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Designing a Lighter Formula Student Suspension Upright

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This public edition omits personal acknowledgements, supplied vehicle appendices, and private supplier correspondence.

Abstract

This project presents the design and optimization of a rear upright for a Formula Student UK (FSUK) electric vehicle, with the objective of minimizing weight while ensuring structural integrity and manufacturing feasibility. A methodical engineering design technique comprising load analysis, material selection, finite element analysis, and topology optimization was used to create the upright, a crucial suspension component. MATLAB computations identified critical forces based on dynamic load circumstances. Because of its advantageous strength-to-weight ratio, fatigue resistance, and machinability, aluminum 7075-T6 was chosen as the final material. In order to reduce mass while keeping the factor of safety above 2.0, six iterations were created. With a maximum von Mises stress of 104.54 MPa and deformation of 0.112 mm, the final version achieved a safety factor of 4.72 and a weight reduction of 20.01%. Despite the fact that this design optimized performance, a cost-benefit analysis found that iteration 5 was the most feasible because of its minimum performance tradeoff and reduced production cost. Because of its accuracy and viability, CNC machining was selected as the manufacturing technique. This study illustrates the importance of designing motorsport components for performance, manufacturing feasibility, and sustainability.

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List of Symbols

F	*Force*
g	*Acceleration caused by gravity*
a	*Acceleration*
z	*Wishbone length*
m	*Mass*
A	*Area*
L	*Length*
E	*Young modulus of elasticity*
I	*Second moment of area*
PI	*Performance index*
ρ	*Density*
δ	*Deflection*

List of Abbrreviations

FSUK	*Formula Student UK*
QMFS	*Queen Mary University of London Formula Student*
FEA	*Finite Element analysis*
SIMP	*Solid Isotropic Material with Penalization*
FOS	*Factor of Safety*
SLM	*Selective Laser Melting*

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1. Introduction

1.1 Background

In the always evolving engineering scene, the Formula Student UK (FSUK) competition is one of the top demonstrators of creativity and quality. Beyond its primary goal of designing and building formula-style vehicles, it forces students to challenge fundamental ideas of automotive performance and design[1]. Starting with a rather basic but important question: How can we design a vital component for a FSUK vehicle that integrates environmental responsibility with optimum performance? Understanding that sustainability will define mechanical engineering's future instead of only speed or design, the project aimed to investigate an environmentally friendly approach.

Founded as Formula SAE in the United States, IMechE brought the competition to the UK in 1998, allowing students to design, manufacture, and formula-style vehicles while also developing teamwork, manufacturing, and complex vehicle design techniques[2]. FSUK keeps changing as the industry moves toward more sustainable practices and modern technologies, including categories for electric vehicles, which is where the emphasis of this project rests. FSUK arms young engineers to meet the changing needs of motorsports innovation[2].

1.2 Queen Mary Formula Student

Figure 1.QMFS at Silverstone 2024 [3]

Living up to these values is Queen Mary Formula Student (QMFS), a team that takes part in FSUK annually. Leveraging the many skills of mechanical, electrical, and software engineering students, QMFS constantly develops in both vehicle design and race-day strategy[3]. Assessed in real-world racing conditions, ensuring theory translates into practice, QMFS has evolved to keep a careful balance of cost-effectiveness, engineering efficiency, and sustainability. For every team member, the combined effort fosters a vibrant culture of technical mastery, innovation, and professional development[3].

1.3 The Upright

From the powertrain to the chassis and suspension, each component of a FSUK vehicle must cooperate in the chase of speed, efficiency, and reliability. The upright is one of the most important structural links directly tying the wheels to the chassis as shown in figure 2. It houses the bearings and wheel hub, therefore allowing the wheel to turn[4]. The upright allows you to fasten the axle, tie rods, and suspension control arms, thereby improving the handling and suspension movement of the vehicle on road. It mounts the brake calipers and discs, supports wheel weights, transmits forces from the tire to the suspension links, making it crucial in handling and safety since it directly absorbs stresses from accelerating, braking, and cornering[4]. Under these difficult conditions, a well-engineered upright preserves stable suspension geometry, preserves cornering stiffness, increases responsiveness, and lowers the possibility of structural failures.

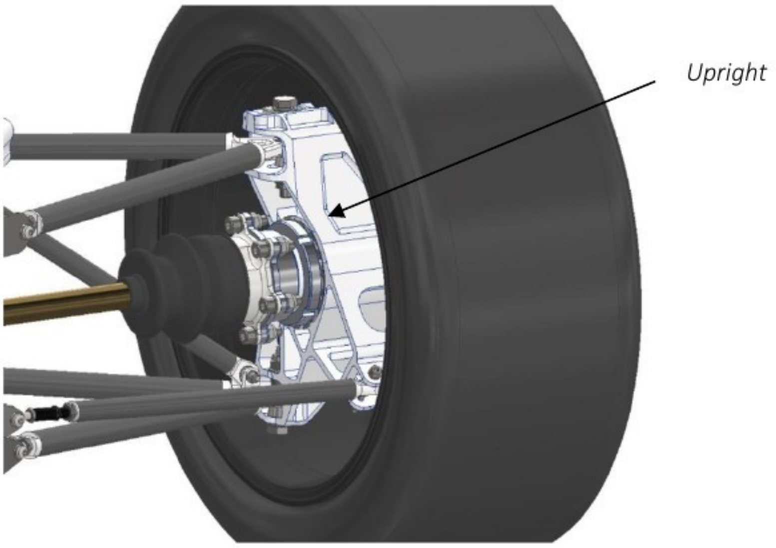

Figure 2. Upright assembly[5]

Camber, caster, and toe angles are controlled by suspension links and uprights. While rear wheels allow toe corrections [4], QMFS manages handling and tire wear by recognizing alignment angles, caster, camber, and toe. Positive casters tilt the axis backward for straight-line stability and centering [6]. Negative casters tilt it forward, reducing stability. Negative camber improves cornering grip but may cause uneven tire wear [6]. Toe measurement reveals tire tilt inward where, it straightens tracking but destroys outside tire or outward improving steering but decreases stability [5].

1.4 Aims and Objectives

Aims

- Design a lightweight and high-strength upright for a FSUK vehicle that enhances overall vehicle performance to reduce the unsprung mass.
- Optimize the upright design for effective force distribution, minimizing stress concentrations under loads such as braking, cornering, and acceleration.
- Ensure the upright meets FSUK safety standards while balancing performance and manufacturing feasibility.

Objectives

- Ensure the upright can withstand vertical, lateral, and longitudinal forces without failure. Perform Finite Element Analysis (FEA) to ensure the design can handle stresses under peak conditions.
- Select materials that offer a balance between strength, durability, fatigue resistance and weight optimization.
- Optimize the design geometry to allow for proper suspension mounting, ensuring adjustable suspension setup for different track conditions and fits within allocated space.
- Ensure the design is suitable for cost-effective manufacturing processes while keeping in mind the competition budget constraints.

1.5 What's to come

Starting with a literature review of past studies assists with development. Design requirements and constraints include performance goals, dimensional constraints, and competition regulations. The upright forces are found in load situations and computations. Material considers strength, weight, fatigue resistance, and sustainability. CAD Design and Geometry provides a complete 3D model with suspension integration. FEA assesses peak load structural performance, whereas topology optimization reduces weight without compromising safety. The production plan evaluates viable manufacturing methods within a budget. After safety and environmental considerations follows. Discussion contextualizes the results. Conclusions and future work identify accomplishments and recommend improvements.

2. Literature Review

2.1-Geometry

Suspension kinematics, load transfer, and general handling qualities all depend directly on the uprights shape. Chauhan et al. underlined the need of precise geometric modeling for uprights in FSUK vehicles, guaranting correct placement of pickup points and to reduce rear toe compliance[7]. It showed that effective suspension work relied critically on the alignment of the upright with other components, including calipers and control arms.

Underlining the need of perfecting upright shape depending on the position of important connection points and structural feedback from simulation data, Khalis et al. strengthened this view[8]. Their design approach underlined the need of iterating between computer aided design (CAD) and FEA technologies to create an upright that satisfies stiffness, packing, and strength criteria[8]. Their results underline how straight geometry has to assist manufacture and serviceability in addition to load distribution.

2.2-Material selection

The weight, stiffness, fatigue life, and manufacturing ease of use of an upright depend much on the materials used. Since the upright is a component of the unsprung mass and its weight must be lowered without sacrificing strength. Because of its excellent mechanical performance and great strength to weight ratio, Bhanderi et al. chose aluminum 7076-T6 for their design[9]. Material mass reduction improved vehicle handling, validated by FEA.

After comparing aluminum 7075-T6, 6061-T6, and Alumec, Wheatley and Popoola found that 7075-T6 was the best appropriate for racing uses because of its high yield strength and fatigue resistance[9]. This alloy provided the optimum trade-off between performance and weight, according to their study, hence it was perfect for parts under constant, high-load conditions like in uprights.

In contrast, Sindhukavi et al. selected mild steel for their upright because to machining practicality and financial constraints[11]. Mild steel was fit, as heat treatment and specialist fabrication tools was limited. Despite its higher density, it offered superb machinability and enough fatigue resistance. Sindhukavi et al shows how design resources, local production capability, and budget could all affect material choice[11].

2.3-Topology optimization

Topology optimization is often used to reduce mass and increase structural efficiency in performance-oriented uses. Hunar et al constructed a rear upright using the Solid Isotropic Material with Penalization (SIMP) technique as shown in figure 3. Under many stress conditions, including cornering and braking, their work attained a 40% mass reduction while preserving a safety factor of 1.8. It made successfully with additive processes, showing no distortion[5].

