

Structural Optimization of a Formula Student Upright Using the BESO Method: Enhancing Performance Through Lightweight Design

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Abstract

This study presents a comprehensive design and optimization approach for the upright component used in Formula Student vehicles, with the primary objective of reducing weight while ensuring structural integrity and manufacturability. The upright, as a critical suspension component, must withstand complex multi-axial loads during braking, cornering, and acceleration. To address these challenges, a combination of advanced engineering tools and methods was utilized. Using SolidWorks for initial CAD modeling and ANSYS Workbench 2023 for finite element analysis (FEA), realistic loading scenarios were simulated based on dynamic vehicle conditions. These included vertical, lateral, and longitudinal forces derived from vehicle dynamics calculations. The mechanical behavior of 7075-T6 aluminum alloy was fully integrated into the simulation environment. A Bidirectional Evolutionary Structural Optimization (BESO) technique was applied to minimize structural mass while preserving stiffness and strength. The topology optimization was carried out within a defined design space, using a mass retention constraint ranging from 30% to 70%. Convergence monitoring and mesh sensitivity analyses ensured accurate and stable optimization outputs. Post-optimization validation included both static structural analysis and fatigue life assessment. The final design demonstrated significant improvements: a 58.75% weight reduction per upright, stress levels well below the material's yield limit, and fatigue life exceeding 10^8 cycles, indicating infinite operational life. These results underline the effectiveness of topology optimization, not only in achieving lightweight designs but also in improving dynamic performance, energy efficiency, and manufacturability in high-performance motorsport applications.

Keywords: BESO; Fatigue life; Finite element analysis; Formula student; Topology optimization; Upright design; Weight reduction

Research Article

History

Received	23.06.2025
Revised	15.09.2025
Accepted	19.11.2025

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To cite this paper: Sandalcılar, H., Çağırankaya, F., Arslanoğlu, M., Dikici, B. Structural Optimization of a Formula Student Upright Using the BESO Method: Enhancing Performance Through Lightweight Design. International Journal of Automotive Science and Technology. 2025; 9 (4): 536-546. <https://doi.org/10.30939/ijastech..1724606>

1. Introduction

In the field of motorsports and high-performance automotive engineering, the development of custom-designed vehicle components plays a crucial role in enhancing overall efficiency, safety, and performance [1]. Unlike mass-produced automotive parts, which are often designed with broad applicability and cost-effectiveness in mind, specialized components for racing vehicles prioritize lightweight structures, high strength-to-weight ratios, and optimized geometries tailored to specific racing conditions [2]. As a result, engineers continuously explore advanced manufacturing techniques, innovative

material selections, and optimization methods to refine the performance of these custom-built vehicles [3].

Custom manufacturing of automotive parts also plays a crucial role in enhancing vehicle performance, safety, and aesthetics. This process goes beyond standard mass-produced components, allowing manufacturers and vehicle owners to design and produce parts tailored to specific requirements and preferences. Customized parts enable vehicles to meet particular performance criteria, accommodate personal tastes, and integrate new technologies seamlessly [4]. Additionally, they are essential for the restoration and maintenance of classic or rare vehicles, where sourcing original components can be challenging. Ultimately, custom automotive parts

manufacturing fosters innovation in the automotive industry while enhancing the overall user experience [5].

Topology optimization techniques have been widely explored in structural design to achieve lightweight and high-performance components [6],[7]. For instance, Tang *et al.* [8] developed a bidirectional evolutionary structural optimization (BESO)-based approach for optimizing lattice structures fabricated via additive manufacturing, demonstrating significant weight reduction while maintaining structural integrity. Although their work focuses on lattice frameworks, similar optimization principles can be applied to solid components, such as the upright in Formula Student vehicles, to enhance performance through efficient material distribution. A comprehensive overview of topology optimization in the automotive industry was presented by Kahraman and Küçük **Hata! Başvuru kaynağı bulunamadı.**, emphasizing its role in reducing vehicle weight and improving structural efficiency.

In addition to its significance in the commercial and restoration sectors, custom automotive manufacturing plays a pivotal role in motorsports, where performance optimization is paramount [10]. Among various student-led engineering competitions, Formula Student has emerged as a prestigious platform for testing and showcasing innovative automotive design and manufacturing strategies [11]. This competition challenges university teams to design, build, and race formula-style cars while applying fundamental principles of mechanical engineering, aerodynamics, materials science, and vehicle dynamics. Given the constraints imposed by race regulations and performance objectives, achieving an optimal balance between structural integrity and weight reduction becomes a critical design challenge [12],[13]. To address this challenge, engineers employ advanced design methodologies, with topology optimization emerging as a key approach to achieve the desired balance between strength and weight [14].

One of the essential structural components in a Formula Student vehicle is the upright, which serves as the primary load-bearing element connecting the wheel assembly to the chassis [15]–[17]. This component must endure substantial forces from acceleration, braking, and cornering while retaining a lightweight and rigid structure **Hata! Başvuru kaynağı bulunamadı.**[19]. Any reduction in the mass of the upright directly contributes to improvements in the vehicle's handling, acceleration, and overall efficiency. Therefore, topology optimization techniques have become an essential tool in designing high-performance uprights by redistributing material in a way that maximizes strength while minimizing weight [20].

