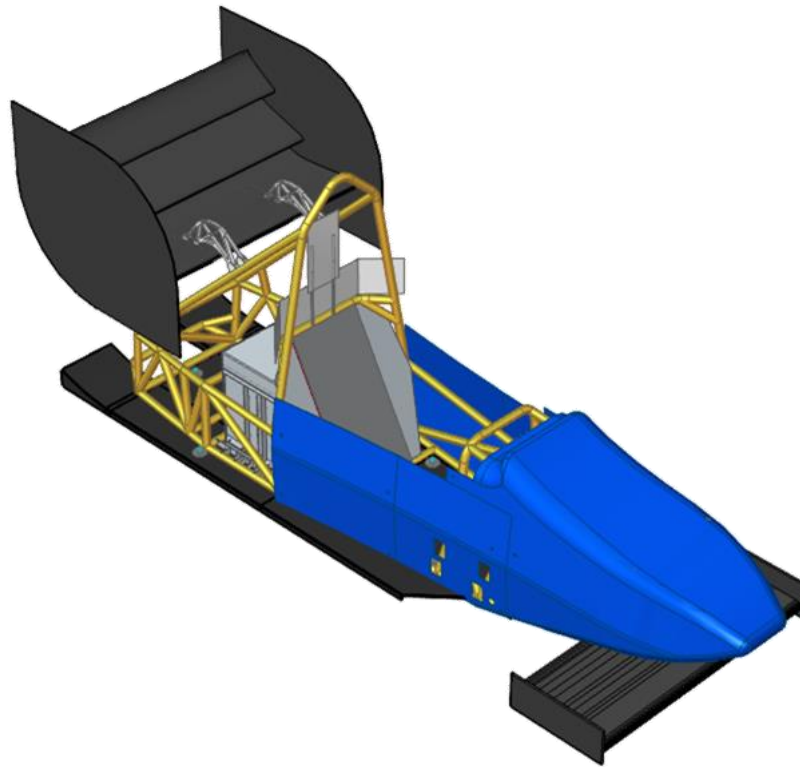


Formula Student 3

Chassis & Aerodynamics (2017-2018)

Critical Design Review (CDR)



Billy Cavanagh 6298087

Tom Clay 6359063

Adam Cox 6302841

Sean Glendinning 6359845

Shahrukh James 6321900

Chris Kyle 6299943

Isaac Lu 6207886

Himanshu Mattoo 6349673

Robert Pemberton 6368443

Supervisor

Dr. Patrick Gruber

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TABLE OF CONTENTS

1	INTRODUCTION (Chris Kyle)	1
2	FRAME (Chris Kyle)	1
2.1	Final Frame Design	1
2.1.1	Front Suspension	2
2.1.2	Rear Suspension	2
2.1.3	Ride Height.....	3
2.1.4	Material.....	3
2.1.5	Jacking Bar	3
2.1.6	Final Changes	3
2.1.7	Future Improvements.....	4
2.1.8	Manufacturing	4
2.2	Torsional Stiffness Testing.....	4
3	ALTERNATE FRAME RULES (Tom Clay)	6
3.1	Design Options Considered	6
3.2	Testing Planned/ Conducted	7
3.3	Algorithms, Formulae and Constraints Considered/ Used	7
3.4	Current Design Concepts.....	8
3.5	Conclusions and Further Improvements	11
4	ACCUMULATOR ATTACHMENT (Sean Glendinning)	12
4.1	Design Options Considered	12
4.2	Constraints Considered.....	12
4.3	Changes since the Initial Design	12
4.4	Final Design and Manufacture	12
4.4.1	Accumulator Mount Design	12
4.4.2	Accumulator Mount Positioning	14
4.5	Cost.....	15
4.6	Test conducted / planned	15
4.7	Conclusions and Future Improvements	16
5	IMPACT ATTENUATOR (IA) (Sean Glendinning)	16
5.1	Design Options Considered	16
5.2	Constraints Considered.....	16
5.3	Changes since the Initial Design	16
5.4	Final design and Manufacture	16
5.4.1	Design of Test Bulkhead and Manufacturing	17
5.4.2	Design and Manufacturing of Impact Attenuator Material	17
5.5	Cost.....	17
5.6	Tests Conducted / Planned	17
5.7	Conclusions and Future Improvements	18
6	DRIVER'S COCKPIT (SEAT + FIREWALL) (Shahrukh James)	18
6.1	Seat	18

6.1.1	Design Constraints.....	18
6.1.2	Design options considered and final design	19
6.2	Firewall.....	19
6.2.1	Formulae and constraints considered.....	19
6.2.2	Design Concepts and Analysis	20
6.2.3	Attachment Method	23
6.2.4	Testing	23
6.2.5	Final Design.....	23
6.2.6	Cost Breakdown	24
7	DRIVER RESTRAINT (Shahrukh James).....	24
7.1	Head Restraint	24
7.1.1	Design Concepts	24
7.1.2	Testing and Design Analysis	25
7.1.3	Final Design.....	25
7.1.4	Configuration Management List and Changes since Initial Design	26
7.1.5	Cost Breakdown	26
7.2	Harness System	26
7.2.1	Mounting Tabs and Analysis.....	27
7.3	Cockpit Conclusion and Potential Improvements	27
7.3.1	Firewall and Seat	27
7.3.2	Head Restraint	28
8	FLOOR (Himanshu Mattoo)	28
8.1	Introduction	28
8.2	Design Options and Possible Improvements	28
8.2.1	Previous Designs	28
8.2.2	Floor Set Goals	28
8.2.3	Floor Design	29
8.2.3.1	Front Section	29
8.2.3.2	Middle Section.....	29
8.2.3.3	Rear Section.....	29
8.3	Finite Element Analysis.....	30
8.4	Mounting Method Used.....	30
8.5	Diffuser.....	31
8.6	Manufacturing and Mould Preparation (Floor Panel and Diffuser)	32
8.7	Design Summary	32
8.8	Testing Plans	33
8.8.1	Floor	33
8.8.2	Diffuser Test Plans	33
8.9	Further Improvements	33
8.9.1	Floor	33
8.9.2	Diffuser.....	33

8.9.3	Computational Fluid Dynamics	33
9	BODYWORK (Adam Cox).....	33
9.1	Design Options Considered:.....	33
9.2	Constraints Considered:.....	34
9.3	Changes since the Initial Design:	34
9.4	Final Design and Manufacture:	35
9.4.1	Geometry	35
9.4.2	Material and Manufacture	36
9.4.3	Mounting Strategy	38
9.4.4	Finalised Design	38
9.4.5	Cost.....	38
9.5	Tests Conducted and Planned.....	39
9.6	Conclusion and Further Improvements	39
10	FRONT AERODYNAMICS (Isaac Lu)	39
10.1	Introduction	39
10.2	Design Options Considered and Changes Made	39
10.3	CFD Setup and Testing.....	41
10.4	Mounting Concepts	42
10.5	Mounting Design Selected.....	43
10.6	Conclusion and Potential Improvements	45
11	REAR AERODYNAMICS (Robert Pemberton).....	45
11.1	Design Options Considered	45
11.2	Constraints and CFD Simulations	45
11.2.1	Post Processing of CFD.....	47
11.3	Final Design.....	48
11.4	Materials Selection	48
11.5	Manufacturing Method	49
11.5.1	Mounting	49
11.5.2	Each Aerofoil.....	49
11.6	Cost Sheet.....	49
11.7	Tests Conducted/Planned	49
11.8	Conclusions and Further Improvements	50
12	DRS STUDY (Billy Cavanagh).....	50
12.1	Introduction	50
12.2	Design Options Considered	50
12.3	Tests Conducted	50
12.4	Conclusion.....	51
13	REAR WING MOUNTING AND AEROFOIL STRUCTURE (Billy Cavanagh)	51
13.1	Introduction	51
13.2	Constraints Considered.....	52
13.2.1	CFD Data.....	52

13.2.2	Loading Conditions	52
13.2.3	Stress Flow in Complex Composite Geometry	52
13.3	Design Changes From Initial Concept.....	52
13.3.1	Swan Neck.....	52
13.3.2	Wing Rib Mounting and Wing Structure.....	53
13.4	Design Analysis.....	54
13.4.1	Swan Neck.....	54
13.4.2	Wing Rib Mounting and Wing Structure.....	54
13.5	Testing.....	55
13.5.1	Testing Conducted.....	55
13.5.2	Testing Planned	55
13.6	Future Improvements.....	56
13.7	Conclusion.....	56
14	PROJECT MANAGEMENT (Chris Kyle + Adam Cox).....	56
15	OVERALL COST	58
16	OVERALL CONCLUSION	58
	REFERENCES	59

Abbreviations

FS – Formula Student
 PDS – Product Design Specification
 PDR – Preliminary Design Review
 DRS – Drag Reduction System
 IA – Impact Attenuator
 FSG – Formula Student Germany
 AIP – Anti Intrusion Plate
 FEA – Finite Element Analysis
 FSUK – Formula Student United Kingdom
 CFD – Computational Fluid Dynamics
 SURTES – Surrey University Racing Technology Engineering Solutions
 CAD – Computer Aided Design
 SE* – Surrey Electric
 NACA – National Advisory Committee for Aeronautics
 AH – Aluminium Honeycomb
 C.S.A – Cross Section Area
 GFRP – Glass Fibre Reinforced Polymer
 CFRP – Carbon Fibre Reinforced Polymer
 ABS – Acrylonitrile Butadiene Styrene
 CDR – Critical Design Review
 CNC – Computer Numerical Control
 ECU – Engine Control Unit
 FIA – Fédération Internationale de l'Automobile
 SIA – Specification Internationale de l'Automobile
 SF – Safety Factor

1 INTRODUCTION (Chris Kyle)

This team has achieved the goal of designing and building a steel spaceframe which satisfies Formula Student (FS) rules and includes a system for supporting and removing an electric accumulator. Particular focus was placed on the integration of critical systems such as the electric motor, suspension and the driver. The completed chassis has achieved a torsional stiffness value of 1665 Nm/deg and a mass of 38.9 kg, meeting the stiffness requirement set out at the beginning of the project but not the mass requirement. An aero dynamic package has also been developed for the whole car including front and rear wings, a diffuser and aerodynamically optimised bodywork. This package is expected to contribute 112 N of downforce at 12 m/s in its maximum downforce setup. All of these components except the bodywork will use high strength and lightweight carbon fibre in order to minimise the weight penalty of adding aerodynamic devices to the car.

2 FRAME (Chris Kyle)

2.1 Final Frame Design

The final version of the frame has a mass of 31.9 kg and a torsional stiffness of 1152 Nm/deg, giving a specific stiffness of 36.1 Nm/deg.kg. These figures were calculated using ANSYS Workbench Finite Element Analysis (FEA) software. The model referred to as the 'frame' does not include the mounting system for the accumulator, the anti-intrusion plate (AIP), the firewall or the floor. The design has evolved in a number of areas since the initial design.

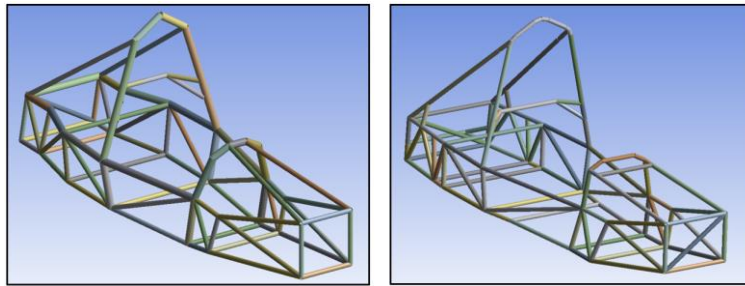


Figure 1: Initial frame design (Left) and final frame design (Right)

The stiffness figure stated above is 633 Nm/deg lower than that stated in the Preliminary Design Report (PDR) despite the design of the frame being improved. The reason for this is that the method used to find torsional stiffness, K , was found to be incorrect by a factor of 2 as the length used to calculate the applied torque from each force used the full distance between the front suspension uprights. Unfortunately, this mistake was only discovered after production of the frame had started. At the time of ordering it was thought that the frame had a torsional stiffness of about 2300 Nm/deg, which would have satisfied the 1500 Nm/deg target set at the beginning of the project. All stiffness values stated in this report have been calculated using the correct equation which is shown below.

$$K = \frac{2FL}{\tan^{-1}\left(\frac{\Delta y_1 + \Delta y_2}{2L}\right)}$$

Eqn 1

Where F is the force applied, L is the distance between each force and the centre of the chassis and $\Delta y_{1,2}$ are the deflections of the nodes at which the forces are applied (Riley & George, 2002). The layout is illustrated in Figure 2.

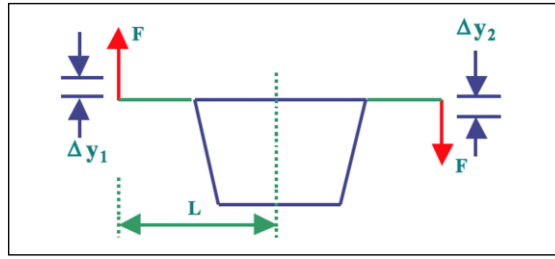


Figure 2: Torsional stiffness calculation setup (Riley & George, 2002)

For our analysis the lower upright nodes were used as these had a tube connecting them, making the deflection at each node easy to measure. The team was advised by our supervisor that this would give the most relevant stiffness estimation. A physical analysis of the completed frame, to be fully explained later in this report, will take place in order to verify the results from FEA.

2.1.1 Front Suspension

In order to mount the rockers and dampers for the front suspension in the required location it was necessary to move the front roll hoop supports to a wider position and add a tube between them. Initially this tube had a raised centre section as the desired location of the damper mounts was higher than the roll hoop supports. In order to keep the damper positions as rigid as possible in use it was decided that the suspension should be redesigned to lower the damper points so that the tube between the roll-hoop supports could be straight. The benefit of using a straight tube is that the majority of the force from the suspension is applied axially to the tube rather than creating a bending moment at the corner which would increase the stress in the tube.

Since the supports were moved to a wider position it was possible to reshape the front roll hoop to remove bends. This shape is simpler to manufacture due to it only having two bends with the same radius. This was a key area for improvement compared to SE3 (Surrey Electric 3). The front roll hoop designed for that car was a continuous curve which had to be manufactured by a specialist and cost around £400. The bend radius used in the new front roll hoop meets Formula Student (FS) regulations and the frame manufacturer agreed that he could produce these bends himself.

2.1.2 Rear Suspension

The initial rear hard points supplied by the suspension team were of a similar width to those used at the front of the car. In order to meet the requested wheelbase, also specified by the suspension team, the frame was forced to narrow quickly behind the space allocated to the accumulator. This greatly reduced the packaging space available behind the driver. The forward rear hard points were moved to a wider position so that the frame could have only one change of direction behind the battery rather than two.

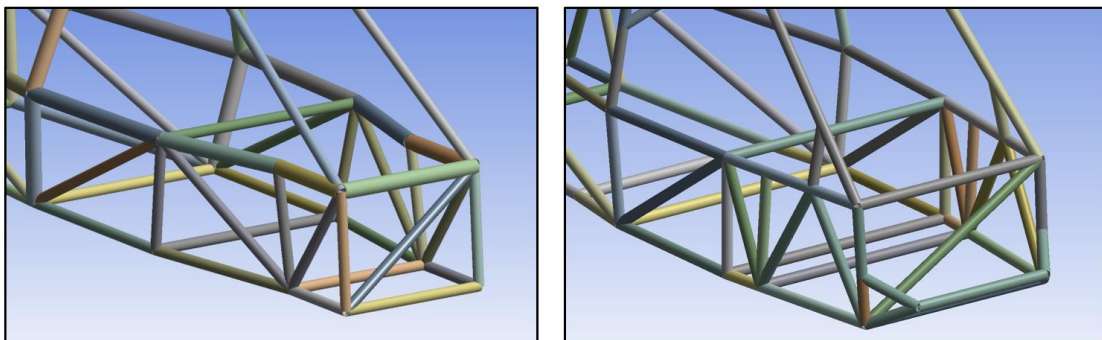


Figure 3: Initial frame rear (left) and final frame rear (right)

This allowed the inverter to be placed low down, beside the motor, rather than above the battery as it was on SE2. This helped to reduce the height of the centre of gravity of the car and therefore minimise load transfer during cornering, acceleration and braking. Following this change the triangulation between the rear uprights also had to change to allow clearance for the drive shafts. This was done using the wheel centre location and the maximum suspension travel to ensure that clearance would be maintained during operation.

2.1.3 Ride Height

The ride height of SE4 was designed to be 65 mm from the centreline of the lowest frame members to the ground. It was calculated using the minimum static ground clearance of 30 mm, the outer tube radius 12.7 mm and the expected thickness of the floor and diffuser 20 mm and rounded up to allow for a margin of error. This is sufficient to allow for the full 26 mm bump travel of the suspension without the floor contacting the ground.

2.1.4 Material

The material chosen for the frame was the steel alloy T45. It has a yield strength of 620 MPa which is higher than the 460 MPa yield strength of AISI 4130 steel which was used in SE3. The main reason for choosing T45 was that, unlike 4130, it does not require heat treatment before or after welding to avoid cold cracking. The material provider also advises that it is inadvisable to weld 4130 to mild steel due to the difference in chemical composition. This is not a problem for T45 so if tabs are produced in mild steel they can be welded to the frame without weakening the surrounding area in the same way as 4130 (Proformance Metals, 2018). The costs of the two metals were very similar so that was not a deciding factor in selection.

The use of ROPT510 was considered for the frame to reduce cost. This is FIA (Fédération Internationale de l'Automobile) approved roll over protection tube and has a yield strength of at least 370 MPa. It would have saved £450 in material cost for the frame but it was not selected due to concerns over durability. As SE4 is expected to complete extensive testing before competing in dynamic events in 2019 it was decided that higher strength alloy was a better option to mitigate the risk of damage to the structure caused by another component failing during testing resulting in a crash. The full material cost spreadsheet is available in Appendix 1.

2.1.5 Jacking Bar

The jacking bar was added behind the rear bulkhead in order to allow space for the diffuser at the bottom of the frame. An area was specified within the team with the goal of keeping the jacking bar out of space which could be useful for the diffuser. Had the diffuser been further developed it may have been possible to integrate the jacking bar into the structure of the diffuser to maximise the diffuser's performance.

2.1.6 Final Changes

To confirm the frame would pass inspection at competition the two minimum size cockpit templates were produced in computer aided design (CAD) and compared to the frame. The first of these templates is required to pass vertically between the two roll hoops and the upper side-impact members. The second must pass horizontally through the cockpit until it is within 100 mm of the pedal face with the pedals in their forward most location. This comparison showed that neither of the templates could pass through as required and the frame had to be adjusted to accommodate them. The cockpit was corrected by widening the main roll hoop where it connected to the upper side-impact members. The width was increased by 50 mm to 650 mm (centreline-to-centreline). The area for the driver's legs also had to be widened as the template would have touched the frame tubes before it had reached the required distance from the pedals. This was done by adding 50 mm to the width of the front bulkhead which allowed the template to move through the majority of the nose. This change could have been optimised if the pedal box design had been available. The wider front bulkhead required the addition of a cross member in order to properly reinforce the anti-intrusion plate. The vertical cockpit template was supplied to the frame manufacturer so that he could ensure compliance as the clearance was kept to a minimum to reduce the impact of the design. These changes increased the mass of the frame by around 0.3 kg and removed around 40 Nm/deg of stiffness.

Once the rear suspension had been fully modelled it was possible to see that the dampers were positioned almost perpendicularly to their mounting tabs. Loading at this angle would have created torsion at the base of the tabs which they had not been designed for. To avoid this, two short members were added in line with the dampers so that the angle between the dampers and tabs could be minimised. These can be seen above the 'A' shaped triangulation in the right image in Figure 3. Due to an updated head restraint design, the upper head

restraint bar could be removed from the frame as it was no longer necessary. These changes had a negligible net effect on mass and stiffness.

2.1.7 Future Improvements

The general form of the frame was constrained early on in the design process with the goal of having it ready for manufacture before Christmas. Due to a lack of aerodynamic components on previous cars, there was little existing knowledge to guide any decisions made with regard to the new aerodynamic package. The nose was raised from the ground in order to allow room for a front wing and with the aim of improving air flow to the diffuser. The research carried out by this team has shown that a lower nose angle of around 5 degrees would be beneficial to diffuser air flow and that the front wing can be packaged in a much smaller area than previously thought. The rear of the frame was kept flat in order to maximise the available packaging space. If beneficial to diffuser performance, this area could be redesigned as the main constraint for packaging here is longitudinal capacity, not vertical. The position of the driver could have been better optimised with regard to centre of gravity if the space required for the pedal box had been declared earlier in the design process. A worst-case estimate had to be used to ensure the cockpit was long enough to meet FS regulation. A possible benefit of the high nose and long cockpit is that SE4 may be more comfortable for driver's than previous Team SURTES cars as more leg room will be available than if the cockpit was at the minimum possible dimensions.

2.1.8 Manufacturing

The frame has been manufactured by welding together steel alloy tubing. Labelled drawings of the frame with each tube numbered were supplied to the manufacturer along with the coordinates of each node in the frame and which nodes each tube connected to. While the drawings are not essential to produce the frame, they simplified the process by providing a target to aim for. Colour coded images of the frame were used to show visually which wall thickness was required for each tube. The tubes were then cut to length and welded together to form the frame. Dimensioned drawings of the front bulkhead and two roll hoops were provided as these parts of the frame required more accuracy. The roll hoops used only 76.2 mm (3") bend radii so that they could be created by the frame manufacturer and meet the requirements in the FS rules. All of the documents sent to the manufacturer can be found in Appendix 2. The final frame used 1" diameter tube in the roll hoops instead of the 1.5" tube specified in the PDR as it was found that the rules had been misunderstood and it was not possible to use the low wall thicknesses previously implemented. This meant that the extra diameter was no longer necessary as the wall thickness had to be increased. The benefit of this is that the manufacturing of tabs for these areas was more straightforward as the standard 1" cutter could be used for all tabs.

Manufacture of the frame was quoted at £3400 plus VAT. The material required for the frame would have cost around £720 but due to the lengths that the tubing was supplied in the total cost was £1008 including VAT and delivery. This price included the extra tubing required to build mock noses to test the impact attenuator (IA).

2.2 Torsional Stiffness Testing

Due to the uncertainty surrounding virtual testing methods such as FEA and computational fluid dynamics (CFD) it is useful to verify results by carrying out physical experiments on the finished products. To validate the results produced during the design of the frame it will be physically tested for torsional stiffness. A rig was designed which would be capable of supporting the frame and applying forces to create the same load case used in FEA. The main difference between the two setups will be the point where the load is applied. In FEA the supports and loads are applied to the central lower suspension upright nodes. For the real test it is more practical to fix the chassis at the wheel hubs and apply a load at the end of a beam attached to the hub mounts. This means the load will be transferred to the chassis via the suspension wishbones. This is illustrated in Figure 4 below however the wheel hubs could not be included as they have not been finished.

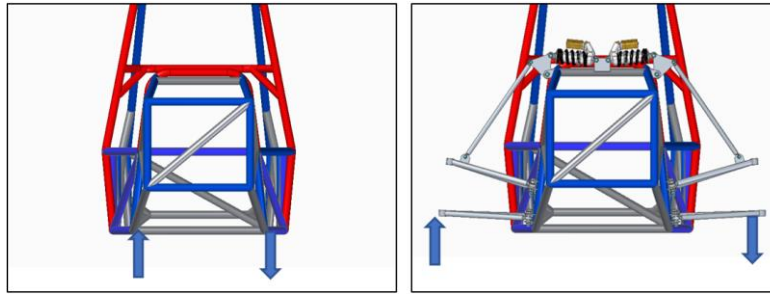


Figure 4: Load to frame in FEA (Left) and in physical test (Right)

Using the wheel hubs to constrain the chassis means that the stiffness of the suspension will have a significant impact on the resultant deflection. Rigid dampers will be used to limit deflection so that the end of the load bar does not touch the ground when load is applied. The measuring technique to be used for the experiment will be 3D Optical Measurement. This will be done by Team SURTES sponsor Physical Digital and can provide live results for the deflection at any point on the frame, accurate to ± 0.0005 mm (Physical Digital Ltd, 2018). The use of this style of measurement will remove the need for dial gauges, reducing the cost and complexity of the testing process.

The rig designed for the testing is shown in Figure 5. It consists of a load beam, a pivot and two mounts for the front wheel hubs. The positions of the mounts and pivot can be changed as needed. These components slide along the load beam so that the equipment can be adapted to fit future FS cars. The mounts are designed to fit in place of a wheel onto the wheel hubs.

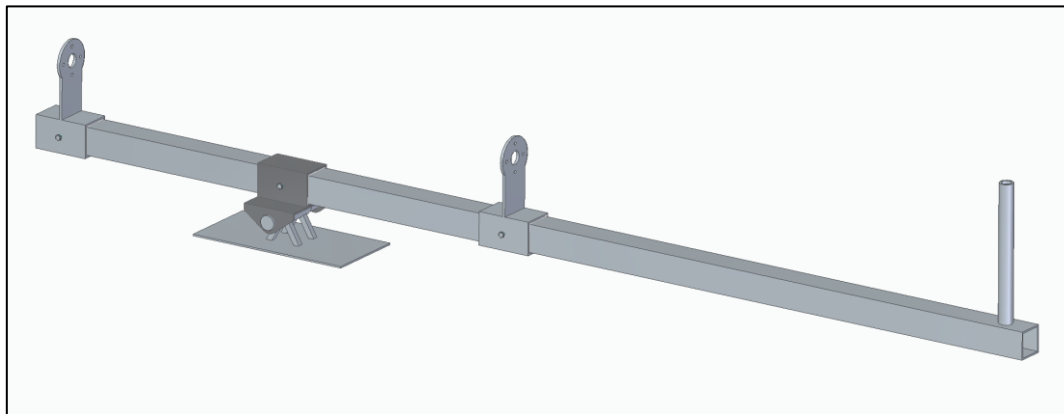


Figure 5: Torsional testing rig CAD

The mounts shown are designed to fit the four-stud pattern used on SE3 and other existing Team SURTES cars but new mounts will need to be designed for use with SE4 once the wheel hubs are completed. The mounts and pivot will utilise nylon bushings to ensure that they can slide along the load beam without the need for excess clearance between the box sections. During operation they will be secured in place by screwing bolts into the front face of the load beam. This added friction will be sufficient to secure the components as axial loads in the load beam will be low due to it being almost parallel with the floor. Not adding holes to bolt these components through the load beam means that the rig remains adaptable to any new chassis. The load beam itself should be stiff enough that it does not deflect under the load of the chassis or bend excessively when load is applied for testing. This is necessary to allow room for negative displacement of the applied load when the frame twists. The section used to construct the rig will be selected from available workshop stock and existing, unused equipment. The majority of the other parts of the rig can be created by repurposing redundant test equipment from the FS store, including the bearing for the pivot and the stands used to lift the chassis off the ground. The goal is to minimise the cost of the experiment as the results will not directly influence the performance of the car.



Figure 6: Rear Hub Mount

The car to be tested will be mounted into the test rig with the front hubs attached to the equipment shown in Figure 5 and the rear hubs attached to similar mounts similar to that shown in Figure 6. The rear of the car will be raised on its mounts so that the front and rear hubs are at the same height. This will ensure that the torsion will be applied in a plane which is perpendicular to both the ground and the wheelbase, as it would be during cornering.