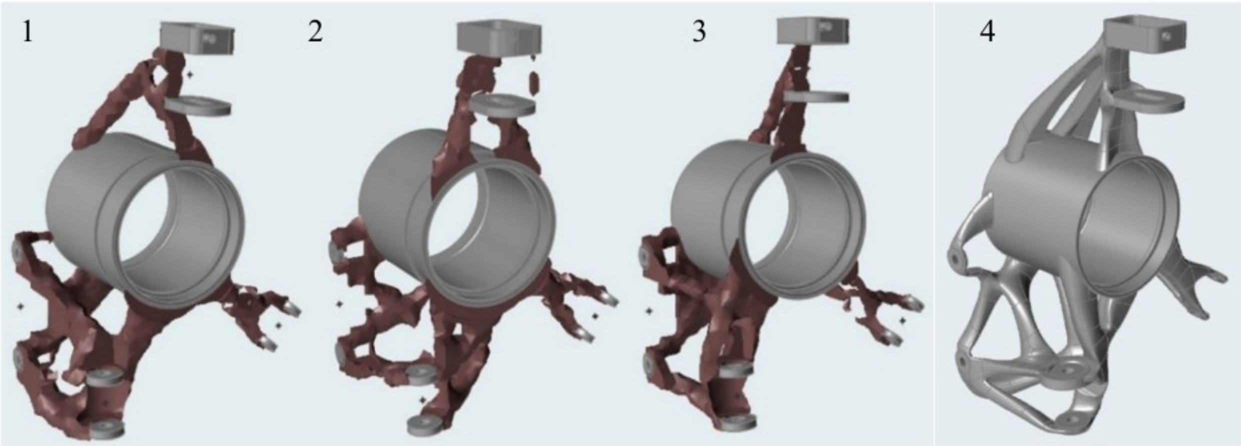

Figure 3. Topology optimization, 1) cornering left and braking, 2) swift braking, 3) cornering right and braking, 4) final outcome [5]

Li et al. similarly utilized variable-density topology optimization to a upright, lowering its mass by 60.43%[12]. FEA verified that during turning, braking, and sharp-cornering conditions, the final design maintained factor of safety (FOS) above 2.2. After dynamic testing and integration into a working electric FSUK vehicle, the upright proved to be really practical.

Both research show the need to combine design tools and simulation to save weight while maintaining performance.

2.4-Manufacturing method

When complicated geometries are involved, the shift from computational geomtry to a physical product offers manufacturing difficulties. Using selective laser melting (SLM), Hunar et al. manufactured their ideal upright by fabricating highly complex structures with minimum material waste[5]. SLM is expensive and less available, despite offering geometric freedom and sustainability.

Due to its accuracy and general availability, CNC machining is employed for upright manufacture. For 5-axis CNC machining, Kozlenok designed a topologically-optimized upright. His work included problems including tool access, undercut elimination, and datum feature inclusion[13]. These changes guaranteed that, without sacrificing structural integrity, the finished product could be manufactured.

Following a similar approach, Mesicek et al. considered CNC constraints and included topology optimization into their upright design[14]. Their design underwent FEA to confirm its mechanical performance, where the last model was modified for five-axis milling. This method guaranteed manufacture with traditional procedures and allowed them to keep weight savings.

3. Design Requirements and Constraints

Within a high-performance FSUK vehicle, the upright is absolutely essential for accurate handling, structural reliability, and suspension system integration. The component must therefore show great structural rigidity to reduce deflection under load, thereby preserving constant suspension geometry throughout rigorous dynamic events like braking, acceleration, and cornering. Targeting a minimum FOS of 2.0 over all important load situations guarantees durability and helps to prevent failure. Deflection has to stay within tight limitations to prevent affecting handling performance or wheel alignment. While supporting the proper kinematic behavior of the vehicle, the upright also connects with important suspension elements including the wishbones, push/pull rods, and steering arms. Geometric restrictions consider mechanical compatibility and demand enough room for the brake disc and caliper.

$$Factor\ of\ Safety = \frac{Maximum\ stress}{Yield\ Strength} \quad \text{(eq. 1)}$$

The upright design requires precise clearances and mounting compatibility with surrounding components to guarantee complete integration with the vehicle assembly and compliance with installing restrictions. Accurate measurements were obtained from the STEP file imported into SolidWorks (figure 4) to better understand the geometrical connections among the upright, wheel, suspension, and braking components.

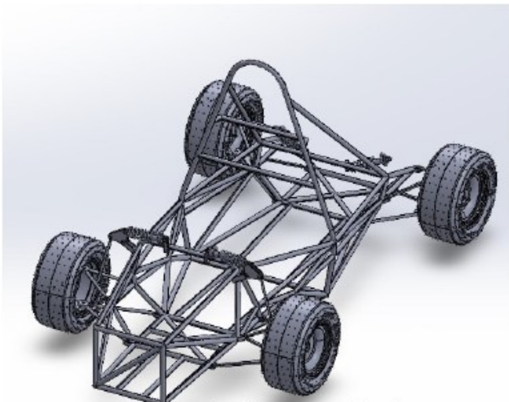

Figure 4. STEP file provided by QMFS [15]

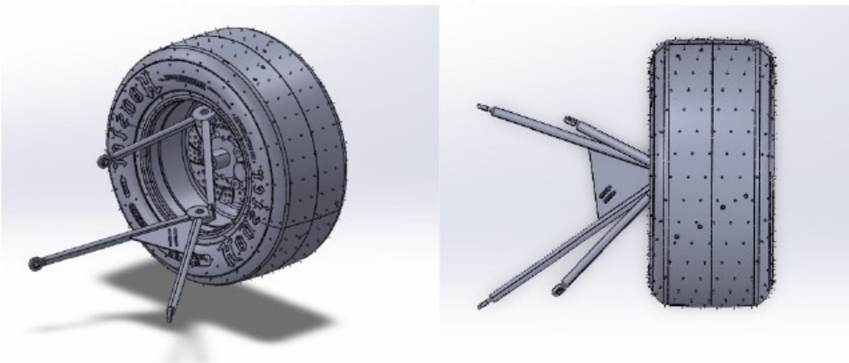

Figure 3. Rear left wheel assembly and wishbones [15]

Identified parameters including the wheel's radii (figure 6), bearing interfaces (figure 7), brake disc (figure 8) and wishbone thicknesses (figure 9) defined the environment for the upright and guaranteed appropriate fitting, load transfer, and manufacturing tolerance.

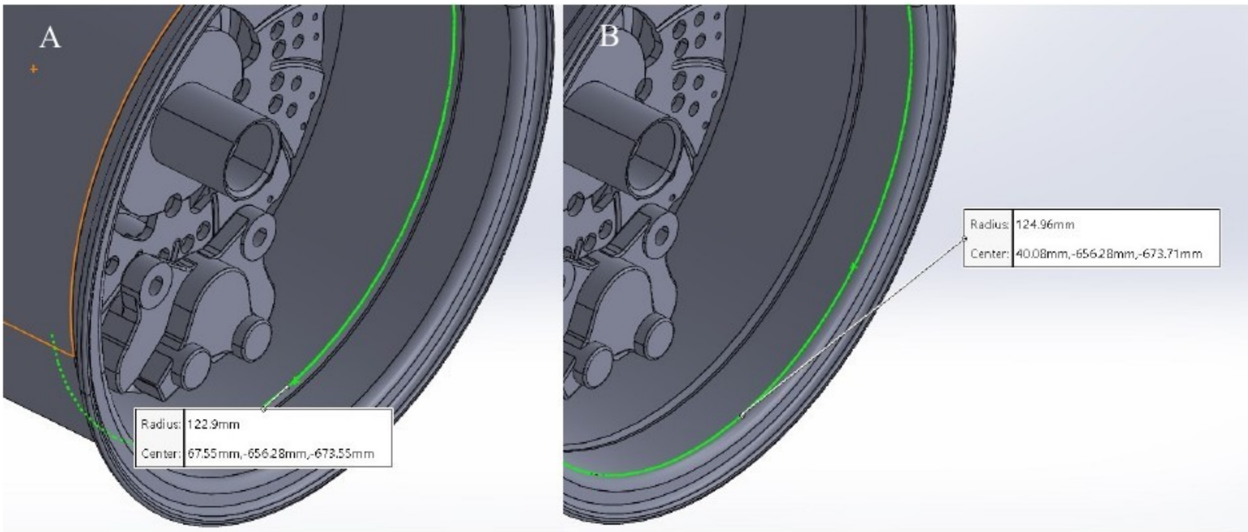

Figure 6. Maximum(A) and minimum(B) radius within the wheel [15]

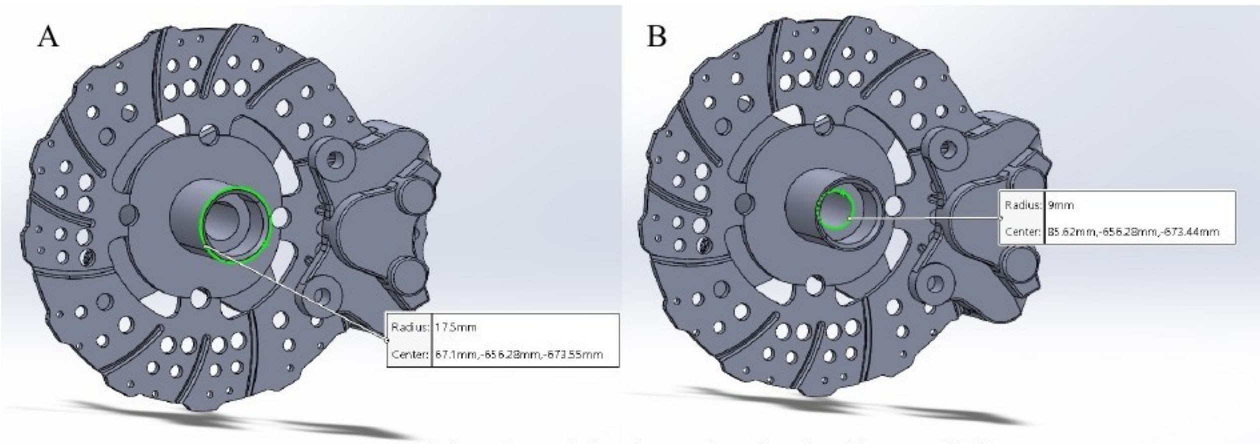

Figure 7. Outer(A) and inner(B) radius within the wheel bearing [15]

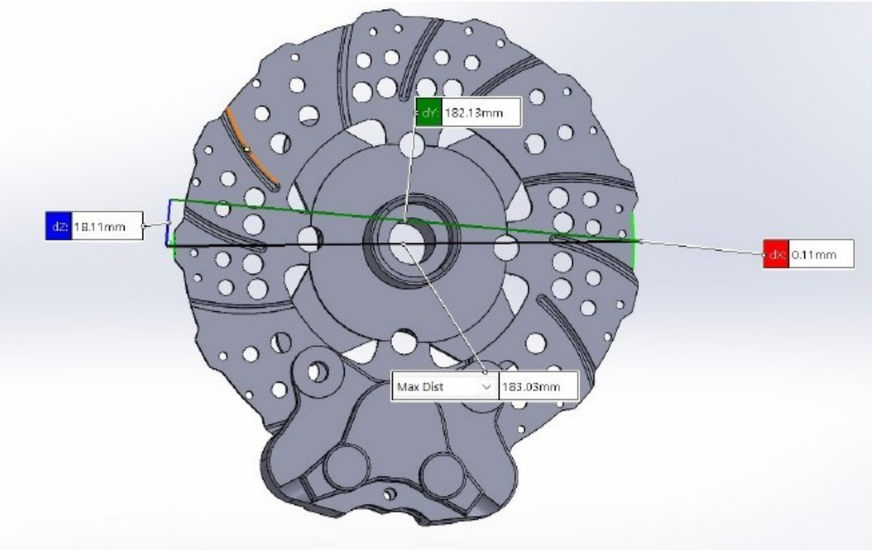

Figure 8. Measurements of brake disc [15]

