Abstract—The research presents the redesign, construction and assembly of the rear suspension system for the German SAE formula type vehicle with topological optimization in its elements. It starts with an introduction of basic concepts of suspension geometry and ends with the justification for the choice of the system type. In the design stage, force, stress, mass transfer, velocity, acceleration, static, dynamic and kinematic analysis are performed with ANSYS software and finite element analysis (FEM); manufacturing involves cutting, CNC machining, threading, welding and analysis of non-destructive tests, penetrating inks, magnetic particles according to ASTM E165, ASTM E1444 and ASTM E709 standards, all these processes respecting the SAE formula regulation which guarantees safety and efficiency in the system. The ideal suspension system was obtained; in addition, a rocker or rocker optimized to 46.6% and the material to be manufactured

Index Terms—Formula SAE, CAD/CAE, Suspension system, optimization. FEM

I. INTRODUCCIÓN

The rear suspension system is a set of elements whose purpose is to absorb the irregularities of the road through which the vehicle travels and to allow control of the vehicle’s trajectory thanks to the quality of the wheel-soil contact, ensuring stability in any circumstance; therefore, it is exposed to damage both in the short and long term, which is why it is necessary to select a material that meets the requirements of good mechanical resistance [1], with a light weight.

The topology of optimization is defined as a numerical method that has captured the interest of engineering and scientists in recent years, since it allows the synthesis of structures with optimal values of one or several physical parameters [3].

Integration of topological optimization tools and knowledge management in the process of developing virtual product development of components in the automotive field [4], is a work carried out with optimization topology where the results show that the design approach can efficiently support the selection of the best conceptual design of a mechanical component. This research focuses directly on a suspension element of a prototype for participation in the German SAE formula; it also clearly defines optimization and graphs the optimization topology in Inventor software.

The proposed project achieves a difference in weight compared to the rear suspension prototype of the previous FESPE 2012 prototype vehicle[5] while maintaining the anchorage points fig. 1, optimizing the geometric configuration that minimizes or maximizes a certain function and integrates the constraints or conditions of the problem; in addition, the suspension system must achieve sufficient elasticity and at the same time the rigidity necessary for the vehicle to achieve performance in comfort and operation.

Fig. 1. Rear suspension system.

Topological optimization seeks the optimal distribution of material in a structure, by suppressing elements or modifying the connectivity of nodal coordinates [6].

The topological optimization method will be used for the elaboration of the rocker or rocker arm, since it has a previous design, which is intended to evaluate with all the loads that it will support and thus obtain an ideal model with
the distribution of optimal material by cancelling nodal coordinates.

Suspension Kinematics Study of the "Formula SAE" Sports Car, is one of the articles that explains that the performance of an automobile in competition depends on the kinematics of its suspension [7].

II. DEVELOPMENT

A. Vehicle suspension systems formula student

The pull and push rod systems are a new technology that in single-seater vehicles have further reduced weights and, above all, aerodynamic resistance because they have arrived with a new era of control [8].

In the present work the push rod system is applied, optimizes the weight and maintains the anchorage points, in fig. 2 we can see the push rod system.

![Fig. 2. Push rod.](image)

B. FESPE 2012 prototype vehicle characteristics

The racing vehicle FESPE 2012 has a Kawasaki ninja ZX-6R engine of 130 hp, a weight of 270 kg without driver, this one must accelerate from 0 to 100 km/h in an average of 6 seconds [5], since it possesses these qualities, stable suspension and steering systems are necessary to avoid any type of accidents.

Table I details the main characteristics of the FESPE 2012 prototype [3].

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Magnitude</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Power</td>
<td>130</td>
<td>CV</td>
</tr>
</tbody>
</table>

| Vehicle weight   | 275       | Kg   |
| Acceleration     | 2,84      | m/s² |
| Tip Speed        | 150       | Km/h |
| Max. G Force     | 2         | G    |
| Wheelbase        | 1570      | mm   |
| Suspended weight (with pilot) | 340 | kg |
| Semi suspended weight | 48 | kg |
| Current weight rear suspension | 1297 | gr |
| Track gauge      | 1,203     | m    |
| Height to swing center | 0,1225 | m |

The front suspension arms of the FESPE vehicle are made of A36 steel tube with a thickness of 2mm. Material optimisation allows you to change to a lighter material with similar characteristics such as aluminium [9]. The previous Rocker are two A36 steel plates welded to a cylinder of the same material, an improvement with respect to weight is obtained through the manufacture of aluminum 7075 by CNC machinery that exceeds in resistance to steel [10].

