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SUSPENSION PROJECT OF A FORMULA SAE: FROM CONCEPT TO FATIGUE CALCULATION

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Abstract. *This work aims the design and analysis of a suspension for the Formula SAE vehicle model. The Formula SAE is a student competition that mobilizes engineering students from Brazil and around the world. To that end, the geometry of the components that compose the suspension were designed with the help of the SolidWorks CAD software, then the main parameters of the front and rear suspensions were defined and simulated in the multi-body Adams/Car software. In order to verify the structural behavior of these components, static simulations were performed using the Finite Element Method (FEM) into the Ansys Mechanical software, as well as vehicle dynamics simulations with the CarSim software to collect the forces in which the suspension is exposed. Finally, the same components were tested for durability using the Ncode Design Life software. The results obtained in the kinematic simulations showed that the parameters varied adequately. From the vehicle dynamics simulation, the history of forces present in the tire/ground contact was extracted, which served as input data in the structural simulations. Regarding the structural behavior of the components, the static simulations by FEM indicated that the Von Mises stresses were lower than the yield strenght. The fatigue calculation was based on the calculated stress in the FEM analysis and the history forces from the vehicle dynamics simulation, resulting in a life upper the defined failure criterion.*

Keywords: *Suspension, Formula SAE, Multi-body, Finite Element Method, Fatigue.*

1. INTRODUCTION

The objective of this work is to design and analyze a suspension system for a vehicle of the Formula SAE student competition. The development of the work begins with the concept, followed by kinematic evaluations and structural analyzes. For this purpose, the following methodology was implemented: define types of suspension systems and required parameters; design the main components of the suspension system using a CAD software; assembly and simulate the suspension geometry into a multibody environment to verify its kinematic behavior; model and simulate the suspension system using the finite element method; and, calculate the fatigue life of the main components of the suspension.

2. METHODOLOGY

The main conceptual aspects of a suspension system and the types of existing geometries were evaluated, as well as the components that compose the system. This research was useful to generate a theoretical basis for the definition of parameters and their respective hardpoints that compose the kinematic model, besides helping in the structural and dynamic simulations of the suspension system.

After defining the hardpoints in the SolidWorks software, a kinematic model was created in the multibody Adams/Car software for simulations of movement and parameter variations of the suspension. Thereafter, the suspension parameters were evaluated relative to displacement and steer of wheels. When the parameters are found unsuitable, the hardpoints are redefined. Otherwise, the next stages of development, which are organized in the flowchart of Fig. 1, in blue color, are followed.

Following the definition of the kinematics, the main constituent components of the system are designed in the SolidWorks program. Once the first CAD modeling is completed, the suspension assembly is exported to the Ansys Mechanical FEM program. Thereafter, the static structural simulations of the components were performed, under calculated loading conditions, in the CarSim vehicle dynamics program. After completing the static analyses, fatigue calculation was performed using the Ncode Design Life durability software. Likewise, these steps are broken down in the flowchart of Fig. 1, in red.

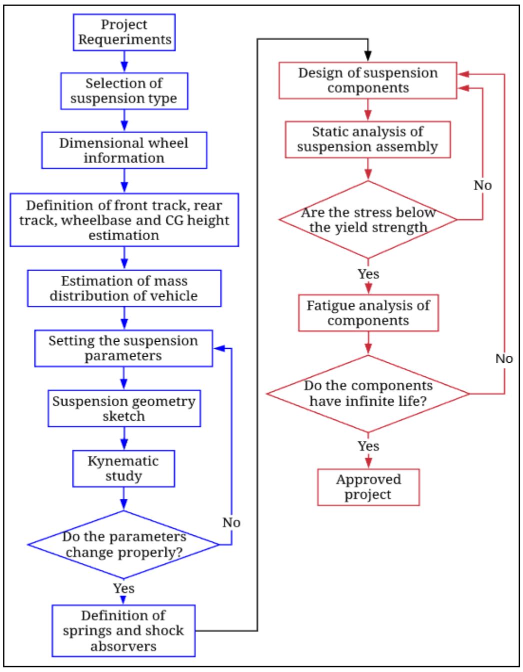

Figure 1. Methodology steps.

