from random input parameters, the optimal values to minimize (or maximize) the objective function in the most efficient way, that is, to obtain the data required with the least number of interactions, reducing the time and cost of the tests.

According to [19], in a planned experiment, the engineer or laboratory technician makes changes in the input parameters and verifies the variation in the output parameters. The input parameters may have different levels of influence on the outputs, showing weak to strong relationships. The author also points out some advantages of applying DoE in an industrial context, such as reductions in the design time, manufacturing costs, defect rates, and the need for reworking and the repetition of tests.

[20] carried out a study on a bucket truck using DoE followed by topological optimization. The authors obtained a

reduction of 2.2% in the mass and a 33% increase in torsional stiffness. In addition, [21] investigated the effects of cutting conditions and surface roughness using a response surface method. The DoE was used with a genetic algorithm to obtain the optimal conditions, where the roughness of the machined surfaces would be minimal. In the tests, they obtained a 10% decrease in the roughness of the cavity of a mold. The technique used in this study was central composite design, which consists of constructing a second-order (quadratic) model to describe the surface of the system response.

2.2 Mathematical model

The loads acting on the steering knuckle can be modeled as a static or dynamic problem. An example of a static model is given by [22]; however, failure may originate mostly from dynamic conditions. [23] presents the case of the fatigue failure of an axle knuckle that caused an automobile accident. The authors identified the fatigue failure characteristics of the component. According to [24], any time-varying load can cause failure due to fatigue. In general, the loading shape over time has no significant effect on fatigue failure in the absence of corrosion. Thus, the significant factors are the mean amplitude and value of the stress within a cycle, as well as the number of load cycles applied to the component. The three types of loads most commonly used as an excitation model for the alternating stress are: (a) mean stress (σ_m) equal to zero, (b) minimum stress (σ_{min}) equal to zero and varying up to a maximum value (σ_{max}), and (c) pulsating stress where the values of the mean, minimum and maximum stress are not zero.

In real cases, the mathematical function for loading over time is not easily defined and it can be random. To obtain loading data in real applications, the vehicle is instrumented and measurements are taken under road conditions. [24] notes that in the automotive industry vehicles are instrumented, for instance, with accelerometers and force transducers, and they are subjected to several tests on different surfaces. In these tests, loading data is obtained as a function of time. In this study, the waveform will be considered harmonic with constant frequency. The amplitude values varied from 0.8 to 1.2 of the stress value determined with the static model.

In order to determine the fatigue strength limit of a material, tests are performed on specimens, which are submitted to increasing bending load until failure. The results for the number of cycles (N) and the stresses (S) applied before the material fails are displayed in an S-N diagram ([25]). Figure 3 shows a diagram based on experimental data obtained for SAE 4140 steel.

3 Vehicle Loads

Table 1 shows the parameters of the SAE Baja vehicle modeled.

For the calculation of the maximum stresses on the steering knuckle, it was considered that in an extreme situation like a small jump, the vehicle weight would be supported only by the two front wheels. The acceleration (a) of the impact was adopted considering 2.5g. Equation 2 was then used to calculate the force acting on the wheels and consequently the load acting on the bearing supports (F_z), where m is the mass of the vehicle.

$$F_z = m.a \tag{2}$$

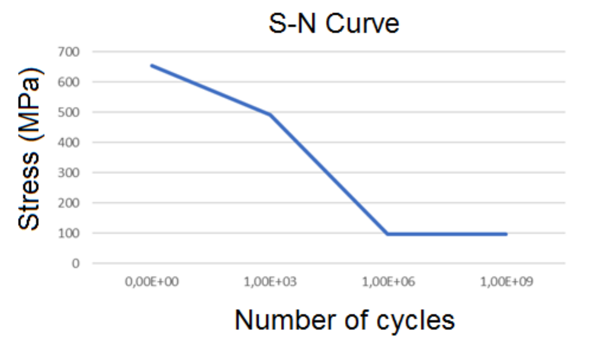

Fig. 3. Steel stress-life curve for SAE 4140.

