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Research Paper

Fatigue Life and Topology Optimization of Racing Car Upright for Formula SAE Electric

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Abstract

This research aimed to reduce the mass of a racing car upright made from Aluminum Alloy 7075-T6 through topology optimization and fatigue life analysis. The design process included stages of meeting over-design conditions, optimizing mass reduction for the uprights, and smoothing critical areas. Finite element simulation was used throughout to analyze strength and fatigue life, considering loading conditions, geometry, and material properties. Special attention was given to critical areas to ensure optimized stress distribution and minimize stress concentration. The results showed that extreme loading conditions occur during braking while turning. The optimization process followed boundary conditions and design requirements, resulting in a 56% mass reduction from 944.39 grams to 416.43 grams while maintaining structural integrity. The optimized design featured a larger fillet radius, reducing stress concentration in critical areas and lowering the maximum stress value. The final design demonstrated a smoother structure with reduced stress concentrations, confirming the effectiveness of the optimization.

Keywords: Topology optimization; Front upright; Fatigue life; Strength analysis; Mass reduction

1. Introduction

Energy efficiency is a solution to prevent the energy crisis currently threatening the transportation sector, such as an increase in cost and greenhouse gas emissions. One form of efficiency is to reduce the weight of vehicle components by improving the formation process [1], the use of lightweight materials [2], [3], and design optimization [4]. For example, an upright or knuckle is a crucial component of a racing car that connects the suspension and wheel assembly. The two types of upright used in the electric racing car are the central and in-wheel motors [5]. Generally, various kinds of loads are received by this vehicle during acceleration, braking, or turning. Therefore, its design should be strong, reliable, and light, especially for an electric car that still has challenges in increasing its mileage due to battery capacity.

The fatigue load that occurs while driving this car on the track will significantly influence the durability of the upright component. Several studies have been conducted on the fatigue analysis of the upright of racing cars [5]–[8]. Furthermore, the fatigue behavior using the finite element method has been studied preliminary by Jhala et al. [9] for three steering knuckle materials and characteristics of upright made of lightweight materials [10].

Weight reduction can improve the performance of this vehicle type, and this is achieved using the finite element method, commonly used for strength and failure analysis of components [10], [11]. Finite Element applications for upright weight reduction are achieved using OptiSctruct [12], shape optimization [13], and minimizing the surface area [14]. Additionally, the weight reduction

Nomenclature

d	: Deceleration (m/s^2)	Wf	: The reaction forces on the front wheel
g	: Gravity Acceleration (m/s^2)	Wr	: The reaction force on the rear wheel
h	: CG height (m)	σ_a	: Amplitude Stress (MPa)
Ha	: Height of aerodynamics force (m)	σ_m	: Mean Stress (MPa)
L	: Wheelbase (m)	σ_e	: Equivalent Strength (MPa)
Lf	: Horizontal distance from front axle to CG (m)	σ_y	: Yield Strength (MPa)
Lr	: Horizontal distance from rear axle to CG (m)	σ_{UTS}	: Ultimate Tensile Strength (MPa)
v	: Velocity (m/s)	CG	: Center of Gravity
Ra	: Aerodynamics Force (N)	FEM	: Finite Element Method
W	: Car Weight (N)	FSAE	: Formula Society of Automotive Engineer

process can be achieved through additive manufacturing [15] and topology optimization [13]–[15].

The weakness associated with the topology optimization process is that it is only based on static loads [16], without considering fatigue conditions on the upright. This racing car traverses a circuit full of turns; hence there will be dynamic loads in the form of acceleration and the application of brakes during the race [17]. The upright design that only considers static loads cannot cover the effects of fatigue due to dynamic loads. Meanwhile, topology optimization, which considers fatigue analysis is still the subject of research by computer scientists and structural engineers [18], [19]. One limitation of this technique is the numerical difficulty found in the fatigue failure criteria. Until now, topology optimization, which considers fatigue, is still limited to simple structures, namely two-bar, cantilever beam, L-bracket, and double L-shape.

This research aims to reduce the upright weight while ensuring the fatigue life of racing cars through the design optimization process. The method identifies critical areas in the existing design under static and fatigue load simulation. The topology optimization process focuses on critical areas optimized to reduce stress concentration. The final design results from topology optimization showed weight and stress reduction with safety in receiving fatigue loads compared to the existing design. This research contributes to forming a light upright design for the racing car that competes in Formula SAE Electric.

2. Method

2.1. Upright Design & Simulation

Upright is the main component of a racing car suspension, which transfers the vehicle's load to the ground through the tires during trips. All suspension components are connected to the upright, including tires, brakes, springs, control arms, shock absorbers, and steering arms. Therefore, any failure of this component will make the car impossible to drive. **Figure 1** shows the initial upright geometry based on several constraints, such as the front wheel hub position, inner and outer bearings, upper and lower arm mounting point coordinates, and tie rod steering position.