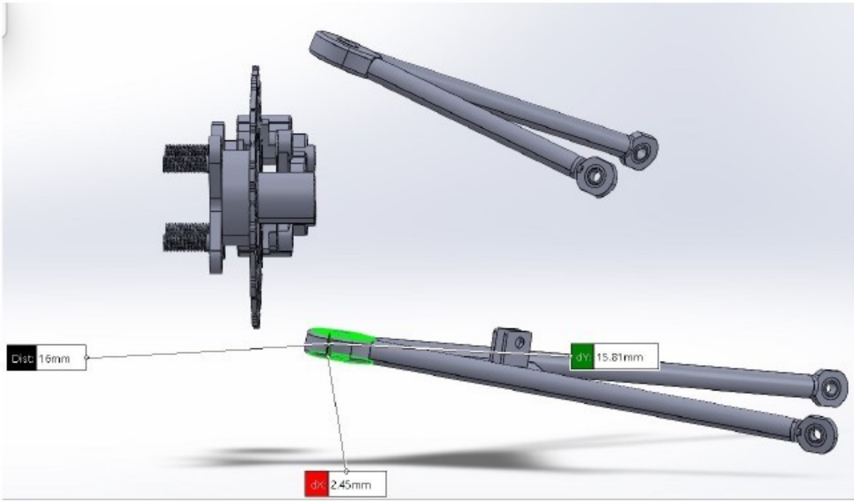

Figure 9. Thickness of wishbone at point of intersection [15]

Balancing technical performance with cost-efficiency and manufacture, all design and manufacturing decisions must follow FSUK rules and stay inside the team's budget as outlined in appendix A.

4. CAD Design and Geometry

Although an initial upright design offered by QMFS (figure 10) with a engineering drawing shown in figure 11. This project builds on that reference by reevaluating geometry, material selection and structural performance with the aim of developing a lighter, more efficient, and competition-ready design customized to the particular performance and sustainability goals of the current car.

Figure 10. View of upright[15]

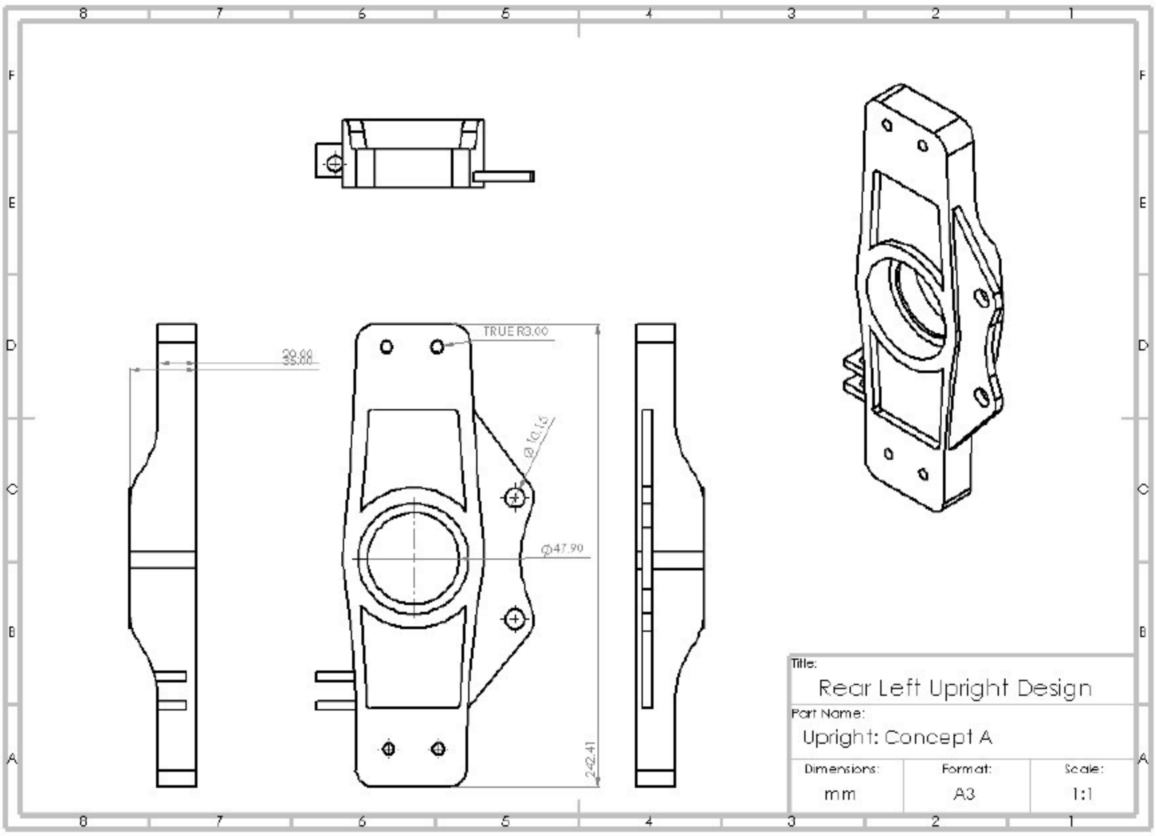

Figure 11. Engineering drawing of upright

5. Load Scenarios and Calculations

It was determined that the upright undergoes longitudingal braking, lateral load and cornerning forces in additon to the wishbone and pushrod forces. Parameters provided in Appendix B[7], shared by QMFS, are required to work out the maximum forces.

$$\mathrm{F}_{\text{static load}} = \mathrm{m}_{rear\ left} * \mathrm{g} \qquad \text{(eq. 2)}$$

$$\mathrm{F}_{\text{brake}} = \mathrm{m}_{rear\ left} * a_{\text{brake}} \qquad \text{(eq. 3)}$$

$$\mathrm{F}_{\text{cornering}} = \mathrm{m}_{rear\ left} * a_{lat} \qquad \text{(eq. 4)}$$

Equation 2 calculates the static vertical load on the rear left wheel using the mass at that corner and gravitational acceleration. Equation 3 gives the longitudinal braking force acting on the rear left wheel due to vehicle deceleration which was taken as 1.2g. The lateral (cornering) force on the rear left wheel from acceleration during a turn, shown in equation 4.

$$\mathrm{F}_{\text{pushrod}} = F_{static\ load} * \sin(push\ rod\ angle) \qquad \text{(eq. 5)}$$

Equation 5 resolves the vertical wheel load into the axial force experienced by the pushrod based on its angle from vertical[17]. Equation 6 calculates the upper wishbone's load by taking a moment around the lower wishbone, balancing both vertical and longitudinal (braking) forces about a pivot point [17]. Newton's 2nd Law in the vertical direction, was used to solve the force in the lower wishbone assuming no other vertical load paths aside from the upper and lower wishbones. These calculations were done on MATLAB, as shown in appendix C.

$$\mathrm{F}_{wishbone\ upper} = \frac{F_{static\ load} * z_{lower} + F_{brake}(z_{upper} - z_{lower})}{z_{upper} - z_{lower}} \qquad \text{(eq. 6)}$$

$$\mathrm{F}_{wishbone\ lower} = \mathrm{F}_{\text{static load}} - F_{wishbone\ upper} \qquad \text{(eq. 7)}$$

```
--- Rear Left Corner Force Analysis ---
Braking Force (X): 1175.32 N
Lateral Force (Y): 1469.15 N
Vertical Load (Z): 979.43 N
Pushrod Force: 1958.86 N
Upper Wishbone Left:  [X: -366.53, Y: 855.23] N
Upper Wishbone Right: [X: 366.53, Y: 855.23] N
Lower Wishbone Left:  [X: 366.72, Y: -244.48] N
Lower Wishbone Right: [X: -366.72, Y: -244.48] N
```

Figure 12. Results gathered from the MATLAB CODE

To mimic the loads on the upright, forces are applied where the wishbones, pushrod, and wheel hub join the upright as seen in figure 13. A more detailed force visualization is seen in appendix D.