TABLE 1
VALUES FOR THE DESIGN VARIABLES

Variable	Value
Track [mm]	1320.0
Center of gravity height [mm]	460.0
Vehicle mass [kg]	300.0
Portion of weight on the rear axle	0.5
Wheelbase [mm]	1400.0

A value of 3678.75 N was obtained for F_z. This force is applied vertically in the position of the wheel bearing. The values obtained were close to those reported by [26], who instrumented a Baja vehicle with extensometers. The authors obtained values between 3200 N and 3300 N acting on the front shock absorber of the vehicle when it jumped over a ramp with a height of 75 cm.

The maximum lateral force acting on the steering knuckle, for a curve of radius ρ and at speed v, is given by the centripetal force (F_c) on the vehicle. The force originates in the tires due to the friction with the ground. Thus,

$$a_c = v^2/\rho \tag{3}$$

$$F_c = m(1-x)a_c \tag{4}$$

where,
a_c is the centripetal acceleration; and
x is the portion of the weight on the rear axle.

Therefore, the road reaction in the right front tire (R_{Ib}) and the friction force (F_{IIb}) are given by:

$$R_{Ib} = \frac{m.g(1-x)\left(\frac{t}{2}+\frac{v^2}{\rho}h\right)}{t} \tag{5}$$

$$F_{IIb} = R_{Ib}.\mu \tag{6}$$

where,
μ is the coefficient of friction between the tire and the ground;
t is the track of the vehicle;

h is the height of the center of gravity of the vehicle.

Considering that the vehicle makes a curve with a radius of 6 m at 25 km/h and that the friction coefficient between the tire and the ground, for a dry road, is 0.7, the value for F_{IIb} will be 933.81 N. This force is applied transversely in the position of the wheel bearing.

With regard to the steering arm, the force of the driver on the steering wheel was considered to be 20 kgf. This force is transmitted through the steering column, rack and pinion, and tie rod, which acts on the steering knuckle. The torque (M) generated by the driver is:

$$\mathrm{M} = \mathrm{F}.\mathrm{d} \quad (7)$$

where,

F is the driver's force; and

d is the distance from the application of the force to the axis of rotation.

This torque acts on the steering column, which is connected to the pinion. Thus, based on the radius of the pinion, the force on the tie rod is 455.56 N. This force acts on the steering knuckle.

4 Numerical Modelling

The initial geometry was modeled in SolidWorks and the design is illustrated in Fig. 4.

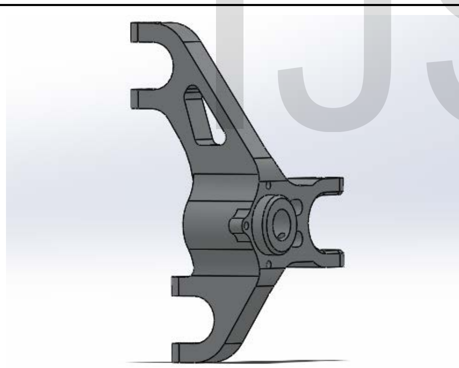

Fig. 4. Initial steering knuckle geometry.

The parameters selected for the optimization and the ranges of their values are shown in Table 2 while the positions associated with these parameters can be seen in Fig. 5.

In this study, the screening method was applied from the DoE, which consists of varying the parameter (factor) values between the lower and upper limits and verifying the effect on the output variables. The objective function of the optimization is the steering knuckle volume to be minimized. This function has a constraint for the maximum von Misses stress at 250 MPa. In total, 1000 samples were generated and the best result, i.e., that closest to the optimization objective, was selected for the FEM analysis.

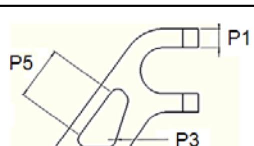

TABLE 2
Values for Steering Knuckle Parameters and Dimensions

Parameter	Initial dimension (mm)	Lower boundary [mm]	Upper boundary [mm]
DS_OSC@Sketch2 (P1)	12,000	6,000	12,000
DS_OIB@Sketch3 (P2)	12,000	6,000	12,000
DS_RAF@Sketch16 (P3)	5,000	5,000	15,000
DS_RFP@Sketch16 (P4)	14,000	14,000	24,000
DS_RFG@Sketch16 (P5)	34,513	34,513	55,000

5 Results

The finite element model applied in the optimization had a mesh of 340,834 elements and the boundary conditions were applied according to Fig. 6.