In straight driving conditions, the load is evenly distributed on each tire. However, the vehicle's maximum weight is shifted to the front axle under braking conditions. The same thing can happen when driving on a curvy or sloping road. The inertial force due to deceleration makes it undergo wheel rolling resistance and wind drag. Therefore, the most challenging condition experienced by the vehicle is when experiencing braking while turning. The free-body diagram of a vehicle experiencing a deceleration on a straight horizontal road is shown in **Figure 2a**. The reaction forces on the front wheel (Wf) and rear wheel (Wr) can be obtained by applying the Equilibrium of the rigid body as illustrated in equations (1) and (2). Meanwhile, when the vehicle is in a turning condition, a centrifugal load is added to the outside using equation (3), as shown in **Figure 2b**.

Finite element simulation is used to acquire an approximate solution to an engineering problem involving loading conditions, geometry, and material properties on the upright. This process

$$\sum M_A = 0; \quad W_r = \frac{R_a.h_a + W.L_f - \left(\frac{W}{g}\right).d.h}{L_f + L_r} \tag{1}$$

$$\sum F_z = 0; \quad W_f = W - W_r \tag{2}$$

$$\sum F_y = \frac{mv^2}{R} \tag{3}$$

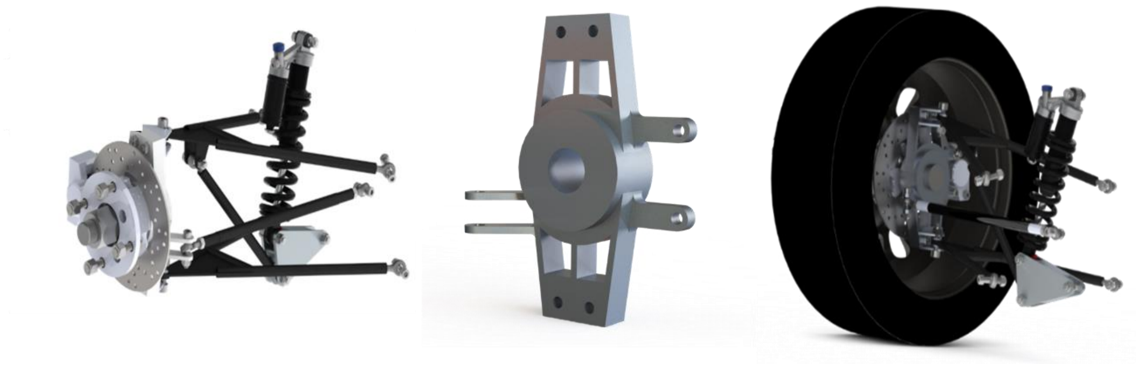

Figure 1. Design structure of the front upright

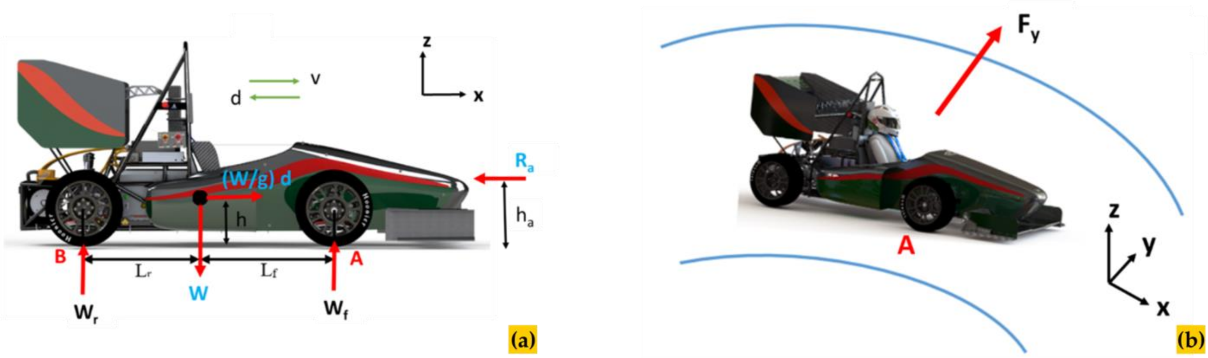

Figure 2. The forces that occur when the racing car experiences conditions: (a) deceleration in a straight line and (b) turning

uses three brackets as support: the upper arm, forearm, and brake calipers. Meanwhile, the loading comes from the force and the moment on the upright when experiencing braking and turning conditions. The material selected for this component is Aluminum Alloy 7075-T6 because it has a high strength-to-weight ratio as shown in **Table 1**.