To perform the test a base chassis position must be measured. This can then be compared to the position of the chassis with load applied to find the deflection. The key points which are to be identified are the lower nodes on the front suspension uprights. These can be used to compare the results directly to those produced in FEA. As the deformation of the frame will be elastic a linear increase of deflection is expected with increased load. To ensure that this linear increase has occurred mass will be increased incrementally in 10 kg intervals from 15 kg to 75kg (maximum disc weight available from FS stores). This is necessary as there may be some play in the testing equipment or suspension members or friction in the pivot bearing which could produce a non-linear change in deflection under light loading (Riley & George, 2002). A graph of torsional load, T , against angular deflection, θ , can be used to display the data. The gradient of a best fit line in the linear section of the results will give an average torsional stiffness, K , for the frame. Angular deflection and torsional load can be calculated using Eqn 2 and Eqn 3.

$$\theta = \tan^{-1} \left(\frac{\Delta y_1 + \Delta y_2}{L_N} \right)$$

Eqn 2

$$T = Fd$$

Eqn 3

The deflections $\Delta y_{1,2}$ refer to the lower nodes of the rearward front suspension uprights and L_N is the horizontal distance between the nodes at rest. F is the load applied and d is the horizontal distance at rest between load F and the pivot centre of rotation. This is illustrated in Figure 7.

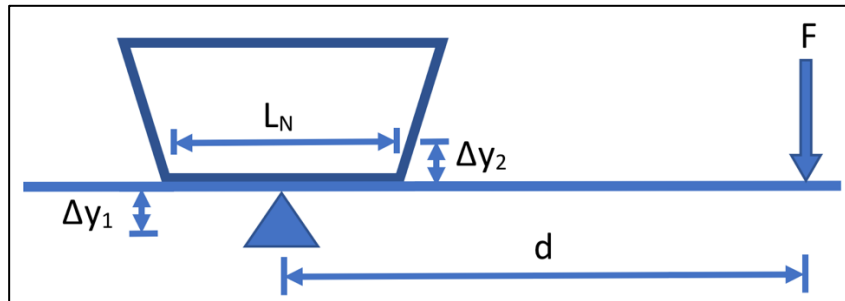


Figure 7: Simplified test layout

3 ALTERNATE FRAME RULES (Tom Clay)

3.1 Design Options Considered

As previously explained in the PDR, the first decision regarding the frame was selecting either a steel tubular spaceframe, a monocoque or a hybrid monocoque with rear spaceframe. There are advantages and

disadvantages to each type which were considered in detail at the start of the design process. Given a larger budget as well as more time for design and modelling, a carbon fibre monocoque design would have been the best option for performance which can be seen from its use in the top motorsport divisions such as Formula 1. However, performance must also be balanced with cost, complexity and time. On this basis, having found that the Team SURTES target of a top 10 finish in the FSUK competition is possible with a spaceframe design, a steel tubular spaceframe chassis was chosen because of the relative simplicity of the modelling and manufacture which would lower the cost and increase the likelihood of the chassis being manufactured on schedule.

With the aim of being able to provide recommendations and further improvements to Team SURTES for future cars, alternative frame designs and materials were also investigated. This mainly included the use of aluminium tubing within the spaceframe to reduce the mass of the car and particularly the use of composite shear panels as these could be added to the SE4 frame to further increase the torsional stiffness before it competes in the 2019 Formula Student Class 1 competition.

3.2 Testing Planned/ Conducted

As the alternative designs considered are hypothetical and will not be manufactured, testing will therefore be done only by using FEA software to model the design and simulate forces in order to estimate the torsional stiffness. To obtain a value for torsional stiffness, fixed supports are placed at the forward rear suspension uprights and equal and opposite forces placed at the rearmost front suspension uprights. Using the values produced for the vertical displacement of each of the front suspension uprights ($\Delta y_{1,2}$), the angle of twist (φ_A) is calculated with the equations below where B is the horizontal length between the applied forces.

$$\varphi_{1,2} = \tan^{-1}\left(\frac{\Delta y_{1,2}}{0.5B}\right) \quad \text{Eqn 4}$$

$$\varphi_A = 0.5(\Delta\varphi_1 + \Delta\varphi_2) \quad \text{Eqn 5}$$

Once the angle of twist has been calculated, the torsional stiffness K can be calculated using Eqn 6 where F is the applied force.

$$K = \frac{FB}{\varphi_A} \quad \text{Eqn 6}$$

FEA can also be used to estimate other parameters such as lateral loading as a result of the centrifugal forces produced during cornering, however, for this the analysis of the alternate frame designs, torsional stiffness was accepted as the best indication of chassis frame performance and was therefore the only parameter calculated.

3.3 Algorithms, Formulae and Constraints Considered/ Used

For the investigation into alternative frame designs, a recent version of the SE4 steel tubular spaceframe design was used as a benchmark. The torsional stiffness value for this frame iteration was 1231 Nm/deg and its mass was 33.02 kg without the battery mounts. Although the alternative designs are not limited by the same cost and time constraints as the actual frame design, they must still be commercially viable as that is a requirement set out in the project brief.

The main constraints for the alternative frame designs were the specifications given in the 2018 Formula Student rules. Notably that there must be a main roll hoop made of steel, a metal front roll hoop and any welded structures within the primary structure must be steel. Although the front roll hoop is allowed to be a welded aluminium structure, these rules limit the extent to which aluminium tubing could be used to replace steel tubing in the frame. If any materials other than steel are used in the primary structure, they must be physically tested to show equivalency to the following properties for steel in Table 1 as well as equivalent buckling modulus and absorbed energy if used for the side impact structure.

Table 1: Minimum material requirements for primary structure

Young's Modulus (E)	200 GPa
Yield Strength (S_Y)	305 MPa
Ultimate Strength (S_U)	365 MPa
Welded Yield Strength	180 MPa
Welded Ultimate Strength	300 MPa

3.4 Current Design Concepts

The first alternative explored was the use of aluminium tubing to replace some of the steel frame members, where permitted, in order to reduce the mass of the frame. To make the design viable, the aluminium tubing would need to be readily available so it was chosen from the range on the ProFormance Metals website. 7075 Aluminium was selected as this was the highest strength aluminium available, offering comparable strength to many steels (Proformance Metals, 2018). For the first iteration using FEA analysis, the front roll hoop and any frame members outside of the primary structure were replaced with 7075 aluminium tubing of the same dimensions as shown in Figure 8 below where the aluminium tubes are coloured blue and the steel tubes are represented by grey. This reduced the mass significantly to 22.77 kg from 33.02 kg for the steel frame. However, the FEA simulation yielded a torsional stiffness of 718.6 Nm/deg compared to 1231 Nm/deg despite the aluminium having higher strength properties than those of the steel used in the FEA analysis. This suggests that the torsional stiffness of the frame is affected most by the modulus properties of a material rather than its strength properties.

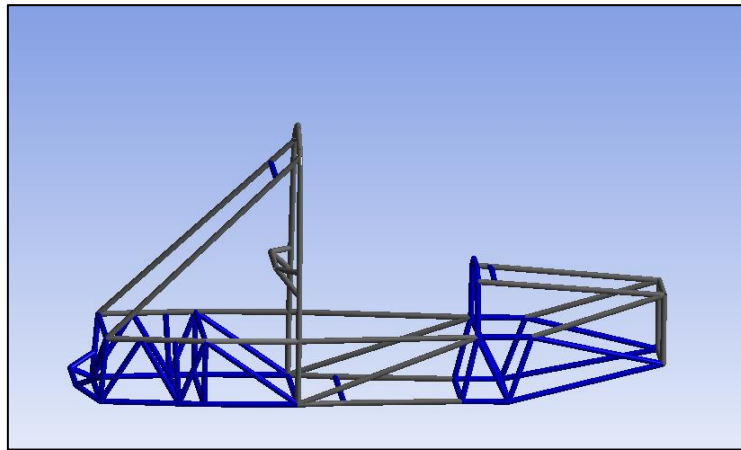


Figure 8: Aluminium frame sections iteration 1

After further iterations, it was found that the best compromise between torsional stiffness and mass was given by the design shown below where just the frame members behind the rear suspension mounts were replaced by aluminium tubing. This is because these frame members do not appear to affect the torsional stiffness as much as the frame members between the front and rear suspension uprights. The FEA analysis for this design produced a torsional stiffness of 1107 Nm/deg and a mass of 28.83 kg, saving 4.19 kg at the cost of 124 Nm/deg torsional stiffness.

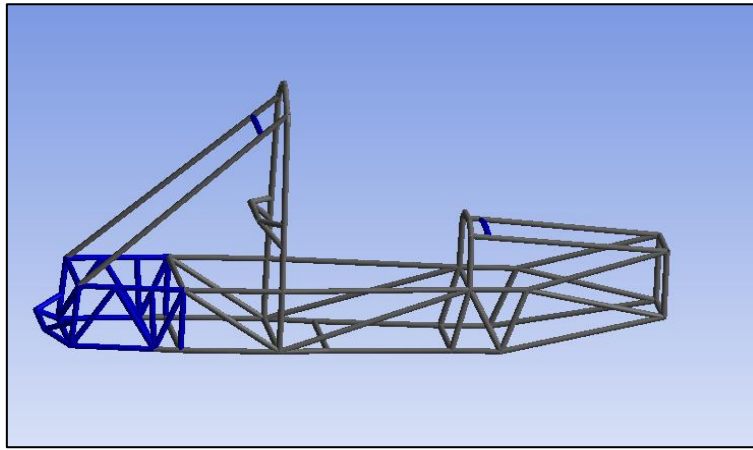


Figure 9: Aluminium frame sections final iteration

However, further research revealed that most of the higher strength aluminium alloys, including 7075 aluminium, cannot be welded. This means that mechanical fastening would be required which adds complexity and cost to the design as fastening all the joints would require multiple different bracket designs. The number of brackets required would also add mass to the frame and likely outweigh the mass saved through the use of aluminium tubing. In light of this, the analysis was repeated using 6063 aluminium which has lower strength properties but can be welded. This change had only a minor effect on the torsional stiffness and mass results, giving values of 1107 Nm/deg and 28.74 kg respectively. Because of the lower strength properties, 6063 aluminium costs between £1.44 per kg and £1.51 per kg of raw material, according to the CES Materials Database, which is roughly half the price of 7075 aluminium and therefore reduces the cost of the frame. Although the aluminium frame members could be welded together, the problem of joining the existing steel frame members to the aluminium remains. Welding steel to aluminium is possible but may be more expensive to do and whilst it is not against the rules, testing would have to prove its equivalency to a welded steel joint which may not be achievable. It is therefore more likely that mechanical fastening would have to be used for this which, as mentioned previously, adds complexity and mass to the design, particularly at the rear of the frame where there are more frame members meeting at nodes.

. For the purpose of the FEA analysis, a single 1 mm layer of woven epoxy carbon prepreg was used for the shear panels as this is readily available from composite suppliers. Again using the torsional stiffness of the steel frame, shown in the table as iteration 0, as a bench mark, shear panels were first added to the side impact structure and around the front roll hoop supports as these are the largest open spaces in the frame and therefore the most flexible. This increased the torsional stiffness from 1231 Nm/deg to 1725 Nm/deg with a mass increase of 1.24 kg, bringing the frame closer to the torsional stiffness target of 1750 Nm/deg set by the team.

Prior to iteration 2 in the table, the total deformation map of the steel frame, shown below in Figure 10, was examined. Shear panels were then placed only in the regions showing the highest levels of deformation. The FEA analysis for this iteration showed a comparatively small torsional stiffness increase of 16 Nm/deg, indicating that adding shear panels to the largest open spaces in the frame offers the best increase in torsional stiffness. Analysis of iteration 3 was then carried out to determine the effect of using shear panels throughout the whole frame with just the rear of the frame left unchanged as not to interfere with driveshaft. This gave a torsional stiffness of 1840 Nm/deg for a mass increase of 2.21 kg from that of the steel frame. However, adding shear panels to the rear of the frame restricts access to powertrain and systems components so the analysis was repeated without any shear panels rear of the main hoop. This produced a torsional stiffness of 1813 Nm/deg which is still higher than the target of 1750 Nm/deg so it was decided that the rear shear panels would not be considered any further.

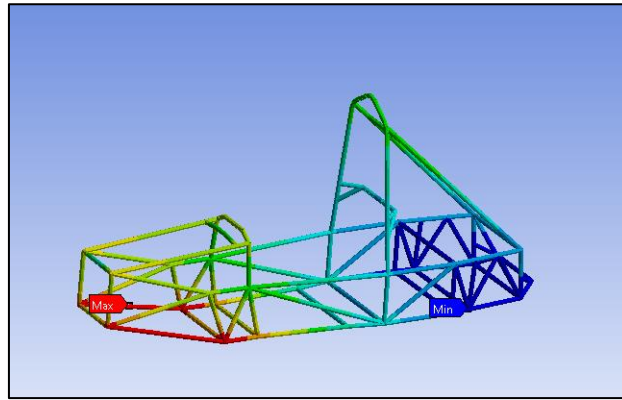


Figure 10: Total deformation map of steel frame

Further iterations were then carried out investigating the effect that the various shear panels have on the torsional stiffness. From which it can be concluded that the shear panel on the top surface of the frame between the front roll hoop and front bulk head is the most effective at adding torsional stiffness to the frame, followed by the side impact structure shear panels. Despite this, iteration 4 still gave a torsional stiffness 158 Nm/deg higher than iteration 9 for a mass increase of 0.98 kg. It was therefore decided that iteration 4 would be the best option if shear panels were to be implemented into the chassis frame design.

After the analysis on alternate frame rules had been carried out, it was revealed that carbon fibre cannot be used on the top surface of the frame between the front roll hoop and front bulkhead. This is because carbon fibre is electrically conductive which means it can interfere with signals from some of the systems components in the car. Analysis was therefore carried to determine the effect of replacing that shear panel with glass fibre which would not interfere with systems components. The results produced showed a torsional stiffness of 1788 Nm/deg, 25 Nm/deg down from iteration 4, for a mass increase of 0.136 kg. Another option that should be investigated is using a smaller carbon fibre panel only extending 400 mm rear of the front bulkhead so as not to interfere with systems components. However, it is thought that using glass fibre would offer higher torsional stiffness.

However, it must be noted that the FEA analysis considers the shear panels joined flawlessly to the steel frame members and therefore produces best case scenario torsional stiffness results which are unlikely to be achieved in practise. The shear panels could be bonded to the steel frame members using an epoxy-based adhesive which are available to purchase from easycomposites.co.uk at a price of £12.20 and are suitable for bonding metals to carbon fibre composites (easycomposites, 2018). The problem with this method is that, because the shear panels are 1mm thick, there may not be enough surface area to effectively bond to the steel frame members. This means the shear panels may need to be manufactured to give the edges more surface area, as shown in Figure 11 below. This could be done by making a mould from a flat sheet of aluminium with strips of aluminium bonded at right angles to the edges.

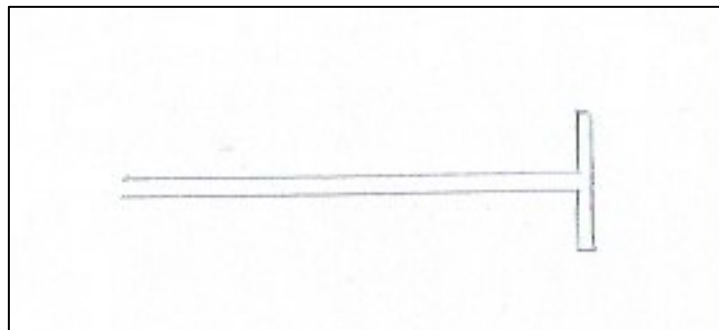


Figure 11: Shear panel edge concept

An alternative to using an adhesive would be to use tabs welded onto the frame, allowing the carbon fibre shear panels to be bolted into place. However, further analysis would need to be carried out on this as the use of tabs and bolts may lead to stress concentrations and therefore reduce the effectiveness of the shear panel. Figure 12 shows a drawing of this concept.

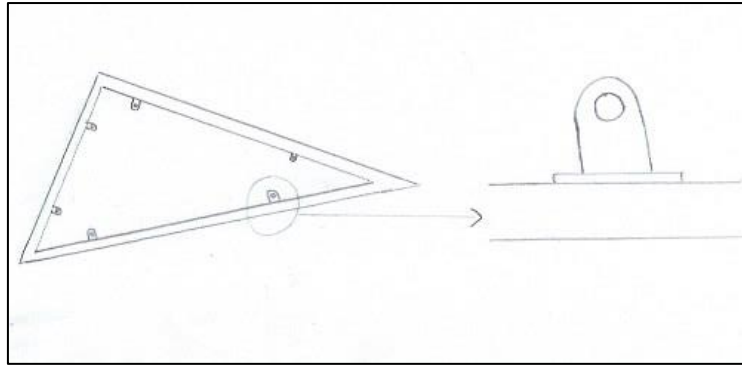


Figure 12: Shear panel tab concept

Manufacturing the shear panels from epoxy carbon prepreg is done by applying the material to a mould, which could be a piece of aluminium cut to the right dimensions, before vacuum packing the mould with the applied epoxy carbon and then curing it in an oven or an autoclave (easycposites, 2018). Because of the simple shape of the shear panels, they could be manufactured by Team SURTES with the use of the facilities at Snapdragon. There are many other options for the materials and layup of composite shear panels but choosing a readily available composite such as woven epoxy carbon would reduce the cost as well as the timescale for the manufacture.

Under the alternate frame rules, composites may also be used in the primary structure provided they are physically tested to show equivalency to steel tubing. This means that carbon fibre shear panels could be used to save weight in the chassis frame by replacing the steel tubing used for triangulation both within and outside of the primary structure. An FEA model of this concept is shown in Figure 13 where, as before, the rear of the frame has been left unchanged for the driveshaft. The analysis produced a torsional stiffness of 1408 Nm/deg and a mass of 31.25 kg compared to the values of 1231 Nm/deg and 33.02 kg for the steel frame. This demonstrates that the use of shear panels instead of triangulation can add torsional stiffness to the frame whilst also decreasing the mass. However, for the analysis, 1 mm thick shear panels were used but physical testing may show that these would not meet the minimum requirements to be used in the primary structure. In this case, a thicker composite, possibly with a honeycomb core, would have to be used which would add mass. The FEA analysis was therefore repeated using a honeycomb sandwich structure, consisting of 5mm honeycomb with 1mm woven epoxy carbon bonded to each side. This gave a torsional stiffness of 1866 Nm/deg and a mass of 34.07 kg. Although this would give a mass 1.05 kg higher than that of the steel frame, the torsional stiffness is significantly higher. If honeycomb composite shear panels were to be used in future frames, the frame would need to be designed with this in mind because, unlike epoxy carbon, honeycomb sandwich panels cannot be moulded into any shape.

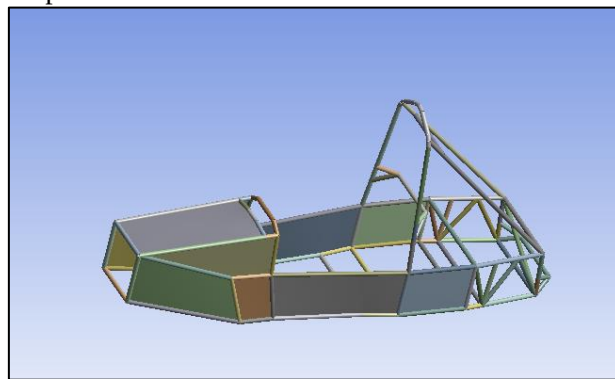


Figure 13: Triangulation shear panels

3.5 Conclusions and Further Improvements

Because of the loss of torsional stiffness and the complications arising from joining aluminium to steel, it was decided that it would not be worth attempting to use aluminium frame sections to reduce the mass of future chassis frames.

As shown by the FEA analysis, the addition of shear panels forward from the main roll hoop offers a torsional stiffness increase of up to 582 Nm/deg for a mass increase of 1.89 kg. This concept should therefore be considered for the SE4 frame as it could be implemented after manufacture and would bring the torsional stiffness of the frame above the 1750 Nm/deg target set by the chassis and aerodynamics team at the start of the design process; although coordination with the other sub-teams would be required to ensure that the shear panels do not interfere with systems components or obstruct tab locations.

The use of shear panels to replace frame triangulation should also be considered for future frame designs as FEA analysis has shown significant improvements in torsional stiffness compared to the triangulated steel frame and also a possible reduction in mass depending on what materials are used for the panels. However, further analysis and physical testing would be required for this because any panels used in the primary structure must be equivalent to the minimum material properties stated in the formula student rules. Based on research throughout the process, it would also be recommended that FEA analysis is carried out on monocoque frame designs for future cars so that the performance advantages can be more accurately compared to the practical disadvantages and therefore allow the best decision to be made on the type of chassis to design.

4 ACCUMULATOR ATTACHMENT (Sean Glendinning)

4.1 Design Options Considered

As discussed in the PDR, there were three initial design considerations when choosing the accumulator attachment system final design. An in-depth analysis into weight, modularity, cost and whether they meet the regulations was discussed in the PDR to determine the best design going forward is the bracket method.

4.2 Constraints Considered

The initial design conditions set were to develop an accumulator mount that reduced the mass of the 2017 design from 5.5 kg to 2 kg and to allow multiple batteries that have different weights and dimensions to be mounted. Additionally, the FSG rules require the mount to withstand forces of 40g, 40g and 20g independently in the lateral, longitudinal and vertical directions respectively (FSG, 2017). The heaviest accumulator considered weighs 78 kg so the FEA tested will be designed to withstand the forces resultant to 78 kg at 40g.

4.3 Changes since the Initial Design

Clarifications of the rules and changes to coinciding components has resulted in changes to the initial design concept. Reviewing the rules with the FSG's questioning body showed an error in their structural equivalency spreadsheet (SES) form. The required force for a bracket to be able to withstand decreased from 125 kN to 5 kN to pass inspection. Additionally, reviewing the rules showed that the forces applied on the mount do not affect the chassis and consequently when modeling in FEA the chassis beams can be simulated as having infinite stiffness. A rule has been added so that all brackets require gussets to maintain safety. Additionally, the expected error for the welding of the chassis components are expected to be ± 1 mm. This error has been considered to ensure minimum requirements are met. To ensure the e-Powertrains team's components fit, the chassis dimensions have increased from the initial model. These changes resulted in longer structural beams being required and increased dimensions for the structural tabs.

4.4 Final Design and Manufacture

4.4.1 Accumulator Mount Design

The final design selection has a structural beam that is positioned due to the minimum size requirements of the Altertek's bracket. The design fits multiple batteries by changing the brackets that are unique to the individual's batteries size and dimensions. As discussed in the PDR the optimal removal method is through the underside of the vehicle. Therefore, there must be a minimum distance between the structural beams of 500 mm to allow the Altertek to pass through. FEA on the bracket for the Altertek accumulator resulted in minimum width of 2 mm to pass SES requirements. However, due to the ± 1 mm build error of the chassis the design will be built assuming a bracket size of 3 mm to ensure minimum dimensions are maintained. The bracket will be connected

to the batteries and structural beams by 20 evenly distributed M8 bolts along the bracket distributing the load and reducing lateral deformation. The applied forces result in large bending of the structural beams. This was resolved by adding eight structural tabs between the longitudinal structural beam and the structural bar.

The bracket was designed to mirror the bolt holes on the structural beams and its individual accumulator to determine the minimum size requirements as shown by Figure 15 (right). The FS rules require all brackets to contain gussets and be made out of at least 1.6 mm thick steel. A CAD simulation of the removal of the bracket using a spanner was created to illustrate the minimum spacing required to avoid clashing as shown in Figure 14 (left). The results were collected and the required spacing conveyed by Figure 14 (right).

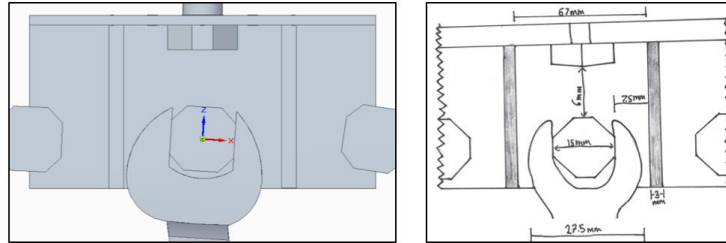


Figure 14: CAD simulation (left) and collected results from simulation (right) of bolt removal

FEA on the bracket, resulted in reduction of the brackets minimum width from 6 mm to 2 mm. Additionally FEA showed mass redundancies that could be removed by cutting triangular segments at stress low points. The fastener holes were changed from a circular shape to an elliptical shape to remove the static strain that would be caused by forcing bolts through misaligned holes. The initial design changes resulted in a mass reduction of 1.9 kg. The changes from the initial design concept can be seen in Figure 15. The bracket designs will be finalised after the mount is constructed to determine the required sizing to connect between the structural beams and the accumulator. The bracket will be milled from a solid block of steel in the Surrey Workshop resulting in an error of ± 0.01 mm. Engineering drawings have been created as shown in Figure 81 in Appendix 4: Engineering drawings.

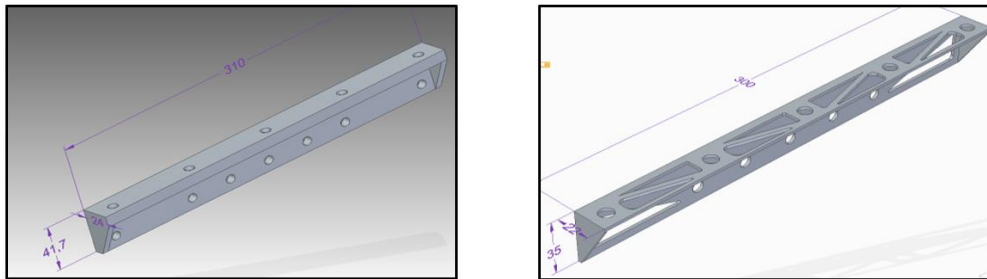


Figure 15: Design changes from initial bracket design (left) to final bracket design (right)

The structural beams consist of two steel bars welded to the chassis. The beams have 8 mm holes, separated by 70 mm to comply with the bolts simulation. Beam theory and FEA was carried out on the beams; loading them with 78 kg at 40g. To determine the minimum required cross sectional dimensions of the steel beam, beam theory calculations were conducted. The load of the accumulator was assumed as an evenly distributed load across the middle 200 mm of the beam to acquire an accurate bending moment. Fixed-fixed end conditions were used to simulate the welded attachment of the frame and convey the resistance of the bending moment. The maximum bending moment was calculated as shown by Figure 84 in Appendix 5.

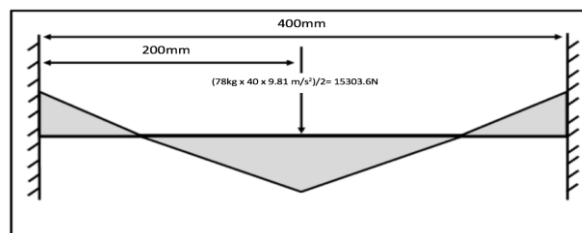


Figure 16: Load case and bending moment diagram of structural beams

A force of 15303.6 N was applied across the 200 mm surface, to show half the 78 kg mass at 40g. From the maximum bending moment, the cross sectional area (CSA) could be found to calculate the width required. The second moment of area for the beam was calculated using the equation for a rectangle. This was then used in Eqn 7 to calculate the maximum stress in the bar in MPa.

$$\sigma_{\max} = \frac{M_{\max} \times Y_{\max}}{I}$$

Eqn 7

$\sigma_{\max}$ is the maximum stress in MPa, $M_{\max}$ is the maximum bending moment, $Y_{\max}$ is distance from the neutral axis and I is the second moment of area of the cross section (Wood, 2005). Using the maximum stress in the beam value, the minimum width was determined to be 1.8 mm. The complete calculations can be found in Appendix 5. Additionally, to confirm the results FEA was carried out on the support beams. FEA showed that the beams passed the required loads at 2.5 mm thickness resulting in a mass reduction of 0.6 kg compared to the previous 10 mm thick bars. The beams will be welded to the chassis by Chris Isaac while the frame is being constructed. Engineering drawings have been made illustrating relevant dimensions as shown by Figure 82 in Appendix 4.