Hasan *et al.* [20] conducted a study on the lightweight design and analysis of the steering knuckle in a Formula Student car using topology optimization. Their research aimed to optimize the component's weight while maintaining its structural integrity and performance. The study considered three different materials (structural steel, 4130 steel, and AISI 1020 steel). The research applied topology optimization with two mass reduction constraints (40% and 48%), with 4130 steel achieving a

significant 42.70% weight reduction while ensuring mechanical strength and durability. The optimized design was analyzed using finite element analysis (FEA) in ANSYS Workbench. The study demonstrated the importance of material selection and topology optimization in developing high-performance, lightweight components for Formula Student vehicles.

Topology optimization has been increasingly applied in critical automotive components such as suspension and steering parts. For instance, Mete and Başak **Hata! Başvuru kaynağı bulunamadı.** optimized an outer tie rod design using FEA and fatigue analysis, demonstrating significant weight reduction while maintaining performance. Sanguri *et al.* [22] examined the design, analysis, and topology optimization of a front upright for an electric solar vehicle. Their study involved determining the dynamic forces acting on the upright, followed by finite element analysis and optimization to reduce unsprung mass while maintaining structural strength. Using Altair Inspire 2020, they performed simulations and iterative improvements to lower the upright's weight without compromising its ability to withstand braking, cornering, and bump forces. The optimized upright was made from Al 6061-T6, chosen for its balance of strength, weight, and ease of manufacturing. The final design led to a 35% reduction in weight, enhancing vehicle performance and efficiency. Their findings highlight the role of topology optimization in developing lightweight and high-performance vehicle components. Samin *et al.* **Hata! Başvuru kaynağı bulunamadı.** conducted a structural optimization of a Formula SAE upright by comparing Al 6061-T6 and Ti-6Al-4V alloys, achieving notable weight reduction and improved safety factors, which highlights the importance of material selection in upright design.

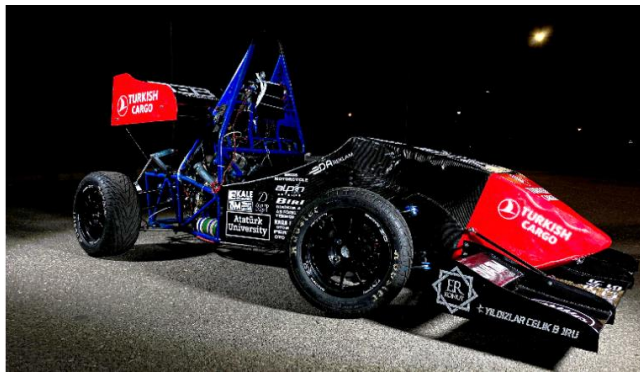
In a study by Gürses [24] on a suspension control arm, a 14.71% weight reduction was achieved through topology optimization followed by shape optimization using the Geyser Optimization Algorithm (GIO). While that study provided insights into the effects and progression of different algorithms, the current study adopts the BESO method. In contrast to stochastic methods, BESO utilizes a deterministic, sensitivity-based process that operates on the principle of element addition and removal.

Yang and Chahande [25] explored topology optimization in automotive design, using MSC/NASTRAN to optimize a truck frame, deck lid, and space frame chassis through a density-based approach, achieving weight reduction while maintaining structural integrity. Similarly, Wu *et al.* [26] optimized an automotive engine bracket using RADIOSS and OptiStruct, reducing weight without compromising performance. Aulig *et al.* [27] introduced a multi-objective topology optimization framework for a vehicle control arm, balancing stiffness and energy absorption under static and crash loads. By comparing individual, sequential, and concurrent optimization approaches, their study found that scaled weighting yielded the best trade-off, allowing engineers to tailor structural efficiency based on performance needs.

The aim of this study was to optimize the upright component used in Formula Student vehicles, which serves as a load-bearing and connection element for the suspension and wheel assembly. The work focused on reducing weight while maintaining structural integrity, thereby improving handling, efficiency, and overall competitiveness. This research contributes by employing the BESO method for the structural optimization of a Formula Student upright, while integrating detailed structural and fatigue analyses to ensure reliability and performance. This study provides a comprehensive design framework for lightweight and high-performance 7075-T6 aluminum alloy uprights. This combined approach underscores both the practical feasibility and performance benefits of BESO-driven optimization in Formula Student race cars.

2. Methodology

The methodology of this study was developed to evaluate and enhance the structural performance of a Formula Student vehicle's upright component through topology optimization and weight reduction strategies. The upright is a critical load-bearing component of the suspension system, connecting the wheel assembly to the chassis and transferring loads during braking, acceleration, and cornering [16],[28]. The main goal was to achieve significant mass reduction while preserving the required mechanical integrity and performance under dynamic conditions typical of Formula Student competitions.