The central concept of finite element simulation is to utilize the discretization principle in the form of a finite number of small elements. The type of element used in the upright simulation is a tetrahedron with a mesh size of 1 mm, and the number is 470,836. The reaction due to the load at the given boundary condition can be calculated with a finite number of small elements. These can be arranged into matrix equations and solved numerically. The matrix results answer the load conditions given to the upright.

2.2. *Fatigue Life and Topology Optimization*

Automotive components often fail due to fluctuating loads even when the stress is below the yield strength. This is because fluctuating loads on the components make it more dangerous and prone to failure than static loads. Furthermore, the fluctuating loads can cause more failure at stress levels than the elastic stress of the material. The failure is usually due to the spread of tiny defects at each receiving stress.

Table 1. Mechanical properties of Aluminum Alloy 7075-T6

Mechanical Properties	Value	Unit
Ultimate Tensile Strength	572	MPa
Tensile Yield Strength	503	MPa
Elongation at Break	11	%
Modulus of Elasticity	71.7	GPa
Poisson’s Ratio	0.33	
Density	2.81	g/cm^3
Hardness	150	Brinell

Fatigue analysis for a component needs to be carried out based on actual loading conditions to determine the right resistance results. Moreover, components that experience varying loading amplitude conditions tend to affect the fatigue life obtained significantly. The fluctuating load analysis in the racing car comes from historical data obtained by dividing the Formula SAE Electric endurance event. The trajectory consists of 5 conditions, including turning while braking with a radius of 6.75 m (green), 10 m (red), 12.5 m (blue), 15 m (purple), and straight without braking (grey), as shown in Figure 3. The time taken for each condition and the stress experienced by the components are calculated by dividing each condition. The stress data experienced by the components each time is then accumulated as input to the fatigue life analysis, as shown in Figure 4. Fatigue failure occurs if the stress amplitude exceeds a material's endurance limit and when the maximum stress exceeds the yield. The mean stress theory is determined as seen in Equations (4) until (6).

Soderberg
$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_y} = 1 \tag{4}$$

Goodman
$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_{UTS}} = 1 \tag{5}$$

Gerber
$$\frac{\sigma_a}{\sigma_e} + \left(\frac{\sigma_m}{\sigma_{UTS}}\right)^2 = 1 \tag{6}$$