The aforementioned changes required the length of the structural strengthening bars to increase from 38 mm to 45 mm to maintain functionality as shown by the engineering drawing Figure 83 in Appendix 4. FEA on the initial assembly showed that the number of strengthening tabs could be reduced by half while still maintaining a safety factor (SF) greater than one. Figure 17 shows the final assembly of the accumulator attachment mount.

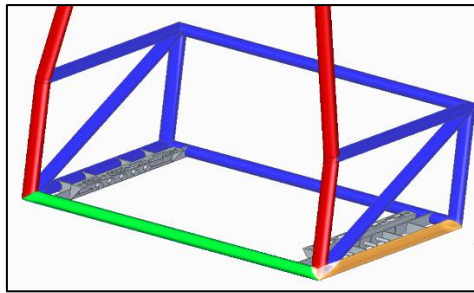


Figure 17: Final accumulator attachment system design

4.4.2 Accumulator Mount Positioning

As discussed in the PDR the battery needs to be mounted low and centrally to ensure a low centre of gravity and to reduce the effect of yaw. In addition, a full assembly of all components was completed to check obstructions with other components as shown by Figure 18. Wire locations were considered with a 30 mm minimum bending radius. The CAD design showed no collisions, so the mounting location was finalised.

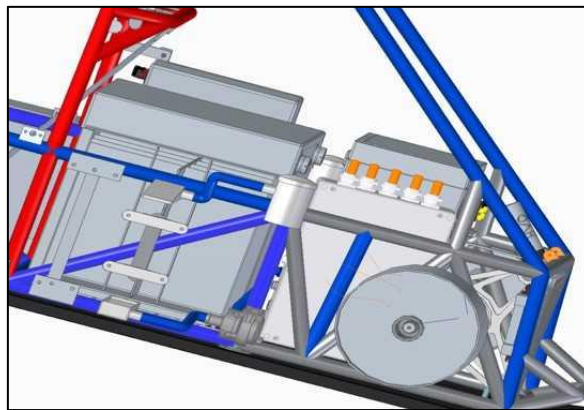


Figure 18: Clash check of components

4.5 Cost

Table 2 shows a cost breakdown of the Accumulator attachment mount components. This includes all costs for materials and manufacture, with a total cost of £225.64. The structural beams will be welded to the chassis simultaneously to its formation by Chris Isaac, while the bracket will be milled in the workshop in the University of Surrey.

Table 2: Cost breakdown of the accumulator attachment

Component	Material Cost	Quantity	Total cost
Structural beams	£12.37	2	£24.74
Structural Tabs	£2.15	8	£17.20
Altertek Bracket	£35.25	2	£70.50
Mingda Bracket	£53.20	2	£106.40
M8 Bolts grade 8.8	£0.34	20	£6.80
Total cost:			£225.64

4.6 Test conducted / planned

To ensure that the accumulator mount passes the scrutineering by FSG, FEA was conducted on the mount. The mount was simulated with the chassis beams having infinite stiffness and cylindrical supports. The design was modeled as half the full assembly due to the ANSYS license not allowing the required mesh size to simulate the full assembly. The forces were applied through the bolts to simulate the effect of the battery. Shear calculations were conducted on the bolts to check if they can endure the required loads, as shown in Appendix 5. The FSG rules require the mount to be able to withstand forces of 40g, 40g and 20g individually in the longitudinal, lateral and vertical directions respectively (FSG, 2017). Figure 14 shows the final design where the red beam is in the vertical axis (underside of the car to the top of the car), the green beam is in the lateral axis (from one side of the car to the other) and the orange beam is in the longitudinal axis (front bulk head to rear of the car). FEA will be conducted to determine the deformation and factor of safety of the assembly at the required loads. For the model to satisfy the FSG requirements, the factor of safety has to be greater than 1. The resultant forces on the final design resulted in a factor of safety of 1.7, 1.2 and 1.2 in the longitudinal, lateral and vertical direction respectively, as shown by Figure 19. The resultant design with the Altertek resulted in a total weight of 1.2 kg.

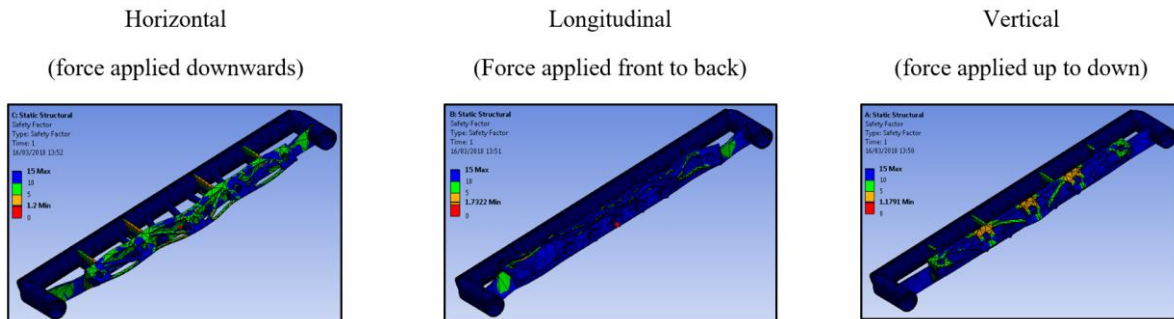


Figure 19: FOS results from FEA of half accumulator mount

In addition to the FEA, a model of the mount will be laser cut from chipboard to a model of the mounting system to ensure that it fits all the batteries and to ensure that it is functional.

4.7 Conclusions and Future Improvements

The main aim was to develop an accumulator mount that reduced the mass of the 2017 design to 2 kg and to create a design that worked with multiple batteries with different weights and dimensions. The final design weighs 1.5 kg which is 4 kg less than the previous design and can fit multiple batteries at a lower production cost. The new mounting system also resulted in an easier and quicker removal process. Overall, the new mounting system is an improvement on the previous model resulting in more efficient design.

In the future it would be beneficial to test using a high-grade polymer to construct the mount. The polymer would plastically deform upon impact resulting in a greater absorption of kinetic energy. The plastic deformation would result in a lower assembly weight. However, problems would occur with conducting FEA, as ANSYS does not simulate plastic deformation completely (Verse, 2010). Additionally, to pass FSG requirements the design would have to be reviewed to ensure that it complies with all the regulations.

5 IMPACT ATTENUATOR (IA) (Sean Glendinning)

5.1 Design Options Considered

As discussed in the PDR, two different materials were considered when designing the impact attenuator (IA). An in-depth analysis of the energy absorption characteristics, weight and cost were conducted resulting in an Aluminium honeycomb from Plascore being chosen. The honeycomb has a lower density and greater compressive strength at 7 m/s than the ROHACELL impact foam resulting in a more efficient attenuation of energy (Damian, 2017).

The honeycomb impact attenuator is adhered to an AIP by Loctite 638 High Strength Fast Cure glue. Aluminium and steel were initially considered to form the AIP due to their physical properties. A breakdown of each material's benefits was discussed in the PDR, resulting in a steel AIP being selected. Steel was chosen instead of aluminium despite it having a better strength to density ratio than steel (Wood, 2010). Also, aluminium requires a specialist to weld it to a steel chassis to prevent brittle intermetallic compounds being formed along the welds (Cooper, 2012). Consequently, to avoid this and mount the AIP to the front bulkhead eight M8 bolts would be required. The additional weight due to the bolts would negate the effect of the improved material properties of aluminium in comparison to steel.

5.2 Constraints Considered

The main constraints considered when designing the impact attenuator were the FSG requirements. These require the IA to be installed forward of the front bulkhead, be at least 200 mm wide and 100 mm high and be 200 mm beyond the front bulkhead. The AIP must be made from either 4 mm aluminium or 1.5 mm steel. When mounted on a vehicle weighing 300 kg and impacting a solid, non-yielding impact barrier with a velocity of impact of 7 m/s, the energy absorbed in this event must meet or exceed 7350 J. The deceleration must not exceed 20g average and 40g peak, which may result in the AIP deflecting no more than 25 mm from initial position (FSG rules, 2018).

5.3 Changes since the Initial Design

Reviewing of rules relating to the chassis design resulted in the initial concept being altered. These changes increased the dimensions of the front bulkhead from 300 x 300 mm to 300 x 360 mm. The increased size required a diagonal cross member to be added across the front bulkhead to prevent deformation. The initial design concept did not have a structural cross member. Therefore, to limit the deformation of the AIP, the base of the IA was increased to 250 x 200 mm to distribute the load applied to the AIP and to decrease its deformation. The addition of the cross member reduced the AIP deformation such that the IA base can be reduced to the minimum required set by FSG.

5.4 Final design and Manufacture

5.4.1 Design of Test Bulkhead and Manufacturing

FSG rules require physical testing to be carried out on two test impact attenuators to ensure it passes the required rules. The rules require the AIP to be attached to a structure similar to the chassis bulkhead and must extend a minimum distance of 50 mm away from the front bulkhead (FSG, 2018). The front test rig will be constructed by steel beam with an outer diameter of 25.4 mm. The cross member has a thickness of 1.2 mm while the rest of the bars have a thickness of 1.6 mm to simulate the chassis beams. The beams will be cut as shown by Figure 20, in the Surrey workshop using a XYZ260 machine. The bars will be welded by the Surrey workshop technicians using a combination of the provided engineering drawings and the CAD model shown by Figure 20. Subsequent to the bulkhead formation the AIP will be welded onto the tubing ensuring the dimensions of AIP, extend to the tubing's centre line.

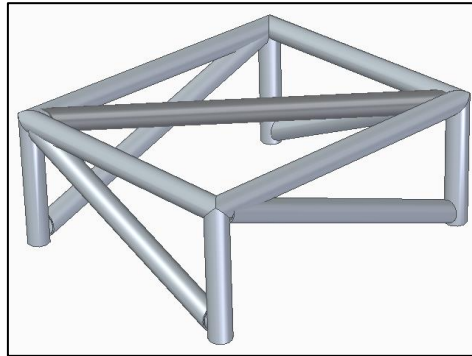


Figure 20: Test bulkhead CAD design for the Impact Attenuator

5.4.2 Design and Manufacturing of Impact Attenuator Material

The aluminium honeycomb impact attenuator from Plascore arrives pre-formed and pre-crushed as a rectangle with base dimensions of 200 mm wide and 100 mm high, extruded by 200 mm. The IA is glued to the centre of the AIP by Loctite 638 High Strength Fast Cure glue with a peel strength of 3 N/mm (Henkel, 2010) as shown by Figure 21.

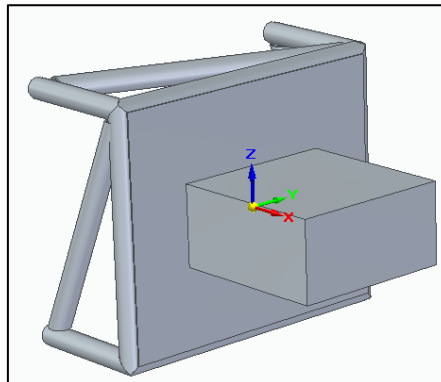


Figure 21: CAD model of Impact Attenuator final design

5.5 Cost

A cost breakdown of the individual components and the expected machining cost has been conducted and accumulated into Appendix 7. The Final expected cost is £474.80, which is £40 more than the 2017 car. However, the increase in cost is due to the change in the bulkhead sizing.

5.6 Tests Conducted / Planned

Theoretical testing was conducted on the IA material to determine if it conforms with the FSG requirements. To ensure that the material is able to absorb 7350 J, Eqn 8 was used.

$$E = \sigma V$$

Eqn 8

In Eqn 8, σ is the maximum stress and V is the volume of the IA material. Aluminium honeycomb has a compressive strength of 2.62 MPa, resulting in a minimum volume of $2.81 \times 10^{-3} \text{ m}^3$. To ensure an average deceleration of 20g and a maximum deceleration of 40g the corresponding max and average CSA were determined by Eqn 9.

$$F = m * a$$

Eqn 9

Mounted to a 300 kg vehicle resulted in a maximum force of 117720 N and an average force of 58860 N. These forces resulted in maximum CSA of 0.0449 m^2 and an average CSA of 0.0225 m^2 . To ensure that the AIP does not deform more than 25 mm, FEA analysis was conducted to ensure that the material and shape conform to the regulations as shown by Figure 24. The FEA resulted in the AIP having a maximum deformation of 17 mm, which is within FS regulations.

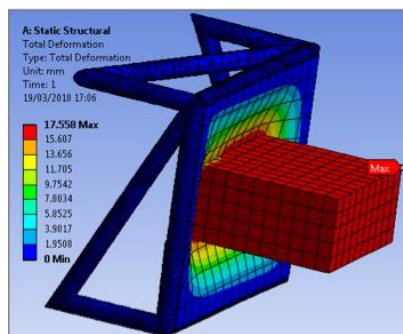


Figure 22: FEA conducted on Impact Attenuator to determine AIP deformation

In addition to FEA, physical testing will be conducted on the IA to ensure that it complies with the FSG regulations. Quasi-static testing will be conducted in order to verify FEA results, however shipping delays have prevented the testing report being produced before the CDR submission date. To test the impact attenuator, it will be placed in an Instron 1175 machine. A plate will be placed on top of it to distribute the load evenly and the machine will be set to $8.33 \times 10^{-3} \text{ m/s}$ (FSG, 2018). The machine will compress the IA taking information about the load, displacement and energy absorbed at even time intervals. The acquired data can subsequently be used to determine the peak and average energy absorption characteristics of the IA.

5.7 Conclusions and Future Improvements

Although no physical testing was conducted, the IA passes the theoretical calculations and the expected FEA deformation. This year's expected design resulted in a weight increase of 0.6 kg compared to last year's design. However, the increased weight is due to the increased sizing of the bulkhead and the weight of the IA material is 0.2 kg less than the previously used ROHACELL IA.

This model could be improved by eliminating the need for a cross member structural beam by using a different material to form the IA with a higher compressive strength and a higher density than aluminium honeycomb. Other teams, for example UCL, use high density foam and carbon fibre to reduce the deformation, however budgetary constrictions prevented this from being a viable option to construct the IA. (Wyatt, 2015)

6 DRIVER'S COCKPIT (SEAT + FIREWALL) (Shahrukh James)

6.1 Seat

6.1.1 Design Constraints

The 2017 Team SURTES car used a rigid seat in an attempt to reduce the mass of the firewall and provide more comfort to the driver. Experimentation at PDR stage and feedback from team SURTES technicians indicated that last year's seat was unable to fit drivers of various body stature. The goal for this year was to

provide a seat which was capable of accommodating drivers whose stature varies as per the 95th percentile male and 5th percentile female template (as required by FSG 2018 Rules).

The design constraints were to comply with restrictions on driver positioning which are stated below:

- A. Driver's head to be a minimum 50 mm from the line connecting top of the main hoop and front hoop.
- B. The distance between driver's hips and the pedal shall be a minimum of 915 mm.
- C. Shoulder harness should be a maximum of 20° below the horizontal.

These restrictions are displayed below in Figure 23.

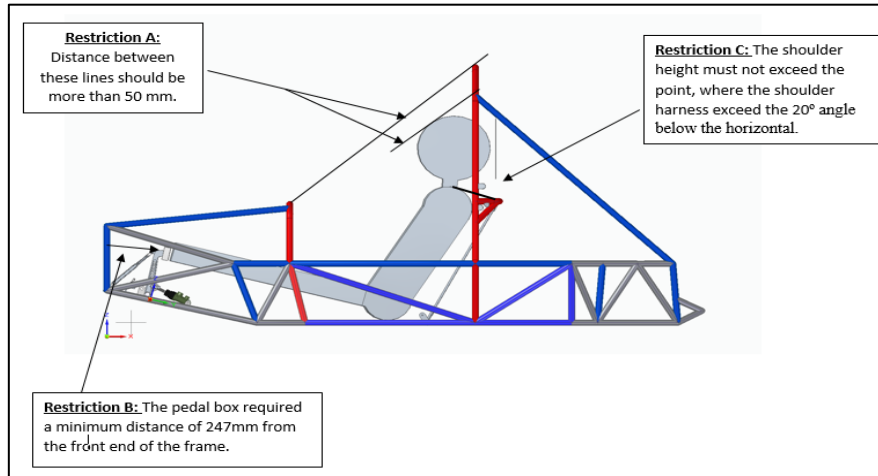


Figure 23: Restrictions on the driver's compartment that rely on the seat's dimensions

6.1.2 Design options considered and final design

Table 3 shows that the foam seat option was the most suitable considering it's lightest mass and the fact that the supplier can send a technician to ensure the seat is customised as required to fit the team SURTES chassis. The foam seat kit will be purchased for £45.50 excluding VAT, and the technician for customization will cost an extra £184.50 costing £230 in total.

Table 3: Design options for seat

Design Option	Conflict with constraints and/or Disadvantages	Advantages	Mass (kg)	Price (£)
Rigid Carbon Kevlar Seat	Mountable back frame increased driver's height.	Make the firewall non-load bearing and decrease it's mass.	2.54-3.6	382
Foam Seat with Fire Retardant Carbon X cover	No conflicts but no customization option and requires additional supports.	Structural firewall not required for the chassis.	2.15	300
Expanding Foam Seat	No conflicts.	Customized by a technician to fit this year's team SURTES chassis	1.54	230

6.2 Firewall

6.2.1 Formulae and constraints considered

At the PDR phase forces on the firewall were identified to be caused by the driver's upper body weight and the reaction force from the force exerted by the driver on the pedal box. In addition to these forces, the tabs were also to withstand the bump load caused by both the driver's and firewall's mass. The mass of the upper body for the heaviest driver (100 kg) was calculated to be 54 kg based on the approximation that upper body weight is 54% of the total mass (Exrx.net, 2018). The final firewall design weighs 2.06 kg. The resultant forces

acting on the firewall are shown in Figure 24. These forces were resolved based on the current angle of 65° between the firewall and the horizontal axis.

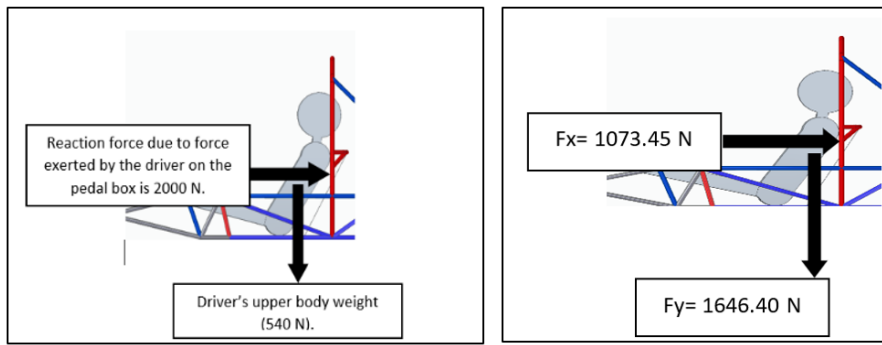


Figure 24: Forces on the firewall and its tabs (left), the resultant resolved forces (right)

6.2.2 Design Concepts and Analysis

Previously in the 2016 Team SURTES car the firewall had gaps in shoulder harness support region preventing the car to participate in the competition. The design goal for this year was to provide a firewall that is successful in completely sealing off the driver's compartment from the battery and tractive systems components.

Two design options were considered.

- The first design option was to build a non-load bearing firewall, which had the advantage of reducing the firewall's mass. This option required a load bearing rigid seat, as described in the seat section. The rigid seat, would increase the overall mass. It was also identified at the PDR phase, that the hollow structure of honeycomb makes it extremely light compared to a rigid seat, and therefore it's more beneficial to reduce the seat's mass.
- The second option was to make the firewall load bearing as discussed in PDR phase. This was also done in the 2017 SURTES car by employing a Nomex Honeycomb sandwich. Two concepts to employ the honeycomb sandwich panel are shown in Figure 25 below.

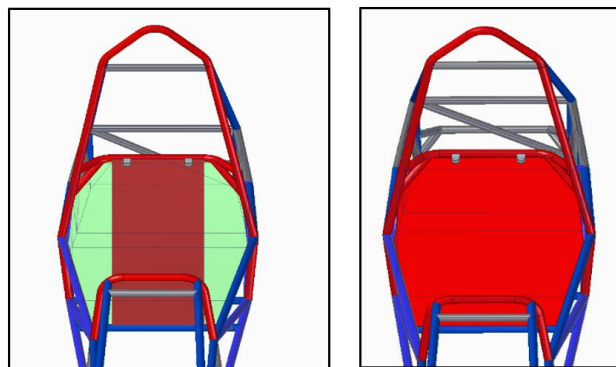


Figure 25: Honeycomb sandwich panel (red region) reduced to driver's back area (left), honeycomb panel (red coloured) will cover the entire composite firewall (right)

For mass savings and manufacturing ease, the design on the left was chosen. FEA was only conducted on the sandwich honeycomb panel and not the composite firewall, as the sandwich honeycomb panel will cover the load bearing region of the firewall. The FEA setup is shown in Figure 26. Forces applied in this setup are the same as discussed in the Formulae and Constraints considered sections.

The sandwich honeycomb assembly consisted of two aluminium plates of 0.5 mm thickness on each side of the honeycomb, forming a sandwich panel as shown in Figure 27.

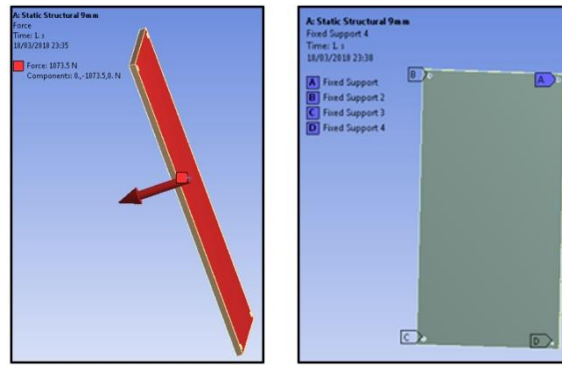


Figure 26: FEA setup in ANSYS workbench. Load applied is shown (left) and fixed supports in the four holes (right)

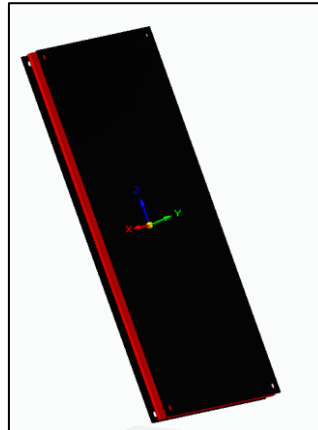


Figure 27: Exploded view of the honeycomb sandwich panel. Black layers are aluminium and honeycomb is in red

The honeycomb materials simulated for the above setup are shown below along with their results, materials were tested in three different thicknesses in which they are manufactured of 6 mm, 9 mm and 12 mm.

Table 4: Honeycomb materials analysis results for the given load cases

Material	Layer Thickness (mm)	Foil Thickness (microns)	Deformation (mm)	Factor of Safety	Max Principle Stress (MPa)	Equivalent (Von Mises) Stress (MPa)	Mass (kg)
Al Compocel	9	70	1.74	1.50	110.52	122.74	0.59
Drilled Aluminum	9	70	2.00	0.65	197.65	191.6	0.58
Nomex Commercial Grade	9	70	3.00	2.13	209.62	221.41	0.54
Polycarbonate	9	70	2.35	0.22	209.63	191.6	0.61
Al Compocel	6	70	3.79	0.76	185.09	221.33	0.56
Drilled Aluminum	6	70	3.95	0.56	238.26	221.68	0.55
Nomex Commercial Grade	6	70	4.20	1.85	185.16	221.68	0.53
Polycarbonate	6	70	4.00	0.19	185.16	221.42	0.58
Al Compocel	12	70	1.04	2.33	69.88	72.86	0.62
Drilled Aluminum	12	70	1.65	0.97	69.94	73.02	0.61
Nomex Commercial Grade	12	70	2.60	5.60	68.29	72.86	0.56
Polycarbonate	12	70	1.85	0.59	68.29	72.86	0.66

As per requirement of 1.5 factor of safety two options were potential solutions, Al Compocel 9 mm and Nomex Commercial grade 6 mm, due to higher factor of safety and lower mass Nomex Commercial grade 6 mm honeycomb was chosen as the final material.

The final composite firewall panel behind the driver will have the following sections as shown in Figure 28. The panels will be sealed with a Methacrylate Adhesive already available in the workshop. Thicknesses of the Aluminium and FR4 Epoxy layers are kept as required by the FSG 2018 rules.

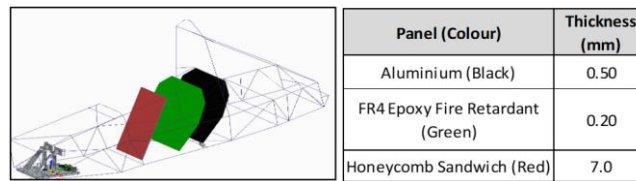


Figure 28: Three layers of the firewall are shown in exploded view (left) with the respective layer thicknesses (right)

As per the regulations, the firewall was required to cover any element at a height less than 100 mm from the bottom of the driver's helmet. The first design option was to extend the existing firewall upwards however, this was an expensive option given that we had to ask the supplier to shape the firewall and buy extra firewall material. The second option was to over the driver's helmet area with a 0.5 mm thick aluminium plate covered in fire retardant powder coating (Resicoat EL). The model for this design is shown in Figure 29 below.

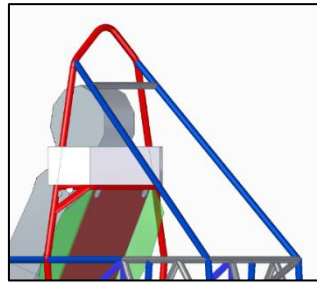


Figure 29: Aluminium plate to cover the driver's helmet region up to a 100 mm height

The only space left to cover was the gap in the shoulder bar region. This gap and its design solution are shown in Figure 30.

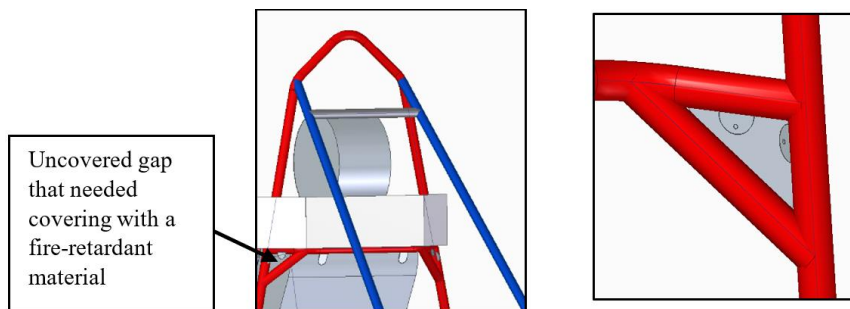


Figure 30: Uncovered gap (left) and the design solution to cover the gap (right)

The design solution proposed above, was employed in the 2016 SURTES car shown in Figure 31, with the concept to cover the gap with a curved aluminium triangular plate, this plate is covered in a fire-retardant paint on the outside and for this year's car, the plate will be covered by an epoxy powder (Resicoat EL) with UL 94 (V-0) fire rating (requirement of FSG rules 2018). Resicoat EL is already available in the Design Centre and the team's technicians have experience in applying this, making it convenient and an economic option.