3. LITERATURE REVIEW

This chapter will cover the main parameters of a suspension system, such as camber, caster, kingpin, among others, as well as fatigue topics.

3.1 Suspension system

The vehicle suspension is the system that establishes the connection between the carbody and the wheel. According to Costa Neto (2006), the suspension is responsible for the vibration isolation of the chassis caused by track irregularities. According to Gillespie (1992), the primary functions of the suspension can be:

- Allow vertical displacement, so that the wheels can travel on an uneven road, isolating the chassis from roughness;
- Keep the wheels behaving properly to the road surface;
- React to the control forces produced by the tires (acceleration and braking, lateral forces, braking and steering torques;
- Resist to the chassis roll;
- Keep tires in contact with the surface with minimal variation in loads.

3.2 Main parameters of a suspension

3.2.1 Camber angle

Camber angle is the inclination of the wheels when viewed from the front of the car, being defined as the inclination between the radial or vertical axis of the wheel and the vertical axis. Such an angle can be positive when the wheel tilts out of the vehicle, negative when the wheel is tilted in, or neutral when the wheel has no inclination.

Staniforth (1999) states that race cars typically have a negative camber angle of up to 2.5 degrees.

3.2.2 Caster angle

This angle is formed between the inclination of the axle under which the wheel steers and the vertical, projected on the lateral plane of the vehicle. It is positive when the upper point is positioned backward, negative when forward and neutral when this angle does not exist.

One of the objectives of this parameter is the self-aligning torque of the steering, which is generated due to the lateral force acting on the tire/ground contact, being responsible for this moment the caster trail (Milliken and Milliken, 1995). The larger the caster angle, the greater the force required to rotate the steering wheel, however if too small it will result in steering instability. According to Staniforth (1999), values between 2 and 6 positive degrees are advisable.

3.2.3 Kingpin angle

The Kingpin angle is the inclination between the axle under which the wheel steers and the vertical, projected on the front plane of the vehicle. Kingpin compensation is the distance between the ground interference point of the caster axle projection and the center of the wheel.

According to Adams (1993), it is recommended to set the kingpin compensation value as close as possible to 0, as a jounce travel of the suspension caused by an impact or bend, for example, can generate a steering torque if it value is different from 0, causing loss of handling. According to Reimpell (2001), passenger cars usually have a kingpin angle in the range from 11° to 15° 30’, similar to that given by Milliken and Milliken (1995). On the other hand, Adams (1993) states that this parameter varies between 5° and 10°. To define the kingpin angle it must be taken into account that it is related to the camber angle variation, therefore, the higher the kingpin value, the greater the camber gain in relation to steering.

3.2.4 Toe angle

This parameter is defined as the angle between the wheel direction and the lateral plane of the vehicle in the centralized steering condition. The angle becomes convergent and positive when the front of the wheel is closer to the center of the vehicle, and divergent and negative when the other way around.

One of the purposes of this parameter is the elimination of steering compliance, in which lateral forces are generated by tensioning the steering system equally on both sides. In addition, this angle influences the vehicle behavior with regard to oversteer or understeer. According to Adams (1993), in a front suspension the understeer behavior is obtained when there is a divergence alignment, whereas for the rear suspension this alignment must be convergent. Typical values for this angle on the front axle when the vehicle has rear wheel drive are between 0 and +30', while on the rear axle -20’ and +20’ (Staniforth, 1999).

3.2.5 Roll center

According to Reimpell (2001), the roll center is the point in the vertical plane that passes through the center points of the wheels at which lateral forces are transmitted from the axle to the sprung mass. Gillespie (1992) adds by stating that the roll center is the point around which the suspension assembly rotates when subjected to a pure roll motion.

The position of the roll center is governed by the suspension geometry kinematics. According to Staniforth (1999), low center roll height performs low mass transfer from the inside of the curve to the outside, however higher roll angles are produced which can be corrected with higher spring stiffness. According to Adams (1993), most cars that perform well in competitions have a center height between 25.4 mm and 76.2 mm, so it is recommended that the value be set within this range. In addition, the position of the roll center should be defined along with the length of the swing arm.