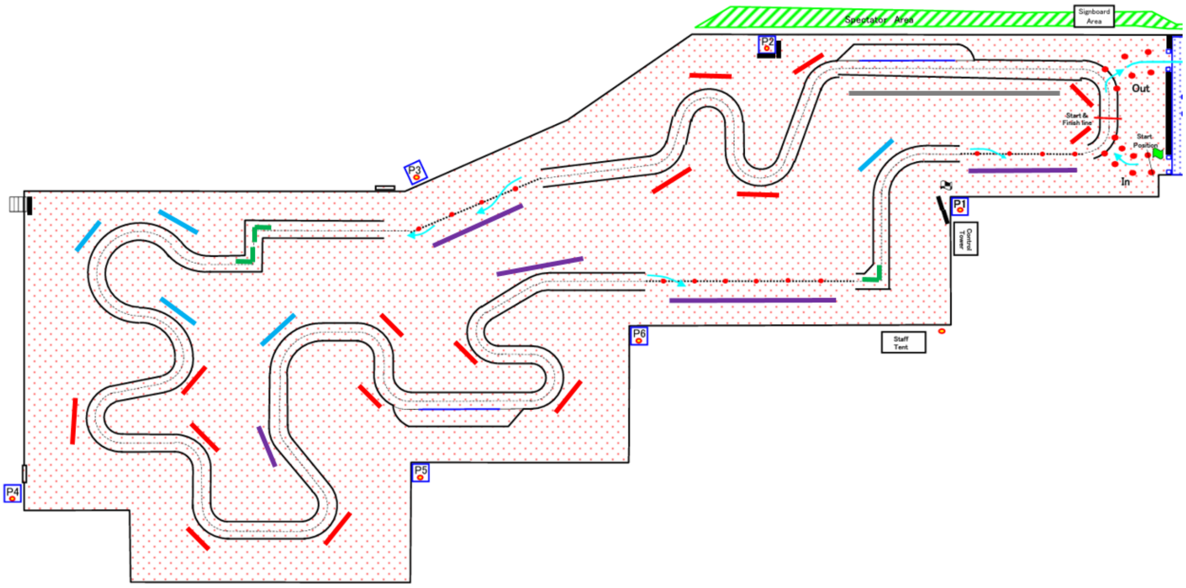

Figure 3. Division of track endurance conditions for fatigue analysis

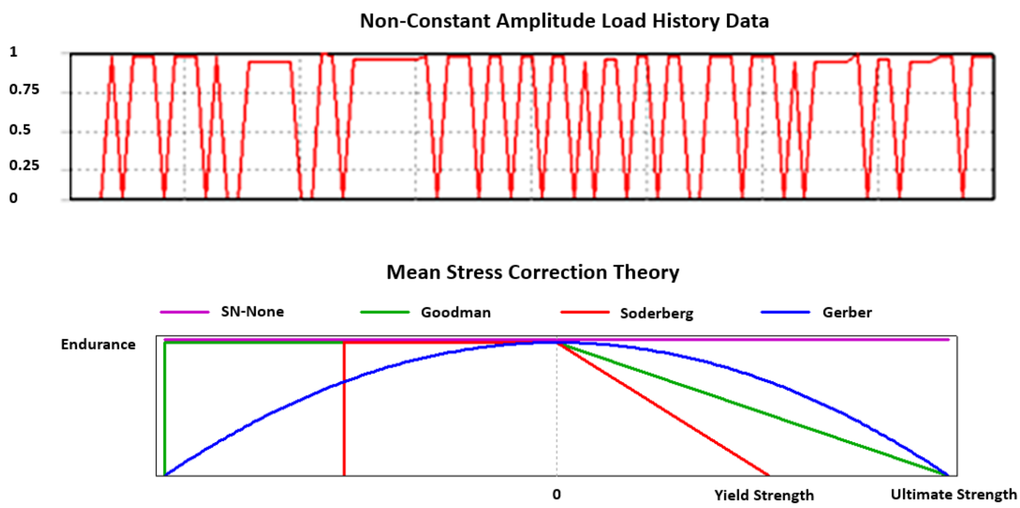

Figure 4. Graph of the history data track endurance event

Topology optimization is a structural process used to determine the optimal mass design of the racing car upright based on the stress distribution of the material used in the design space. The process carried out is to reduce the mass according to the boundary conditions applied and the design requirements. Before performing topology optimization, the constraints, such as the areas that need to be optimized, were first determined. The constraint is defined as a solid volume maintained from the optimization process to reduce the mass. Some parts of the front upright should not be removed because they are places for installing components such as the brake caliper, steering arm, wheel axle, and A-arm. Therefore, the constraint parts should be separated at the modeling stage.

The procedure for carrying out fatigue life simulation and topology optimization in designing the racing car upright consists of several stages. The first stage is fulfilling the over-design condition from the existing design indicated by fatigue life and safety factor values. The second is the optimization process for several variations of mass reduction on the front and rear upright. Furthermore, the final stage is smoothing the optimization results, especially in the critical area of the upright. The fourth is a static simulation to calculate the occurring stress with a limit that is not higher than the stress in the existing design. Finally, the fifth state calculates mass reduction and fatigue life from the final design to ensure that the simulated upright remains safe.

2.3. *Fatigue Life of Existing Design*

The fatigue simulation results of the existing design for the front and rear upright when the racing car is braking on straight and turning tracks are shown in Figure 5. This process uses the mean stress correction theory from Goodman, Gerber, and Soderberg to simulate a fatigue life of 100 million cycles, which occurs for all loading conditions. In all mean stress correction theory, the safety factor value when a racing car experiences braking and turning conditions is always smaller than when it goes straight. When turning, there will be an addition of centrifugal force on the front and rear upright with safety factor values of 1.9 and 2.55 compared to the braking and straight values of 2.16 and 2.76 for Goodman's theory.

Table 2 shows the strength simulation results of the racing car when receiving a static load, similar to the fatigue life simulation process. It shows the stress and deformation values due to static loads on straight and turning braking track conditions for the front and rear upright. In the front upright, the maximum stress value occurs in braking and turning conditions where the maxim-