The von Mises stress obtained for the steering knuckle, under the static condition using the forces calculated in Section 3 - Vehicle Loads, is shown in Fig. 7 for the initial geometry (before optimization). It can be observed that the maximum stress obtained was 70.5 MPa where the geometry has a hole for the suspension arms. The safety factor under this condition is 5.9, demonstrating that the component is oversized.

Figure 8 shows the response of the sensitivity analysis, indicating which input variable has the greatest influence on the output. The values can vary from -1 to 1 and a higher value indicates a stronger positive relationship between the input parameter and the output variable. The negative values indicate an inverse (negative) relation between the input and output data. From the graph, it can be observed that the parameters with the greatest influence on the stress are P1 and P2. These parameters are inversely proportional to the stress, that is, an increase in their values leads to a decrease in the stress value. In relation to the volume of the steering knuckle, the most important parameters were P1 and P5. An increase in the dimension of P1 caused a volume increase in the component, and an increase in the P5 dimension caused a reduction in the volume, since P5 removes material from the structure.

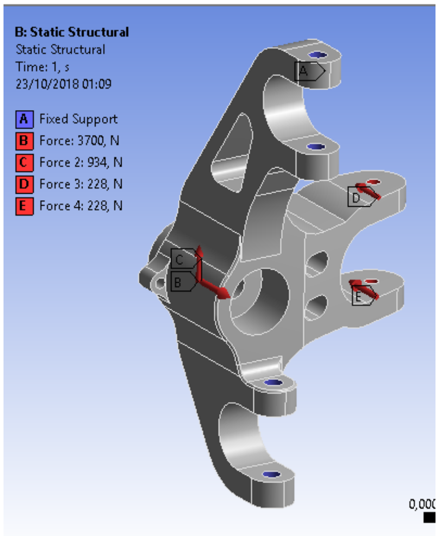

Fig. 6. Boundary conditions of the steering knuckle.

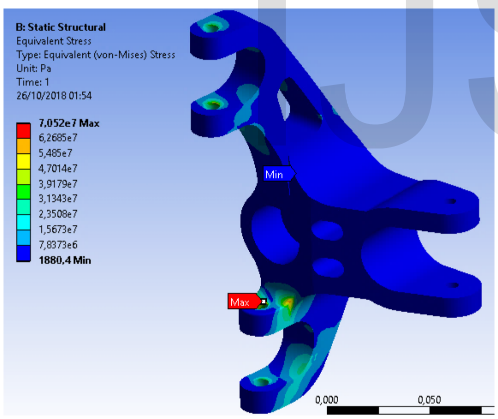

Fig. 7. Von Mises stress under static condition.

The volume and mass of the initial geometry were 28.3x10^{-4} m^{3} and 2.2 kg, respectively, and after applying the optimization tool the corresponding values were 25.1x10^{-4} m^{3} and 2.0 kg. Thus, the optimization process achieved a decrease of 11.3% in relation to the initial values (Fig. 9). The maximum von Misses stress for the optimized geometry was 97.59 MPa (38.4% higher than that of the original geometry). Figure 10 shows the static stress distribution of the von Mises stress for the new geometry.

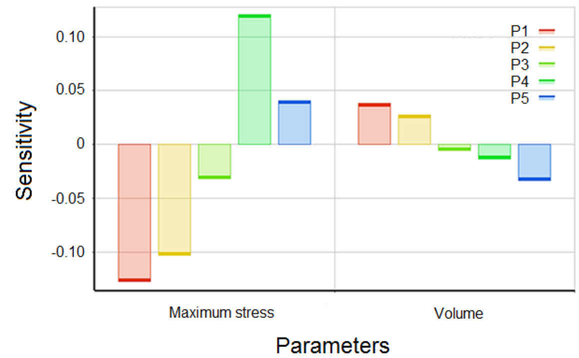

Fig. 8. Graph showing the sensitivity of the parameters of the steering knuckle.