Total Weight: 350 kg (280 + 70)
Weight Distribution: 50% – 50%

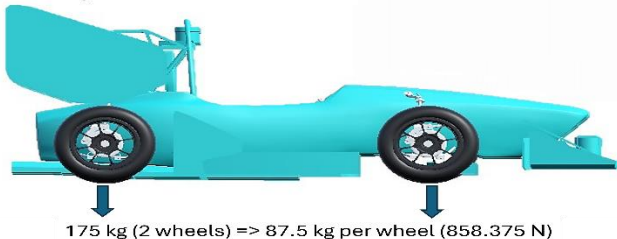


Figure 1. Schematic and actual view of the Formula Student vehicle

2.1. Geometrical Modeling and Baseline Analysis

Figure 1 illustrates the general configuration of the Formula Student vehicle used in this study. The upright component is

located within the suspension system, connecting the chassis to the wheel hub assembly. Figure 2 provides a detailed view of the rear upright and hub assembly, highlighting its structural features, including the bearing housing, brake caliper mount, and attachment points for the suspension and drivetrain components.

The initial 3D geometry of the upright was developed using SolidWorks 2023. This baseline model reflected the existing upright component used in the current vehicle configuration, with an individual part mass of approximately 2.45 kg. Since each vehicle utilizes four uprights, this configuration results in a total unsprung mass contribution of 9.8 kg. The baseline model served as a reference for subsequent optimization steps.



Figure 2. Close-up images of the rear upright assembly, showing the integration of the wheel hub, bearing housing, brake caliper, and suspension linkage interfaces.

2.2. Material Properties and Baseline Definition

In the current vehicle system, the upright component is manufactured from 7075-T6 aluminum alloy, a material known for its high strength, low density, and excellent machinability. As such, no material substitution was considered in this study. Al 7075-T6 has long been favored in motorsports and aerospace applications due to its superior mechanical properties and its ability to withstand high loads while minimizing weight. Therefore, the focus of this study was solely on achieving weight reduction through topology optimization, without altering the material itself. The mechanical properties of Al 7075-T6 used in the simulations are as follows: a yield strength of 503 MPa, an ultimate tensile strength of 572 MPa, and a density of $2.81 \text{ g}\cdot\text{cm}^{-3}$. Table 1 summarizes the mechanical properties of the 7075-T6 aluminum alloy employed in this study. These parameters served as the foundation for structural and topological analyses conducted throughout the study.

Table 1. Mechanical properties of 7075-T6 aluminum alloy used in this study

Property	Value
Yield strength	503 MPa
Ultimate tensile strength	572 MPa
Elastic modulus	71 GPa
Poisson's ratio	0.33
Density	$2.81 \text{ g}\cdot\text{cm}^{-3}$
Fatigue limit (10^8 cycles)	159 MPa

2.3. Load Conditions and Boundary Constraints

Realistic loading conditions were defined based on Formula Student vehicle dynamics, incorporating vertical loads, braking forces, cornering forces, and steering inputs. These loads were derived from vehicle geometry, weight distribution, and race track characteristics.

Boundary conditions were applied in ANSYS Workbench 2023 to simulate fixture points, suspension mounting locations, and wheel hub interactions. These constraints ensured accurate representation of operational stresses and deformations. The loading conditions were determined based on standard Formula Student vehicle parameters. Table 2 summarizes the key geometric and mass-related properties of the vehicle, which served as input values for calculating dynamic forces acting on the upright component under various maneuvering conditions.

Based on the vehicle specifications in Table 2, dynamic loading conditions, including braking, cornering, and vertical loads, were computed. These forces were applied to the upright using ANSYS Workbench to simulate real driving conditions. The load values were derived using classical vehicle dynamics equations and distributed across defined boundary surfaces.

Table 2. Technical specifications of the Formula Student vehicle used for load calculations

Parameter	Value
Wheelbase (mm)	1675
Center of Gravity Height (mm)	265
Front Track Width (mm)	1200
Rare Track Width (mm)	1300
Vehicle Mass (kg)	350
Front Weight Distribution	50%
Front Weight Distribution	50%
Tire Diameter (mm)	550

To model dynamic loading scenarios such as braking, cornering, and longitudinal acceleration, the parameters listed in Table 3 were employed. These values were based on performance data and assumptions derived from standard Formula Student race conditions.

Based on the dynamic vehicle conditions and geometry, the resultant forces acting on the upright were calculated. Table 4 presents the detailed force components and reaction loads used as input for the FEA simulations.

In addition to braking and longitudinal forces, lateral loads were also calculated for cornering scenarios. These forces were used to simulate the worst-case multi-axial loading on the upright. The associated values are summarized in Table 5.

Table 3. Dynamic conditions used for loading scenario definitions.

Condition	Parameter	Value
Braking Phase	Speed (u)	100 km/h (27.7 m·s ⁻¹)
	Braking Distance (s)	13.85 m
	Deceleration (a)	5.54 m·s ⁻² (1.3 g)
Cornering Phase	Braking Efficiency	56.5%
	Speed (v)	36 km·h ⁻¹ (10 m·s ⁻¹)
	Lateral Acceleration (a)	13.5 m·s ⁻² (1.5 g)
Longitudinal Acceleration	Final Speed	100 km·h ⁻¹
	Time	5.5 s
	Acceleration	5.04 m·s ⁻² (1.85 g)

Table 4. Calculated loading conditions applied to the upright during braking.