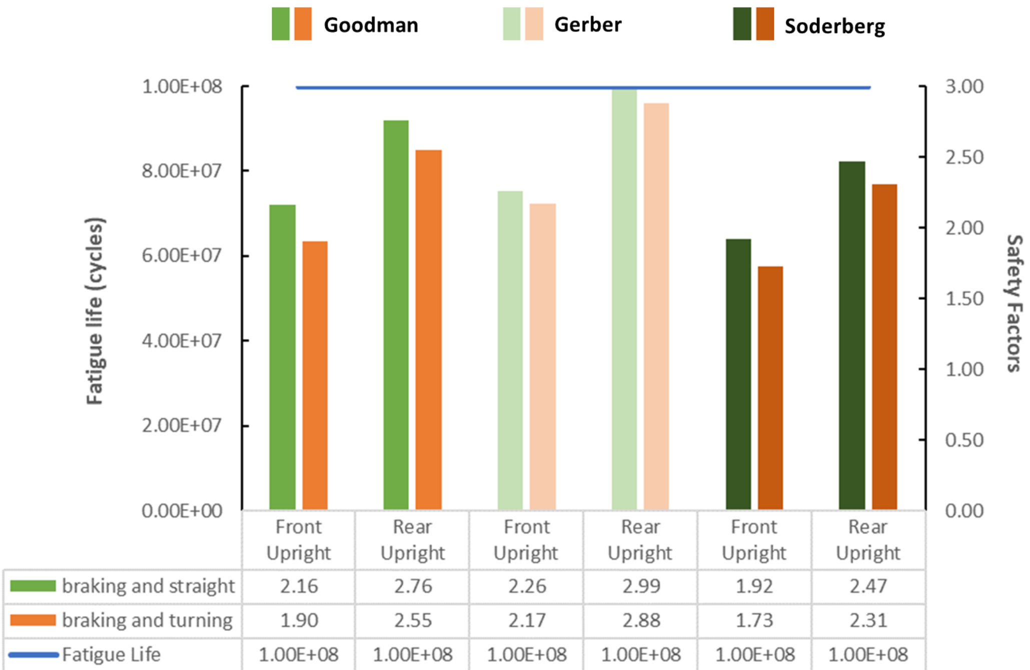

Figure 5. Fatigue simulation results for designing the front and rear upright

Table 2. Existing design simulation results in the form of stress and deformation

	Front Upright		Rear Upright	
	Braking and Straight	**Braking and Turning**	**Braking and Straight**	**Braking and Turning**
Maximum Equivalent Stress (MPa)	183.91	245.15	155.6	184.11
Maximum Deformation (mm)	0.22	0.37	0.19	0.22

-um equivalent stress and deformation are 245.15 MPa and 0.37 mm, respectively. Meanwhile, the maximum equivalent stress and deformation for the rear upright are 184.11 MPa and 0.22 under braking and turning conditions. The material used to design the front and rear upright is aluminum 7075 with a yield strength of 503 MPa. Therefore, the existing design is still in a safe condition, and the topology optimization process for mass reduction can be conducted.

The fatigue life and strength analysis results showed that extreme conditions occur when the car brakes while turning. This is because, during this process, the fatigue safety factor is smaller, with more significant maximum equivalent stress compared to straight road conditions. Therefore, the topology optimization needs to be based on the braking and turning conditions because if these extreme cases are safe, the other conditions will be free from harm. Furthermore, due to the additional load from steering connected to the front upright, the front upright has a smaller safety factor and more significant maximum stress than the rear upright. Therefore, the mass of topology optimization in the front upright is supposedly more significant than in the rear upright.

3. Results and Discussion

3.1. Topology Optimization in Front and Rear Upright

The finite element simulation results showed that the maximum stress value is lower than the allowable strength of the material in the existing upright design. The fatigue simulation is also safe in the existing design because it is greater than the minimum age limit of the material. Therefore, topology optimization needs to be reduced in the weight of the upright. **Figure 6** shows the topological optimization process for the FR30 design on the front upright consisting of meshing, determination of design region, optimization result, final design image, and equivalent stress and deformation contours. The implementation of topology optimization on the existing design is carried out by setting the purple area as a design region that can be optimized. This process results in a final design with maximum stress and deformation values of 208.24 MPa and 0.45 mm.

Table 3 shows the maximum equivalent stress values for the front and rear upright designs from the topology optimization results, which are all smaller than the existing design. However, the topology optimization process is discontinued when getting a maximum stress value more significant than the existing one. **Figure 7** shows that the maximum stress value in the FR50 design of 225.1 MPa is smaller than the existing design. The same thing also happens to the rear upright RR50 with a stress value of 164.22 MPa, which is smaller than the existing 184.11 MPa, as shown in **Figure 8**.

The larger fillet radius in the upright design due to topology optimization causes a decrease in stress concentration in the fillet area, thereby reducing the maximum stress value. Furthermore,

Table 3. Maximum equivalent stress of front and rear upright for various optimized designs

Types		Maximum Equivalent Stress (MPa)
Front Upright	Existing	245.15
	FR30	208.24
	FR40	212.03
	FR50	225.1
	FR60	235.99
Rear Upright	Existing	184.11
	RR50	164.22
	RR60	178.84
	RR70	238