3.2.6 Bump steer and roll steer

Bump steer is a parameter that indicates the relationship between the vertical movement of the suspension and the variation of the steering angle of the wheel. Roll steer, on the other hand, relates the variation of the chassis roll angle with the variation of the wheel steering angle.

Lateral, vertical or longitudinal accelerations cause the suspension to act and, depending on its geometry, may negatively affect the stability of the vehicle. This is because the construction geometry of the suspension has a relationship with the steering system. The value generally analyzed for bump steer is given in wheel steering angle by vertical wheel displacement, whereas for roll steer is given in wheel steering angle by body roll, so the lower these values, the more accurate the curved vehicle.

3.2.7 Camber roll coefficient

This parameter indicates the relationship between the body roll angle produced by lateral acceleration and the camber angle variation. With this value it is possible to know the camber gain for a given roll angle. Therefore, for racing vehicles the ideal is to maintain maximum tire/ground contact and the camber never becomes positive, and to meet these conditions it is necessary that the camber gain is not very high, preferably less than 1.

3.3 Fatigue

Fatigue is a structural failure caused by crack formation and propagation due to repeated loading cycles, even though these cycles generate stresses that are much lower than the ultimate strength of the material. According to Alves Filho (2011), from 80% to 90% of the failures in machines and structures subjected to variable loading in time are caused by fatigue. This justifies fatigue analysis at the design stage to be of utmost importance for reliable products. It has been demonstrated that fatigue life was affected not only by the variation of cyclic loads, but also by the effect of mean stresses.

The mechanism of crack initiation to component rupture can be understood in three stages: *Initiation* - when the onset of one or more micro cracks caused by cyclic plastic deformation followed by crystallographic propagation occurs. The dislocations move parallel to the maximum shear stress (τ_{max}) (45° in relation to the surface) acting at the critical point of the work piece; *Propagation* - Crack propagation runs perpendicular to normal stress at a rate of *da/dN*. *Fracture* - The remaining section cannot support loading, so component fracture occurs.

There are three main methods used to predict life in number of cycles to failure (N) for a specified stress level: stress-life (S-N), strain-life (ε-N), and the linear elastic fracture mechanics method. According to Shigley et al. (2005), life when less than 10^3 cycles is considered low cycle fatigue, and when it exceeds this range is called high cycle fatigue. It further points out that the S-N method is the least accurate procedure, especially for low-cycle fatigue, although it is the most traditional as it is the simplest and adequately represents high-cycle applications. The ε-N method deals with a detailed analysis of deformations in localized regions where stresses and deformations are considered for life estimates. The fracture mechanics method studies the growth of a pre-existing crack, combined in a practical manner with periodic inspections.

3.4 *S-N* method

In the high cycle study, the S-N method is commonly used with the Wöhler curve or S-N curve as is typically known. It relates the amplitude of the stress to the number of cycles associated to material failure, in which the total life of a component subjected to cyclic loading is a combination of the cycle numbers of the three stages of the crack mechanism.

Figure 2 shows a graph of the S-N curve of a ferrous material, where the abscissa axis represents the number of cycles (N), and the ordinate axis the fatigue stress S_f.