Parameter	Front Corner	Rear Corner
Friction Coefficient	2.31	2.59
Longitudinal Load Transfer (per tire)	67.79 kg	-67.79 kg
Vertical Load (per tire)	858.37 N	858.37 N
Friction Force	2157.05 N	1388.87 N
Braking Force (vertical)	3538.9 N	3538.9 N
Caliper Bolt Diameter	0.17 m	0.17 m
Distance: Hub to Lower A-Arm	0.090 m	0.120 m
Distance: Hub to Upper A-Arm	0.094 m	0.095 m
Reaction @ Upper Clamp	-2015.20 N	-927.82 N
Reaction @ Lower Clamp	4148.52 N	2317.64 N

Table 5. Calculated lateral loading conditions on the upright during cornering.

Parameter	Front Corner	Rear Corner
Lateral Load Transfer	87.5 kg	87.5 kg
Vertical Load	858.37 N	858.37 N
Friction Coefficient (μ)	2	2
Friction Force	2739.65 N	2739.65 N
Distance: Hub to Lower A-Arm	0.090 m	0.120 m
Distance: Hub to Upper A-Arm	0.094 m	0.095 m
Reaction @ Upper A-Arm	-2238.18 N	-1826.44 N
Reaction @ Lower A-Arm	4649.56 N	4566.09 N
Lateral Load Transfer	87.5 kg	87.5 kg
Vertical Load	858.37 N	858.37 N

The dynamic forces presented in Tables 2 to 5 were derived using classical mechanics and vehicle dynamics principles. The acceleration during each maneuver was assumed to be constant, and values such as deceleration (for braking), lateral acceleration (for cornering), and longitudinal acceleration (for speeding) were defined based on standard FSAE race car performance. Load transfer due to braking and cornering was calculated using the following formulas, respectively: $\Delta F = m(\Delta V / \Delta t)$ and $F = m(V^2 / r)$

where m is the vehicle mass, $\Delta V/\Delta t$ is the acceleration and r is the turning radius. Friction forces at the tires were obtained by multiplying the normal forces by the estimated friction coefficient (μ), based on tire specifications. These calculated forces were applied at specific nodes of the upright geometry in ANSYS Workbench to simulate realistic boundary conditions.

Braking forces were calculated using Newton's second law, ($F=m \cdot a$), based on deceleration from 100 km/h to 0 in 5 seconds, resulting in a total braking force of 1939 N per wheel. Similarly, lateral forces during cornering were computed using the centripetal force equation ($F=m \cdot v^2/r$), yielding approximately 3835.6 N for a turning radius of 9.125 meters at 36 km·h⁻¹. These updated force values were used to define boundary conditions in the FEA model.

2.4. Topology Optimization Procedure

The topology optimization process was conducted using the Bidirectional Evolutionary Structural Optimization (BESO) method, as highlighted in previous studies [8]. The BESO method was selected due to its capability to iteratively refine the material layout based on stress distribution, enabling lightweight yet structurally robust designs. The objective was to minimize structural mass while retaining essential load paths and ensuring compliance with stiffness and stress constraints. The optimization was carried out under a mass reduction constraint of up to 60%, with the design domain and non-design regions clearly defined to preserve functional interfaces. Mesh sensitivity analysis was conducted using mesh sizes ranging from 0.17 mm to 2.8 mm. Final mesh quality was selected based on convergence stability and computational efficiency.

The optimized design was then evaluated through structural validation using finite element analysis (FEA). FEA is a computer-based method that predicts how a design will react to real-world forces, vibrations, heat, and other physical effects. FEA allows for more complex designs that can be tested quickly, leading to better and safer products [29]. It also helps save time and money. Simulations were conducted to assess stress distribution, displacement, and factor of safety under all defined loading conditions. This step ensured that the topology-optimized geometry met the mechanical requirements for dynamic loading scenarios such as high-speed cornering and braking. Figure 3 shows the chart of the methodology adopted in this study

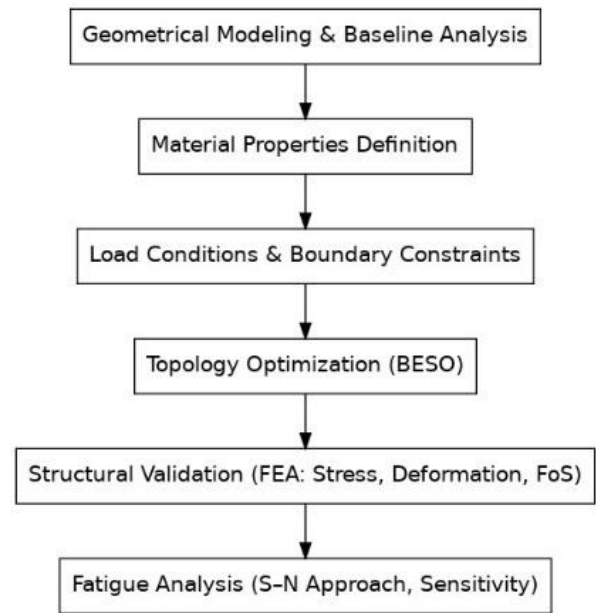


Figure 3. Flow chart of the methodology adopted in this study

3. Results and Discussion

The optimization process was carried out with the objective of enhancing the efficiency of the upright component by improving its structural performance while minimizing its weight [30]. In this context, prior to conducting the analysis, the forces acting on the upright were calculated using vehicle dynamics principles. These forces were then amplified with a safety factor and applied to the component in the finite element simulations, as illustrated in Figure 4.