Prior to stress calculations that the system must withstand, weights and measures are considered in accordance with article A1.2.2 of the SAE formula regulation [11].

III. DESIGN

A. Mass calculation in the student formula vehicle FESPE 2012

Opting for a conservative design, a total mass of 380 kg is applied to the vehicle. With center of gravity and wheelbase due to preserving the attachment points in the bodywork is applied (1).

$$m_{tt} = \frac{mt \times l_2}{L}$$

Where: $m_{tt}$ = total rear mass, $mt$ = total vehicle mass, $l_2$ = distance from centre of gravity to front axle, $L$ = wheelbase.

B. Rear axle force calculation

$$F_{st} = m_{tt} \times g$$

Where: $F_{st}$ = Rear axle force, $g$ = gravity.

The resulting force obtained with (2) is 2076.19 N, a force of 1038.095 N on each rear wheel.
C. Longitudinal mass transfer – Acceleration

As soon as the vehicle accelerates it produces a mass transfer to the rear axle, it must support the rear suspension and prevent the vehicle from losing its optimum configuration.

The traction force is obtained by replacing the mass transfer equation in (3) of traction.

\[ f_t = \frac{F_{at} \times u}{1 - \left(\frac{u}{1 + u}\right)} \]  \hspace{1cm} (3)

Where: \( u \) = coefficient of friction = 1.5, \( l \) = centre distance = 1570 [mm], \( f_t \) = traction force [N]

The traction force is used to calculate the mass transfer under acceleration with (4).

\[ \Delta w_{xa} = \frac{f_t \times h m}{l} \]  \hspace{1cm} (4)

Where, \( \Delta w_{xa} \) = Mass transfer under acceleration.

D. Calculation of force transfer in curves

To obtain curve results it is advisable to calculate the loads of each wheel on a SAE formula vehicle with a negative center of gravity (CG) shift inclined and left turn. The car is accelerated and subject to aerodynamic forces [1].

The summary of all the loads that the vehicle can support is shown in Table III.

Therefore, in a left turn curve, the rear right tyre is the most heavily loaded, with a force of:

\[ F_{curva} = 1945.797 \text{ N} \]

Curved force \( F_{curva} \) data is used to study the components of the rear suspension system in simulation programs.

Using graphical method and breast law, the force is obtained in each arm of the suspension system. In then fig. 3 it is observed that \( \beta \) is the angle between the suspension arms FAD-vrt with FAC-vrt and \( \alpha \) is the angle between the horizontal force of the Hpush-vrt and the suspension arm FAD-vrt; the same data you have from \( \beta = 100^\circ \) and \( \alpha = 40\degree \), experimentally tested[5].

E. Force distribution in the Rocker

Already obtained the forces detailed in table III, the free-body diagram of the rocker arm is made, fig. 4.

Tabla III. RESULTING FORCES MAXIMUM IN THE ROCKER

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Push Bar (C)</td>
<td>2850 N</td>
</tr>
<tr>
<td>Spring-damping assembly (A)</td>
<td>1321 N</td>
</tr>
<tr>
<td>Torsion bar (B)</td>
<td>700N</td>
</tr>
<tr>
<td>Pivot point (D)</td>
<td>Fixed</td>
</tr>
</tbody>
</table>
IV. CAE STUDY (ANSYS).

A. Topological Optimization in CAE software

A structural study is initiated in the software ANSYS applying the loads of the rocker and to obtain a mesh with a Jacobian the closest to 1 representing the center of the straight edges that increases as the curvatures of the edges increase. In this study, a value of 1.006 was obtained, fig. 5.

Mesh size, and forces are taken from the structural study. The solution obtained is shown in fig. 6, with a mass reduction of 46.6%.