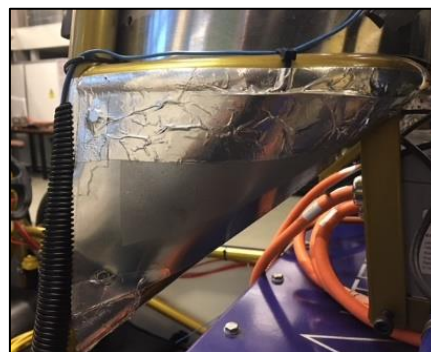


Figure 31: 2016 SURTES car triangular firewall plate

6.2.3 Attachment Method

Firewall tabs were initially designed to mount the firewall, the dimensions were chosen according to the material available in the workshop (3.18 mm thick). Forces acting on the tab, were calculated in the Constraints and Formulae considered section, these were applied in FEA on the tabs (see Appendix 8). The FSG 2018 rules define the yield strength of welded steel to be 180 MPa, to maintain a safety factor 1.5 for this strength thickness was kept the same (3.18 mm) but length was changed to 40 mm. Hand calculations (see Appendix 8) were completed to check what size of bolt diameter was required to prevent failure due to shear, a minimum of 6 mm is required, and hence M6 bolts will be used. See Appendix 10, for the final tab dimensions.

6.2.4 Testing

As per the FSG 2018 rules a test has been planned to be carried out on various thicknesses of firewall material. The test is to drop a 4 mm wide screw driver at 250 N force on the firewall material, the sample with sufficient thickness to prevent penetration will be chosen as the final option. The test has been delayed due to delay in delivery time of test samples and the busy schedule of the Concrete Lab at the University of Surrey. The test samples and jig are available and once the test date is confirmed the test will be carried out. The test jig design can be seen in Appendix 9. FEA was carried to ensure no bending occurs when 250 N of point force is applied on the jig. The resultant bending was a minimal 0.1 mm on the current jig design. (See Appendix 10).

6.2.5 Final Design

The final design of the firewall consists of 3 segments i.e. a composite firewall with sandwich honeycomb, triangle firewall plate and a firewall panel for the driver's helmet region. The complete final design is shown in Figure 32. The total mass breakdown of the cockpit region is given in Table 5, compared to last year's design it weighs 53.7% lighter. Last year's design weighed 4.66 kg (Bayatti, et al., 2017).

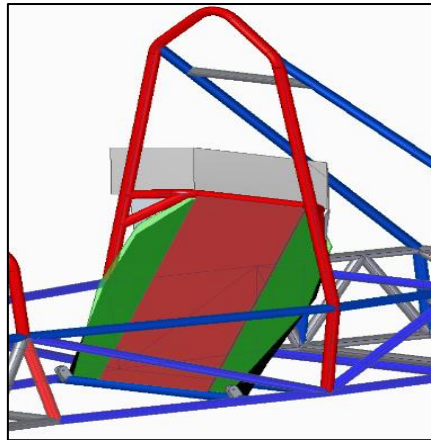


Figure 32: Final design of the complete firewall

Table 5: Total mass of the seat and firewall

Panel (Colour)	Mass (kg)
Aluminium (Black)	0.890
FR4 Epoxy Fire Retardant (Green)	0.640
Honeycomb (Red)	0.530
Tabs	0.240
Bolts	0.220
Seat	1.540
Head Restraint Cover	0.165
Triangle Firewall	0.003
Total	2.060

6.2.6 Cost Breakdown

The cost breakdown of the firewall design is detailed in Table 6. Aluminium layer required costs £20 excluding VAT, but since it's available in the University of Surrey workshop it will be free of cost for Team SURTES.

Table 6: Cost breakdown of the firewall design

Component	Price (GBP)	Supplier
Epoxy FR4 Firewall	40	GaaTech
Nomex Honeycomb	55	Cel Components
Aluminium Layer	0	University Workshop

7 DRIVER RESTRAINT (Shahrukh James)

7.1 Head Restraint

7.1.1 Design Concepts

The design concept that was proposed at the PDR stage required 3 steel plates for its mounting and an additional circular tube at the top to be mounted to the chassis. After the aero team analysed the design, the plates and the circular tube were causing turbulent flow on the rear wing and hence causing a decreased downforce. At this stage new design concepts were needed.

The area where the head restraint is attached to the chassis is shown in Figure 33. The aim was to introduce a design where we would not need an additional circular tube in the main roll hoop support braces, this is the grey tube in Figure 33:

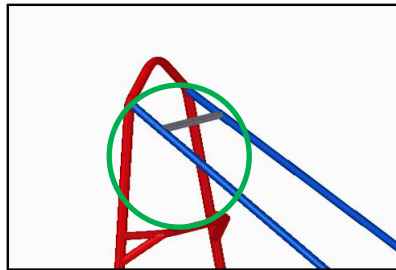


Figure 33: Head restraint is to be installed in the region above the shoulder bar shown in green

The new design that was considered, was to use steel plates in a box section which would help in eliminating the circular tube at the top. This is a design currently being used by the University of Michigan formula student team. However, this design had to be varied to fit this year's Team SURTES frame. The design is shown in Figure 34.

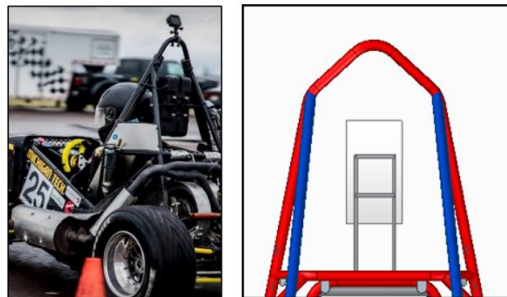
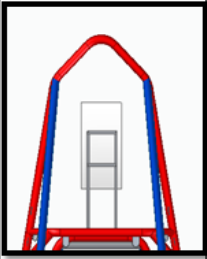
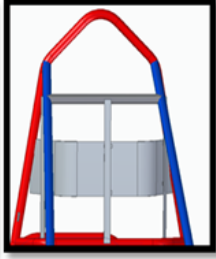


Figure 34: University of Michigan Head restraint design (left), optimized design to fit the Team SURTES frame (right)

7.1.2 Testing and Design Analysis

The FSG Rules 2018 stated that the head restraint must be able to withstand 890 N, applied in the reverse direction. Ansys Workbench was used to carry out the FEA on the two design concepts. Table 7 shows the comparison of various designs.

Table 7: Head restraint design analysis

Design Name	New Design	Initial
Design Concept		
Mounting Requirements	Box section made up of 4 steel plates welded on shoulder bar. No circular tube needed at the top.	3 steel plates needed which are welded on shoulder bar and an extra circular tube at the top.
Airflow on Rear wing	No circular tube and/or plates needed at the top and hence least interference with the airflow on the rear wing.	Three plates reaching to an extra circular tube at the top. Maximum turbulence in the airflow.
Total Mass (kg)	0.47	1.24
Factor of Safety	1.13	1.17

The new head restraint design has a lower mass and satisfies the required SF of 1.10. Hence, this was chosen as the final head restraint design.

7.1.3 Final Design

The final design of the head restraint will be the box section head restraint design. This design's details are tabulated below in Table 8 along with a picture on how it will be mounted on to the chassis in Figure 35.

Table 8: Components of the head restraint design

Component	Mass (kg)
Aluminium Plate	0.171
Box Section Steel Supports	0.171
M6 Nuts and bolts	0.134
Total	0.476

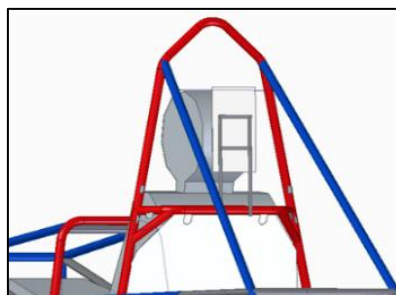


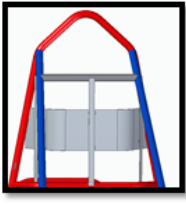
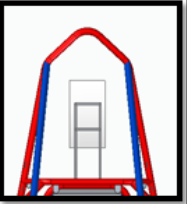
Figure 35: Final head restraint design assembly

Last year's design had a weight of 0.695 kg (BAYATTI et al., 2017). This year's design at 0.476 kg weighs 31.5% lighter than previous years.

7.1.4 Configuration Management List and Changes since Initial Design

The changes from initial design along with the reason for change are detailed in Table 9.

Table 9: List of changes made to the initial design and their reasons

Design	Initial	Final	Reason for change
Picture			At the PDR stage mass optimization and airflow on the rear were not being considered, after the aero team's concern with the airflow a new head restraint design was needed. The final design provides better airflow to the rear wing and weighs 0.767 kg lighter while having a safety factor of 1.13.
Components	1mm thick curved steel plate 3 steel plates of 0.5 mm thickness with a circular tube at the top.	1mm thick Aluminium plate with 2.5 mm thick steel plates for support.	
Benefits	Only steel required to manufacture.	Lighter in mass and better airflow on the rear wing.	

7.1.5 Cost Breakdown

The FSG 2018 rules require padding of the head restraint by a material that is Specification Internationale de l'Automobile (SFI) 45.2 approved. This will need to be purchased along with self-adhesive Velcro, to attach it to the head restraint. All other material required is available in the workshop at the University of Surrey and will be free of cost, with the detailed cost breakdown below in Table 10.

Table 10: Head restraint cost breakdown

Component	Price (£)
Self-Adhesive Velcro Tape (25mm x 25 Metres Loop Tape)	18.33
SFI 45.2 Approved Foam Sheet- Lightweight - EIS-W18	35
Total	53.33

7.2 Harness System

Table 11 shows the 6-point harness options for the driver which qualify for the SFI 16.5 specification, these have been narrowed down after acting various suppliers. Williams club harness was chosen for its lighter mass and cheaper price.

Table 11: Six Point Harness options that satisfy the requirements set by FSG Rules 2018

Product Name	Mass (kg)	Price (£)	Delivery Time (days)
Williams Club 6 Single Seater Harness	2.05	147.29	3-4
Sabelt Formula Harness	2.11	245.91	7-8

7.2.1 Mounting Tabs and Analysis

The dimensions of last year steel tabs were used as a starting point, these tabs were then optimized to ensure they will satisfy this year's requirements as per the FSG 2018 rules and the team's chassis. The final dimensions chosen after FEA on tabs are shown in Table 12.

Table 12: Design requirements set by FSG 2018 rules vs Current Shoulder Tab Dimension

Tab Type	Quantity	FSG 2018 Required Cross sectional Area (mm ²)	Final Design Cross sectional Area (mm ²)	FSG 2018 Required Thickness (mm)	Final Design Thickness (mm)
Shoulder Harness	4	60	80.00	1.60	3.18
Submarine and Lap Harness	4	90	93.98	1.60	3.70

The submarine and shoulder harness tabs were to withstand 19.5 kN and 13 kN respectively. FEA was done on tabs of various thicknesses. Table 13 shows the results, safety factor was calculated (minimum advised safety factor was 1.3) while keeping 180 MPa (strength of welded steel) the reference, as per the FSG 2018 rules. Final tab thickness was 3.18 mm and 3.7 mm for shoulder and submarine harness tabs respectively.

The current harness tabs are sandwiching the harness attachment, and creating a double shear loading condition (Figure 18). Hand calculations (see Appendix 10) were completed to ensure failure does not occur due to shear. After considering results shown in Figure 36, M6 bolts were decided to be used for the tabs, as they are also the standard size bolts being used. The final tabs are shown in Appendix 10.

Table 13: FEA analysis of tabs

Tab Type	Thickness (mm)	Safety Factor	Maximum Principle Stress (MPa)
Shoulder Harness	3.18	1.38	116.00
Shoulder Harness	2.20	0.95	166.14
Submarine and Lap Harness	3.70	1.43	118.36
Submarine and Lap Harness	4.70	1.50	119.50

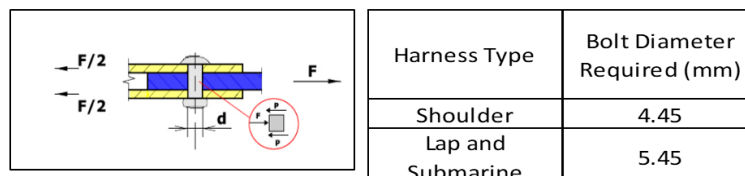


Figure 36: The double shear loading condition (Amesweb.info, 2018) (left) and the results of hand calculations (right)

7.3 Cockpit Conclusion and Potential Improvements

7.3.1 Firewall and Seat

The firewall currently weighs lighter than previous years, however further improvements can be done by considering a lighter material for the sandwich panel instead of aluminium layers. Carbon fibre is an ideal option however, this option has to be assembled in house as most manufactures don't provide honeycomb with carbon fibre sandwich panels. The seat can be improved by considering to design and make a carbon fibre seat that incorporates a firewall, this will highly reduce the overall mass of the required firewall and seat. A similar

process has been used by Brown University (Team-BHP.com, 2018). However, the reliability of the manufacturing process and our workshop facility has to be taken into account.

7.3.2 Head Restraint

The current head restraint design has been implemented for the first time in team SURTES, which does not require an additional circular tube at the top to be mounted. This will improve airflow and decrease mass. However, further improvements can be made by looking into lighter materials such as carbon fibre and aluminium, however, welding aluminium supports on a steel chassis can be expensive.

8 FLOOR (Himanshu Mattoo)

8.1 Introduction

The floor of the car serves two main purposes – load bearing and aerodynamics, which improve quality of flow to the diffuser. Loading bearing sections of the floor are defined as areas where the floor would experience loads such as the driver and equipment weight, and pedal reaction forces, while the aerodynamic aspect is defined as the floor needed to cover areas under the chassis to improve flow under the car and into the diffuser and will not be experiencing any loads. The floor design for the current was split into 3 sections – front, middle and rear, followed by the diffuser as shown below in Figure 37.

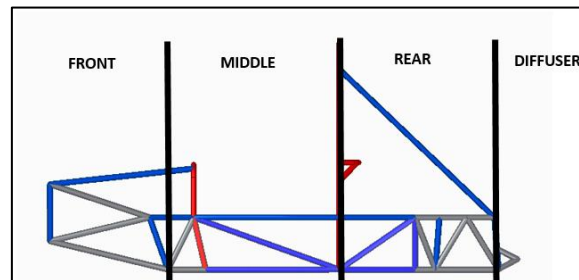


Figure 37: Floor sections

Furthermore, the floor is designed to be light weight but structurally capable to experience the loads under severe driving conditions such as driving over bumps and chassis vibration simultaneously ensure unobstructed flow to the diffuser.

8.2 Design Options and Possible Improvements

8.2.1 Previous Designs

Due to lack of documentation regarding the previous floor, a physical examination and analyses of the previous year's designs was conducted, and potential design improvements were noted. Previous years design was a composite-sandwich panel had a thickness of 12 mm and were bonded together. This floor design was only structural which meant it was located only in the front and mid-sections of the chassis while the rest remained uncovered as there was no diffuser used. The weight of the front and middle section was a total of 6.9 kg. This design was further optimised to reduce weight, material use whilst maintaining required strength.

8.2.2 Floor Set Goals

Aim during designing the floor was to maximise the area covered under the chassis to ensure better airflow quality to the diffuser and avoid water ingress or water splatters to come into the rear that holds the battery and other electronic equipment while enduring loads and reaction forces under driving conditions with the front section able to withstand 2 kN force as required in the FSG 2018 rules as maximum force the pedals can withstand and mid-section being able to withstand drivers weight with a 3g SF. (Bheemaiah Meederira, 2018). Access to the battery bay was taken into consideration during designing the floor as it's vital to be able to access the battery bay quickly and easily. To minimise material use, the aim was to ensure the load bearing and non-load bearing floor act as a single entity.

8.2.3 Floor Design

8.2.3.1 Front Section

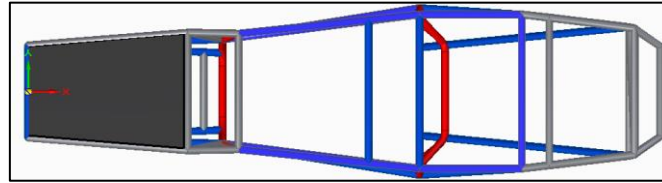


Figure 38: Bottom view of the chassis including the front floor

The front section of the floor as shown in Figure 38 above, will be tapered to fit in between the chassis tubes. This was done to allow bodyworks to be placed over. As mentioned in the previous section 8.2.2, the front floor was designed to experience reaction forces of a maximum force of 2 kN applied onto the pedals. After FEA iterations, (Section 8.3), the front floor section was designed to have composite panel with the new specification and materials to be as shown below in Table 14.

Table 14: Composition of front panel

Location	Material	Thickness (mm)
Top Skin	Carbon fibre woven	2
Core	Aluminium Honeycomb 70 microns	4
Bottom Skin	Carbon fibre woven	1

8.2.3.2 Middle Section

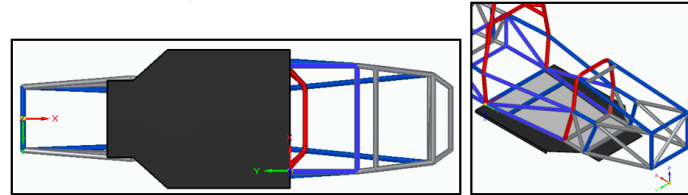


Figure 39: Bottom (left) and isometric (right) view of middle section of the floor

As shown in Figure 39 above, the middle section of the flushes under the chassis tubes followed by an extension sideward to maximise the volume of airflow passing through to the rear floor and finally to the diffuser. As shown in Figure 39 above in the isometric view, the grey zones in the middle floor have load bearing intends. After FEA in Section 8.3, the following floor specifications were selected as shown in Table 15:

Table 15: Composition of middle panel

Location	Material	Thickness (mm)
Top Skin	Carbon fibre woven	2
Core	Aluminium Honeycomb 70 microns	4
Bottom Skin	Carbon fibre woven	1

8.2.3.3 Rear Section

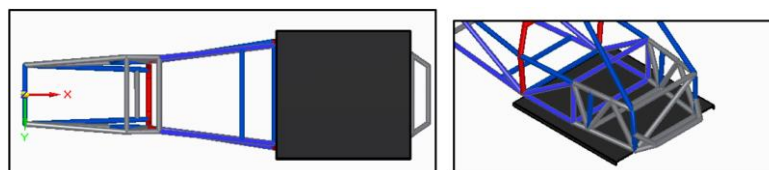


Figure 40: Bottom (left) and isometric (right) view of rear floor

Due to the rear floor, as shown above in Figure 40 being non-load bearing the floor was design to allow airflow from the middle section to flow by seamlessly into the diffuser, hence the outer shape is similar to the middle section. The rear floor was designed to be 1 mm thick.

8.3 Finite Element Analysis

Finite element analysis was conducted for all structural designs above on ANSYS-Workbench. The front section was tested for vertical forces and pedal reaction forces that it had to withstand. While the middle floor was tested for three times drivers weight and a heel test was conducted to imitate a step into the car with forces three times of the driver as shown in Figure 41 with the data recorded in Table 16. Driver's weight was assumed to be 75 kg.

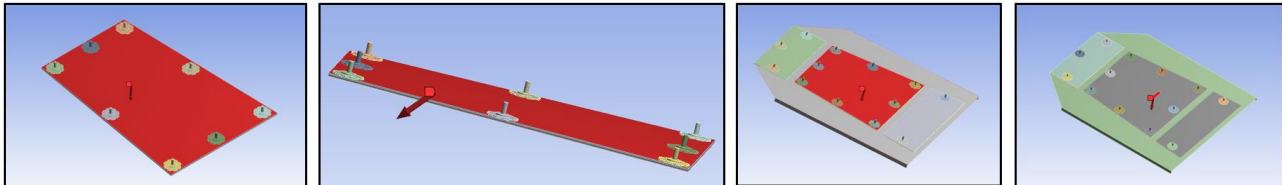


Figure 41: Four loading methods

Table 16: FEA results of load on floor

Section/Type	Net Load (N)	Deflection (mm)	Max. Shear (MPa)	Max. Equivalent (MPa)	Shear Safety Factor	Max Equiv. Safety Factor
Front/Vertical Loading	2000	0.58	23.2	23.1	1.21	1.58
Front/Angular loading at 70 degrees	2000	0.51	12.9	23.3	1.35	1.50
Middle-Central/Vertical Loading	2250	0.02	0.38	0.72	15	15
Middle-Shoe Test/Vertical Loading	2250	1.55	38.1	33.1	1.04	1.11

The shear and max equivalent stress safety factors were based of the epoxy (3M 9823) maximum strength for each. Despite the safety factor for the front panel being just over 1, it is deemed enough as 2 kN was specified in the FSG rules. The above data also includes optimised number of mounts required to minimise stresses near and around each mount. Further information regarding the mounting will follow in Section 8.4.

8.4 Mounting Method Used

Mounting methods used to mount the floor in the car was using carbon fibre clips which were then epoxied over the chassis tubes and onto the composite panel. This method was expensive due to the mould that was needed to manufacture the clips, and once the panels and clips were epoxied, they couldn't be removed from the frame which restricted any changes that could be implemented.

As the current car was designed to have a diffuser, it was essential to have quality airflow flowing under the car. Hence, the goal was to have a smooth surface under the vehicle which meant drilling holes through the floor wasn't feasible due to the rough surface finish as seen in previous cars.

As seen in Figure 42, Spida 53 mm M8 was chosen as it could be epoxied in-between the sandwich structure of the composite floor, at the locations shown, and there after the tab of the floor would be welded to the chassis, reducing alignment problems and ease of mounting the floor through the tabs. Mounting the Spida fixings within the sandwich structure provides 2 times the strength as stated by Spida Fixings Ltd. Further

details on the fixing can be found in Appendix 12. Due to the use of adhesive, stress is evenly distributed across the bond line while mechanical fasteners create stress concentration zones which lead to premature failure. Improved aesthetics of the final assembly with no bolt heads protruding out. Lastly, eliminates or minimises secondary operations like punching holes required for mechanical fasteners. (Design Guide for Bonding Metals, 2018)

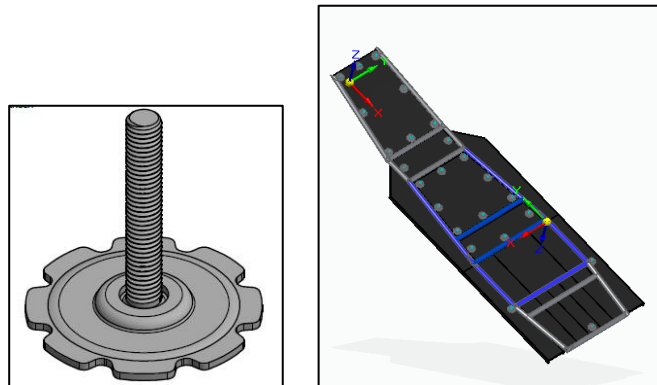


Figure 42: Spida 53 mm M8 (left) and number and position of mounts with related chassis tubes (right)

8.5 Diffuser

The diffuser's outlet angle remained unchanged, as mentioned in PDR section 3.3. However, the diffuser was split into two separate channels. This was done as per FSG supplementary rules for the jacking bar whereby the bar was to be clear for use from the back, top and bottom view. The length of the bar was required to be 300 mm. Hence the final design can be seen below, in Figure 43.

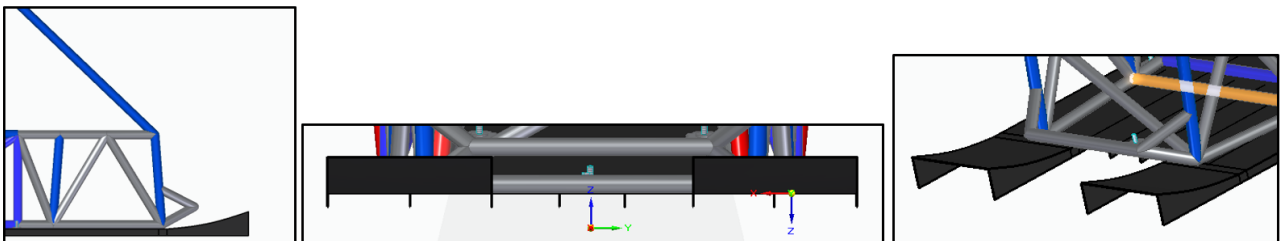


Figure 43: Diffuser Design from side (left), rear (middle) and isometric (right) views

To improve diffuser performance several techniques were implemented- adding skirts, vanes to channel the flow into the 2 channels and lastly the middle section was extended ahead to reduce a separation region as air came under the floor. To reduce turbulent airflow to disrupt the airflow under the floor, skirts with 20 mm height were added along the middle and rear floor.

A padding, as shown below in Figure 44, was placed under the floor to assist in not only separating the flow, but also further accelerating the flow through the rear until it approaches the diffusers. The mild taper on the floor shape was to help accommodate for airflow that would not be coming head on.

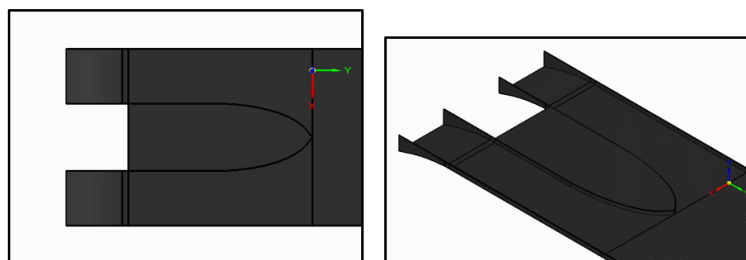


Figure 44: Padding to separate airflow into the channels

Lastly, the middle section was extended further ahead so it protrudes through the chassis, causing a region of high pressure on top and a lower pressure at the bottom with a small wake region results in increased downforce. The following can be explained with Figure 45, as shown below:

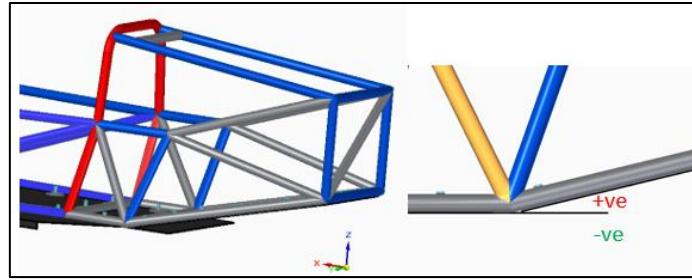


Figure 45: Middle floor extension

8.6 Manufacturing and Mould Preparation (Floor Panel and Diffuser)

Snapdragon was contacted for assembling the floor panel, the procedure for assembly was to cure the carbon fibre skin with the selected honeycomb with a peel ply around the panel and placed in a vacuum bag then in the autoclave.

The Spida mounting method quality is as good as the pre-treatment done for it. There are several methods of pre-treating the surfaces which include – mechanical, chemical and energetic. Further information on the methods can be found in Appendix 13. Method selection were based on variables such as cost, production, durability and feasibility. It should be noted that this process requires a clean environment. Peel ply surface preparation method was selected as it's the most commonly used method to treat composite materials. The peel ply works by placing a fabric material over the external surface of the composite and during the cure cycle parts of the matrix will flow and penetrate the fabric and later, post curing, becomes part of the laminate. The peel ply is removed before the area of the surface is required for bonding which produces a clean, rough surface finish whereby the adhesive can be applied.