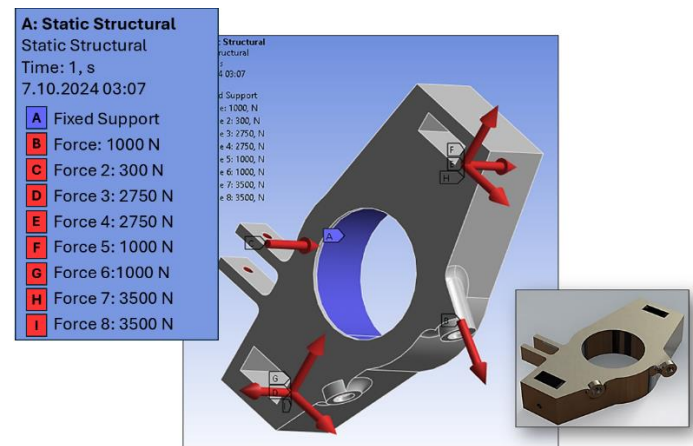


Figure 4. Applied load directions and locations on the upright under real dynamic conditions.

As seen in Figure 4, the upright is subjected to multiple forces transmitted through the suspension system, including those from the upper and lower wishbones, the axle hub via the bearing housing, the braking system through the caliper mount, and the fixing shaft. The upright is connected to the chassis through a bearing-mounted bolt located at the joints of the wishbone arms,

and the boundary conditions were defined accordingly based on these attachment points.

Despite some localized regions exhibiting slightly lower mesh quality, the overall meshing of the upright geometry was considered to be of high quality (Figure 5), which ensuring accuracy in the simulation results

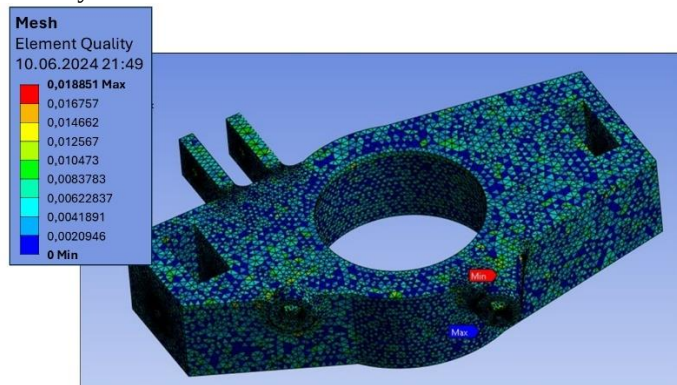


Figure 5. Finite element mesh structure of the upright with high overall mesh quality.

Following the application of the calculated loads, the total deformation was determined, with the maximum value measured at approximately 9 μm (Figure 6). This minimal deformation confirmed that the upright maintained sufficient stiffness under operating conditions.

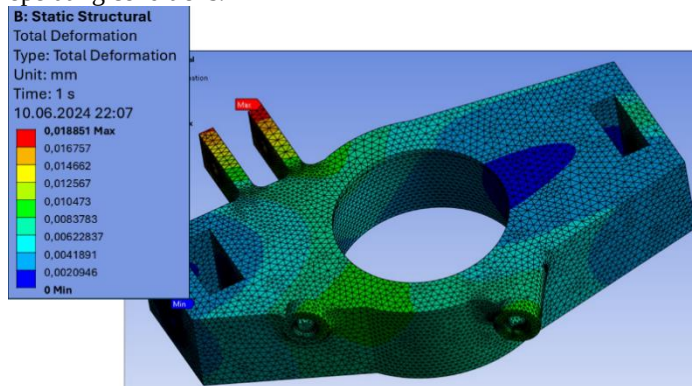


Figure 6. Total deformation distribution under applied forces, maximum deformation is approximately 9 μm .

The deformation and stress values obtained from the initial finite element analysis were used as input parameters for the topology optimization phase. The optimization setup, as illustrated in Table 6, was configured to ensure that the resulting design would maintain sufficient strength under applied loads while achieving significant weight reduction.

Following the established methodology, a weight reduction range between 30% and 70% was defined. The objective was to determine the optimal reduction percentage that would result in the most efficient material distribution without compromising mechanical performance.

Table 6. Topology optimization parameters used in the study.

Category	Parameter	Value
Scope	Scoping Method	Optimization Region
	Optimization Region Selection	Optimization Region
Definition	Type	Response Constraint
	Response	Mass
	Define By	Range
	Percent to Retain (Min)	30%
	Percent to Retain (Max)	70%
	Suppressed	No
	Maximum Number of Iterations	500
	Minimum Normalized Density	1.0×10^{-3}
	Convergence Accuracy	0.1%
	Penalty Factor (Stiffness)	3
	Region of Manufacturing Constraint	Include Exclusions
	Region of Minimum Member Size	Exclude Exclusions
	Region of AM Overhang Constraint	Exclude Exclusions
Solver Controls	Solver Type	Program Controlled
Output Controls	Store Results At	All Iteration

After setting the necessary parameters according to the desired output criteria, the topology optimization process was initiated. The resulting outputs were analyzed in the subsequent steps. The convergence behavior during the optimization iterations is presented in Figure 7, which demonstrates a steady decrease in the objective function value until convergence was achieved.