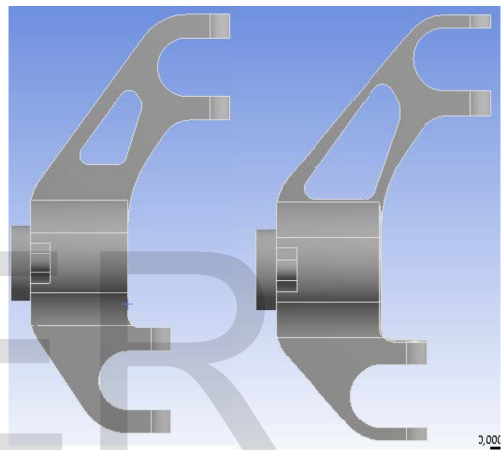

Fig. 9. Comparison between original geometry (left) and optimized geometry (right).

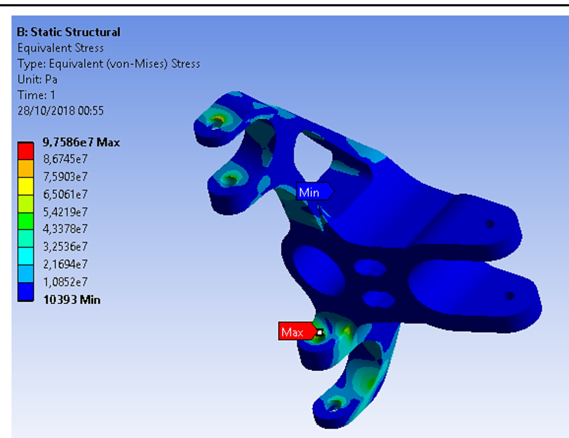

Fig. 10. The von Mises stress under static conditions for the optimized geometry.

In the fatigue analysis, the new geometry provided the results for the safety factor for infinite life presented in Fig. 11. The lowest value for this factor was 3.5 for the loading conditions described in Section 3 - Vehicle Loads. This lowest value was found at the points of greater stress concentration in the structure, over the suspension arm hole. At the other points of the structure, the values for the safety factor were considerably higher (Fig. 11).

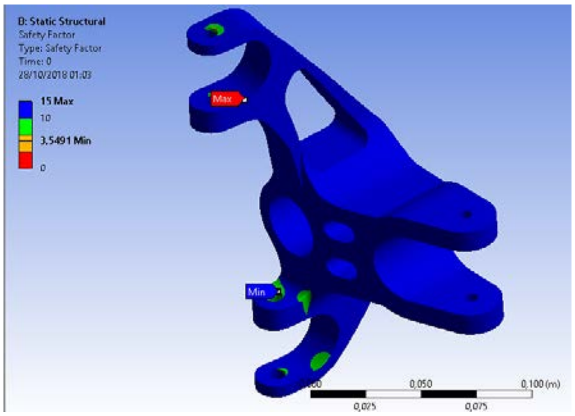

Fig. 11. Fatigue safety factor obtained for optimized geometry.

6 Conclusions

A practical methodology for the design of a complex geometry is presented herein. Based on a numerical model obtained applying the finite elements method, it is comprised of parameter adjustment by DoE and an optimization process. The results obtained indicate that the optimized geometry of the steering knuckle resists the applied forces and, therefore, it is viable for manufacture and use. As regards the magnitude of the applied forces and the safety factors obtained, it can be concluded that the applied approach was conservative. Even so, the safety factors were relatively high, which is of interest since the vehicle will be used on irregular surfaces with random loads and impacts, that is, conditions that could subject the steering knuckle to different types of dynamic forces.

Finally, it was verified that the finite element method in conjunction with numerical optimization can play an important role in the design of a mechanical component. This approach facilitates the engineering analysis and can decrease the number of prototypes required in the validation phase. Therefore, it considerably reduces the design time and costs. However, it was observed that despite the reasonable facility associated with producing results using the numerical method, it is imperative that the user has application experience and a good knowledge of this type of methodology to produce a reliable product design.

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