8.7 Design Summary

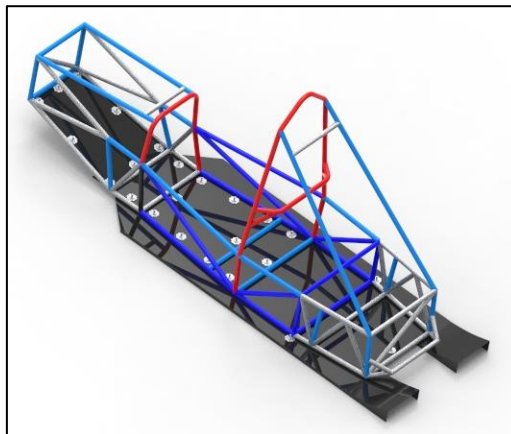


Figure 46: Complete floor and diffuser assembly

The floor was successfully designed to withstand the required loads. Figure 46, as shown above displays the complete floor and diffuser assembly. This design has a total weight of 4.2 kg including use of epoxy and the Spida fixings. This displays a weight drop of 39.8% and accommodates for maximised volumetric flow under the floor by covering the whole car in comparison to previous years design. Post-optimising the diffuser, the downforce performance improved by 6.22% in comparison to not having vanes and channel separation. The design also helps levelling the floor by using have the bottom panel flush along the lower plane of the chassis as shown in Figure 47 below.

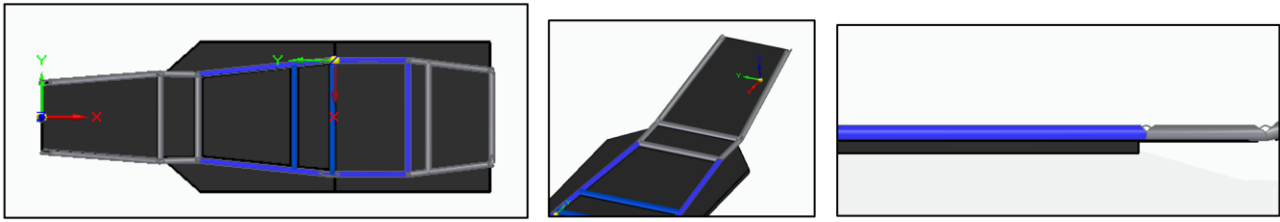


Figure 47: Complete floor layout along the bottom chassis tubes

8.8 Testing Plans

8.8.1 Floor

Due to the complexity of the floor design and mounting method, it would be beneficial to validate the FEA analysis and mode of failures by subjecting a sample section of the floor to physical testing – tensile, shear loads. Recording maximum stresses, the sample can withstand and validate against experimental data with the technical specifications of the panel skin, epoxy or the Spida fixing.

8.8.2 Diffuser Test Plans

Due to limited experimental data for diffusers at the speeds that are being used for optimising the diffuser for the current vehicle, an experimental setup is required to validate the design. Manufacturing a scaled down model of the diffuser whilst maintaining the Reynolds number in the wind tunnel. Using smoke visualisation and the force balance, flow separation and effective downforce produced can be measured.

8.9 Further Improvements

8.9.1 Floor

Multi-channel diffuser should be further investigated as they are relatively more efficient at lower speeds but due to time constrain could not be considered. To further increase downforce, lateral expansion of the diffuser should be investigated to maximise the downforce with the given regulations.

8.9.2 Diffuser

Multi-channel diffuser should be further investigated as they are relatively more efficient at lower speeds but due to time constrain could not be considered. To further increase downforce, lateral expansion of the diffuser should be investigated to maximise the downforce with the given regulations.

8.9.3 Computational Fluid Dynamics

Before conducting 3D CFD simulations, designs should be put through 2D CFD simulations and after analysing the results, should 3D simulations be run. This method would increase data reliability and dependency of the CFD setup. Validation of the setup can further be done against existing experimental data available in books and research papers.

9 BODYWORK (Adam Cox)

9.1 Design Options Considered:

As discussed in the PDR it was determined that a curved front bodywork was the concept to be taken forward for detailed design. Along with this it was decided that the rear bodywork would be designed in further detail as this showed increased aerodynamic performance. Four materials were considered for manufacture, glass fibre reinforced polymer (GFRP), carbon fibre reinforced polymer (CFRP), aluminium and Acrylonitrile Butadiene Styrene (ABS) polymer. Each material requiring a different manufacturing process. The attachment techniques were another area being considered. The three options were cable ties, quarter turn fasteners or p-clips.

9.2 Constrains Considered:

When designing the bodywork, the main constraints considered were the rules and regulations, as defined in the Product Design Specification (PDS), the mass, aerodynamic performance and packaging of the chassis, suspension and powertrain components. The air flow to the rest of the car, including the diffuser, as well as overall downforce and drag generated were the main aerodynamic factors considered. Ensuring that the flow coming in under the nose and around the body was not turbulent was a key factor in the design, as this would reduce the efficiency of the cooling and aerodynamic devices of the car. Additionally, some components, such as the front wing, side pods and wishbones will require mounting points through the bodywork to the frame, these must also be accounted for in the design of the bodywork. Along with this the dampers were mounted above the frame. Moreover, time was critical in the design phase as the car was required to be ready for the 2018 FSUK class 2 competition. This meant that the designs needed to be manufactured in the shortest time at a high quality.

9.3 Changes since the Initial Design:

The CFD model of the initial design is shown in Figure 48 with the front bodywork sitting underneath the dampers. The dampers on top of the car provides a region of disturbed flow, causing turbulence and drag on the vehicle. An example of flow separation around a circular body is shown in Figure 49. The dampers are more complicated bodies than normal cylinders and therefore will cause more turbulence in the flow. A further concept was developed which has an aerodynamic flick on the top of the bodywork to divert the airflow above the dampers and the cockpit to reduce the drag generated by the dampers, and as a secondary benefit increase local downforce production. Figure 50 shows the static pressure plot on the first iteration of the bodywork design.

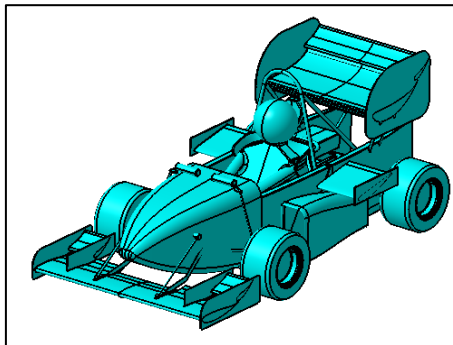


Figure 48: Showing the initial bodywork design on a CFD model

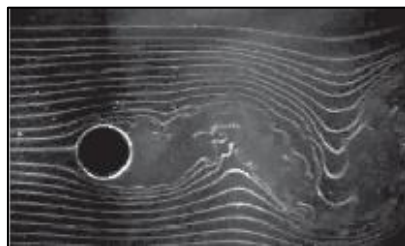


Figure 49 showing separation of flow around a cylinder (Cengel & Cimbala, 2006)

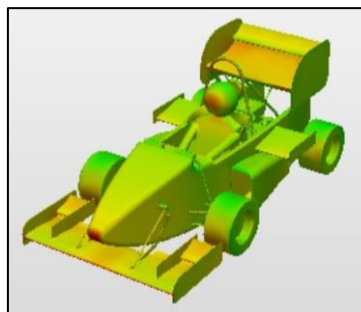


Figure 50 showing the CFD plot of the first concept of bodywork with the “aerodynamic flick”.

The new concept was run in Simscale CFD and showed an increase of 10 N of downforce with no increase of drag compared to the initial design. The increase in downforce is due to the redirection of the flow over the dampers as well as an increase of static pressure on the top of the car. The drag was initially expected to decrease with the new concept due to the air being redirected over the dampers and cockpit, but this was not shown in the CFD. Further analysis showed that one reason for this is that the increased static pressure on the top surface of the nose increased pressure drag on the car, but reduced the overall drag which acted on the dampers, resulting in a neutral net drag performance of the vehicle. The new concept increased the mass of the nose compared to the initial concept by 0.75 kg. Using the results from the CFD further simulations were carried out to determine the overall impact of the change. The straight-line performance was evaluated using the ePowertrain team's acceleration event simulation. The new concept was shown to increase the acceleration event time by 0.01 seconds (Martins, 2017). Along with this a basic lap time simulation was done by determining the maximum cornering speed at a constant tyre-road friction coefficient, assuming the tyres were not the limiting factor for grip. This was then used to determine the time saved on each corner of the 2017 Formula Student UK track. Knowing the length and radius of each corner, the time taken to complete each corner can be found by dividing the corner length by the theoretical maximum cornering speed of the car. This was done for each corner and a time saving calculated by summing the individual time savings. A factor of safety was then considered to account for the driver not being able to push to the limit of the car and a decrease in straight line speed due to the increased drag and mass, so the corrected time was assumed to be 75% of the theoretical time saving. Using this calculation, the increase in downforce of the new bodywork flick showed a time saving of 0.1 seconds per lap of the endurance event. As the endurance event is worth more points than the acceleration event, and the increase in endurance performance of 2.2 seconds over the event and the minimal increase in acceleration event time, the bodywork flick was chosen to go into further detailed design.

Along with this, due to time constraints and the number and size of the holes required in the rear bodywork to allow for the radiator pipes, rear suspension and master switches which will impact the quality of flow over the bodywork, the rear bodywork will not be used on the car for the class 2 event. However, this will be considered for the car for the class 1 event in 2019.

9.4 Final Design and Manufacture:

9.4.1 Geometry

The final design was the nose with the aerodynamic flick without rear bodywork. The flick on the top of the nose could not go above the maximum height of the dampers as this would impact the visibility of the driver. The nose design was iterated 5 times and each model run on the same CFD set-up to have a direct comparison. The design chosen was the biggest gain in endurance event lap and skid pad time was chosen as these events provide more opportunity for points than the acceleration event. The performance of each iteration of nose is shown in Table 17. The initial concept was built around the SE3 chassis, which has a lower nose than frame for SE4. A new geometry was the first design built around the SE4 frame and impact attenuator which was used as the new baseline for further simulations. The angle of the front portion of the frame was designed such that it could act like a nozzle into the floor, accelerating the air flow and improving the efficiency of the floor and diffuser (Katz, 2006). By fitting the bodywork to the car to the frame, all iterations of the nose design allowed for clean airflow into the floor. Iterations 1 and 2 raised the tip of the nose compared to the baseline and 3, 4 and 5 dropped the tip of the nose below the baseline vertically. The simulations were carried out using the rear wing from SE2 and an initial front wing concept for SE4 to simulate the impact to the aerodynamic package.

Table 17: Showing the simulated performance of the nose cone iterations

Iteration Number	Lap time delta from Baseline (seconds)	
	Endurance	Skid Pad
1	0.03	0
2	0	0
3	-0.02	-0.01
4	-0.07	-0.01
5	-0.04	-0.01

It can be seen from the table that dropping the nose relative to baseline showed an increase in performance of the vehicle. This is because as the nose of the car was lifted the static pressure on the underside of the nose increased. This was expected as there was more surface area which the free stream flow stagnated against. The aim of raising the nose was to act like a nozzle and direct more air flow under the nose into the floor, however the velocity of air underneath the car did not increase enough to increase downforce under the car. The increase of static pressure under the nose also increased the local lift generated by the nose of the car and increased drag. This increase of static pressure was due to the flow stagnating on the underside of the nose as shown on iteration 2 in Figure 51. As the nose was dropped the local downforce produced increased as less airflow was stagnating on the underside of the nose. Along with this the nose became more streamlined with the flow reducing the drag on the car, this is shown with iteration 4 in Figure 51. The red indicating high static pressure. Furthermore, dropping the tip of the nose did not greatly affect the nozzle effect into the floor/ this demonstrated that the sensitivity of the low-pressure region generated under the car due to the Venturi effect of the floor was not as sensitive to the curvature of the nose as to the angle.

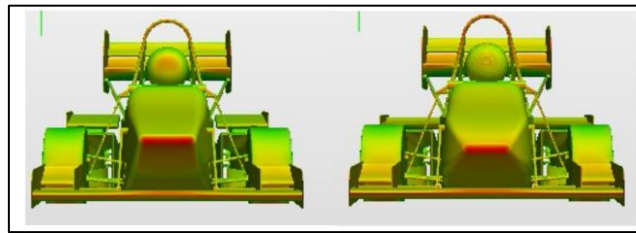


Figure 51: Showing the static pressure plots of iteration 2 (Left) and iteration 4 (Right)

The final bodywork geometry chosen was iteration 4 shown in Figure 52. This bodywork geometry is split into the nose and side bodywork components. This was taken into the detail design for manufacture and practical application. The bodywork was divided into a nose cone and side body to allow for easier access to mounting of components. The wishbones would need to be mounted to the frame after the attachment of the bodywork. If the bodywork was made in one piece, the wishbones would need to be removed for the team to access the inside of the cockpit where the pedal box was mounted. To allow for easy access to this area the nose cone was designed as separate piece to the side panels.

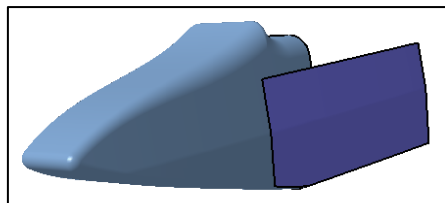


Figure 52: Showing the geometry for the bodywork

9.4.2 Material and Manufacture

The main factors in choosing the materials for the bodywork were having a low mass and low cost of manufacture. The bodywork is not a structural component therefore the strength of the material is not a major factor, but the material needs to be rigid enough to hold its shape. Four materials were considered, ABS plastic, aluminium, GFRP and CFRP. The density of the materials was used to determine which materials would create the lightest bodywork. The average density of GFRP is 1750 kg/m^3 , CFRP is 1500 kg/m^3 , aluminium is approximately 2700 kg/m^3 and ABS is approximately 2000 kg/m^3 (Ashby, 1999). Carbon fibre is recommended to be a minimum of 3 plies thick, which is approximately 0.8 mm thick (XPREG, 2017). To achieve the same mass GFRP bodywork would need to be 0.7 mm, ABS would need to be 0.6 mm and aluminium would need to be 0.4 mm thick, all of which are commercially available.

The next step to determine which material was to be used was the ease and cost of manufacture. A sponsorship agreement with Snapdragon Composites meant that Team SURTES was allowed the use of their composite facilities for manufacture of components for free. This meant that the only cost of manufacturing composite parts was the cost of the moulds and materials. As there were not facilities within the university to form ABS or aluminium into the designed shape the nose and moulds would need to be manufactured externally, adding further cost and time to the manufacturing, therefore these were ruled out for the nose.

The average cost for CFRP is approximately £150 per kg (Carbon Mods, 2017) and the cost of GFRP is approximately £100 per kg (East Coast Fibreglass supplies, 2017). To determine the final material choice the cost of the moulds was also considered. CFRP body panel required a negative mould made from modelling board and milled to the correct shape. The maximum size of modelling board which could be milled at the University of Surrey is 650 x 400 mm. This means that multiple boards would have been needed to create the nose section. Assuming 6 boards would be needed to create the nose and 4 for the side bodywork the cost for the moulds would be £530 (Easy Composites, 2018). The mould needed for a GFRP design would be created from 2 mm thick MDF ribs with the voids filled with expanding foam and cut and sanded to shape, an example of this is shown in Figure 53 (Ludemann Engineering, 2013). Using this mould, the GFRP sheets would be layered over the top, and sealed with resin, this would create the negative moulds ready for the final GFRP lay-up of the bodywork. The MDF ribs would be laser cut using the University's laser cutter. This mould will cost approximately £150 both the nose and the side panels (Hobarts, 2018).



Figure 53: Showing the mould strategy for the bodywork (Ludemann Engineering, 2013).

As the GFRP cost per kg was approximately 66% lower, the moulds were approximately half the price and the size of mould required for CFRP nose cone and bodywork would mean multiple modelling boards due to the size restrictions of the machining tools at the University, GFRP was chosen for the final design. The CAD for the mould for the nose and left-hand side panel are shown in Figure 54. The mould for the nose was split into the top and bottom surfaces of the nose to get the correct curvature on the upper side of the nose.

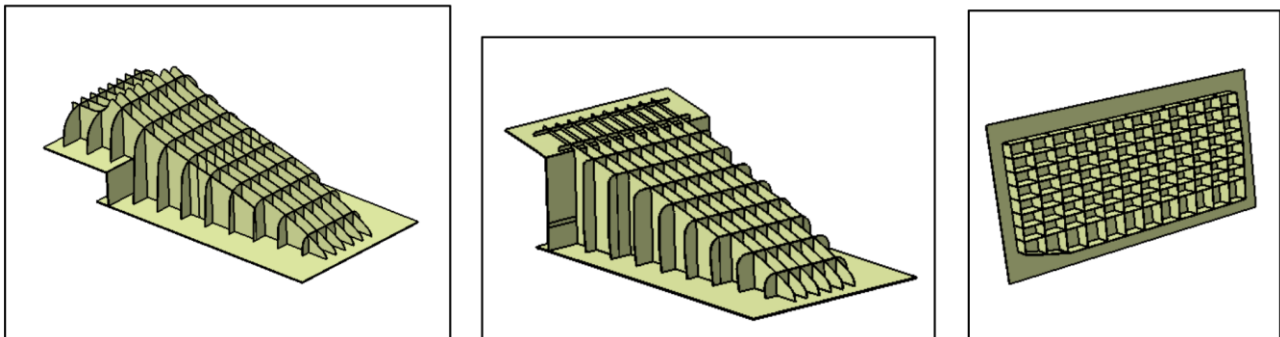


Figure 54: Showing the moulds for the top of the nose (left) bottom of the nose (middle) and left-hand side body panel (right)

At the time of writing this document the moulds were yet to be manufactured. The plan for the mould is that once the MDF buck has been created, the voids filled with foam and then sanded smooth, a release agent will be applied to the outside to allow the GFRP negative mould to be removed from the buck when cured. The GFRP sheets would be laid over the top of the bucks and sealed with resin and cured to form the negative moulds. The cured GFRP mould will then be removed, and the top and bottom halves of the nose bolted together to form the negative mould. The negative moulds for the nose and the side body will then be sanded smooth and have the release agent applied to the inside, ready for the final GFRP lay-up of the bodywork. When the nose lay-up has been completed, the mould will be removed and the body panels sanded smooth, ready for cutting. The holes necessary for mounting of components and the bodywork itself will then be created. Diagrams of the negative moulds to be used are shown in Figure 55.

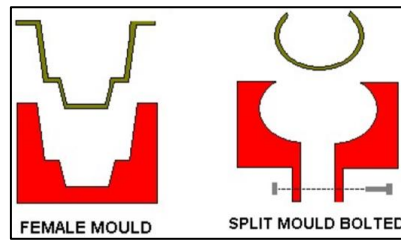


Figure 55: Showing the moulds to be used for the manufacturing of the bodywork (East Cost Fibreglass Supplies Ltd., 2016)

9.4.3 Mounting Strategy

In previous years the bodywork has been mounted using cable ties to the car through drilled holes, which is not a very repeatable or reliable way of mounting the body. For this year quarter turn fasteners or P-clips mounting options were being considered. This was so that the bodywork could be removed and reattached in the same place after each time. Due to the difficulty to access the interior of the car's nose, p-clips were not a feasible option for the nose cone, therefore the fasteners were chosen. The fasteners come in two parts, the fastener and the fixing spring, shown in Figure 56. The fastener will be adhesively bonded to the fibreglass after a hole was drilled into the bodywork. The spring will be riveted to a tab, a CAD example of this tab is seen in Figure 56.



Figure 56: Showing the quarter turn fasteners being used to secure the bodywork (left, middle) (Demon Tweeks, 2018) and the tab on the frame to secure the spring (right)

9.4.4 Finalised Design

Appendix 14 shows the final bodywork design with the major dimensions defined along with an example of a tab used to secure the bodywork to the frame. The height of the nose is 1188.5 mm with a maximum width of 513.2 mm. The radius at the front of the nose is 38 mm, the minimum allowed. The side panel has a maximum height of 457.5 mm and a length of 1045.62 mm.

The GFRP sheets and resin chosen for the manufacture of the parts were a bi-axial glass cloth 320 g/m² and EL2 Epoxy resin, as recommended by the manufacturer. The bi-axial sheets are in a weave of -45°, +90°, 0°, +45°, which allows for tensile and compressive strength in all directions, however as the bodywork is not load bearing this is not a major factor in the design of the body. The minimum recommended thickness, 3 layers of GFRP sheets, will be used for the lay-up. The mass of resin required for this is equal to the mass of the layers, giving the bodywork a total projected mass of 5.7 kg (CFS, 2016).

The GFRP sheets and resin chosen for the manufacture of the parts were a bi-axial glass cloth 320 g/m² and EL2 Epoxy resin, as recommended by the manufacturer. The bi-axial sheets are in a weave of -45°, +90°, 0°, +45°, which allows for tensile and compressive strength in all directions, however as the bodywork is not load bearing this is not a major factor in the design of the body. The minimum recommended thickness, 3 layers of GFRP sheets, will be used for the lay-up. The mass of resin required for this is equal to the mass of the layers, giving the bodywork a total projected mass of 5.7 kg (CFS, 2016).

9.4.5 Cost

Appendix 15 shows the cost breakdown of the bodywork materials. This includes all costs for mounting and manufacture, with a total cost of £348.22. The laser cutting of the MDF is to be done by the workshop at the University of Surrey and the manufacture of the mould and nose will be done at Snapdragon, who provided their facilities to Team SURTES free of charge, therefore there was no additional cost for the manufacture of the components.

9.5 Tests Conducted and Planned

As discussed in the PDS and PDR a coast down and downforce test was planned on SE2, however due to electrical issues and delays in production of SE3 the testing has been delayed. The plan is to complete the testing on SE4 when all parts are manufactured.

9.6 Conclusion and Further Improvements

The finalised concept of the bodywork has a 1% increase in drag, a 15% increase in downforce and an increase of 0.75 kg in mass, including all tabs and fixings, compared to the initial design. Simulations show that despite the increased mass and drag the new bodywork has an increased performance in endurance event lap time of 3.74 seconds with a minimal decrease in straight line performance, assuming the same powertrain performance. This shows that the new bodywork design provides an overall performance increase in terms of competition points. However, these results must be validated against real life data as currently no real-world aerodynamic data is available to Team SURTES for CFD evaluation.

In the future, if space frame chassis will continue to be used, CFRP could be used for weight saving of the bodywork. Assuming 3 layers of CFRP, and 500 g/m² of resin, the mass of the bodywork can be reduced to 5.4 kg including all tabs and attachment (East Coast Fibreglass Ltd., 2016). However, this would further depend on the machining tools at the university as the cost for manufacture of the mould externally will further increase the costs.

10 FRONT AERODYNAMICS (Isaac Lu)

10.1 Introduction

As discussed in previously in section 9 of this report and section 1.1 of the PDR, it was decided that an aerodynamic package comprising of aerodynamic bodywork, front and rear wings along with a diffuser would help to achieve the targets set by Team SURTES. The primary purposes of the front wing were to increase flow rate into the diffuser/ floor of the car, increasing its effectiveness, balancing the downforce generated by the rear wing to keep the front of the car stable, as well as managing turbulent wake produced by the rotation of the front tyres as detailed in section 2.2 of the PDR. The first two objectives had to be revised however, as it was researched that the front wing's purpose would be to move the coefficient of pressure (C_p) through downforce and would often not influence the diffuser. These objectives had to be met in the most aerodynamically efficient manner as possible, with minimum drag, as a drag value of any more than a third of the downforce produced would render an aerofoil too inefficient to be of practical use. (McBeath, 2017; Anderson, 2010)

10.2 Design Options Considered and Changes Made

In the PDR, it was explained that a two-element wing, extending to cover the width of the tyres was the preferred option, with downforce values of ~110 N and ~35 N of drag. However, upon further research and more detailed studies conducted such as observations by (The Flying Lap, 2017) and (Newey, 2017), it was realised that upon the front wing crossing the tyres, no downforce is produced at the ends of the wing along the width of the tyres and therefore, initial CFD downforce values were higher than the actual value.

The initial design was made to the width of the whole car (1550 mm) as it was assumed that it would have a benefit with managing tyre wake and therefore increasing downforce on the rest of the car which was copied from current racing car designs, but as (Newey, 2017) explained, modern professional racing cars only do so as regulations in most competitions enforce wing width, making them cross over the tyres. Aerodynamicists therefore use this area to attempt to manage tyre wake caused by the turbulence of the wing but as explained in literature and observed in older racing cars before regulations enforced width, wings almost always stopped before crossing the tyres, with a simple endplate sufficient to generate a high energy vortex on the inside of the wheel to divert the tyre wake.

Another possibility within the regulations, due to the lack of a width limit would be to extend the wing pass the outside of the front tyres, increasing the wing area (variable S_w in Equation 3, the lift equation, as explained in the PDR) to achieve a large amount of downforce. This however, would result in difficulty for the driver to

perceive the width of the front, as well as resulting in having to take a wider racing line while turning to prevent the wing from knocking over the marking cones placed at the side of the track to measure turning accuracy.

For these reasons, the additional elements on the initial two-element wing were decided to be redundant and would only add to the complexity of manufacturing with no downforce benefit. The length of the wing was also shortened to 1030 mm, to fit on the inside width of the tyres. The initial design compared against the new simplified design decided upon are as shown in Figure 57 below.

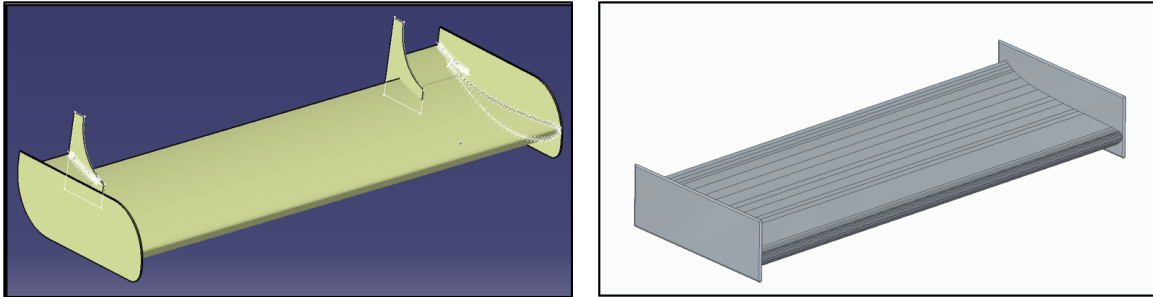


Figure 57: Initial two-element wing (1550mm width) (left) against new single element wing (1030mm width) (right)

The main aerofoil selected was unchanged as described in the PDR (Section 3.2), using an online aerofoil database at (AirfoilTools, 2017) to obtain the most efficient design from the operational Reynolds number range with reference to the height (Re_H) of the wing that the aerofoils (of varying chord length) will be exposed to, considering an average vehicle speed of approximately 12 m/s, is between 150,000 - 350,000 as calculated and explained by (Morton, 2007) and (McBeath, 2017).

It can also be observed that the endplates were made simpler, as the initial design did not increase vorticity as much as a sharp frontal edge, as further detailed in Section 10.3 below.

The most efficient aerofoil (as explained in the PDR, was dictated by the highest ratio of C_L/C_D), within the Re_H range, ‘high lift’ aerofoils were recommended. Upon comparing the charts of C_L and C_D against the angle of incidence (α), the CH10 was selected as it provided a good efficiency ratio as shown in Figure 58 below.