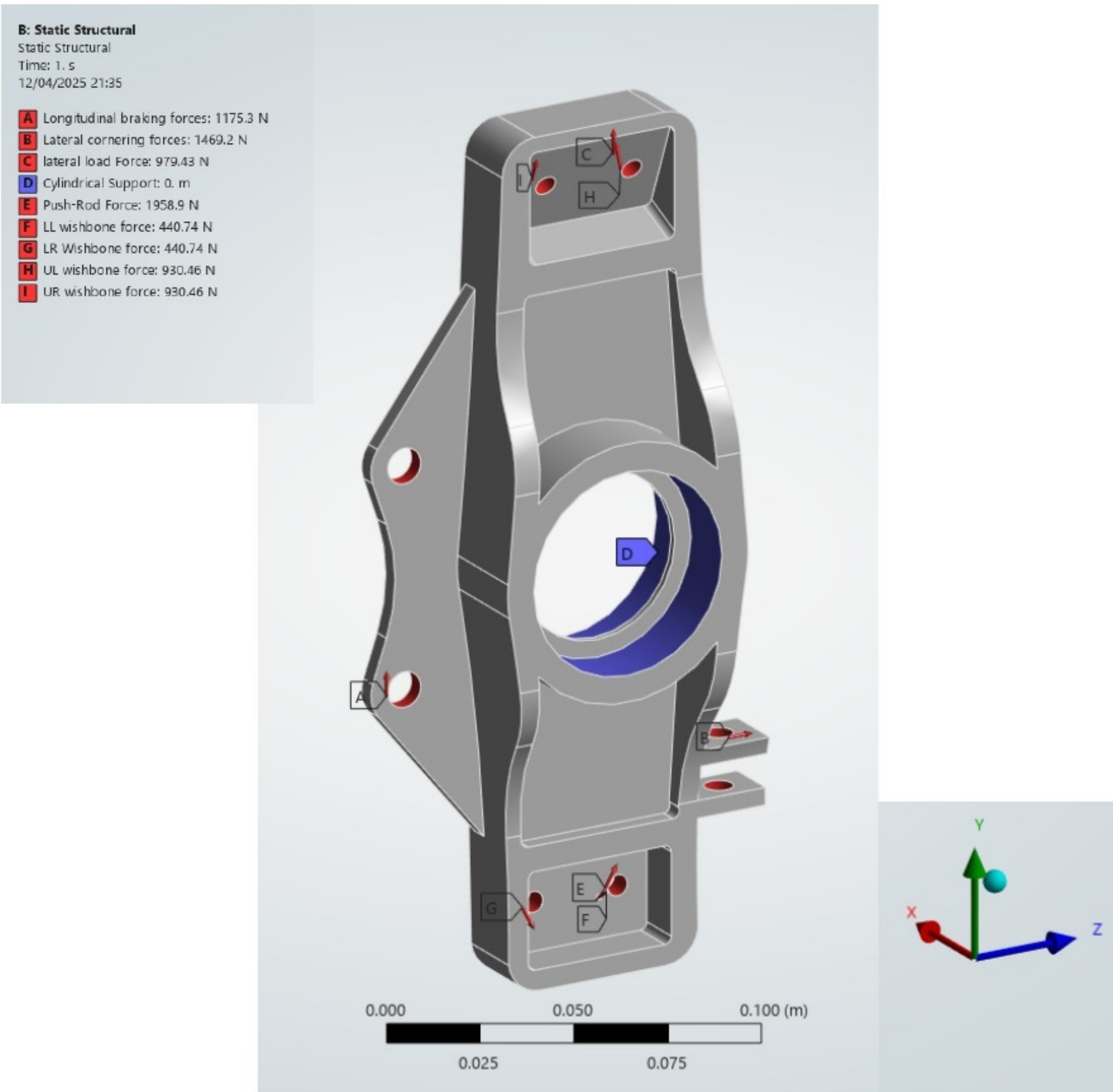

Figure 13. Calculated loads applied on the upright

6. Material Selection

The whole performance, safety, and manufacturability of the component depend critically on the chosen upright material. The upright must have excellent strength and stiffness due to complicated dynamic forces including braking, acceleration, and cornering while yet being as lightweight as possible to minimize unsprung mass. It also needs to have strong fatigue resistance to endure repeated loads all through the vehicle's running lifetime. Mechanical properties needs have to be matched with practical factors like machinability, cost, availability, and environmental impact. The chosen material also has to fit the manufacturing technique as well as the team's budget and the sustainability objectives.

The material selection was carried out using a systematic engineering design approach involving analysis through Granta EduPack, evaluating stiffness to weight and strength to weight trade-offs. This started by creating a primary chart (figure 14) comparing density to Young's modulas enableing identification of materials offering high stiffness at low mass. A secondary chart (figure 15) of Yield Strength vs. Density further ensured structural safety under high loads. Yield strength is important for structural integrity, especially under high axial and lateral stresses and the stiffness controls deformation.

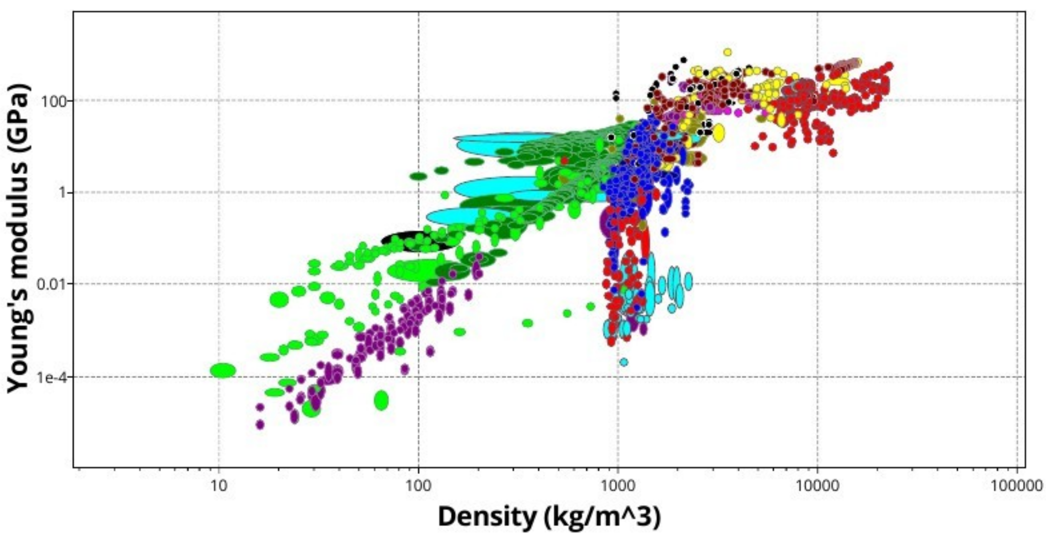

Figure 14. Ashby Chart: Young's Modulus vs. Density - used to identify materials with high stiffness-to-weight ratios suitable for suspension uprights.

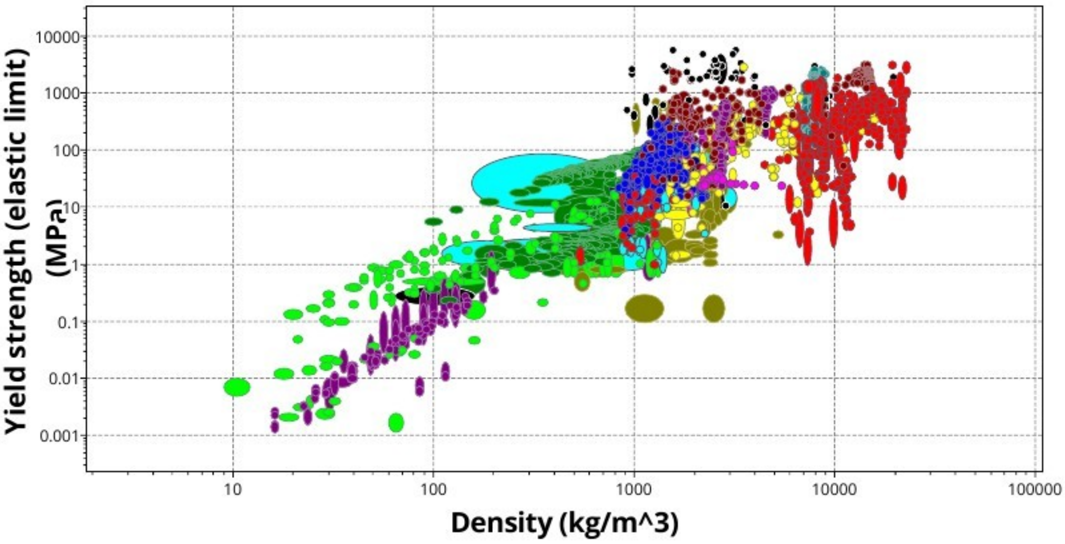

Figure 15. Ashby Chart: Yield Strength vs. Density - used to evaluate structural safety by identifying materials with high strength at low density.

Materials for the suspension upright had to endure two main loading scenarios: bending caused by lateral centrifugal stresses and vehicle's mass and axial compression due to vertical loads like braking. The material should function well in both circumstances.

When subjected to lateral cornering forces and the load of the car, the upright can be modeled as a beam under bending. The objective is to maintain sufficient flexural stiffness while minimizing the component's mass. A step by step derivation was done starting with the mass in equation 7 and bending deflection equation in equation 8[18] to find the performance index PI_1.

$$m = \rho A L \qquad \text{(eq. 7)}$$

$$\delta = \frac{FL^3}{3EI} \qquad \text{(eq. 8)}$$

For a rectangular cross-section, the second moment of area is directly proportional to the area squared, assuming geometric constraints length and force are fixed, minimizing deflection becomes proportional to maximizing Young's modulus and the second moment of area.

$$PI_1 = \frac{E^{\frac{1}{2}}}{\rho} \qquad \text{(eq. 9)}$$

During breaking, the upright can be modeled as a buckling force. The Euler's buckling load force is shown in equation 10[19].

$$F_{cr} = \frac{\pi^2 EI}{L^2} \qquad \text{(eq. 10)}$$

The second moment of area is directly proportional to the area squared, and for a fixed buckling force load the requaired area is inversly proportional to young modulus resulting in the performance index PI_2.

$$PI_2 = \frac{E^{\frac{1}{2}}}{\rho} \qquad \text{(eq. 11)}$$

Both bending and buckling analyses lead to the same performance index, making it highly effective and convenient for evaluating lightweight, stiff materials for the upright. The performance index was then applied to the EduPack charts, in addition to EduPack filters where only non-ferrrous metals due to their lower density and recyclable materials would be shown as seen in figure 16 and 17.

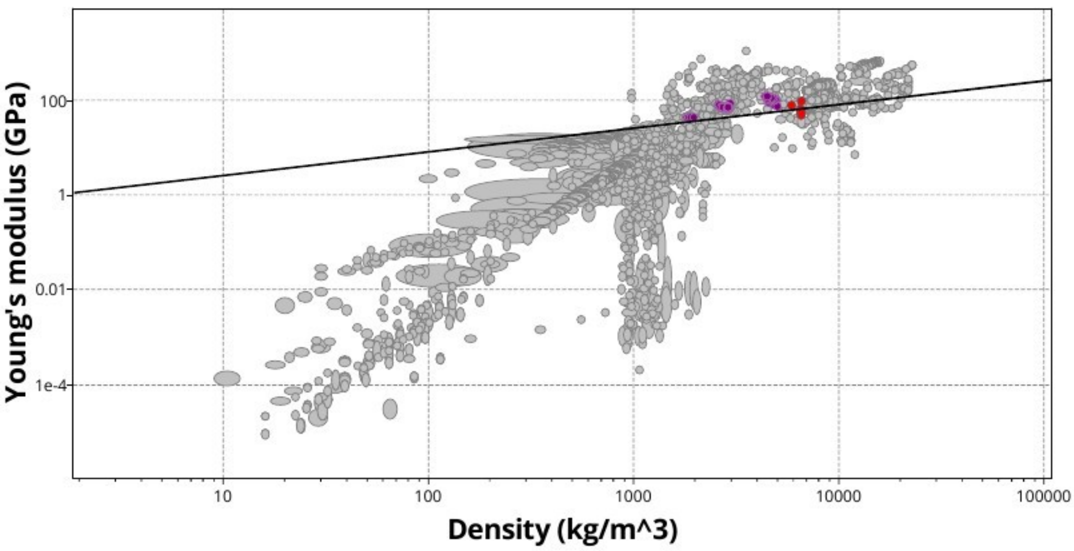

Figure 16. Filtered Ashby Chart: Young's Modulus vs. Density - filtered for non-ferrous, recyclable materials to align with sustainability and weight objectives.