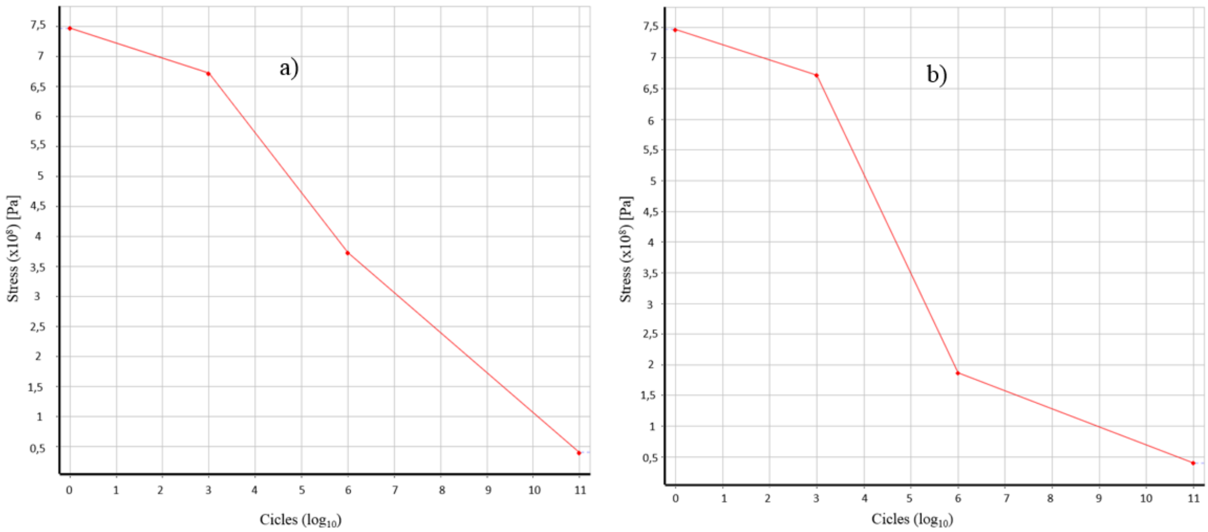

Figure 2. a) Estimated S-N curve for SAE 1045. b) S-N curve calculated for SAE 1045 affected by weld.

In the literature it is possible to find S-N curves of the most diverse materials, which were developed from fatigue tests. However, it is often not possible to find a reliable curve for the material under study, so Castro and Meggiolaro (2002) suggest a method to estimate it without the need for an experimental test. What supports this idea are the data obtained in the studies by Juvinall (1967). The fatigue stress, when the number of cycles is 10^3, is equal to 90% of the ultimate stress of the material; the fatigue limit (S_e) equals 50% of the ultimate stress of the material when the number of cycles is 10^6. Mathematically, these definitions are expressed in Equations 1 and 2:

$$S_f(N = 10^3) = 0.9\, S_{ut} \tag{1}$$

$$S_e(N = 10^6) = 0.5\, S_{ut} \tag{2}$$

3.5 Fatigue strength modifying factors

The determination of fatigue curves is carefully done in controlled environments and on polished surface specimens. But it is known that the conditions under which everyday applications are subjected are not consistent with those of the tests. Therefore, to correlate the applications with the tests in relation to the fatigue limit, it is necessary to combine the modifying factors with the fatigue limit value of the specimens (S'_e). According to Shigley et al. (2005), the main factors that should be taken into account, when applicable, are: surface condition (k_a), size (k_b), load (k_c), temperature (k_d), reliability (k_e), miscellaneous effects (k_f). Mathematically these definitions are expressed in Eq. (3):

$$S_e = k_a.k_b.k_c.k_d.k_e.k_f.S'_e \tag{3}$$

A manufacturing process present in most applications is the welding process. This in turn significantly impacts the fatigue limit. There are standards that refer to the welding process as a modifying factor, which may vary according to the quality with which it is made. A standard that deals with weld quality is the "Swedish Regulations for Welded Steel Structures" 74 StBK-N2, National Swedish Committee on Regulations on Steel Structures", which is commonly used in projects subject to metal structure fatigue in general mechanical construction. (apud. Alves Filho, 2011).

3.6 Effect of mean stress on fatigue strength

The *S-N* curve search tests of the materials are performed by compressing and tractioning the specimen in such a way that the tensile and compression load are the same, however in the opposite direction, resulting in a mean stress equal to zero. The effects of mean stress should be properly considered in fatigue analysis (Lee et al., 2012).

3.7 Palmgren-Miner rule

The life of a given component can be defined as the number of cycles required to start a crack for a given stress amplitude (Alves Filho, 2011). Any stress in a given component that is above the fatigue limit will accumulate some damage. Palmgren-Miner states that when the amount of accumulated damage equals one, fatigue failure may occur. This rule is mathematically represented in Eq. (4):

$$D = \sum \frac{n_i}{N_i} \tag{4}$$

where,
D: is the accumulated damage;
n_i: is the number of stress amplitude cycles σ_i in a certain time interval;
N_i: is the number of stress amplitude cycles σ_i required for the failure to occur.

4. PARAMETERS DEFINITION AND KINEMATIC SIMULATIONS

In this section the design and kinematic simulation of the front suspension system will be presented, based on the theoretical references presented in section 3.