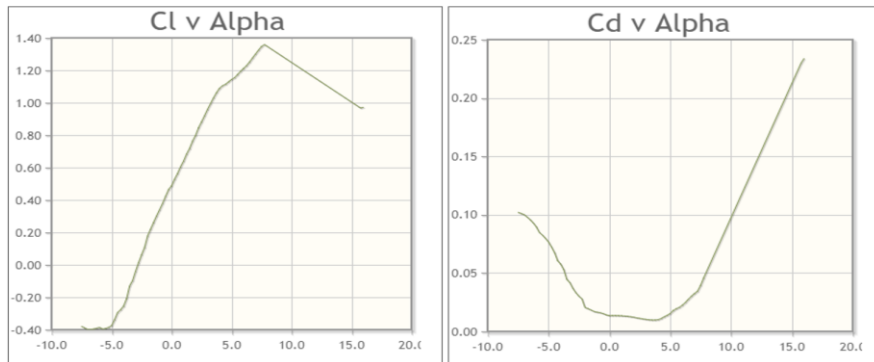


Figure 58: C_L (left) and C_D (right) plot of CH10 aerofoil against α (AirfoilTools, 2017)

As it could be observed that an increase in drag would begin at above an α of more than 3° , and with the assumption that average cornering speeds would be approximately 12 m/s, the priority of the front wing; as considered in the PDR was decided to be to accelerate flow into the diffuser with a secondary consideration being downforce generation. For this reason, the α was selected to be 3° with the downforce only needing to be sufficient to balance the downforce generated by the rear wing but moving the C_P forward by 500 mm (Section 11).

The main element of the rear wing used an identical profile, to ease manufacturing as detailed in Section 12, allowing for the same moulds and insert designs for endplates to be used. These were accounted for and explained in Section 13.4.2 below along with detailed engineering drawings for manufacture in Appendix 21 with an estimated cost of ~£300.

10.3 CFD Setup and Testing

To obtain initial values of downforce and drag to verify the selected design, CFD simulations were performed using Simscale CFD software. The setup of the initial conditions were mentioned previously in the Bodywork section (Section 9.2) and Appendix 4 of the PDR, with a speed of 12 m/s for downforce and flow velocity calculations as the assumed average cornering speed, and 30 m/s for vorticity calculations as an estimate for the car's top speed. For this geometry, the grid independent study showed that a mesh expansion ratio of any lower than 1.11 would provide the same results, which should be noted when recreating the mesh.

As previously stated in Section 10.2, the initial multi-element wing downforce value of ~120 N was incorrect and therefore not a deciding factor in the comparison. In Figure 59 below, it can be observed that flow speed increases to ~20 m/s with the lowest allowed ground clearance (or z-axis coordinate) of 30 mm from the bottom of each endplate and therefore providing downforce via ground effect. In Figure 60, the magnitude of vorticity (frequency or strength of a vortex measured at a point in $1/s$ or Hz), at $> 175 \text{ } 1/s$ before encountering the endplate, increases to $\sim 700 \text{ } 1/s$ immediately behind the endplate and drops to $\sim 350 \text{ } 1/s$ roughly 400-600 mm back from it, in contrast, the initial endplate design only increased vorticity to $\sim 400 \text{ } 1/s$ immediately behind it. While it is difficult to determine without physical testing if this is sufficient to deflect tyre wake as rotating bodies were not able to be simulated, the initial model proves that the endplates have sufficient vorticity at 30 m/s and above, and therefore can be further developed or improved upon in the future to aid in engine cooling.

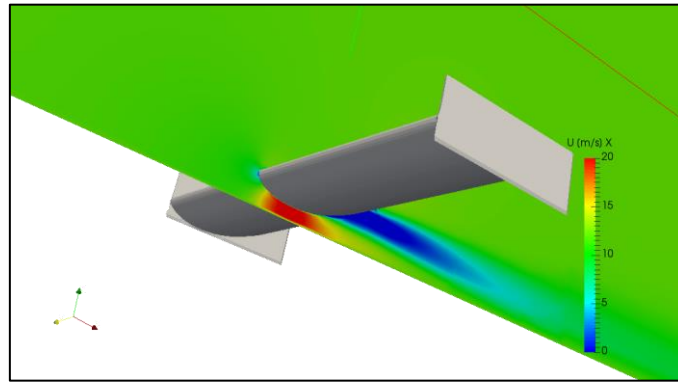


Figure 59: Flow acceleration/ ground effect under wing at 12 m/s with 30 mm ground clearance

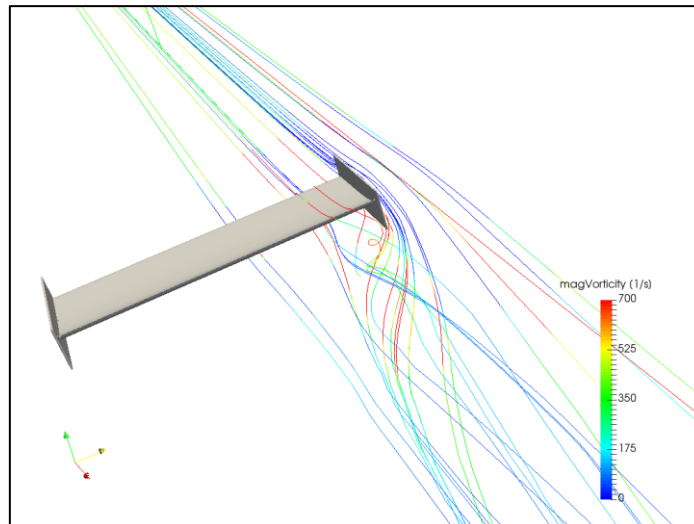


Figure 60: Vorticity streamlines on endplates at 30 m/s with 30 mm ground clearance

Isolated downforce values for the front wing were measured by Simscale to be ~50 N as shown in Figure 61 with the convergence plot for accuracy verification in Figure 62. This value agrees with the theoretical value calculated using the lift/ downforce equation (equation 3 in PDR), with the C_L value at 3° in Figure 58 being ~ 1.1 . In the current CFD setup, it was not possible to accurately determine total drag values due to the limited understanding of the topic.

Pressure drag force plots however, showed a total value of ~ 6.3 N. While the theoretical value using the drag equation (equation 3 in PDR) provides a value of ~ 0.65 N, with the C_L value at 3° being ~ 0.02 , it was decided to assume the drag to be ~ 10 N, using the theory that viscous drag accounts for a third of the total drag. The CFD setup should therefore be improved in the future to obtain more accurate results and reliable data.

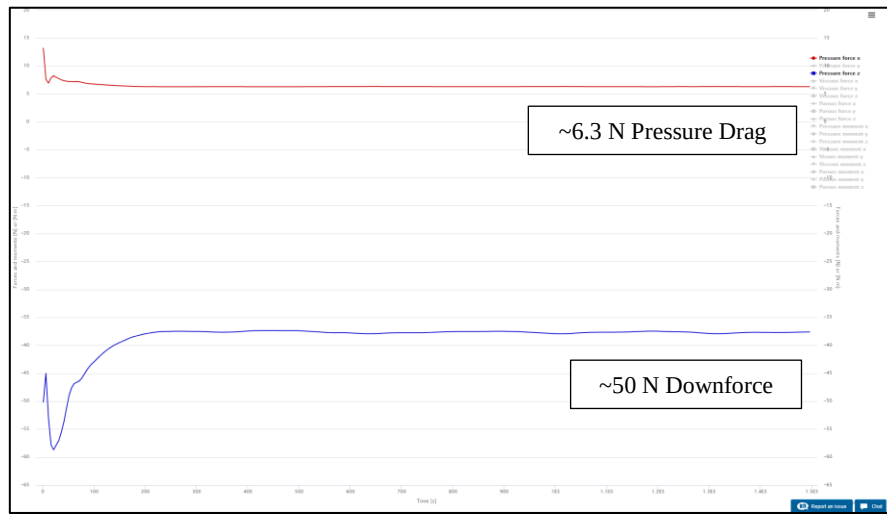


Figure 61: Force plot

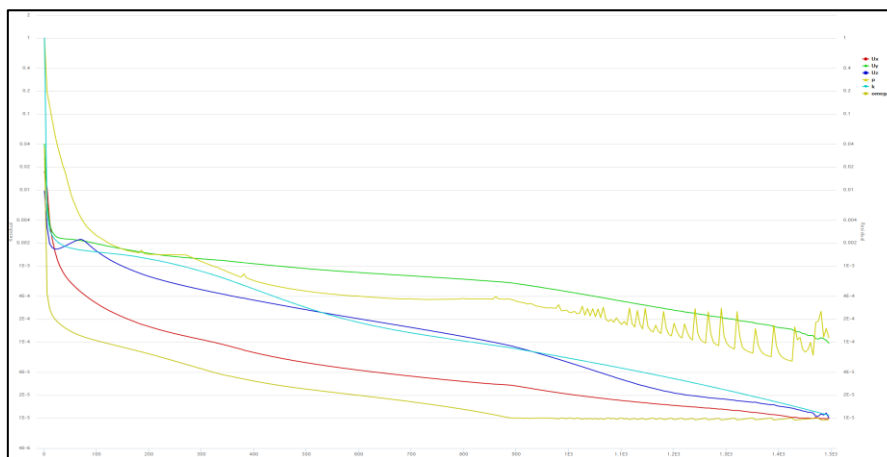


Figure 62: Convergence plot

When the front wing was tested on a model with the diffuser and rear wing attached, the downforce increased by 70 N, producing 160 N from 90 N. This was mostly due to the optimisation of the C_p location, therefore allowing the rear wing to perform more effectively, on top of the ~ 50 N isolated downforce as shown above, therefore, showing that its primary purpose was fulfilled.

10.4 Mounting Concepts

The primary objective of the mounting design was to keep a rigid structure, with additional care taken to minimise the impact on aerodynamic performance. For this reason, mounting points were concentrated close to the middle of the wing and frame, which has most of the front bodywork behind it. As the bodywork would prevent the central section of the wing to perform optimally in any case, the inclusion of mounting points in that area would therefore not hinder the actual performance of the wing greatly, allowing for the maximum span of the wing to operate freely at the sides.

Another consideration on the location of the wing itself along the x-axis, depended on the location of the IA (Section 5 above). As the IA extended roughly 200 mm from the front of the frame, the leading edge of the wing was placed approximately 185 mm in front of the frame, sitting 15 mm behind the IA. This allows the IA to absorb the impact before the wing and therefore as per regulations, a separate IA would not have to be produced for the wing.

10.5 Mounting Design Selected

On the 2016 Team SURTES car, which used a horizontal mounting concept, as shown in Figure 63 below. This used vertical bars bolted on and extending from the middle of the wing being fastened on to mounting tabs protruding from holes in the side of the nose cone from the frame. The design relied upon attaching an endplate tabs welded onto the chassis through bolts. These bolts, therefore, had to pass through the nose cone to reach their assigned mounting tabs. This made the process of mounting the front wing tedious as mounting the nose cone before the wing would require climbing to the front of the cockpit facing upwards and tightening the locking nuts. To solve this problem, the 2016 team welded the nuts onto the tabs, which removed a significant amount of mounting difficulty as the nuts on the inside of the nose would no longer need to be tightened manually.

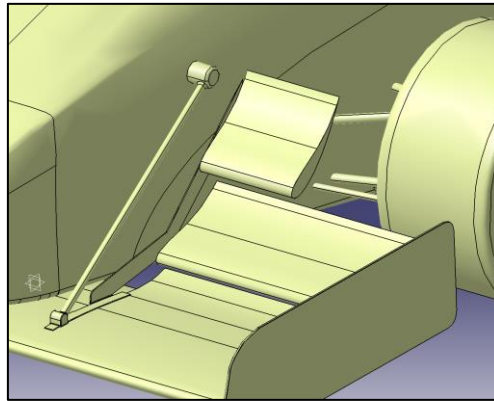


Figure 63: 2016 Car front wing mounting method

While the problems with the mounting were mitigated, the vertical bars produced a stream of turbulent air, therefore reducing the effectiveness of the aerofoil by reducing the area of which the aerofoil could manage laminar airflow. This had to be solved by adding multiple wing elements to achieve the downforce target, therefore increasing manufacturing complexity, and it also appears that the front wing was created independently of the rest of the car, as it extended to the full width outside of the front tyres, therefore not producing downforce across the tyre region as previously described in Section 10.2.

To solve the remaining problem of flow separation, mounting points were redesigned to be contained within the nose. Bluff bodies would therefore be concentrated on the lower plane of the wing, minimising the impact on airflow. Mounting was also chosen to remain vertical as horizontal mounting would add the consideration of bending stresses, limiting the length of the bolts that could be used. With the estimated weight of the wing being ~ 4 kg, considering the reduction in mass from the 2016 Team SURTES front wing weighting ~ 4.5 kg, with a maximum of ~ 50 N (~ 5.01 kg) in downforce at 3g, the mounting tabs and bolts would have to support a load of ~ 28.5 kg in total in a vertical loading manner.

An M3 bolt of a property class 4.6, the lowest available, was calculated to have a minimum ultimate tensile load of ~ 2 kN (~ 200 kg) by (Advanced Mechanical Engineering Solutions, 2013), and therefore would be of sufficient strength to mount the wing, however, it was decided to use four mounting points on the wing to reduce flutter and other deflections on the wing to meet the regulations which state that ‘any aerodynamic device must be able to withstand a force of 50 N applied in any direction at any point and not deflect more than 25 mm. Therefore, from a loading perspective only, this results in a safety factor of 28, but this does not take into consideration inconsistencies in the welding strength of the tabs as well as flutter/ deflections on the wing and therefore is not fully representative of the reasons for using four mounting points. The tab locations are displayed in Figure 64, and the mounting nut and bolt design in Figure 65.

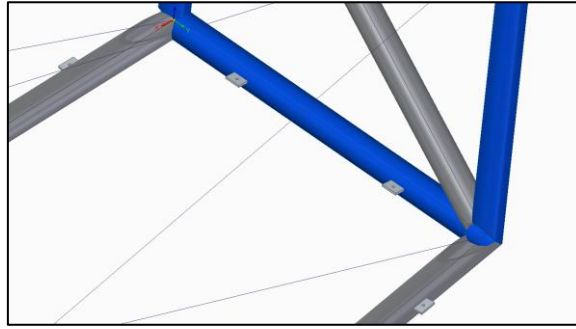


Figure 64: Front wing tab locations (front of frame)

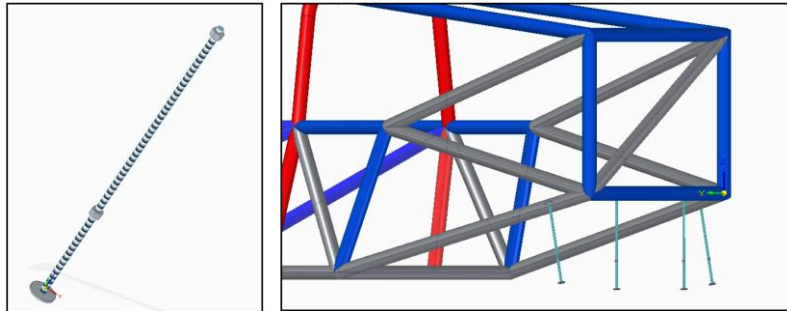


Figure 65: M3 nut and bolt design (left) and mounted on frame without wing (right)

The location and design of the mounted wing can be seen in Figure 66. As the wing profile is curved, to fully tighten the holding nuts at the top and bottom of the wing profile, rubber washers will be added to fill the gaps on surface, also preserving the wing from physical damage from over tightening of the bolts. Refer to Appendix 16 for engineering drawings of the tabs required.

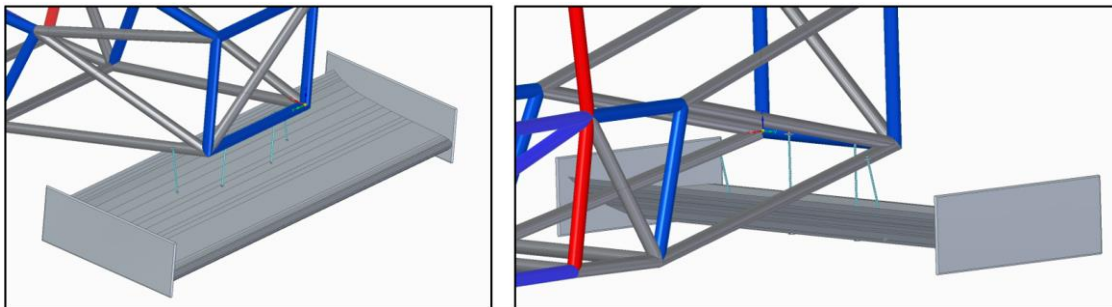


Figure 66: Front (left) and rear (right) view of mounted wing

Longer tabs were considered (Figure 67 below) for the tabs moving outwards of the Y-axis, extending through holes in the nose cone, allowing for the locking bolt at the top to be removed without the nose cone being removed, however, this design would only be feasible if the nose cone was two separate pieces vertically down the middle as in its current form, being a single piece, makes it difficult to remove, which is a more important factor when accessing the inside of the front of the car.

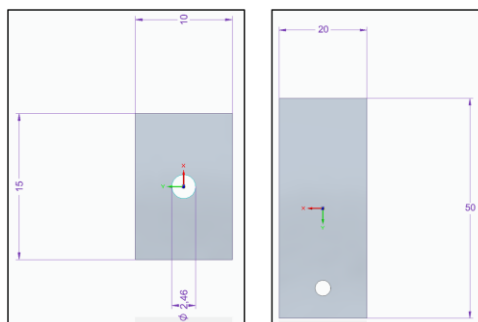


Figure 67: Selected tab (left) vs longer tab design (right)

10.6 Conclusion and Potential Improvements

With the main objectives of the front wing to move the C_p of the car closer to the middle from the rear by producing at least one-third the downforce when isolated to balance the rear wing, as well as the generation of vortices to minimise the impact of tyre wake, it can be concluded that the objectives have been met with the design selected. At a speed of 12 m/s, the isolated front wing produced ~50 N of downforce to the rear wing's ~112 N while the diffuser showed an estimated increase of ~72 N, therefore outweighing the estimated drag penalty of ~10 N.

To further improve the design in the future, more research can be performed on endplate shapes and how they can increase vorticity. Without the availability of physical testing and limited experience of the team, it is difficult to determine if the present vorticity is sufficient for its purpose of diverting turbulent tyre wake. The easiest improvement would be to understand the CFD initial conditions to obtain rotating bodies for the tyre simulation, as well as using heat transfer simulations to check battery cooling conditions.

For the main aerofoil, should the resources be available, a creation of a second element might aid in the downforce generation, however, initial research into this area showed a higher increase in drag without higher additional downforce. A simpler option would be to place a 'Gurney', a small right-angled strip attached to the trailing edge of the aerofoil to increase the effective area of the wing and to create a larger stagnation point at a different location due to the introduction of a recirculation region at the trailing edge. In theory this would create a higher downforce value, however, when attempted with the rear wing, a 100 mm flap produced less than an additional ~2 N of downforce and ~5 N of drag in CFD simulations. Not much literature and/or formulae is available about such devices however, making them difficult to justify without experimental data as theoretical data will be limited. They are known to have high sensitivity to changes in air density which limit their practicality, especially in colder climates where air is of higher density; for this reason, most professional racing teams only add them for high speed applications in hot climates with lower air density. This option can be explored further if wind-tunnel or other forms of physical testing is made available to the team in following years (McBeath, 2017; Li, et al., 2015; McLaren, 2017).

11 REAR AERODYNAMICS (Robert Pemberton)

11.1 Design Options Considered

Design Requirements as stated in the PDS:

1. Positive net benefit of manufacturing a rear wing
2. A minimum downforce of 130 N at 12 m/s
3. Cost of manufacturing no more than £660
4. Total mass (not including drag reduction system (DRS) and Swan Neck) no more than 8 kg
5. Ensure a high quality surface finish

11.2 Constraints and CFD Simulations

As specified in the PDR, the constraints stated in the rulebook refer to the working area, minimum edge radii, and stability and strength.

Leading up to the optimal 3 element aerofoil design, several options were considered. As the use of a DRS was initially uncertain, static designs were generated, and the priority was the aerodynamic efficiency, as a high constant drag would significantly reduce the endurance of the whole car (325 points) and efficiency (100 points). A single element wing was the first option, being relatively inexpensive, and simple to manufacture. The downforce was easily controlled by adjusting the angle of attack, however, flow separation began on the bottom surface at 16° incidence at 12 m/s, and the downforce of 43 N did not meet the requirement. To increase downforce without flow separation, a double-layered wing (one wing stacked above the other) at 15° incidence was considered, however, the downforce of 80 N at 12 m/s did not meet the target. Interference between the two layers made the arrangement less efficient than the single aerofoil design, and this finding eliminated the possibility of a triple or quadruple stacked wing, as the drag penalty due to interference would only increase, and instead shifted the focus towards a multi-element design with DRS (either actuated or slotted design). As a greater downforce allowed a higher cornering velocity, the rear wing was essential for the

Skidpad (75 points) and Autocross event (100 points), while the DRS would ensure a minimal impact on the endurance and efficiency events.

By observing the flow in a 2 element wing design, the flow approaching the second aerofoil was given an upward velocity by the preceding aerofoil, allowing the flow to stay attached to the relatively steeper (32° incidence) downstream aerofoil at 12 m/s. It is for this reason that multi-element configurations are common in F1, FSC and FSE.

F1 teams exclusively use a 2 element design, and this arrangement was initially assumed to be optimal for FSE, however a comparison between F1 and FSE shown in Table 18 inferred that the rear wing had to be designed from the ground up.

Table 18: F1 and FSE comparison

F1 Technical Regulations 2017 (FIA, 2017)	Formula Student Rules 2018 (PDS)
3.6.3 Maximum depth to around 520 mm (depth dependent on height).	T7.3.3 Maximum depth to 850 mm (accounting for swan neck).
3.6.8 Only a single DRS component.	Number of DRS components not restricted.
3.6.1 Maximum height of 800 mm and minimum 200 mm.	T7.3.1 Maximum height to 1200 mm and minimum 500 mm.
3.6.2 Maximum width to 950 mm.	T7.3.2 Maximum width to 1022 mm.

Thus, the working volume of FSE is 205% greater than F1, and the two element design is an optimized solution to the F1 restrictions, and an average cornering velocity of 70 m/s as opposed to the 12 m/s cornering velocity in FSE.

A 3 element and 4 element wing design was tested and allowed an even steeper angle of attack of the furthest downstream aerofoil, however, as the manufacturing of each unique aerofoil required a separate mould to be created, designs requiring four or more elements were eliminated. As the behaviour of flow around a multi-element wing design often produced counter-intuitive implications, a selection process based of CFD analysis had to be created.

Leading up to the PDR, CFD analysis on 3 element rear wing designs using uncambered (symmetric) NACA 4412 aerofoils were tested to determine the general position, incidence, chord, and thickness distribution of each element. Since the final design in the PDR, numerous cambered aerofoil profiles were tested using a similar arrangement, and the exact incidence, chord, and thickness were optimized.

The aerofoil coordinates were manipulated on excel using a rotation transformation matrix, translation, and scaling factor. These coordinates were then extruded to form 3D wings, which were assembled to the endplates. Using SimScale, a unique fine mesh was generated, and the wing was placed in a 12 m/s flow. See Appendix 18 for exact formulae/algorithms, and boundary conditions. The resulting viscous and pressure forces acting in the direction of downforce and drag were tabulated. Further analysis was carried out on Autodesk Flow Design to advice adjustments in position/size/angle of attack by observing the flow and modifying the geometry such that flow separation was minimized.

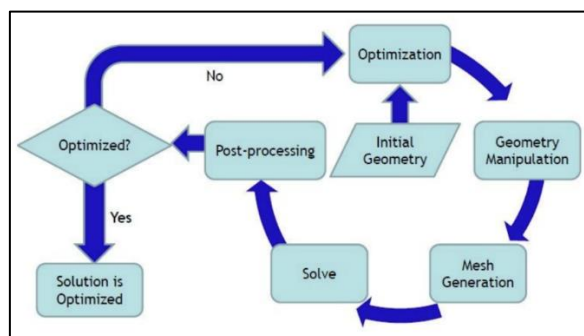


Figure 68: Optimization algorithm process

- Simulate wing in 12 m/s flow on Simscale and add the viscous and pressure forces in the drag and downforce direction to compare it with the previous iteration.
- Simulate wing on Autodesk Flow Design and analyse the flow: Fine tuning the aerofoil positions and angle of attack to affect attachment, downforce, and up wash.

The aerofoil coordinates were manipulated on excel using a rotation transformation matrix, translation, and scaling factor. These coordinates were then extruded to form 3D wings, which were assembled to the endplates. Using Simscale, a unique fine mesh was generated, and the wings were placed in a 12 m/s flow. See Appendix 18 for exact formulae/algorithms, and boundary conditions. The resulting viscous and pressure forces acting in the direction of downforce and drag were recorded in Table 19. Further analysis was carried out on Autodesk Flow Design – see Figure 69 to advise adjustments in position/size/angle of attack by observing the flow, and modifying the geometry to minimize flow separation.

11.2.1 Post Processing of CFD

- Summation of viscous and pressure forces in the drag and downforce direction.
- Comparison of drag and downforce to previous iteration.
- Simulate on Autodesk Flow Design and analysis the flow: Fine tuning the aerofoil positions and angle of attack (a few mm, and degrees at a time) could affect flow attachment, downforce, and up wash.
- Results are displayed in Figure 69 and Table 19 below

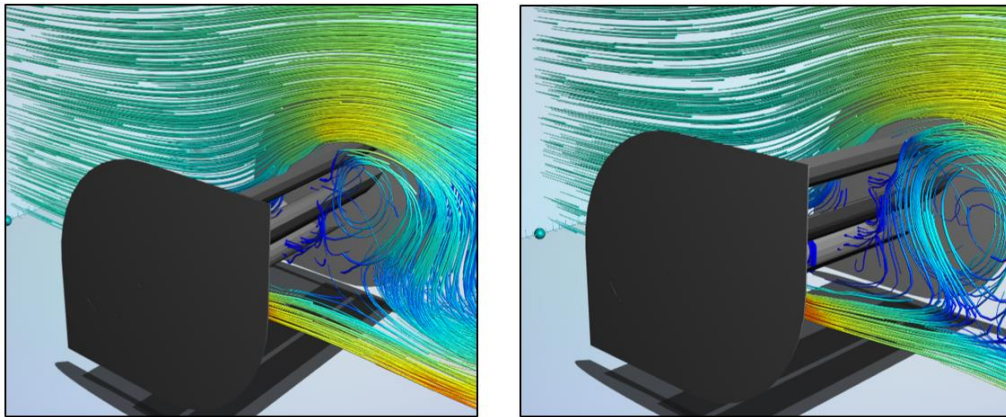


Figure 69: Autodesk Flow Design: Separated flow around B6 (Left) and attached flow around final B2 design (Right)

Table 19: Rear wing CFD results

Design	Moulds (#)	Total Downforce (N)	Total Drag (N)	Aerodynamic Efficiency
B1	3	159	88	1.8
B2	3	176	97	1.8
B2.1	3	169	104	1.6
B2.2	3	179	110	1.6
B2.3	3	184	119	1.5
B2.4	3	180	112	1.6
B3	3	203	141	1.4
B3.1	3	180	113	1.6
B4	3	136	66	2.1
B4.1	3	133	63	2.1
B4.2	3	126	57	2.2
B5	3	205	142	1.4

B5.1	3	187	122	1.5
B5.2	3	213	170	1.3
B6	3	194	127	1.5
B6.1	3	177	114	1.5
B6.2		182	118	1.5

11.3 Final Design

As with the other designs, the B2 was comprised of highly cambered aerofoils (NACA, 2018) suitable for racing applications, and the geometry was optimised as shown by Figure 70. In isolation the B2 design was chosen due to its high downforce (180 N) at a cornering speed of 12 m/s, see Table 19. Visually, the geometry permitted the greatest flow attachment and it performed at a high aerodynamic efficiency, as it caused significantly less drag (97 N) compared to wings producing similar downforce, see Table 19. With respect to the overall performance the design increased downforce over the rear tires. Traction was increased proportionally and a greater cornering velocity was permitted with a reduced risk of skidding. The impact of drag to cornering velocity is small, with a possible DRS mechanism to be implemented in the future (reducing drag to 7 N), the impact in the endurance and efficiency events will not be significant. With the main section geometry to be used on the front wing, the number of moulds required for the aerodynamic components is decreased, reducing manufacturing time, and cost. See Figure 71 for final design.