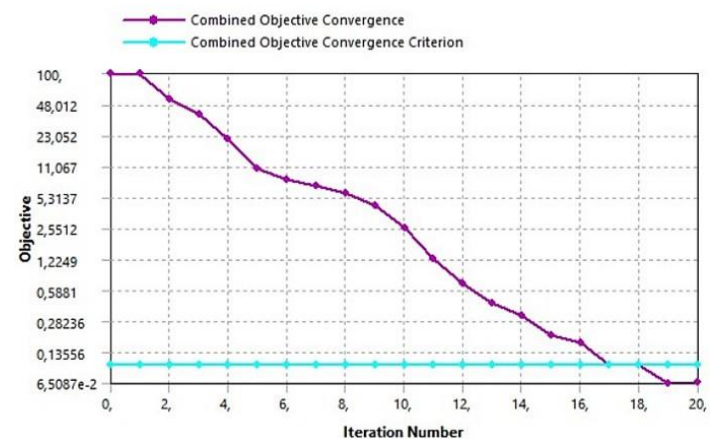


Figure 7. Combined objective convergence and convergence criterion over iterations.

Upon completion of the input definitions and boundary condition adjustments, the optimization was carried out with respect to the applied loads. This process generated a reference

geometry representing the most efficient structural layout. As shown in Figure 8, the resulting configuration served as a guiding design for the following stages of development.

While the final topology was primarily guided by mechanical efficiency, attention was also given to manufacturing feasibility. Design alterations, such as filleted edges, minimum member thickness, and simplification of complex contours, were introduced post-optimization to ensure CNC machinability [31]. These modifications contributed to the manufacturability of the component without compromising its structural performance.

Following the topology optimization process, the resulting geometry was further valuated in terms of manufacturability and structural feasibility. As observed in Figure 9, the design evolved through four stages, beginning with the initial full-mass model and progressing toward a highly efficient, lightweight structure. Each iteration removed unnecessary material while preserving critical load paths and connection points. To ensure that lightweight and high-performance structures resulting from topology optimization are manufacturable, it is essential to either integrate manufacturing constraints during the optimization process or apply post-optimization design modifications that enhance production feasibility [32].



Figure 8. Optimized geometry of the upright component after topology optimization.

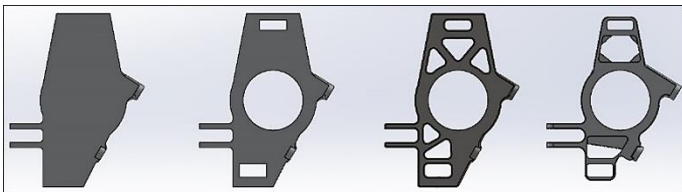


Figure 9. Evolution of the upright geometry during the optimization process. From left to right: (1) initial full-mass model, (2) intermediate mass reduction, (3) refined structure with optimized internal cutouts, and (4) final geometry revised for manufacturability.

To evaluate the mechanical integrity of the optimized upright under real-world conditions, a new round of structural analysis was conducted by applying the same loading conditions using the same loading conditions as in the initial simulation. The results of total

deformation for the updated design are presented in Figure 10. The maximum total deformation was found to be approximately 0.136 mm, which remains well within acceptable limits for high-performance applications.

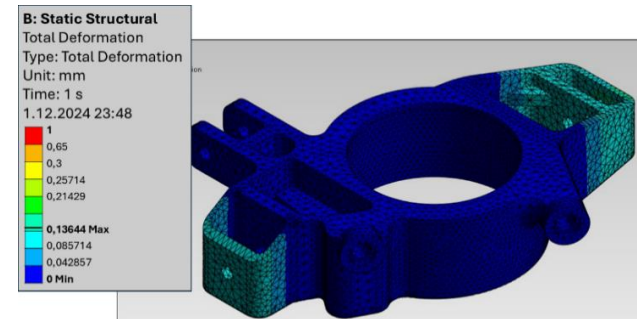


Figure 10. Total deformation contour of the optimized upright under real-world loading conditions.

The equivalent von Mises stress distribution, shown in Figure 11, indicates that the peak stress reached 107.52 MPa, which is significantly below the yield strength of the material (503 MPa for 7075-T6 aluminum alloy). This confirms that the optimized geometry retains sufficient structural rigidity under the applied multi-axial loading conditions. Engineers must consider these forces to ensure that components like the upright can withstand the stresses of racing while remaining lightweight and effective. This knowledge helps in creating safer and more efficient vehicles [33].