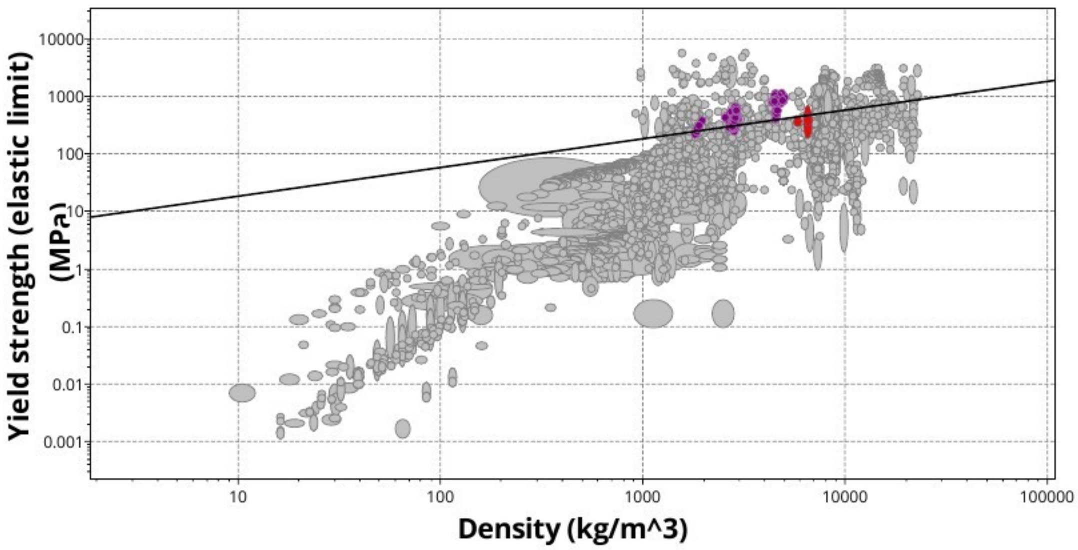

Figure 17. Filtered Ashby Chart: Yield Strength vs. Density - showing strength performance of non-ferrous, recyclable candidates for the upright.

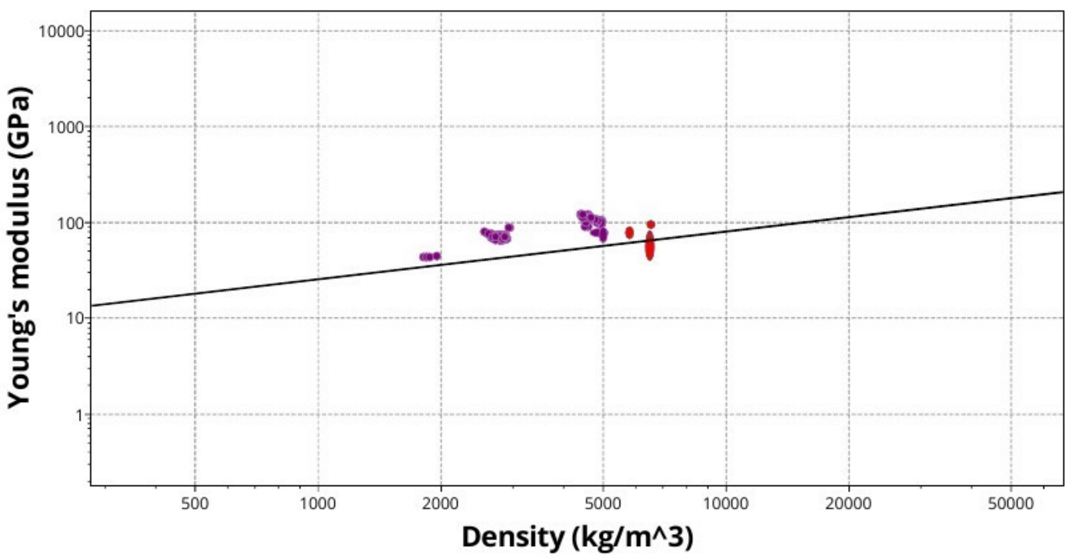

Figure 18. Final Material Selection Chart - stiffness-to-weight performance comparison of shortlisted non-ferrous candidates including aluminium, titanium, and magnesium alloys.

The final chart (figure 18) shows materials that passed the performance index and strength filters. To quantitatively compare the final shortlisted materials, a weighted decision matrix was made, assessing each candidate across key performance and practical criteria as shown in table 1.

Table 1. Decision matrix of the shortlisted materials

Material	Specific Stiffness [NM.m/kg]	Yield Strength [MPa]	Fatigue Strength [MPa @ 10^7 cycles]	Manufacturability	Cost [USD/kg]	Fracture Toughness [MPa·√m]	Total Score
Aluminum 7075-T6	25.9	495	160	Excellent	6.915	26.7	**4.45**
Titanium Alloy (Ti-6Al-4V)	25.85	848	354	Moderate	25.3	95.5	4.36
Aluminum 6061-T6	25.15	260	121.5	Excellent	3.47	33	2.94
Magnesium EA55RS-T4	23.4	403	212.5	Moderate	13.85	16	2.73
Nickel-Cu-Si alloy M-25S	19.75	725	147.5	Moderate	29.95	18.7	2.17

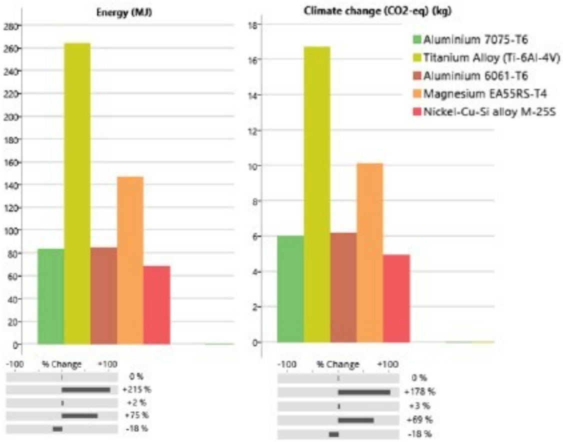

Figure 19. Eco-Audit of shortlisted material

Based on mechanical characteristics, cost, and manufacturability, a weighted decision matrix was used to evaluate and rank the top five shortlisted materials, as table 1 shows. Every material was assessed against six main criteria, with scores based on its relevance. In addition to the decision matrix, an Eco-Audit was conducted to evaluate the environmental impact of each material, particularly in terms of energy consumption and climate change potential, further informing material selection from a sustainability perspective, as shown in figure 19.

Although Titanium Alloy showed the best yield strength and fracture toughness, its high material cost and low machinability greatly reduced its practical viability within the confines of a FSUK vehicle. Furthermore, even if titanium's mechanical performance is outstanding, given financial constraints the cost to benefit ratio did not support its adoption.

Identified the best option, Aluminum 7075-T6 scored 4.45 overall. While keeping great machinability and a fair cost per kg, it presents a better mix of high specific stiffness, great yield strength, and good fatigue performance. Its mechanical characteristics make it rather appropriate for load-bearing parts. It offered the best trade-off between performance and feasibility.

Although the alternative materials, Aluminum 6061-T6 and Magnesium EA55RS-T4, provided high specific stiffness and reduced cost, they lacked either the necessary strength or fatigue resistance for a key safety component like the upright. Though robust, Nickel-Cu-Si alloy M-25S suffered from both density and cost.

7. FEA

Ansys Workbench was used in FEA to test the structural integrity of the upright under load scenarios. The goal was to determine whether the design could resist the applied forces while preserving, a minimum FOS of 2. Since the upright is a complex, load-bearing component, the accuracy and dependability of the analytical findings are very important.

7.1-Mesh Convergence

A mesh convergence study was done to guarantee consistent and accurate results in the FEA of the upright. With element counts from 14,078 up to 1,138,160, 6 different mesh sizes from coarse (mesh size 5mm) to fine (mesh size 0.6mm) were examined, as shown in appendix F. The maximum deformation and maximum von Mises stress were recoreded for every mesh size. Figures 20 and 21 show that the data reveal that both deformation and stress values started to converge above about 400,000 elements. This stage was crucial to guarantee that the next simulations were not examples of inadequate meshing.