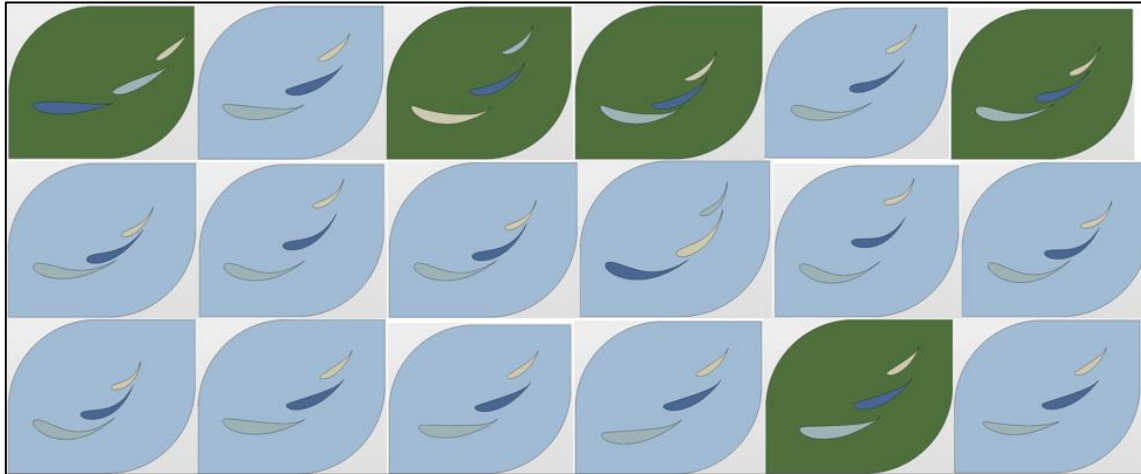


Figure 70: Aerofoil Geometry B1-B6.1 left to right

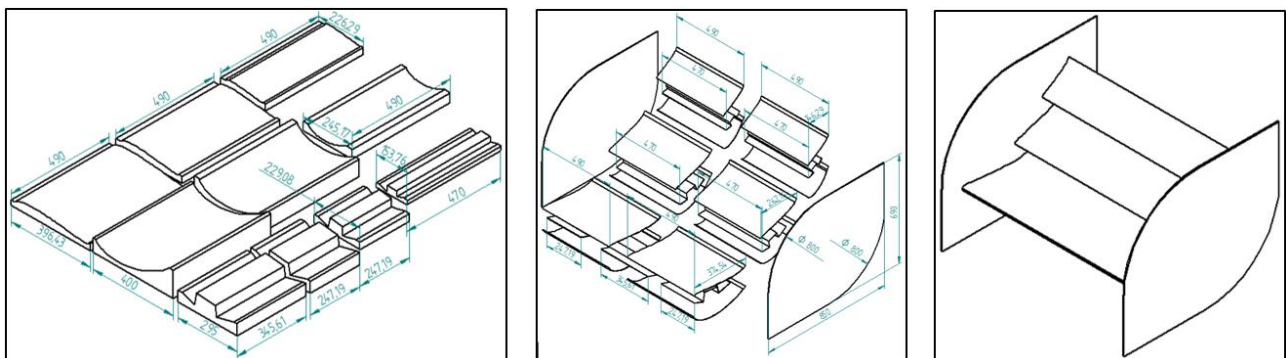


Figure 71: left - moulds left to right – upper surface, lower surface, corrugated section (s), middle - exploded view of 1 mm CFRP components, right – full assembly

11.4 Materials Selection

In order to choose an appropriate material for the rear wing, key factors were ranked in order of importance.

1. Strength and stability: As stated in the rulebook - Any part of the wing can deflect no more than 10 mm under a force of 200 N over 225 cm² in the load carrying direction, and no more than 25 mm under a 50 N point load in any direction.
2. Density: Minimize the total mass of the wing, such that the benefit in increased cornering velocity is not counteracted by the additional power required to overcome the increased rolling resistance.
3. Manufacturing: Material had to be easily manufactured, and within £660 (One third of the £2000 budget for aerodynamic package).

With the above criteria, CFRP, and GFRP were the most appropriate options. The cost of GFRP was 33% less than CFRP. However, with a lower density of 1500 kg/m³, CFRP is 15% lighter than GFRP 1750kg/m³, and the estimated mass is 7.6 kg, (a mass saving of 1.6 kg). As the most popular material in F1 and FS, CFRP was the ideal material.

11.5 Manufacturing Method

11.5.1 Mounting

The endplates were supported by the main section, and the two upper elements were supported by endplates. The aerofoils were reinforced by the ribs, and the main section was supported by the Swan Neck (See section 13.3.1).

11.5.2 Each Aerofoil

A mould for both the top surface, bottom surface, and corrugated section was created (3 corrugated sections for main section). The open female design of the moulds allows the cured 3-ply prepreg carbon to be peeled off. The top and bottom surface are bonded together at the leading edge, and trailing edge using adhesive (all trailing edges are flat and 2 cm long), and to the corrugated section in the centre. Due to the mould board size limit of 650 x 400 mm, most components will be assembled from two parts, and glued side by side. An optional £40 surfacing finish kit is available as a contingency plan should the surface finishing be too rough.

11.6 Cost Sheet

Table 20: Rear wing cost sheet

PRIORITY	SUPPLIER	DESCRIPTION	QUANTITY	UNIT COST (exc. VAT)	Line Cost
1	Easycomposites	1980 x 980 mm (6 mm thick)	1	£329.00	£329.00
2	Permabond	50 ml	2	£15.39	£30.78
3	Easycomposites	Carbon Fibre rod	2	25.8	£51.60
6	Easy Composites	XPREG XC130 Prepreg Carbon 3K, 210 g, 2/2 Twill (1250 mm) - 1m Roll	1	£54.57	£54.57
7	JD Multi Metals LLP	3 mm thick Aluminium Sheet plate guillotine cut model making - All Sizes	1	19.99	19.99

11.7 Tests Conducted/Planned

Wind tunnel testing to be scheduled in the future (delayed due to electrical issues) will be conducted to provide performance characteristics of the wing. Firstly, the reaction forces on the wing in isolation will be measured at 12 m/s. Then, the same measurements will be taken for the full car, and for the car without the rear wing. Should a slot mechanism/DRS be implemented, the same measurements will be taken to determine the effect of the aerofoils at low angles of incidence, and compare it with the 7 N drag as on CFD. If the DRS is driver controlled, the flow reattachment time will be determined, and this value will inform the driver when to activate DRS to utilise the full benefit of the wing.

Static pressure and point load testing will also be conducted to ensure that the strength requirement (stated above) is met - particularly at the centre of each aerofoil, where bending moment and thus deflection is at a maximum.

11.8 Conclusions and Further Improvements

Originally, wind tunnel testing of the SE2 rear wing would provide validation for the Simscale CFD results, however, wind tunnel testing was delayed due to electrical issues. In place of this testing, CFD simulations on alternate software could have been conducted. By using the design cycle iterative process, the overall flow around the wing was improved, and the downforce was increased from the initial 80 N to 180 N, surpassing the initial target by 50 N.

12 DRS STUDY (Billy Cavanagh)

12.1 Introduction

After analysing initial CFD and acceleration data during the PDR and showing that DRS could provide a benefit to the lap time and acceleration of the car, further analysis on the system was undertaken. The results from the CFD data were reanalysed for the final version of the wing and then compared back to updated acceleration data and the updated lap time simulation from the MDDP team. From this a final concept was drawn up, providing a strong base point for the kinematics linkage to be designed and for DRS to be implemented on future Team SURTES cars.

12.2 Design Options Considered

After doing some benchmark analysis it was found teams tend to implement one of two main methods, gas compressed cylinder actuation or electrical motor actuation. Due to the complexity of compressed systems and the higher mass of the components compared to an electrical system this wasn't chosen to be analysed. An endplate mounted DRS system similar to that of KA Racing shown in Figure 72 below was chosen to be the basis of the DRS concept, of which the sketches are shown in detail in Appendix 20.

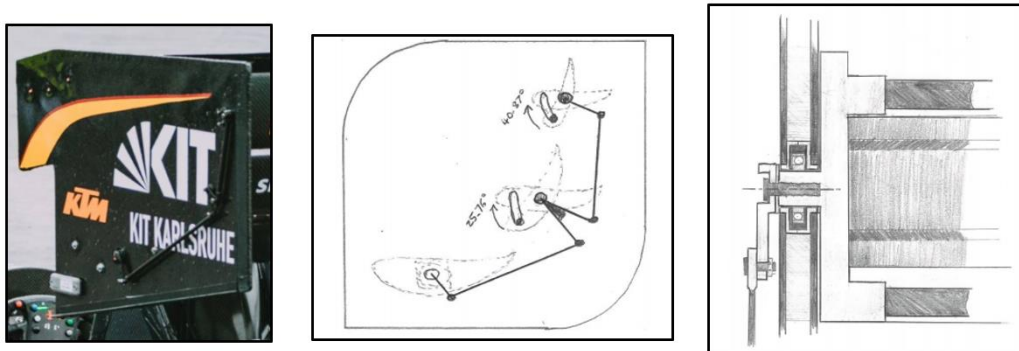


Figure 72: (left) shows KA-Racing DRS linkage (KA-RaceIng, 2017), (middle) DRS schematic, (right) endplate concept connection used to rotate the wing elements

12.3 Tests Conducted

Using the final rear wing design as a basis, the geometry was adapted to simulate the effect of DRS by changing the angle of attack of the middle and top element to 0° with respect to the horizontal, pivoting about the centre of the chord length. The pivot was chosen due to failure mode analysis, as if the system were to fail, the centre of pressure would be in front of the pivot, forcing the wing closed to the high downforce position. Estimating the centre of pressure for each element to be $\frac{1}{4}$ of the chord length (NASA, 2015). The torque provided on the top and middle element was calculated to be 12.1 Nm and 23.1 Nm respectively at 27.9 ms^{-1} . This torque would then have to be countered by a motor with added torque to accelerate the linkage system.

The geometry was uploaded to Simscale and run in free stream velocity to find the advantage of using DRS for the final wing design. Wing parameters were evaluated against updated acceleration data from the

powertrain team and then against lap time simulation data provided by Optimum Lap used by the suspension team. Acceleration data showed a 0.03 second reduction in 0 – 75 m sprint time. The data was also analysed against Optimum Lap simulation, assuming the driver uses the DRS on all maximum longitudinal acceleration points during the lap, the overall improvement in lap time is 0.344 seconds per autocross lap. Over an endurance event this would equate to a 7.559 second advantage.

12.4 Conclusion

Estimating the rough cost of a DRS system including carbon rods for the linkage, bearings, rod ends, and the maximum sized motor that could fit in the main plane of the wing, the total cost would roughly be £300. Assuming this to be accurate a DRS system would provide a £872.10 per lap second improvement. If the same rear wing design is continued for SE5 design, the linkage system could be finalised and evaluated against other potential improvements using this cost per second evaluation as justification.

13 REAR WING MOUNTING AND AEROFOIL STRUCTURE (Billy Cavanagh)

13.1 Introduction

The final rear wing mounting and structural components were designed to reduce the weight of the rear wing compared to the previous mounting design used in SE1, whilst also aiming to improve the aerodynamic efficiency of the rear wing.

To mount the rear wing to the chassis previously, simple support struts were attached to the wing in six positions (four positions on the underside of the main plane at the outside of the span and two times in the middle of the span at the leading edge of the main plane). The mounting system design for this years' wing design is based around swan neck top surface mounting, rather than the lower surface mounting on SE1. This method of mounting provides many added benefits including reduced weight and drag, whilst not disrupting the attached flow on the lower surface of the wing, improving downforce. The swan necks design incorporated industry structural design, as well as benchmarking against existing Formula Student designs. The previous mounting supports weighed 1.25 kg (not including tabs). A personal design goal of a 50% weight reduction was set after the PDR stage, with the finalised design weighing 0.61 kg (including tabs), a 51.2% reduction in weight. The other goal was to improve the aerodynamic efficiency by increasing the lift to drag ratio. The old mounting system from SE1 in combination with this years' final wing design has a ratio of 0.685. In comparison, the swan necks increases this ratio to 0.708 by reducing drag by 2 N and increasing downforce by 2 N at 12 ms⁻¹, meeting the initial goal to improve the aerodynamics of the rear wing.

Due to a swan neck mounting system being used, the internal structure of the rear wing elements had to be adapted from the previous SE1 design. This was done using industry techniques of wing design in collaboration with subject specific experts in carbon composite manufacturing. The previous SE1 wing was made by filling the inside of the wing with ROHACELL structural foam with steel inserts for bolting into. The design for this year has been based around a conventional wing design using ribs and spars, verifying this against existing Formula Student designs and industry design. The aim of redesigning the internal structure of the rear wing was again to reduce mass by 50%, hence reducing the centre of gravity, roll forces during cornering and thereby increasing lateral grip and reducing lap time. The final design for the rear wing structure weighs in total 2.28 kg (without the corrugated carbon rib or carbon wing skin). Using a ROHACELL 71 (Evonik, n.d.) filled core wing would have resulted in an internal wing weight of 11.76 kg. Neglecting the weight of the final carbon wing skin and the endplates, the new internal structure shows a mass reduction of 80.61%. The final design of the wing is bolted through the endplates using M5 bolts, with the wing endplate ribs having M5 Keensert Inserts (Clarendon, 2018) so the aluminium can be bolted into. There is also the possibility to reposition the top two elements for the dynamic sprint event to zero angle of attack, gaining 0.03 second advantage calculated from simulation results. The final design showing the full car with swan necks and internal wing structure can be seen in Figure 73 with the engineering drawings for the components shown in Appendix 2.

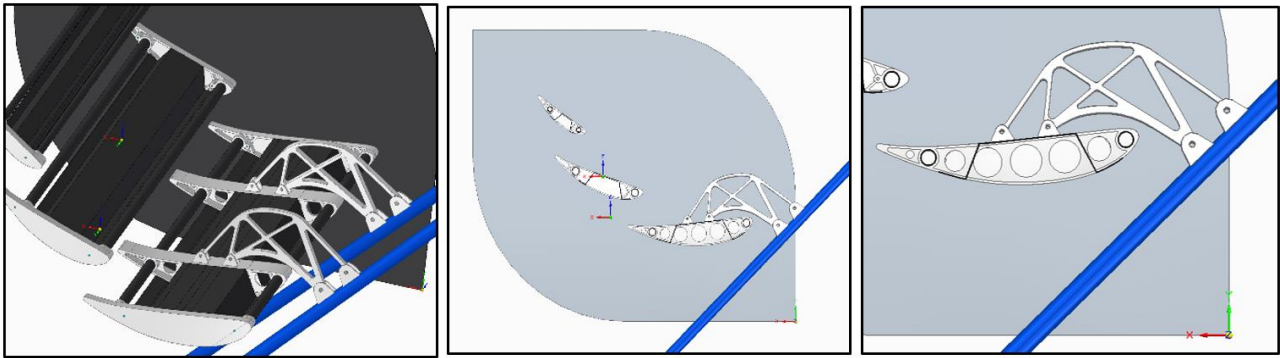


Figure 73: Screenshots of the final internal structure of the wing with swan neck attachment, (right) close up cut away showing corrugated central spar and front/rear carbon rod spars in black

13.2 Constrains Considered

13.2.1 CFD Data

Due to aerodynamic testing being delayed and cancelled, there has been no validation between the CFD and real-world results. Any results from the CFD have therefore been assumed to be accurate to real world applications for the purpose of FEA analysis.

13.2.2 Loading Conditions

Failure loading conditions from PDR stage have been modified to incorporate a 3 g bump and 1.52 g max lateral acceleration calculated by the suspension team (both assuming 10 kg rear wing),. This has been added to the CFD drag and downforce values for the final wing design at 27.9 m/s (calculated max speed by the powertrain team). The two failure loading cases for the rear wing are shown in Table 21 below with a screenshot of the applied loading cases shown in Appendix 22. These loading conditions have been applied to all swan neck and rib iterations throughout the design improvement process.

Table 21: Max loading cases assuming worst failure scenario

	Force components (N) relative to master origin		
	X	Y	Z
Loading Case 1 – max speed + 3g bump	-259.2	0	-1215.9
Loading Case 2 – max speed 45deg slide +1.52g max lateral acceleration + 3g bump	-439.0	588.1	-294.3

13.2.3 Stress Flow in Complex Composite Geometry

Due to limited analysis of the carbon composite wing layup and the data available for pre-cured composites, bending rigidity testing will have to be done prior to full car assembly. The details of this testing will be explained in the testing section below.

13.3 Design Changes From Initial Concept

13.3.1 Swan Neck

Due to time constraints placed on the tab manufacturing and positioning, the initial tab positions from the PDR have not changed with respect to the hole positioning relative to the master origin on the full car assembly. Because of these time constraints, the outer dimensions of the swan neck have not changed considerably since the PDR. The initial concept shown below in Figure 74, assumed bonding could be used to attach the rear wing to the frame. However, after considering bond strength factors and using failure modes analysis, it was decided that the swan necks should be detachable at the wing and at the frame to protect the carbon wing from potential damage.

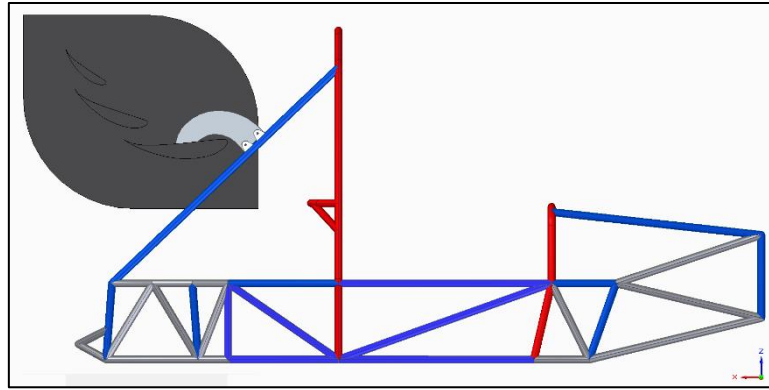


Figure 74: Original side cut away design of the swan necks bonded onto the wing

After speaking to Tu Fast racing regarding the attachment of their swan necks to their rear wings, they have found the best solution is to use carbon ribs with aluminium tab inserts to bolt the swan neck to the wing, as shown in Figure 75 below.



Figure 75: Tu Fast racing swan neck and mounting system highlighted (TuFast Racing, 2017)

After incorporating this bolting design into the swan neck and analysing the two loading cases, a mass optimisation study was undertaken using the topology tool in ANSYS Workbench. Each iteration undertaken was analysed against loading case 1 and 2. The outcomes from each iteration can be seen below in Table 22 below with regard to the reduction in mass and the max stress and deformation seen on a single swan neck. Screenshots of an example iteration can be seen in Appendix 22.

Table 22: Comparative mass reduction using topology base

Iteration no.	Weight (kg)	Max Equivilant Stress (MPa)	Max Deformation (mm)
1	0.984	19.79	0.21
2	0.380	56.26	1.02
3	0.302	104.67	1.21
4	0.269	369.59	1.85
5	0.276	166.36	1.72

Inspiration for the triangulation was taken from benchmarking research against other Formula Student teams. It was found the best way to triangulate around the geometry was to keep all internal angles close to 60°. The increase of weight between iteration 4 and 5 was due to the second loading case causing the swan neck to fail due to a tight radius in the triangulation. The thickness of the failing member was increased to 10 mm thick and the radii of the triangulation enlarged, until the max stress reduced below the yeild strength of the material to a SF of greater than 1.5. The final design of the swan neck can be seen in the Appendix 21.

13.3.2 Wing Rib Mounting and Wing Structure

After researching wing manufacturing techniques, it was found that the wings weight could be reduced by not using ROHACELL 71, instead using carbon composite components as spars and ribs, similar to the design of high performance glider wings. Further use of aviation wing design was used when reducing weight in the swan neck mounting ribs. Here holes were cut from the centre section to reduce the weight of the wing rib.

After discussing the concept with Dr John Watts regarding the bonding of carbon profiles together, it was advised to make the wing ribs out of aluminium as this can be CNC milled, thereby closer matching the inside profile of the wing. It was also advised to manufacture the aluminium components with a 0.1 – 0.5 mm offset to the profiles of the mating carbon components. The offset would allow for a suitable bonding thickness from his experience to provide a good quality bond using a proven adhesive such as Araldite 2015 (Araldite, 2013). The final design for the ribs and aluminium endplates have a 0.3 mm offset with respect to the carbon components. The bonding case used has been designed specifically to minimise areas of peel, with all areas of bonding experiencing lap shear or tensile adhesion.

13.4 Design Analysis

13.4.1 Swan Neck

The final Swan Neck design can be seen in Appendix 21 with the final FEA results from both loading cases shown in Appendix 22. The max stress in the swan necks was seen from load case 2 indicating 166.36 MPa. This loading case is very unlikely to happen, as almost simulating a crash. The swan neck has been overdesigned for the application, however ensures the safety of the driver, providing no external impact during a crash on the wing. The final full assembled CAD showing the assembly of the swan necks and the rear wing can be seen below in Figure 76.

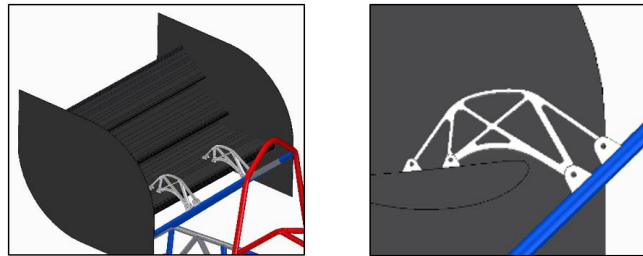


Figure 76: Final swan neck design attaching to the rear wing on the main roll hoop supports

The final weight for the swan neck design is 0.276 kg being made of 6082 T6 Aluminium, with the tabs weighing 0.058 kg and being made of mild steel. The overall cost for the Aluminium is £41.18. The full breakdown of the cost and the suppliers can be found in Appendix 23. The material choice was based on high strength to weight ratio of the Aluminium available from our supplier, corrosion resistance and low cost.

13.4.2 Wing Rib Mounting and Wing Structure

The final structural wing design can be seen in Appendix 22 with the engineering drawings, the final FEA results from both loading cases are shown in Appendix 22 with the max stress 10 MPa seen due to loading case 1. The final full assembled CAD showing the assembly of the structural wing components and the rear wing can be seen below in Figure 77. Araldite 2015 was chosen for its known strong bonding properties with carbon fibre. Careful consideration has to be undertaken when preparing the carbon fibre before bonding to the aluminium refereeing to the instructions regarding bonding (Araldite, 2013). The carbon rods used are 20 mm outer diameter and 17 mm inner diameter on the main and mid element and 14 mm outer diameter and 10mm inner diameter for the top element. The rods are pultruded 90° carbon fibre with 37° braiding, designed as high-performance carbon rods with respect to bending and tensile forces. The mechanical data properties for the carbon layup is shown in Appendix 24.

The final total weight of the structural design (not including corrugated rib weight) is 2.28 kg. The overall cost for the aluminium and the bonding epoxy is £221.44. The full breakdown of the cost can be found in Appendix 23.

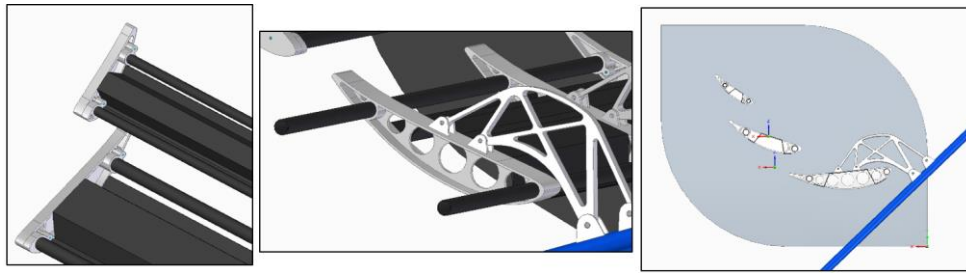


Figure 77: Final internal wing structure attaching to the swan necks. (Left) Wing endplate middle and top element with carbon rod and corrugated carbon spar attached, (Middle) Wing rib without corrugated spar showing weight reduction holes, (Right) side on cut away view

13.5 Testing

13.5.1 Testing Conducted

To analyse the true benefit of the using a swan neck mounting system the theoretical benefits of a swan neck design had to be evaluated. Using a CFD simulation of final wing with the SE1 mounting system and with the swan necks, it could be shown that the design provided an added benefit to performance. The results of the full car CFD simulation show that by mounting the rear wing using swan necks, it reduces drag by 2 N and increase downforce by 2 N at 12 ms^{-1} .

From the Full car CFD testing it could be seen that there was a 40.3% reduction in downforce and 20.1% increase in drag on the rear wing when compared to CFD analysis of the rear wing in free stream flow. This is likely to be due to the interaction of the flow with the main roll hoop and the driver. However, the loading of the rear wing was still analysed in free stream velocity air, as this will provide the worst case scenario base figures. This will therefore incorporate a base SF into all the designs which should account for unknown stress factors such as wind gusts, material defects and mishandling of parts during assembly.

Assuming the force from the wings is transmitted purely through the front and rear carbon rod spars the max bending loading case 1 was applied to single carbon rod. It was found through hand calculations that the max bending moment was -102750 Nmm at the swan necks. Using this value, assuming Isotropic properties for the carbon rod, the max tensile stress experienced in the rod is 552.7 MPa , 3.85 times lower than the rated tensile yield strength of the rod, 2128.12 MPa . The calculations for this value can be seen in Appendix 25. As there are two carbon rods in each wing section it is assumed that this approximation will hold until further physical testing can be undertaken.

13.5.2 Testing Planned

Physical load testing of the swan necks will need to be undertaken to validate FEA. A mount will be designed to simulate the frame and angle at which the swan neck would be mounted in reality and then the two loading cases will be applied to the swan neck in the correct geometry.

The carbon fibre tubes that act as front and rear spars in the final wing design, will have to have a 4 point flexure test shown in Figure 78 below to provide valid strength against bending, assuming all the loading from the downforce and drag transmits through the carbon rod spars. A test sample spar has been ordered from PFT Profiles (PFT Profiles, 2018) and will be tested before manufacture of the full wing structure.

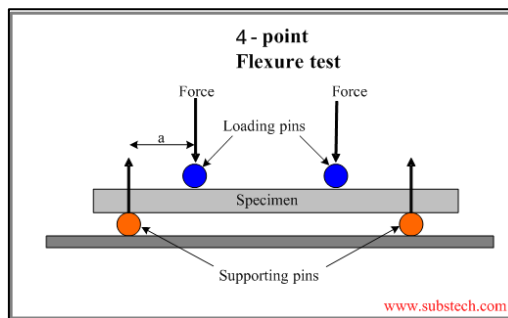


Figure 78: Flexure Test: Loading case 1 total force will be applied over both loading pins located at the swan neck positions, supporting pins will be placed where the wing endplate ribs are located (SubsTech, 2012)

13.6 Future Improvements

The final design of the rear wing mounting could have been improved given the rear wing geometry further in advance. Having this geometry earlier would have meant that further iterations could have been undertaken to minimise the cross sectional width in certain areas of the design. The current design has a constant cross sectional thickness of 14 mm which from the stress distribution could be reduced in areas. Material selection was based on available materials and ease of construction, hence the choice of 6082 T6 Aluminium for all the components. Regarding the swan neck in particular, carbon shear panels on the outside faces could be used to improve the strength of the component. Due to receiving the wing geometry late into project timeframe, the wing spars and endplates have had no mass optimisation undertaken. The final safety factor of the wing ribs and the wing end plates is over 15. Knowing this future mass optimisation should be undertaken to reduce the mass of the wing rib endplates and the central mounting ribs further.