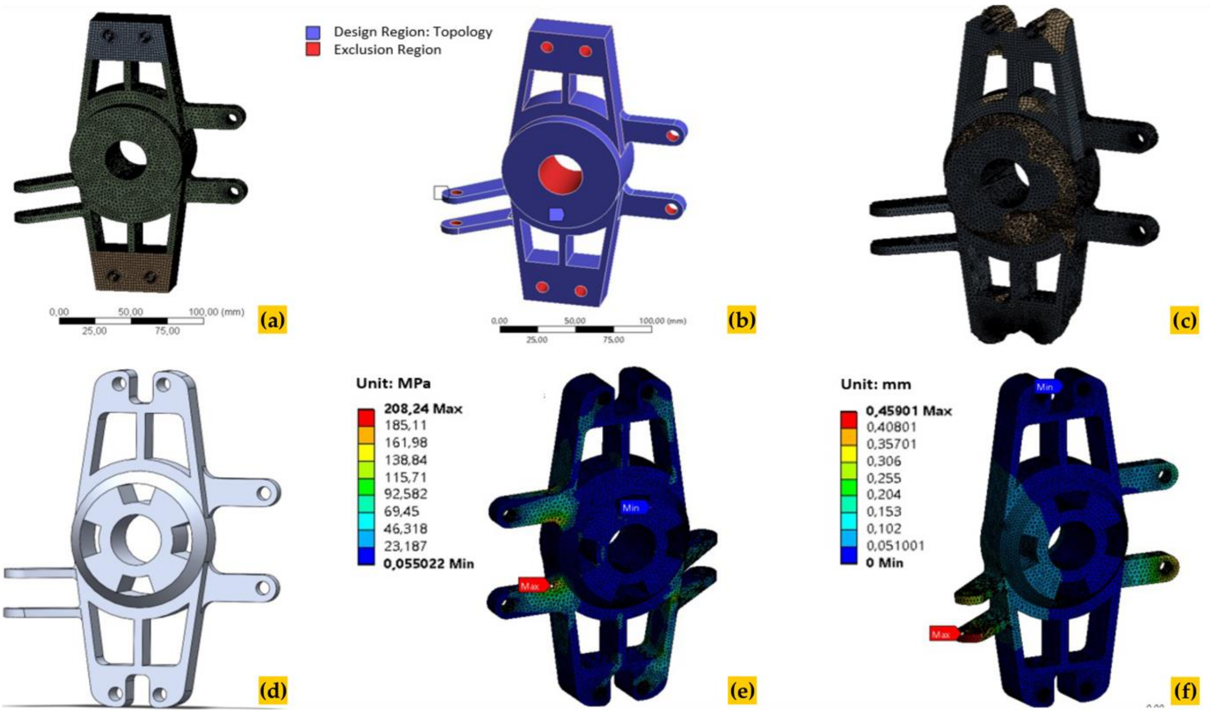

Figure 6. Topology optimization process of the front upright with a mass reduction of 30% (mass retain of 70%): (a) Meshing; (b) Design & exclusion region; (c) Topology result; (d) Final design; (e) Equivalent stress contour; (f) Total deformation contour

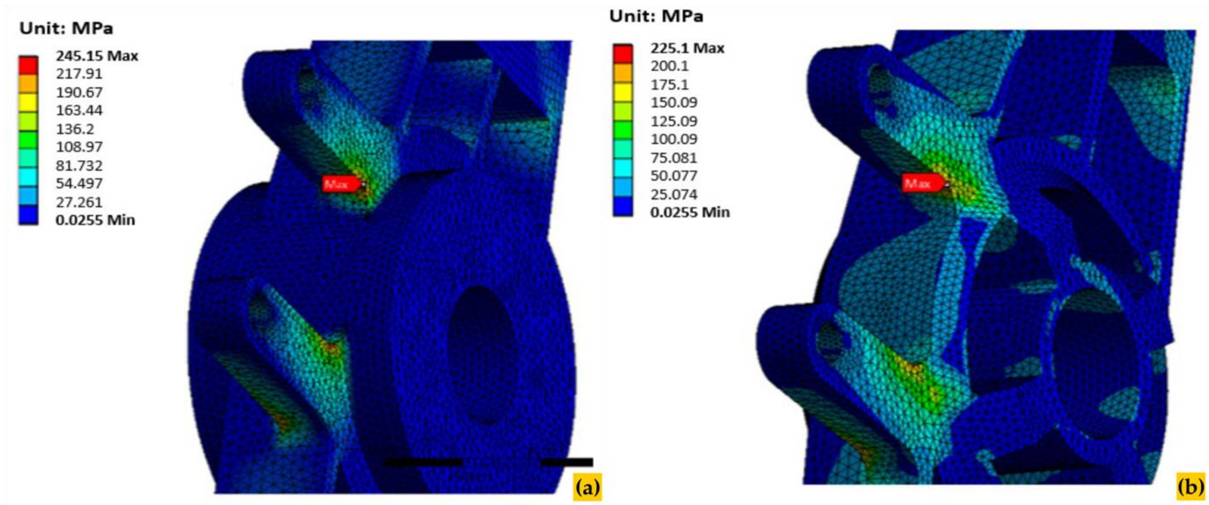

Figure 7. Equivalent stress contour of front upright for (a) existing product and (b) FR50 design

during the topology optimization process, the radius of the fillet can be designed to be larger due to the reduced outer diameter of the main body on the rear upright. Therefore, it illustrates the advantages of topology optimization, which reduces the maximum stress value through a smoother design.