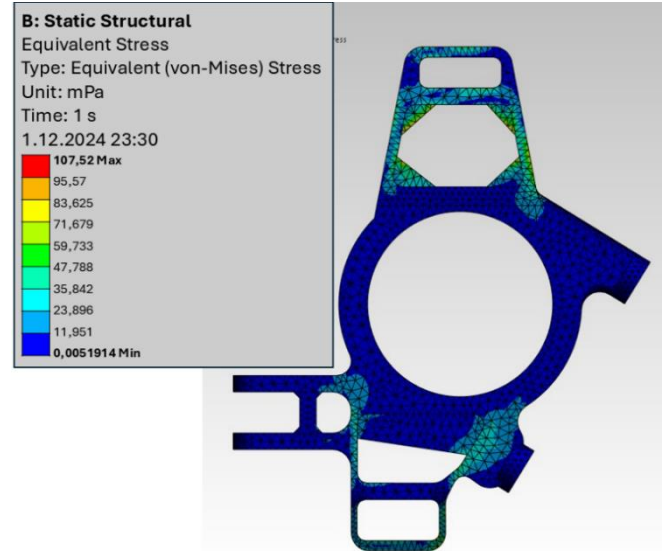


Figure 11. Equivalent von Mises stress distribution for the final design. Maximum stress remains safely below the material's yield strength.

The fatigue sensitivity illustrates the influence of load multipliers on the component's fatigue life (Figure 12). It is observed that the upright design exhibits infinite life when subjected to loads up to approximately 1.25 times the nominal design loads. Beyond this threshold, the fatigue life begins to decline significantly. For instance, at 1.5 times the load (equivalent to an additional ~1000 N in certain locations), the fatigue life drops

to approximately 1.07×10^5 cycles. This emphasizes the importance of maintaining loading within defined safety limits to ensure long-term reliability in dynamic conditions.

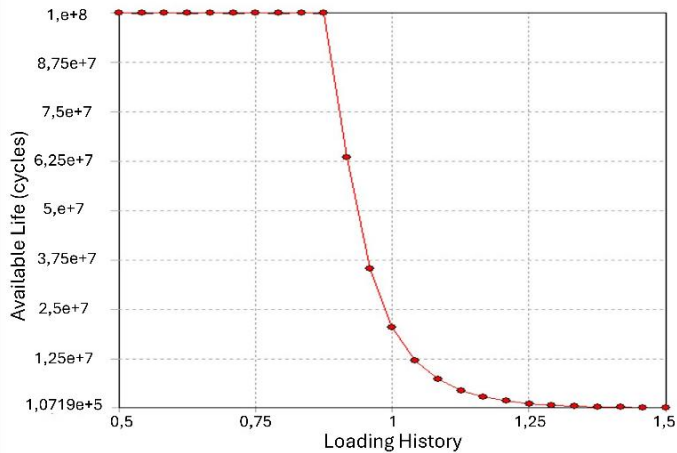


Figure 12. Fatigue sensitivity chart of the upright under varying load conditions.

The factor of safety distribution (Figure 13) clearly demonstrates that most regions of the upright maintain a FoS well above unity, indicating a conservative and robust design. While some localized areas, particularly those subjected to combined loads during acceleration, braking, and cornering, experience higher stress concentrations, they still remain within acceptable limits. The visualization confirms that the component is structurally capable of withstanding forces greater than those anticipated during typical Formula Student race conditions.

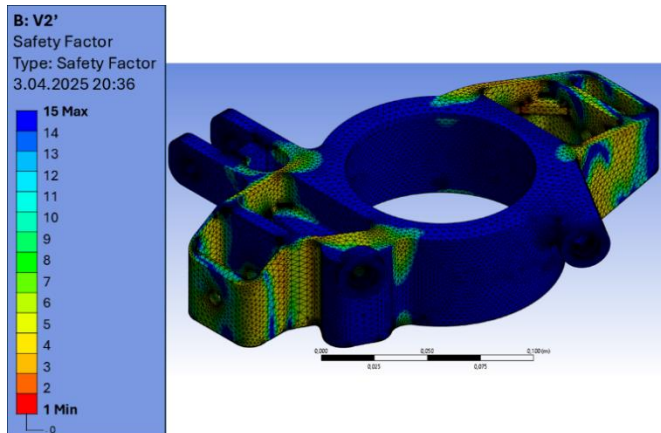


Figure 13. Factor of safety (FoS) distribution of the final upright geometry.

Fatigue performance was also assessed based on a stress-life (S-N) approach. As illustrated in Figure 14, the entire upright geometry demonstrated fatigue lives exceeding 108 cycles, which qualifies it as an infinite-life component under the specified conditions. Additionally, all stress values remained below the fatigue limit of the material, which is approximately 159 MPa for 7075-T6 aluminum alloy.

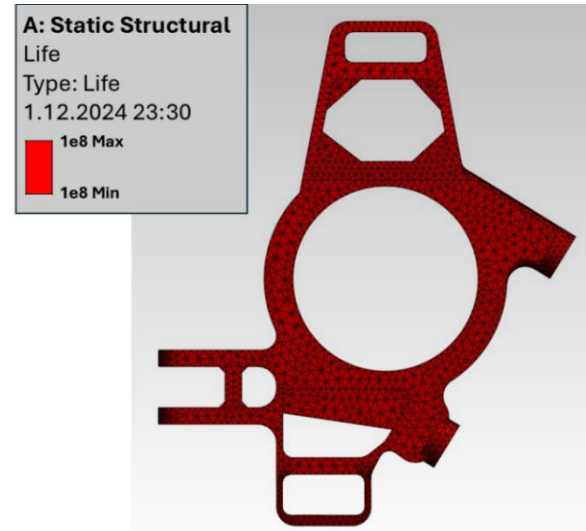


Figure 14. Fatigue life distribution of the final design, indicating infinite life behavior under specified loading cycles.