B. Analysis and simulation of suspension system elements

In the load analysis of the optimized rocker, fig. 7, the loads generated on the critical points of the piece were applied, the values used are shown in table IV.

Fig. 8 shows the static safety factor of the rear rocker with the material 7075 T6 and a plate thickness of 4mm gives a minimum safety factor of 3.012 as this greater than 1 indicates that the rocker is in the safety range and will not fail when applying static loads-distribution of forces in rocker.

Fig. 9 shows as a result the equivalent effort - Von Mises in aluminium rocker obtains a maximum equivalent of...
158.17MPa in the plates and a minimum of 20.659 KPa in the pivot point.

C. Simulación y análisis del rocker a fatiga

Perform the analysis of the rocker optimized to dynamic load (fatigue), since the material to be used in this element has linear properties it was configured under the failure criteria of the Goodman Modified theory [12].

Fig. 10 shows the fatigue design factor of the rear rocker with mechanical properties of material 7075 T6 with a minimum of 1.61 generated in the plate. This design factor of 1.61 indicates that in the working process the rocker can withstand between 5 to 10 % more in the maximum forces applied in the design.

Table V shows the summary of results obtained in the computational analysis of static and fatigue rocker.

D. Results obtained in Ansys software

Dynamic study in is important in designing a mechanism to interpret the behavior of bodies interacting with each other and to avoid unwanted interference [13].
to the ground in a frontal view of the vehicle, it is recommended that a SAE formula prototype has a negative camber, Fig. 12 shows that at the starting point has an angle of -1°; if the tyre rises 50mm the camber angle varies to 1.3° [13].

![Displacement vs Time](image1)

**Fig. 13. Displacement – String force vs Time.**

The displacement and the force supporting the spring is visible in fig. 13. In a time of 6.5 seconds it makes a total displacement, starting from 190mm which is the length at rest, 180mm when compressed and a length of 210mm when expanded. This spring is designed to withstand a force of 6000N.

V. NON-DESTRUCTIVE WELDING TESTS

A. Test for penetrating liquids

According to ASTM E165 Test Method for Liquid Penetrant Examination, the penetrating liquid tests consist of applying a liquid on the surface of the body to be examined, which penetrates by capillary action into the weld imperfections [14].

The penetrating liquid Magnaflux SKL-WP2 is sprayed on the part, fig 14. It must comply with the main characteristic of being difficult to remove once inside the discontinuity, it is important to wait for the penetration time recommended by the manufacturer of 15 minutes.

![Part bathed in penetrating liquid](image2)

**Fig. 14. Part bathed in penetrating liquid.**

Magnaflux developer liquid SKD-S2 is sprayed very carefully as it is a volatile liquid. This liquid is transparent but once applied, wait a time of 10 minutes, recommended by the manufacturer to react and turn it into white color; then, the areas of the piece containing traces of penetrating liquid will stand out to the naked eye, fig. 15.

![Part bathed in developer liquid](image3)

**Fig. 15. Part bathed in developer liquid.**

The imperfections are clearly and accurately marked along the weld bead, the observation for this type of liquids will be made under the ASTM E165 standard, with surface red dots appearing in the lower triangle but in an operational state, fig. 16.

![Pores and cracks in the welding process](image4)

**Fig. 16. Pores and cracks in the welding process.**

B. Magnetic particle testing

According to ASTM E1444, magnetic particle tests have the mission of detecting possible discontinuities in a welded part that are not only found superficially, but also in the vicinity of it [15].

The element must be completely clean to start the test, in fig. 17, the steps to be followed are observed; apply the SKD-S2 spray of visible magnetic particles suspension (a), magnetize the element with the help of the magnetic yoke MAGNAFLUX Y7-AC-DC-77 for 5 minutes (b), apply the white contrast paint (c) and then wait until irregularities are visible on the weld seams.

![Element with magnetic particles](image5)
Finally, in Fig. 18, the irregularities in one of the weld seams can be observed.

Once the process was finished, all the test results were observed, which were evaluated according to the ASTM 1444 standard of magnetic particles, which defines that for the welding cord to be in an operational state it must only have 10% of discontinuities in its total length; in this case, the upper arm had a 30cm cord with 3 imperfections, fig. 18, where each fault represents 1%; i.e., the process is qualified as welding in operating condition with slight surface defects [16].