Figure 9 shows the mass reduction process in topology optimization. The existing design of a solid front upright around the center hole of the wheel hub connection experiences more mass reduction from the design of FR40 to FR60. The rear upright does not experience any mass reduction in the center hole of the wheel hub connection because it is relatively more minor. This is in contrast to the conditions of the existing design of the RR60. Therefore, the mass reduction due to the topology optimization process occurs more in the rear and front upright upper and lower bracket areas. It means that the mass reduction from the topology optimization results from the rear upright is smaller than the front upright.

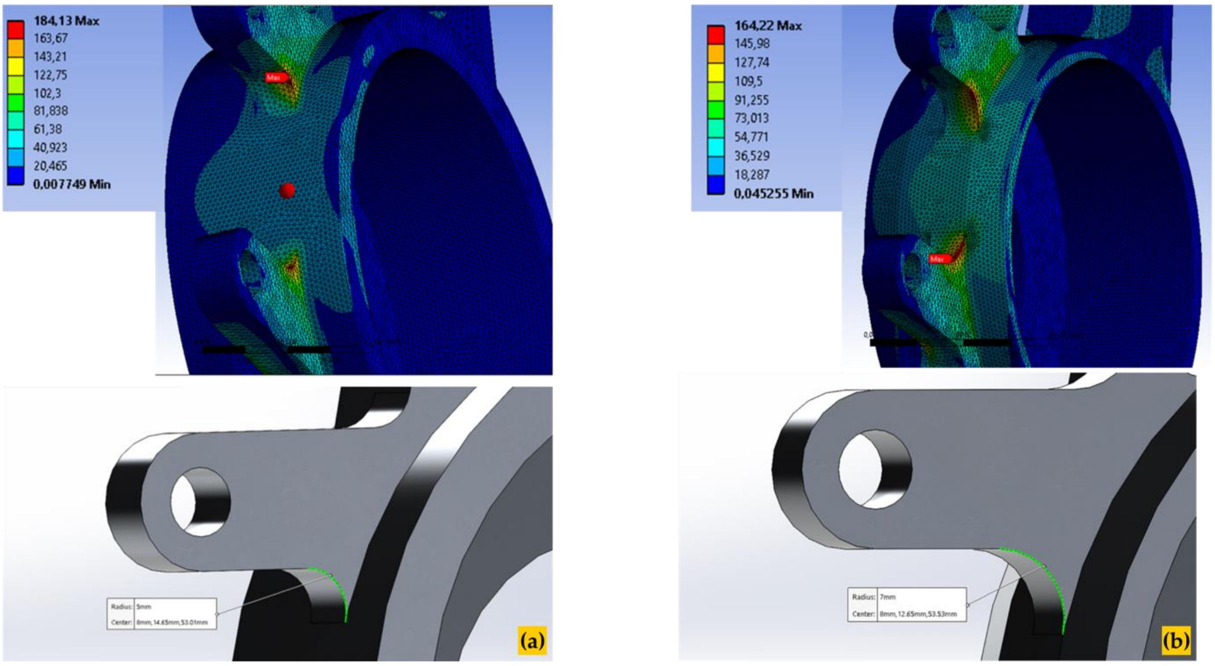

Figure 8. Equivalent stress contour of rear upright for: (a) existing product and (b) FR50 design