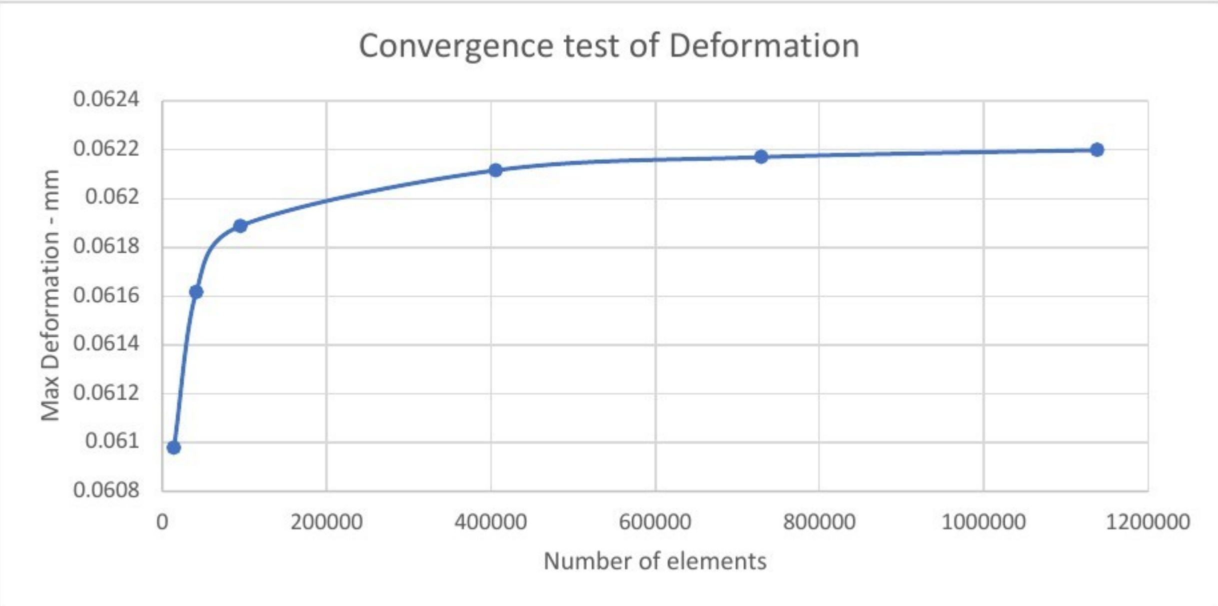

Figure 20. Convergence test of number of elements vs maximum deformation

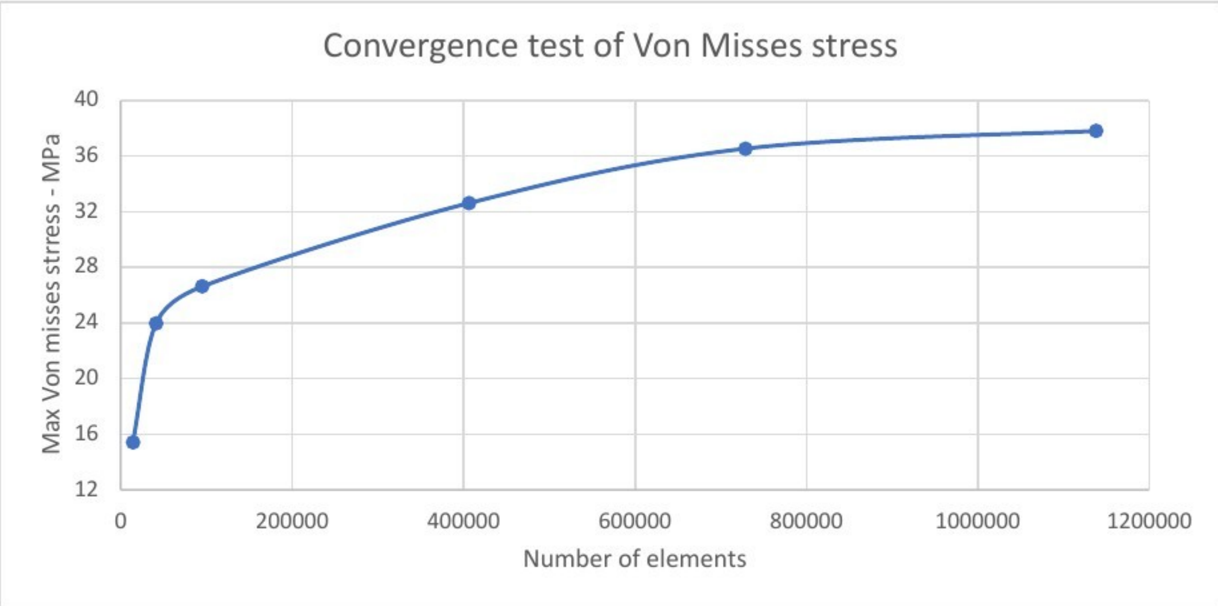

Figure 21. Convergence graph of number of elements vs Maximum Von Misses Stress

Subsequently, all simulations were conducted with a mesh size of 0.75 mm since it resulted in accurate and converged findings for von Mises stress as well as deforrmation.

7.2-Results

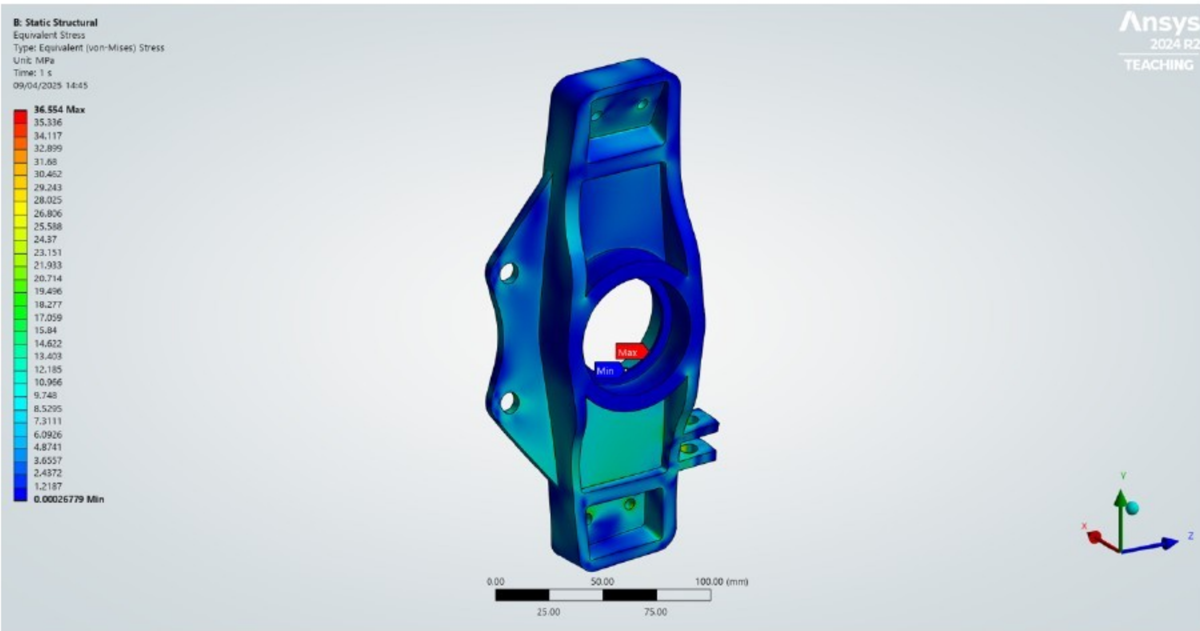

Figure 22. FEA result – Equivalent von-Mises stress distribution in the upright

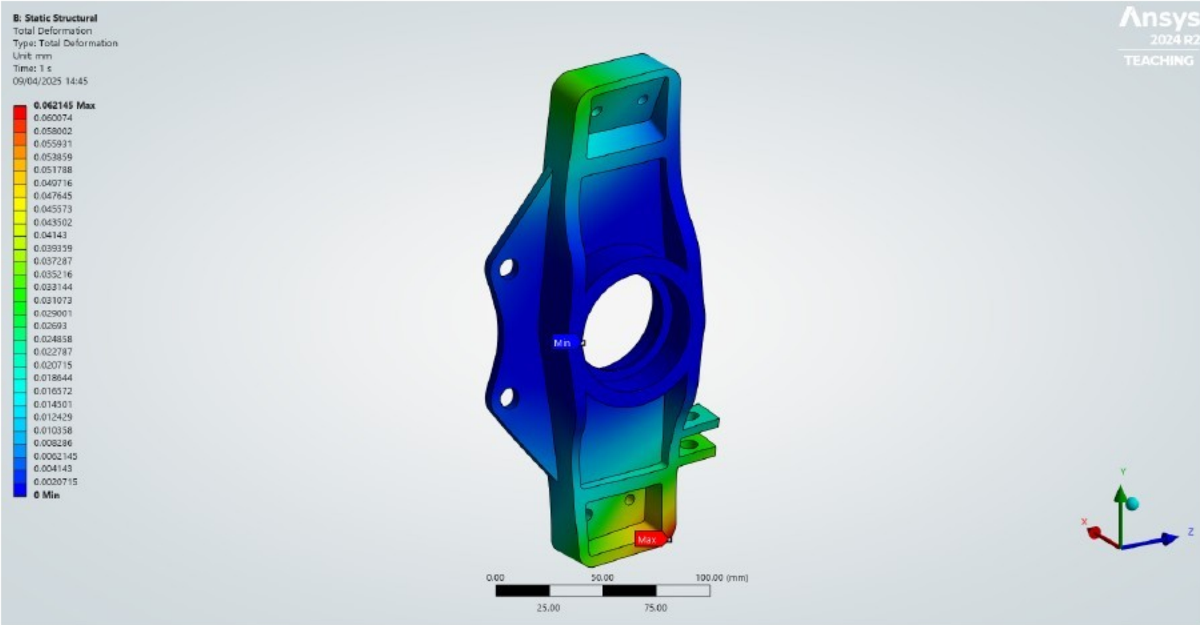

Figure 23. FEA result –Total deformation in the Upright

The orignial geomtry achieved a maximum von-Mises stress was 36.554 MPa, much below the yield strength of aluminum 7075-T6, guaranteeing a safety factor above 2 as shown in figure 22. With a maximum deformation of 0.062145 mm, figure 23 shows how the suspension geometry is unaffected. While deformation mostly in the bottom area, stress is focused around mounting points. These findings verify structural integrity of the design under maximum loads. This means that topological optimization, material removal from low-stress regions without sacrificing strength, will lead to a lower weight.

8. Topology Optimization

A topological optimization attempt was made to increase the weight efficiency and performance of the FSUK upright. Several iterations (geomtries shown in apendix G) were developed and analyzed using ANSYS workench beginning with the baseline model (iteration 0). With the aim of minimizing mass while yet maintaining structural integrity under maximum loads.

Table 2. Topology optimization results across upright design iterations

Iteration	Upright Mass (kg)	Mass Difference from Iteration 0	Mass Difference from Previous iteration	Maximum Deformation (mm)	Maximum Equivalent Stress (MPa)	Factor of Safety	
0	0.67248			0.06215	36.55	13.51	Pass
1	0.64117	4.66%	4.66%	0.07511	46.19	10.69	Pass
2	0.54458	19.02%	15.06%	0.23164	218.18	2.26	Pass
3	0.53378	20.63%	1.98%	0.33576	381.55	1.29	Fail
4	0.58479	13.04%	-9.56%	0.11140	94.37	5.23	Pass
5	0.54852	18.43%	6.20%	0.08923	102.92	4.80	Pass
6	0.53793	20.01%	1.93%	0.11245	104.54	4.72	Pass

With a mass of 0.67248 kg, iteration 0, provided the reference model. Its maximal overall deformation was 0.062145 mm and its von Mises stress was 36.55 MPa as seen in figure 22 and 23. 13.51 was the FOS computed against the yield strength of Alaminium 7075 T6, 493.8 MPa[20], therefore verifying an overengineered yet safe geometry.

Next, low-stress areas were identified for material removal using circular cuts based on the stress distribution in the baseline model, creating iteration 1. Mass dropped 4.66% due to this. The deformation grew to 0.07511 mm, and the maximum stress went up slightly to 46.20 MPa. The FOS stayed high at 10.69 and the design easily satisfied structural criteria.

Inspired by the first iterations results, iteration 2 used more aggressive internal cutouts focusing on the central spine (figure 24). This approach dropped mass by 19.02% from the baseline but raised stress to reach 218.18 MPa. The FOS declined to 2.26 and total deformation likewise rose greatly to 0.23164 mm. The design focused on meeting key performance goals, even though it was still limited by material choices.