VI. ANALYSIS OF RESULTS

One of the characteristics to be considered is the previous and current masses of all the elements that make up the rear suspension system, which are detailed in table VI.

<table>
<thead>
<tr>
<th>Element</th>
<th>Previous Mass (gr)</th>
<th>Current Mass (gr)</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper triangle</td>
<td>839</td>
<td>448</td>
<td>2</td>
</tr>
<tr>
<td>Bottom triangle</td>
<td>712</td>
<td>555</td>
<td>2</td>
</tr>
<tr>
<td>Push rod rod</td>
<td>195</td>
<td>41</td>
<td>2</td>
</tr>
<tr>
<td>Rocker</td>
<td>487</td>
<td>253</td>
<td>2</td>
</tr>
<tr>
<td>Summation</td>
<td>2233</td>
<td>1297</td>
<td>8</td>
</tr>
</tbody>
</table>

The percentage that was achieved to reduce in weight is 41.92% (936gr) in all the rear suspension system; the same, is a representative number so that the vehicle can develop better in the competition SAE formula, table VI.

<table>
<thead>
<tr>
<th>Material</th>
<th>Deflection (m)</th>
<th>Equivalent Stress Von Mises (Pa)</th>
<th>Static safety factor</th>
<th>Maximum main effort Safety factor</th>
<th>Fatigue</th>
<th>Life Cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum 7075</td>
<td>0.000463</td>
<td>1.58*10^8</td>
<td>3.01</td>
<td>1.26*10^5</td>
<td>1.61</td>
<td>5*10^8</td>
</tr>
<tr>
<td>Steel A36</td>
<td>0.000210</td>
<td>1.66*10^8</td>
<td>2.76</td>
<td>1.46*10^5</td>
<td>2.19</td>
<td>1.08*10^11</td>
</tr>
</tbody>
</table>

The comparison of materials can be seen in table VII where the equivalent stress in aluminum ASTM 7075 (1.58*10^8 Pa) is 5.06% lower than in ASTM A36 (1.66*10^8Pa) steel, benefiting the static safety factor of the element by having a breaking strength of 572 MPa in aluminum and 495 MPa in steel.

It was analyzed that the welded aluminum alloy suspension arms meet the application of penetrating liquids and magnetic particles, which determines their operability according to the acceptance and rejection criteria seen in both ASTM E165/270M and ASTM E1444.

VII. CONCLUSIONS

• The rear suspension system of the FESPE prototype was redesigned and built in accordance with the norms and parameters established in the SAE 2017-2018 formula regulation, and the maximum efforts supported by the vehicle.
• The maximum forces entering the tyre are a vertical force of 2703.42 N and a lateral force of 3784.78 N, which are distributed on the upper and lower arms and the push rod, when the vehicle takes a curve at 40 km/h with a radius of 150m.
• The maximum deformation in the optimized aluminum rocker ASTM 7075 is 0.46mm represented a 119% increase over the deformation that occurs in steel A36
(0.21mm) due to the fact that Young’s aluminum module (71.7 GPa) is lower than that of steel A36 (169 GPa).

The weight ratio of the suspension system made of aluminum with respect to steel is lower by 41.92% (936 gr); in addition, aluminum 6061-T6 is 2.2% higher in strength than A36 steel.

The fatigue safety factor in aluminum rocker ASTM 7075 is 1.61 which compared to ASTM A36 steel with a safety factor of 2.19 is 26% lower because of the fatigue strength characteristic which is 159 MPa for aluminum and 270 MPa for steel, this characteristic also influences the life cycles increasing by 1*103 steel life cycles with respect to aluminum.

When performing the non-destructive welding tests on the suspension arms, it was determined that the aluminum alloy welded suspension arms comply with the application of penetrating liquids and magnetic particles, which determines their operability according to the acceptance and rejection criteria seen in ASTM E165/270M and ASTM E1444 respectively, qualifying these weld beads as OPERATIONAL state.

REFERENCES