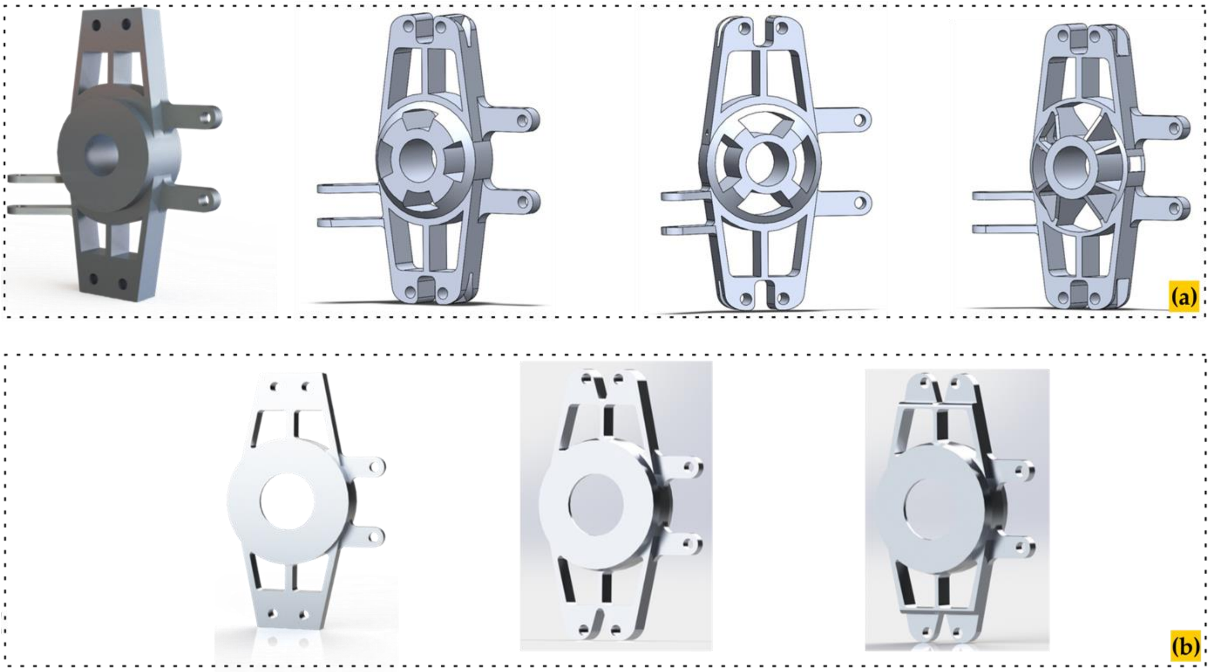

Figure 9. Results of mass reduction in topology optimization for: (a) the front upright, (b) the rear upright

3.2. *Mass Reduction of Optimum Design*

The front and rear upright mass of the existing design is 944.39 grams and 451.47 grams, respectively. **Table 4** shows the optimum mass of the fatigue life simulation and topology of several areas. The front upright's optimum final mass experiences a more significant mass reduction of 56% compared to the rear upright's mass reduction of 34%. This contributes to the required increase in energy efficiency. **Figure 10** shows the graph of fatigue life and mass reduction for several optimized design models for the front and rear upright. When the mass decreases, the fatigue life value also reduces and is still above the design life of the Aluminum series 7 materials used in the upright. Therefore, it can be concluded that the topology optimization results from this upright component are safe against fatigue and static strength.

Table 4. Mass of topology optimization results

	Existing Design	Optimum Design	Mass Reduction
The total mass of front upright (gram)	944.39	416.43	56 %
The total mass of rear upright (gram)	451.47	295.97	34 %

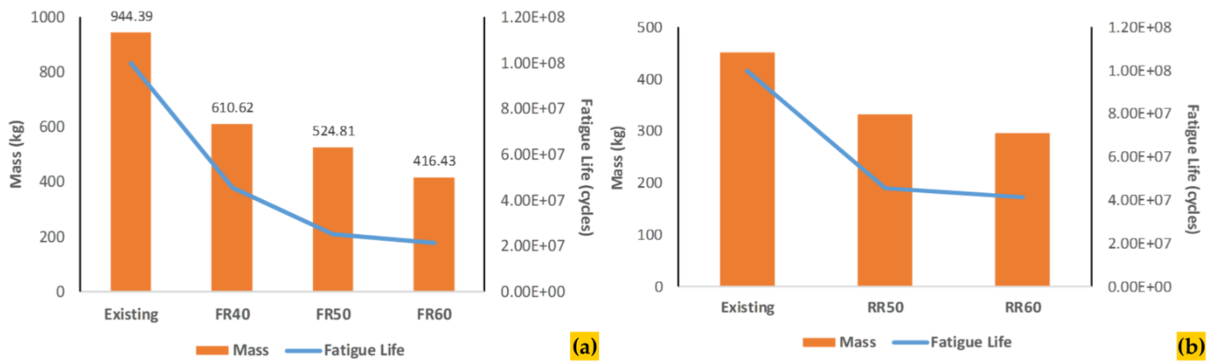

Figure 10. Fatigue life and mass reduction for: (a) the front upright; (b) the rear upright

This research has numerous advantages compared to other studies on topology optimization, which were limited to static loads. Some current research has integrated topology optimization processes when exposed to fatigue loads but is still limited to simple geometry. Therefore, the approach proposed in this research is relevant enough to ensure that the upright from topology optimization results are safe against fatigue loads.

4. Conclusion

In conclusion, this study achieved a 56% mass reduction of a racing car front upright made from Aluminum Alloy 7075-T6, decreasing from 944.39 grams to 416.43 grams. The topology optimization process, which included increasing the fillet radius, reduced the maximum stress values. The FR50 design exhibited a maximum stress of 225.1 MPa, lower than the existing design. Similarly, the rear upright RR50 design showed a maximum stress of 164.22 MPa, lower than the existing 184.11 MPa. The final design showed a reduction in stress concentration and achieved a fatigue life above the design life. These improvements highlight the effectiveness of topology optimization in achieving substantial weight reduction in racing car components.

Author's Declaration

Authors' contributions and responsibilities

The authors made substantial contributions to the conception and design of the study. The authors took responsibility for data analysis, interpretation and discussion of results. The authors read and approved the final manuscript.

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Availability of data and materials

All data are available from the authors.

Competing interests

The authors declare no competing interest.

Additional information

No additional information from the authors.

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