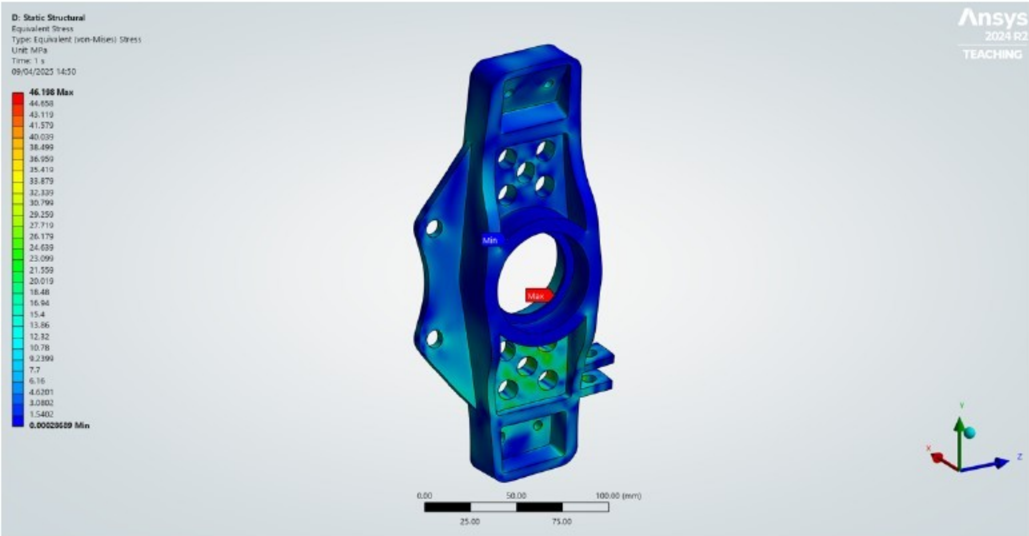

Figure 24. FEA result – Equivalent von-Mises stress in the 1st iteration of the Upright

Seeking further weight loss, iteration 3 (figure 25) expanded the cutout approach. This iteration suffered from excess stress concentrations (381.55 MPa) and deformation (0.33576 mm), even if it achieved a 20.63% mass reduction. The design was evaluated unsuitable for operational usage and fell short of the structural standards with a FOS of just 1.29.

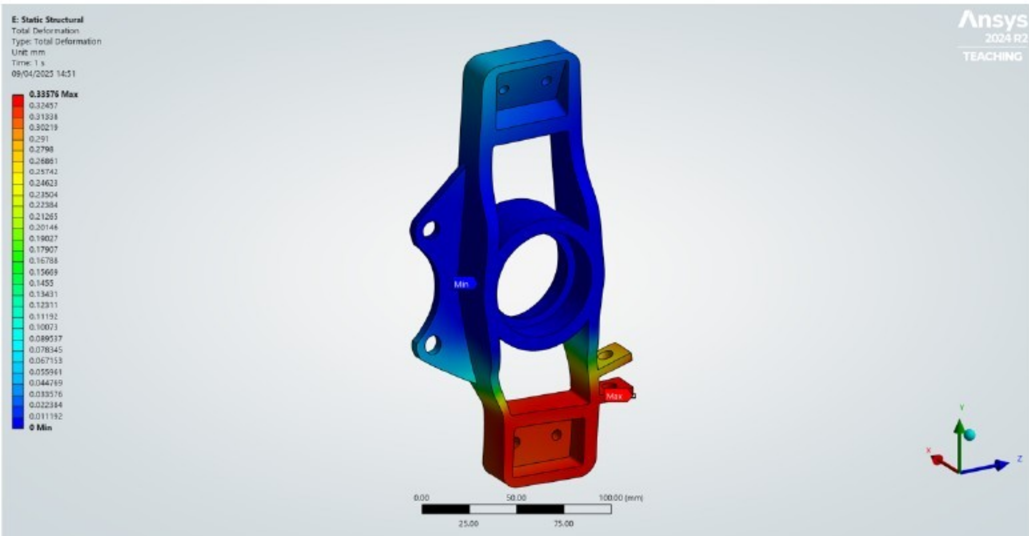

Figure 25. FEA result – Total Deformation in the 3rd iteration of the Upright

Iteration 4 took a fresh approach with internal truss reinforcement in response to iteration 3 problems. This structural improvement somewhat raised mass to 0.58479 kg but caused a significant decrease in deformation (0.1114 mm) and maximum stress (94.37 MPa). FOS increased to 5.23, restoring design dependability.

Iteration 5 improved the internal truss design and included fillets to help to reduce stress concentrations. Maintaining a maximum stress of 102.92 MPa and deformation of 0.08923 mm, the mass was further dropped to 0.54852 kg. The FOS of 4.80 demonstrated the strength of the design.

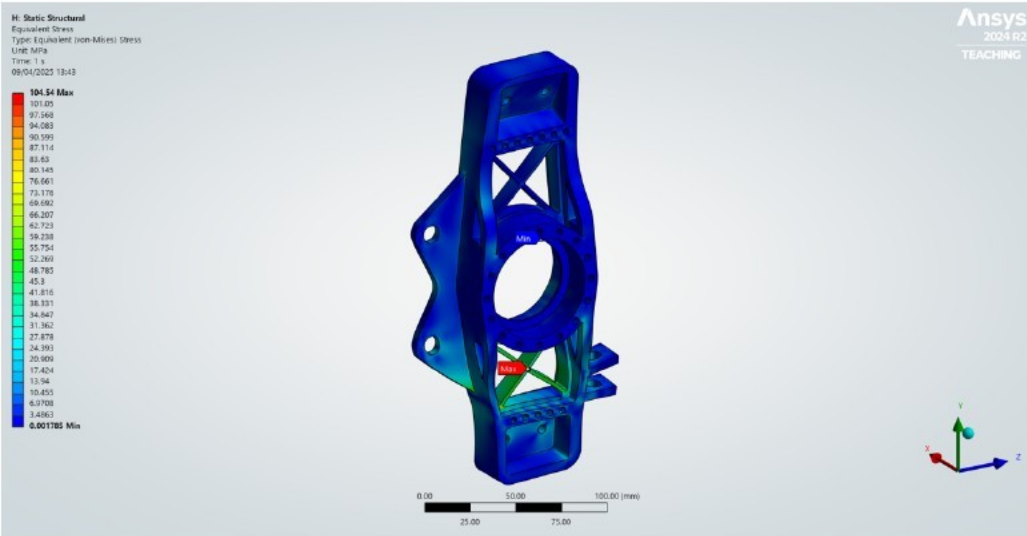

Figure 26. FEA result – Equivalent von-Mises stress in the 6th iteration of the Upright

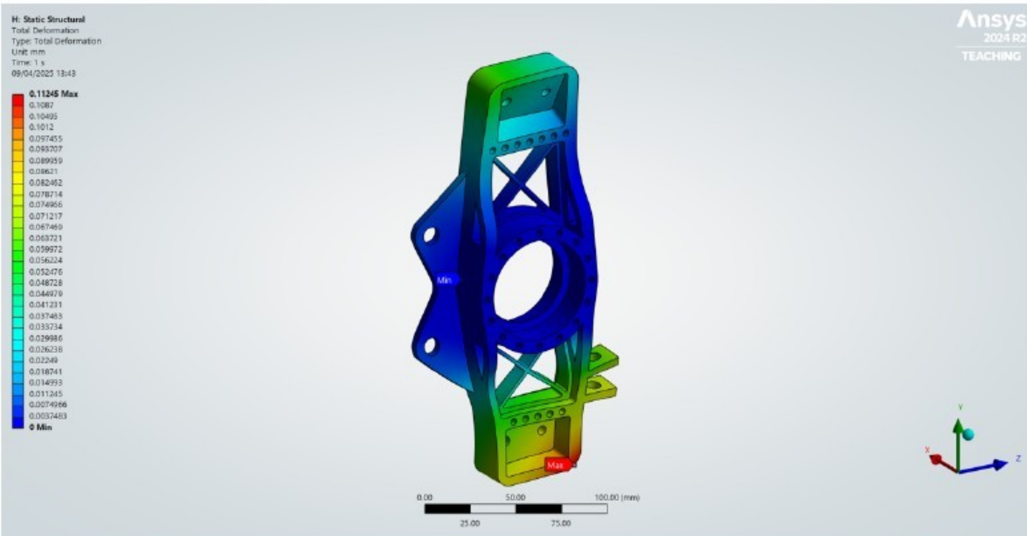

Figure 27. FEA result – Total Deformation in the 6th iteration of the Upright

Iteration 6 produced a final design with more geometry smoothing in high-stress areas and filleting around bolt holes. Maintaining appropriate stress (104.54 MPa) and deformation (0.11245 mm) as shown in figure 26 and 27, this iteration attained a 20.01% weight reduction (0.53795 kg). With a factor of safety of 4.72, the design was validated as being mass-efficient and structurally robust.

The full set of results, including total deformation and equivalent von-Mises stress plots for each design iteration, are provided in Appendix H and Appendix I, respectivly.

9. Manufacturing Plan

A FSUK upright's production process calls for a mix between economic viability, material efficiency, and accuracy. Several possible manufacturing methods were under thought. Casting is mostly employed for its scalability in high-volume manufacturing, it sometimes suffers from dimensional limitations inappropriate for suspension components needing great accuracy. Forging has geometric restrictions and more tooling expenses even if it offers great mechanical qualities. Although additive manufacturing gives design freedom, it is still less sensible for the selected aluminium alloy and the necessary mechanical load conditions and remains expensive.

CNC machining was chosen as the most suitable manufacturing technique considering the criteria for tight tolerances, complicated geometries, and limited production volume. With limited post-processing, CNC machining guarantees repeatability, great surface smoothness, and the capacity to produce complex details including bearing housings, bolt holes, and strengthening features.

Made from 7075-T6 aluminum, a high-strength alloy, the raw material was quoted at £165.56, as shown in appendix J. Four iterations of design versions of the upright manufacturing costs were evaluated, machine and technician times (appendix K) were determined as shown in table 3.

Table 3. Cost breakdown of CNC machining for upright design iterations

Upright Iteration	Machine Time	Technician Time	Cost (Excl. VAT)	VAT (20%)	Total Cost
0	3 hours (£15.00)	2 hours (£42.94)	£57.94	£11.59	£69.53
4	3.5 hours (£17.50)	2 hours (£42.94)	£60.44	£12.09	£72.53
5	4 hours (£20.00)	2 hours (£42.94)	£62.94	£12.59	£75.53
6	5.5 hours (£27.50)	3 hours (£64.41)	£91.91	£18.38	£110.29

The last iteration had the most expensive geometric complexity and reflected this in design. Features like the deep side pockets and interior fillets, forced greater setup times and longer machining distances. These small details naturally slowed down the cutting operation and required more expert input to guarantee dimensional correctness and surface polish.

10. Environmental and Safety Considerations

Looking at envirementai considerations, a high strength to weight ratio and recyclability made Aluminium 7075-T6 the right option. Though CNC machining is a subtractive technique, careful toolpath planning reduced waste. Offcuts will be gathered for recycling. Topology optimization further cut unneeded material, hence lowering part mass and energy use during operation.