4.1 Parameters definition

The type of suspension chosen for the project is multilink because it performs well compared to other options. Assumptions were done to estimate the gravity center height, vehicle mass and weight distribution.

According to the theoretical framework of this work the authors Staniforth (1999), Reimpell (2001), Milliken and Milliken (1995) indicate and relate values for the parameters that involve the kinematic design of the suspension. The chosen parameters are listed in Tab. 1:

Table 1. Suspension parameters.

Parameter	Front	Rear
Camber	-2º	-1°
Caster	4.3°	-
Kingpin	12º	-
Toe	+30'	-20'

4.2 Kinematic simulations

This section presents the kinematic simulations of the front and rear suspension systems as defined previously. The simulations were performed in the MSC Adams/Car multibody software, from multibody models created according to the geometric definition of the suspensions.

The front suspension system is represented in Fig. 3. To simulate the kinematic behavior of the suspension parameters, a parallel wheel travel test was performed, in which the program applies a vertical wheel displacement and then calculates the coordinate displacement. Thus, all variables were analyzed in relation to the wheel travel, which was plotted on the horizontal axis of the graphs. It is observed that the values in the horizontal axis range between -35 mm and 35 mm, being negative the rebound and positive the jounce.

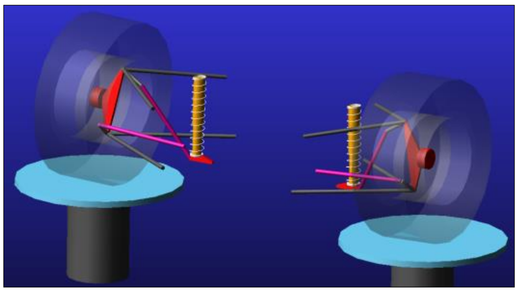

Figure 3. Multibody model of the front suspension.