As a result of this design iteration, the mass of a single upright was reduced to 1.01 kg, compared to the pre-optimization weight of approximately 2.45 kg, corresponding to a weight reduction of 58.75%. This reduction leads to a total unsprung mass savings of 5.76 kg across all four uprights, significantly enhancing vehicle performance by lowering inertial forces and improving dynamic response. It is well known that in motorsport applications, reducing vehicle weight is paramount for enhancing performance metrics such as acceleration, cornering agility, and braking efficiency. The 58.75% weight reduction achieved per upright in this study contributes significantly to lowering the vehicle's unsprung mass, which in turn improves suspension responsiveness and tire contact with the road surface. By reducing the unsprung mass through weight optimization, the suspension system responds more effectively to road irregularities, maintaining better tire contact with the track surface. This improved contact increases lateral grip, allowing for higher cornering speeds and enhanced vehicle stability. Furthermore, a lighter upright decreases the overall inertia, leading to more efficient acceleration and deceleration, which is crucial in competitive racing environments where energy efficiency directly impacts performance. Studies have shown that reducing vehicle weight contributes to improved handling and reduced energy consumption, underscoring the importance of component optimization in motorsport applications [34]. Compared to similar studies such as [20],[22], this study achieved a greater weight reduction (58.75%) while maintaining structural and fatigue reliability. These enhancements are crucial for high-speed maneuvering and overall energy efficiency on the track. Studies have shown that optimal weight distribution and reduction can lead to better handling and increased safety in race cars [35].

The optimized upright not only reduces vehicle mass but also plays a role in enhancing cornering stability and reducing energy consumption in dynamic scenarios. By making the upright lighter, vehicles can achieve higher speeds, improved handling, and greater energy efficiency. This is especially important in racing, where every second counts. Understanding these benefits helps us appreciate the engineering behind high-performance vehicles [36]. Considering that uprights are unsprung masses, their reduction has a compounded effect on suspension response and tire-road contact stability, which are crucial in high-speed maneuvering situations commonly encountered in Formula Student races. To provide a clear comparison between the initial and optimized upright designs, Table 7 summarizes the key mechanical outcomes and performance metrics obtained through the finite element and fatigue analyses.

Table 7. Comparison of mechanical and performance parameters before and after topology optimization of the upright component.

Metric	Pre-Optimization	Post-Optimization
Max von Mises Stress	—	107.52 MPa
Max Deformation	—	0.136 mm
Fatigue Life	—	>10 ⁸ cycles (Infinite life)
Mass per Upright	2.45 kg	1.01 kg
Total Upright Mass (4x)	9.8 kg	4.04 kg
Total Mass Reduction	—	5.76 kg (58.75%)

Future studies may extend this work by integrating additive manufacturing (AM) constraints directly into the topology optimization process, enabling the production of even more complex and weight-efficient geometries [37]. Additionally, coupling structural optimization with aerodynamic analysis could further enhance the overall performance of Formula Student vehicles, particularly in high-speed cornering and drag-sensitive conditions. Finally, extended fatigue life validation under variable amplitude loading and thermo-mechanical coupling for aluminum alloys such as 7075-T6 aluminum alloy would provide deeper insights into real-world durability and long-term performance of the optimized components.

4. Conclusions

This study aimed to develop a lightweight yet structurally robust upright component for Formula Student vehicles through the application of topology optimization techniques, focusing on material efficiency, mechanical integrity, and manufacturability. The optimization process, conducted using the BESO method, successfully addressed the challenge of reducing unsprung mass without compromising performance, as confirmed by comprehensive structural, stress, and fatigue analyses. The results obtained from structural and fatigue analyses confirm the effectiveness of the topology optimization strategy employed in this study. The maximum stress remained significantly below the material's yield strength, and fatigue analysis indicated infinite life under typical loading conditions. Moreover, the

overall mass of each upright was reduced from 2.45 kg to 1.01 kg, achieving a substantial weight saving of 58.75%. This led to a total unsprung mass reduction of 5.76 kg across all uprights. Such weight reduction not only enhances handling and responsiveness but also contributes to energy efficiency and dynamic performance. Optimizing the wheel upright significantly enhances a Formula Student vehicle's cornering stability and reduces energy consumption during dynamic scenarios. Additionally, the use of 7075-T6 aluminum alloy allowed the final design to benefit from high strength-to-weight ratio without introducing new material processing challenges. These outcomes highlight the suitability of BESO-based topology optimization for high-performance motorsport applications, especially when combined with strategic material selection and manufacturability-focused design refinement.

Acknowledgment

This research was supported by The Scientific and Technological Research Council of Türkiye (TUBITAK) under the 2209-B – Industry Oriented Research Project Support Program for Undergraduate Students (Project No: 1139B412303851). The authors gratefully acknowledge TUBITAK for its financial support and encouragement of undergraduate research and innovation.

Conflict of Interest Statement

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Credit Author Statement

Hamit Sandalcılar: Conceptualization, Methodology, Formal analysis, Investigation, Visualization, Writing – original draft.,

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