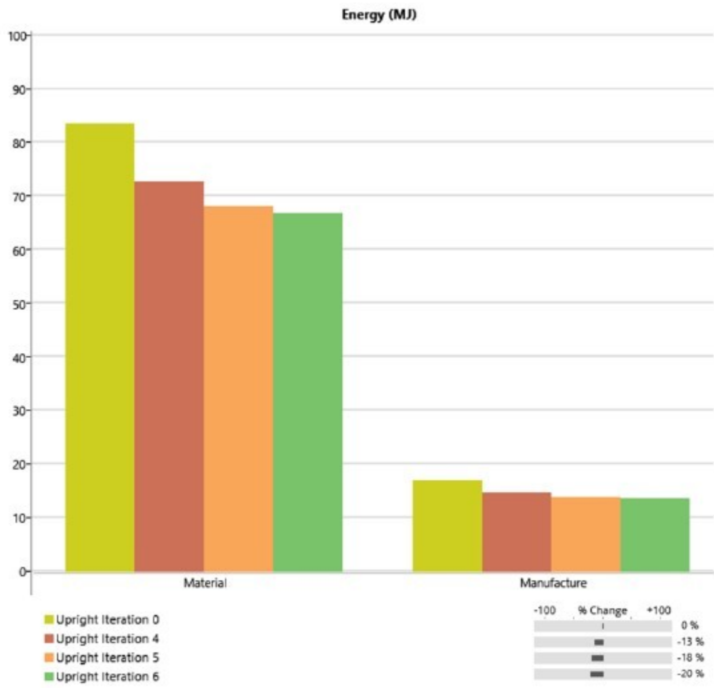

Figure 28. Energy consumed for each iteration in MJ

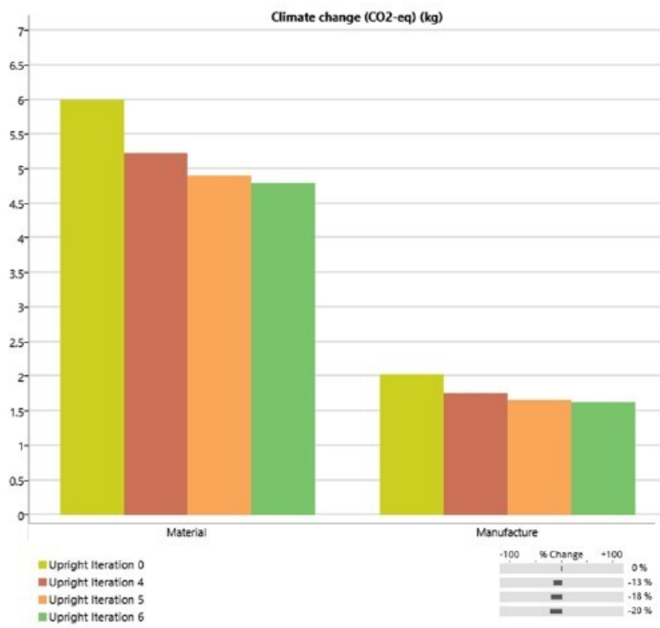

Figure 29. CO_2 released for each iteration

In figure 28, the energy audit reveals that successive upright design iterations resulted in a consistent reduction in total energy consumption, with Iteration 6 achieving the lowest energy usage in both material and manufacturing phases. In figure 29, The carbon footprint assessment shows a progressive decline in CO_2 emissions across the upright iterations, with Iteration 6 producing the least environmental impact. EduPack was used to make these.

Hazards were found throughout the FSUK upright's design and optimization that, if improperly controlled, would impact safety, manufacture, or performance. Four main areas, simulation accuracy, material selection, production feasibility, and operational integration, saw a structured risk assessment as seen on table 4.

Table 4. Key Risks and Mitigation Approaches in FSUK Upright Development

Risk	Impact	Likelihood	Mitigation Strategy
Inaccurate load assumptions in FEA	- Underestimated stress - Potential failure under load	Medium	- Use conservative assumptions - Safety factor > 2.0
Mesh dependency affecting simulation accuracy	- Unreliable stress/deformation - False validation	Low	- Mesh convergence study (0.75 mm mesh)
Excessive material removal in topology optimization	- Reduced stiffness - Structural failure risk	Medium	- FEA review of each iteration - Reject unsafe designs
Complex geometry increases CNC time/cost	- Delays in production - Budget overrun	Medium	- Select simpler, optimized design - reject unsafe designs
Incorrect manufacturing tolerances	- Misalignment - Bearing fitment issues	Medium	- Apply precise tolerances - Use accurate CAD data
Low safety margin in high-stress zones	- Local yielding - Fatigue failure	Low	- Reinforce mounting areas - Validate with FEA
Material substitution or supply issues	- Reduced material performance - Safety compromise	Low	- Source from certified supplier - Cross-check properties

11. Discussion

Designing a lightweight, high-strength upright that fits FSUK competition criteria and efficiently interacts with the suspension system was the primary goal of this project. This included maximizing the component for structural integrity, ensuring it could be produced within financial and time restrictions, and significantly lowering unsprung mass. Under peak stress conditions, the last design, iteration 6, achieved a 20.01% mass reduction, maintained a factor of safety of 4.72, and fit CNC machining. These results show that every main goal was satisfactorially achieved.

Manufacturing assessments and FEA helped analyze the last design, iteration 6. Within safe and useful limits for aluminium 7075-T6, it recorded a maximum von Mises stress of 104.54 MPa and a deforrmation of 0.11245 mm. Indicating enough resistance to failure under worst-case scenarios, the FOS stayed well over the goal threshold of 2.0. Furthermore showing its practical fit in the FSUK, the design maintained important suspension geometry.

The phase of topological optimization produced a sequence of iterations wherein mass was lowered while control of stress distribution and deformation was maintained. Early iterations (2 and 3) sacrificed structural integrity in suspension mounting points, while achieving dramatic weight cuts. Iteration 3 was inappropriate for use because of its FOS of barely 1.29 and significant deformation. Iterations 4 through 6, brought structural bracing and strategic reinforcements, hence reducing weight while nevertheless preserving safe performance limitations. Balancing mass efficiency with mechanical robustness proved dependent on this change from aggressive cutouts to intelligent optimization.

In the last phases of the design process, the performance to cost trade-off grew ever more important. Every iteration created by topology optimization sought to lower mass while preserving structural integrity and feasibility within manufacturing and financial limits.

Iteration 4 mass was 0.58479 kg and manufacturing cost was £72.53, offering a balanced structural profile and a strong safety factor of 5.23, it presented a maximum von Mises stress of 94.37 MPa and a deformation of 0.1114 mm making this version dependable and effective.

Iteration 5 refined geometry in high-stress areas and strengthened internal bracing design, therefore improving on iteration 4. Reflecting an 18.43% weight loss from the baseline, it cut the mass to 0.54852 kg and cost £75.53 to manufacture. The performance stayed strong despite the extra complexity, the von Mises stress rose to 102.92 MPa and the overall deformation droped to 0.08923 mm. At 4.80, the FOS remained somewhat high achieving a good mix of production practicality and mechanical performance.

A further optimized geometry produced Iteration 6, the final design. Although it came at a great cost of £110.29, attaining the lowest mass of 0.53793 kg, a 20.01% decrease from the original. A 1.93% mass decrease over iteration 5, the cost rose by 46%. With a safety factor of 4.72, stress dropped to 104.54 MPa and deformation climbed to 0.11245 mm.

The trade-off between Iteration 5 and 6 is paying an extra £34.76 for a 10.59 g weight reduction. Due to QMFS tight budget, the added cost and complexity may not justify the small benefit. In deformation, another important factor in maintaining suspension geometry under load, iteration 5 also showed superior performance.

Improvements in deformation and stress resistance allowed a 6.2% weight loss for an extra £3.00 between 4 and 5. But the weight savings dropped to under 2% from 5 to 6, while cost surged by £34.76. These numbers show how performance stagnated even as design complexity and resource constraints grew.

Iteration 6 was successful in stretching the boundaries of optimization overall, but its marginal performance gains come at a disproportionately higher cost. With its competitive mass, cheap cost, and outstanding mechanical performance, iteration 5 offers a more sensible and effective design choice. This emphasizes the need of assessing in design decision-making not only structural measures but also manufacturing consequences and economic ones.

The performance of the upright owed much upon material choice. Excellent strength to weight ratio, fatigue resistance, and CNC machining compatibility helped aluminium 7075-T6 stand out. EduPack investigation confirmed the high stiffness to density and yield strength to density performance indices, therefore validating the choice. Titanium alloys were rejected because of their greatly higher cost and poor machinability. Although their strength and corrosion sensitivity were lower, magnesium alloys were considered about but turned down.

The chosen manufacturing technique, CNC machining, offers outstanding surface polish and accuracy. Every design element was specifically for CNC operations, but the intricacy of Iteration 6 produced longer machining times, more tool wear, and more setup needs, all of which directly affected its higher manufacturing cost. Early generations of simpler geometries needed less toolpath refining, so they were more affordable. As the cost study shows, a practical final design depends on balancing part complexity with ease of fabrication.

12. Conclusions

This research aimed to create a lightweight, structurally sound, manufacture-able upright meeting FSUK competition criteria. Fulfilling the initial goals, the last design, iteration 6, obtained a 20.01% weight reduction while keeping a factor of safety of 4.72 under peak load.

Important implications are the importance of balancing structural integrity with weight reduction. Iteration 3 fell short of safety standards. Iterations 4 through 6 brought reinforcements that lowered mass while yet improving performance. Practical trade-offs are highlighted by cost-performance study showing iteration 5 had almost the same mechanical advantages as 6 but at a far lower cost.

The project presented many difficulties. Modelling restrictions in FEA meant fatigue behaviours were not captured, so challenging long-term dependability. Given limited access to track telemetry and real-world data, obtaining reasonable load instances was challenging as well. Limited lab access hampered material testing and assembly validation, time limits prevented physical prototyping. These difficulties made clear how different theoretical modeling is from actual application.

Using more sophisticated tools like ANSYSn nCode, future research should stretch into fatigue simulation. Long-term dependability assessment would be enhanced by full fatigue life prediction under cyclical loads. IPG automotive software can be also used to carry out a optimal lap simulation. Following bench testing for strength and stiffness validation, physical prototyping of the final upright should be carried out using CNC-machined aluminium 7075-T6. Real-world stress and deformation data from in-track testing with instrumented suspension arms could enable more exact validation of simulation results. Reviewing the geometry to simplify non-critical elements might also assist lower machining time and cost without compromising performance. Investigating hybrid topology optimization methods or composite reinforcement can provide even more weight savings in further projects.

The study fulfilled its objectives and provided insight into performance and cost trade-offs in upright design.

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