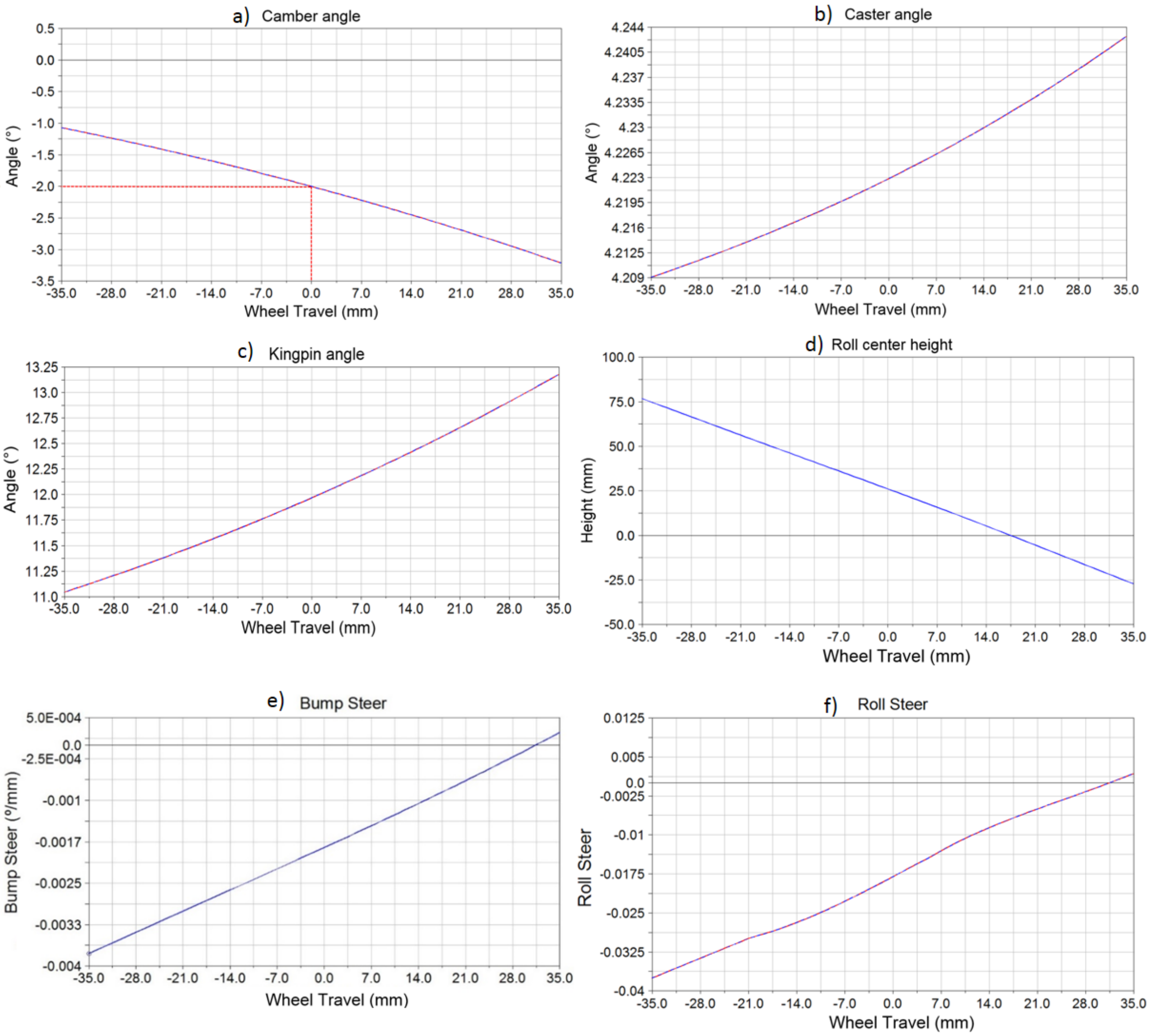

Figure 4. Graphs of the suspension parameters behavior.

Figure 4-a shows that the angle variation is almost linear assuming values between approximately -1º (-35 mm) and -3.25º (35 mm). Thus, it is possible to analyze that in the jounce (0 mm to 35 mm) there is a negative camber gain, and in the rebound (0 mm to -35 mm) the camber gain is positive, which is important for the handling performance.

Regarding Fig. 4-b, it is observed that the caster varies approximately 0.033 degrees, which can be considered negligible. Fig. 4-c shows the kingpin angle varying similarly to the camber, but in the opposite direction.

In Fig. 4-d, a practically linear rate of change is identified between the height of the roll center and the wheel travel.

In the vertical axis of the graph of Fig. 4-e, the values corresponding to the bump steer effect are observed, which relates the variation of the steering angle for a given wheel displacement, and the values analyzed in the vertical axis are very close to zero. This means that this effect is minimal, so it is considered that the suspension travel will not negatively affect the handling of the vehicle. The same interpretation holds for the graph in Fig. 4-f, which relates the chassis roll angle to the wheel steering angle. As with bump steer, roll steer will not affect handling performance. The same analyzes were done for rear suspension.

5. DESIGN AND STRUCTURAL ANALYSIS

This chapter will deal with 3D design of suspension system, vehicle dynamics analysis and structural simulation by the FEM. Finally the fatigue calculation will be performed based on the results of the FEM and the vehicle dynamics.

5.1 Design 3D

Based on the kinematic definition it was possible to design the main components of the front and rear suspension system with the aid of the SolidWorks software as can be seen in Fig. 5.

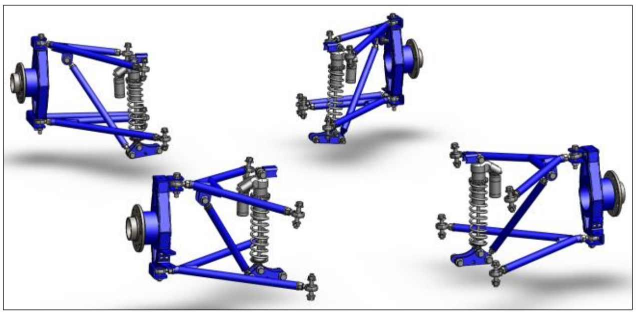

Figure 5. Perspective view of full suspension system.

5.2 Data acquisition

To perform the structural analysis by the FEM it was necessary to know the conditions under which the suspension would be exposed. An alternative to experimental methods is the modeling of vehicle dynamics, which consists of defining some relevant characteristics of the vehicle and when performing tests it is possible to extract results, in this case, the force history acting on the tire/ground contact.

Thus, CarSim software was used to create the vehicle in study, using a template of a Formula SAE and a standard road (Fig. 6), changing suspension mass, parameter and kinematics information. It was also possible to change the driver action on the steering. In this case it was opted for the limited steering by the traction, that is, the driver drives to the limit speed before the tires slip.

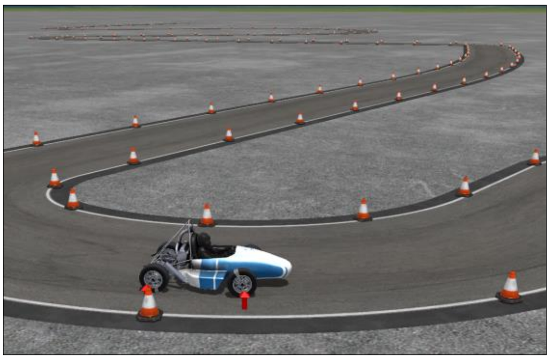

Figure 6. Vehicle dynamics simulation into CarSim.

From the CarSim analysis it was possible to export the force history present in the tire/ground contact of the front right wheel, which can be observed in Fig. 7.

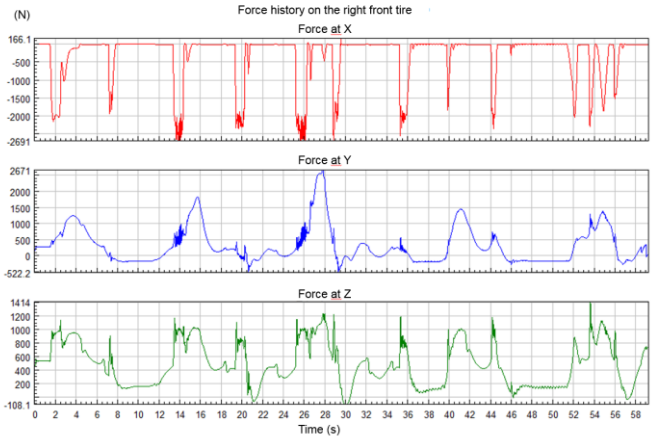

Figure 7. Force history of right front wheel tire/ground contact.

5.3 Material properties

The mechanical properties of the material define the structural characteristics of each component for a simulation. The most relevant for this study were the specific mass (ρ), elasticity modulus (E), Poisson ratio (ν), yield strength (σ_e) and the ultimate stress (σ_u), which are detailed in Tab. 2:

Table 2. Material properties.

Material	ρ (kg/m³)	E (GPa)	ν	σ_e (MPa)	σ_u (MPa)
SAE 1045	7850	209	0.3	510	747

5.4 Static structural analysis

The forces applied in the static analysis were identified in the force history at the moment they generate the largest resulting force, which are detailed in Tab. 3:

Table 3. List of obtained forces.

	Force at X (N). Longitudinal	Force at Y (N). Lateral	Force at Z (N). Vertical	Resulting Force (N)
Front right wheel	-2691	797	1129	3025

Figure 8 shows the equivalent stress result of static analysis of the front suspension. It can be verified that the critical point has stress in the order of 481 MPa, and the yield stress of the material is 510 MPa. Thus, it can be said that the components of the front suspension are sized to withstand the stresses resulting from the road condition without the yield of material.

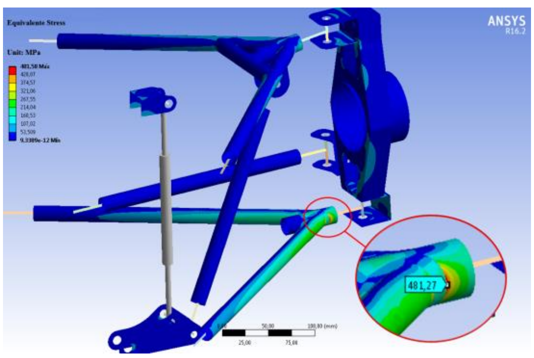

Figure 8. Equivalent stresses from the static analysis of the front suspension.

5.5 Fatigue calculation

After concluding that the components support the static loads without material yield, the fatigue calculation is performed in order to quantify the durability of these components. For this calculation, the specific durability software Ncode Design Life was used. The flow of the calculation can be understood in Fig. 9. From the stresses related to a load of 1 N, they were multiplied by the force history resulting in a stress history; Thus the program performed the cycle count by the rainflow method and calculated the damage generated by the signal.

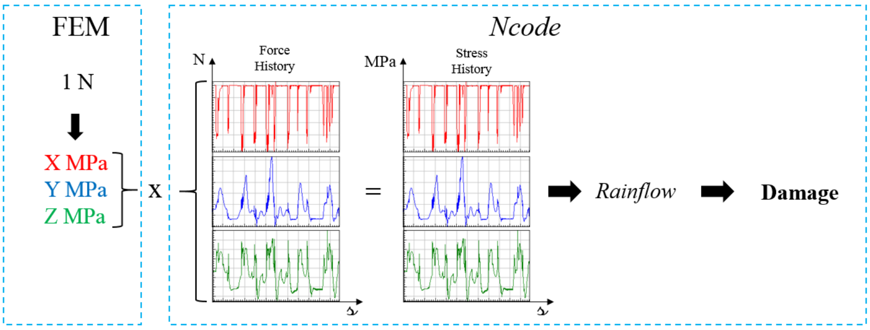

Figure 9. Fatigue Calculation Flow Chart.

As a failure criterion for this analysis, it was considered that the components must have a longer life than necessary to perform tests and competition tests. In the endurance race, the vehicle must complete 20 laps on the track, plus one lap for autocross. Therefore, the set value of laps that the vehicle needs to endure is 21 laps in the competition plus 20 times for the tests and a safety factor of two times, and the vehicle can be used in two competitions. Thus, the fatigue failure criterion is a minimum life of 1,764 turns. Fig. 2-a and 2-b show the representation of the S-N curve for raw material and fatigue reducing factor modification, respectively.

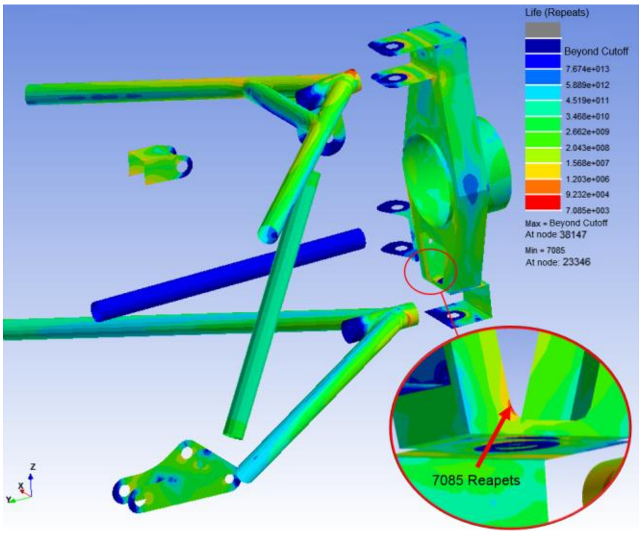

Figure 10. Right front suspension component life.

Figure 10 shows the result of the durability of the front suspension in terms of life in number of lap repeats on the road. The minimum value found for the front suspension was 7,085 turns for the knuckle suspension component in the region indicated by the arrow in Fig. 10.

Thus, it can be said that the front knuckle is vulnerable to fatigue. However, its 7,085 lap front suspension durability is longer than its required life of 1,764. Therefore, the front suspension will not fail due to fatigue, leading to the conclusion that the design is dimensioned against fatigue.

6. CONCLUSIONS

In this paper an approach for the design and evaluation of fatigue life of suspension components has been presented. The results of the static simulations showed that the Von Mises stresses of the suspension components are below the material yield limit. As such, the front suspension components are sized to withstand the maximum stresses calculated in the force history. Suspension component durability calculations were performed with the Ncode Design Life software. Results indicated that the life of such components is longer than the defined failure criteria. Thus, it can be considered that the design was dimensioned against fatigue.

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