13.7 Conclusion

Looking at the overall initial goals set at the start of the CDR stage, outlined in the introduction, the final designs either meet or exceed all of these goals. This has ensured that the rear wing on SE4 will be lighter and be more aerodynamically efficient than if using previous mounting and manufacturing techniques used on SE1. For future Team SURTES aerodynamic development, it is suggested to re-evaluate the initial tab positioning, to maximise tensile and compressive strength of the material, instead of the bending strength. It would also be suggested to re-evaluate the swan neck and structural design as suggested in further improvements above.

14 PROJECT MANAGEMENT (Chris Kyle + Adam Cox)

Over the course of the project the team worked well together and has produced designs for all of the components required. Communicating clearly was key to the progress of the project and this included informative weekly presentations in formal meetings, constructive informal meetings when required and constant sharing of information through group instant messaging and email. The content of formal meetings improved as the project progressed making them more useful and productive. These meetings were particularly helpful to review individual designs and allow the team to contribute to improvements and highlight problems. Informal meetings were largely used to ensure progress was being made with components which affected each other. For example, these regularly regarded how a given component would integrate with the frame.

Throughout the final phase of the project there were delays to various areas. Aerodynamic testing of the SE2 was cancelled due to an inverter sensor issue, this has meant that the aero tests will be conducted on the completed SE4 chassis. This delay has meant that the validation case for the CFD runs is not available at the end of the project but will be available before the car competes in the dynamic events in the 2019 Class 1 event. Along with this, clashes in the design of components have meant that although the concepts were finalised, minor details had to be changed on parts such as bodywork and frame. This has meant that the finished designs were not completed at the original planned time. The original planned completion and manufacture of designs has not been met, however the plan at the start of project had leeway for delays, as defined in the mitigation table in the PDS, which has meant that the designs have been completed by the Team SURTES design freeze on the 25th March 2018 and are on track to be manufactured by the competition. The final project Gantt chart is shown in Figure 79.

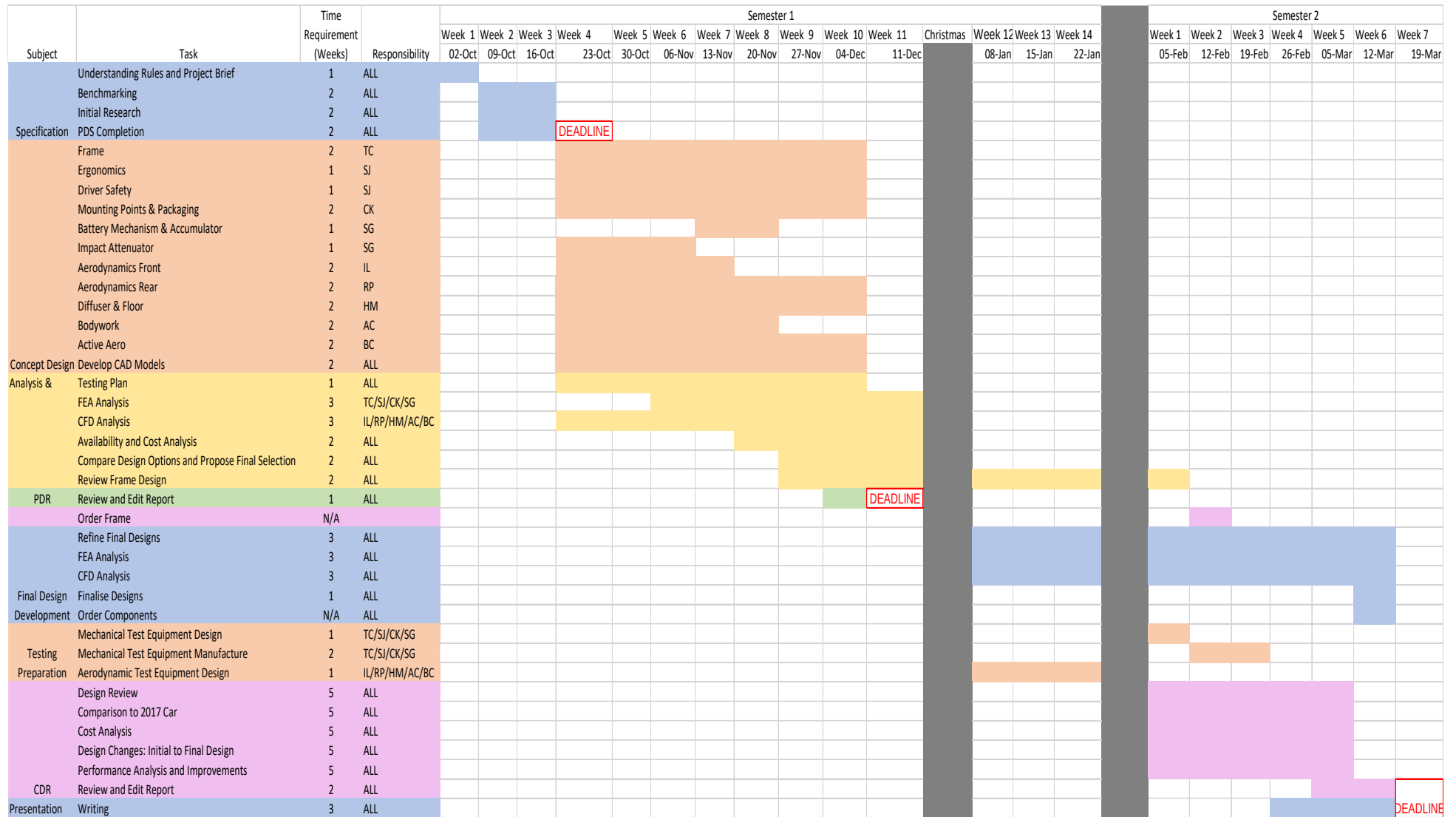


Figure 79: Gantt chart

15 OVERALL COST

The overall cost of all components can be seen in Table 23. Where possible, the cost of components and process has been minimised using in-house manufacturing and sponsored suppliers. However, quality and performance improvements were prioritised over cost.

Table 23: Cost Table

Area	Material Cost	Manufacturing Cost	Total Cost
Battery mount	£225.64	£0.00	£225.64
Bodywork	£348.22	£0.00	£348.22
Frame	£720.00	£4,080.00	£4,800.00
Front Wing	£337.26	£0.00	£337.26
Rear Wing	£460.00	£0.00	£460.00
Impact attenuator	£474.80	£0.00	£474.80
Rear wing mount/Structure	£434.74	£0.00	£434.74
Seat	£230.00	£0.00	£230.00
Firewall	£95.00	£0.00	£95.00
Head Restraint	£43.24	£0.00	£43.24
Floor	£798.00	£0.00	£798.00
Diffuser	£225.00	£0.00	£225.00
Overall Cost:			£8,471.90

16 OVERALL CONCLUSION

Overall the final design can be seen to meet most of the initial project requirements set out in the brief. The final chassis met the torsional stiffness requirement of 1500 Nm/deg, exceeding this by 165 Nm/deg. The mass of the final frame did not meet the initial target of 27 kg, exceeding this by 11.9kg, however this does include the mass of the structural floor, anti-intrusion plate and battery mount.

The current ergonomics were designed around the Percy template set out by the 2018 FS rules. The final position of the driver along with harness, seat and the head restraint have been shown using CAD and FEA to meet these requirements in all areas. The final angle of the driver (65°) has been optimised to minimise mass of the overall frame by improving the overall packaging. The final driver position was also considered with respect to the aero flow around the car, reducing drag and increasing aero efficiency.

After reviewing last years' IA design it was found that the previous material choice did not crush under loading, so all force was transferred to the AIP. This years' design used a pre-crushed aluminium honeycomb structure that crushes under impact absorbing the required 2018 FS energy set out in the rules. Although the test product has yet to arrive, a test plan has been created to validate theoretical results. This years' accumulator mount was designed around adaptability as well as compatibility with 2018 FS rules. Last years' design used an 8 mm thick plate mounted underneath the accumulator bolted to the frame. This design didn't have any capability to be fitted with different accumulator and had an overall weight of 6 kg. The new design used a new bracket design with variable bolting geometry to mount the different accumulators that could be used in the car. This resulted in a reduced weight of 1.5 kg and the benefit of improved packaging and assembly.

An aerodynamic package has been designed for this years' car aimed at reproducing 80% of Team Bath Racing's aerodynamic figures using aerodynamic devices including a floor, diffuser, front wing and rear wing. The initial figure of 238 N of downforce outlined in the PDS was not met for this year aerodynamic package but this can be improved upon for class 1 competition next year. The final full car analysis using Simscale showed an overall downforce figure of 112 N at 12 m/s. The manufacturing of this years' aerodynamic package has been re analysed against the SE1 design. This years' design has improved the aerodynamic efficiency of the overall package with a focus on improving manufacturing techniques to provide an improved aerodynamic flow and reduced weight. Rear wing mounting has been changed to an over wing design, improving air flow, whilst the wing structure has been redesigned around carbon composite spars and ribs.

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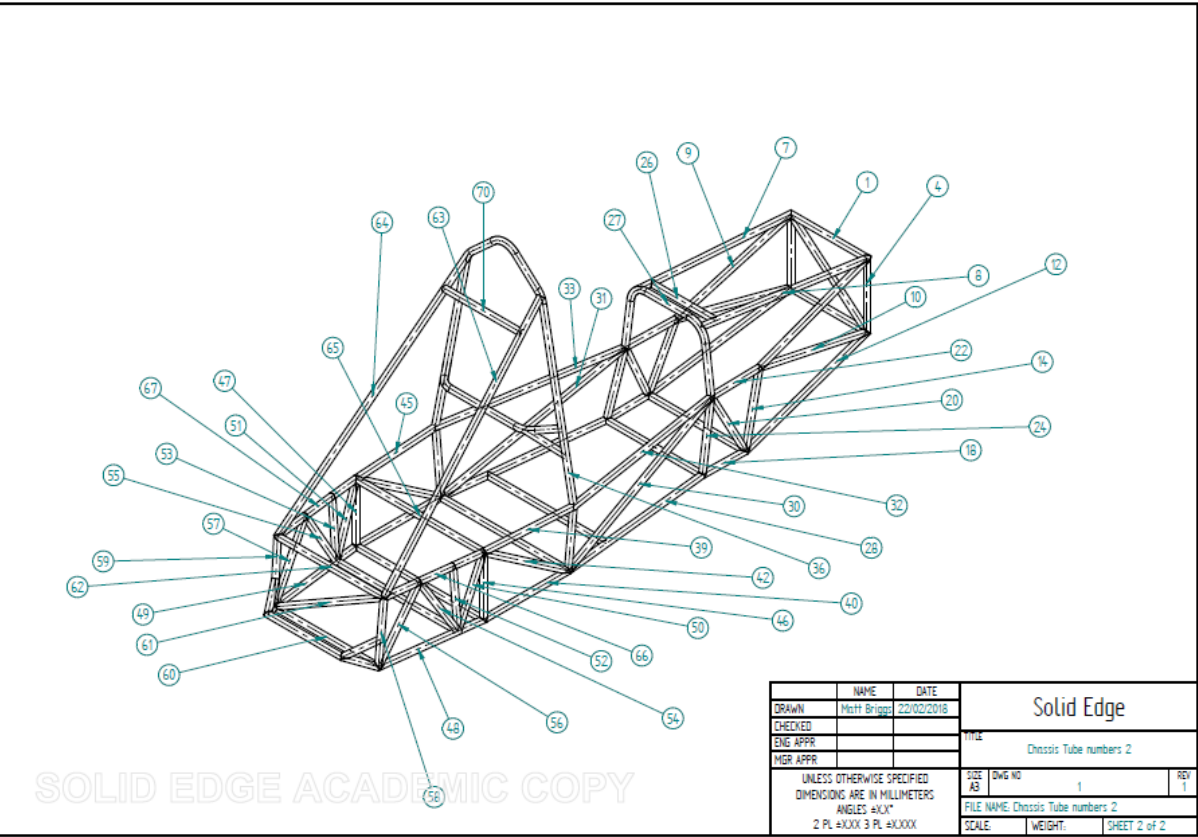
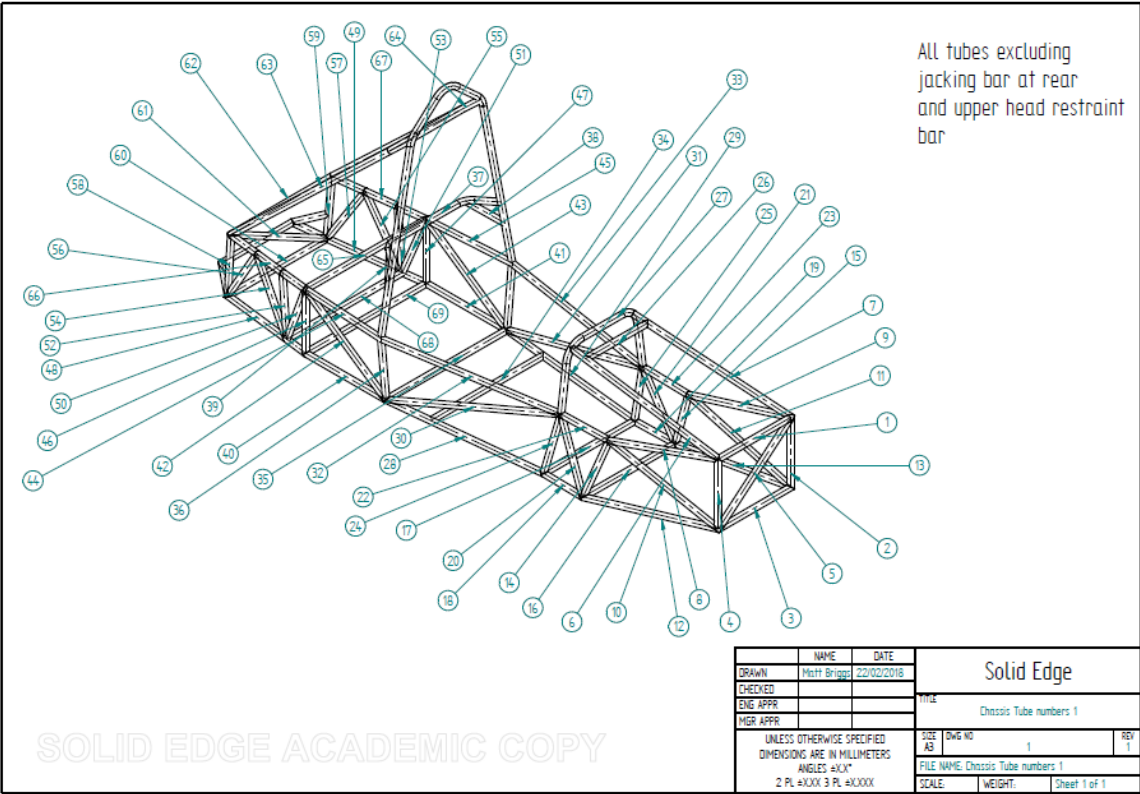
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Appendix 1 – FRAME COST TABLE

1.2mm Tube		1.6mm Tube		2.4mm Tube	
Upper Front Bulkhead Supports	1112.26	Front Bulkhead	1200	Rearward Front Uprights	579.96
Middle Front Bulkhead Supports	1084.96	Front Roll Hoop Supports	1523.08	Front Roll Hoop	824.95
Lower Front Bulkhead Supports	1291.76	Horizontal Infront of FRH	466.5	Main Roll Hoop	2583.63
Damper Support	333.94	Forward Front Upright	587.06	Shoulder Harness	596.19
1st X-Member	458.84	Upper Side Impact Member	1554.32	Shoulder Harness Supports	336.9
2nd X-Member	450.9	Diagonal Side Impact Member	1648.76	TOTAL	4921.63
3rd X-Member	573.54	Lower Side Impact Member	1422.2	Cost Per Metre T45	22
4th X-Member	491	Lower Harness Bar	558		108.2759
Upper Rear Horizontals	844.72	Lateral Battery Supports	1800	Two 4m lengths	176
Lower Rear Horizontals	887.92	Longitudinal Battery Supports	1600	CPM ROPT510	7.8
Battery to Upright Diagonal	590.76	Diagonal Battery Members	970.82		38.38871
Rear "A" Triangulation	1289.22	Forward Rear Upright	551.72	Two 4m lengths	62.4
Upper Rear Bulkhead Member	501.24	Rearward Rear Upright	551.72		
Lower Rear Bulkhead Member	491	Main Roll Hoop Supports	2271.66		
Diagonal Rear Bulkhead Member	567.62	TOTAL	16705.84		
Jacking Bar	350	Cost Per Metre T45	19.8		
Jacking Bar Upper Support	283.12		330.7756		
Jacking Bar Lower Support	268.3	Five 5m lengths	495		
Upper Head Restraint Bar	364	CPM ROPT510	7.2		
TOTAL	12235.1		120.282		
Cost Per Metre T45	20.15	Five 5m lengths	180		
	246.5373				
Three 5m lengths	302.25				
CPM ROPT510	8.75				
	107.0571				
Three 5m lengths	131.25				
P&P	35				
Total T45	720.5888				
Real Total	1008.25				
Total ROPT510	265.7279	Difference	454.8609		
Real Total	373.65				

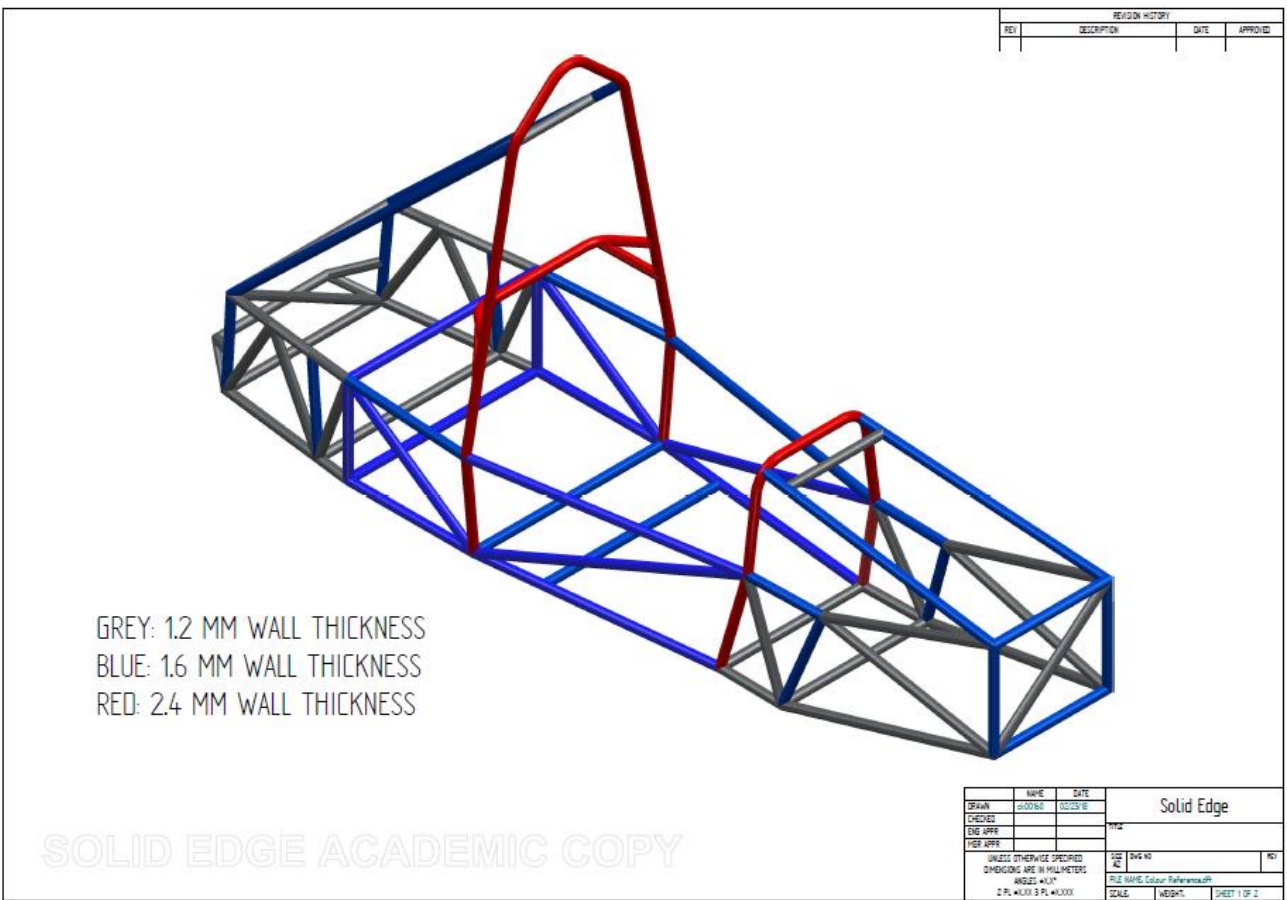
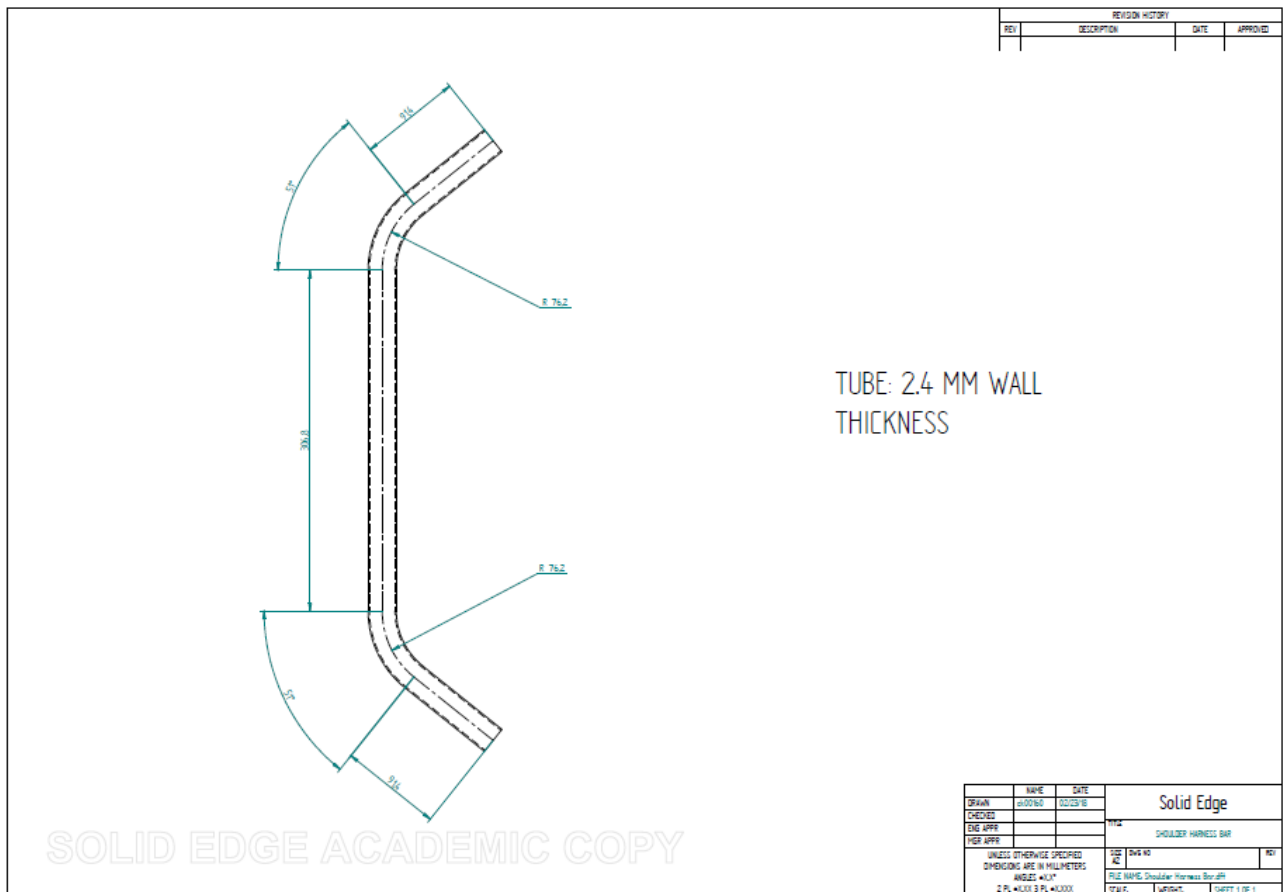
Appendix 2 – FRAME MANUFACTURING DOCUMENTS

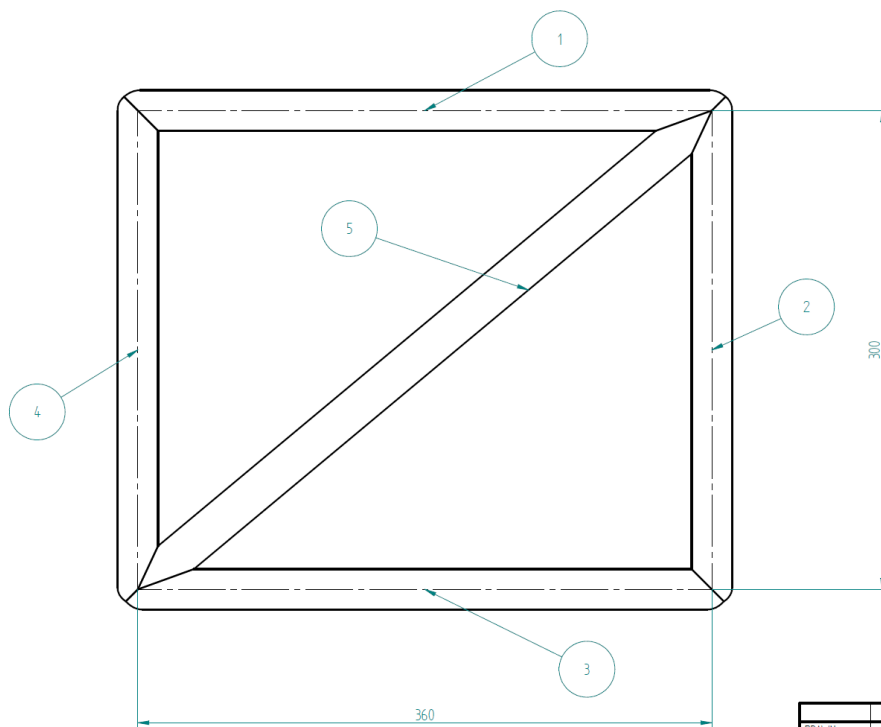


Coordinate System	x	y	z
2	0.0	180.0	150.0
3	0.0	180.0	450.0
4	623.2	229.4	0.0
5	525.8	197.1	275.0
6	822.8	225.5	0.0
7	759.0	202.7	275.0
26	0.0	-180.0	150.0
27	0.0	-180.0	450.0
28	623.2	-229.4	0.0
29	525.8	-197.1	275.0
30	822.8	-225.5	0.0
31	759.0	-202.7	275.0
56	1530.0	300.0	0.0
57	1530.0	-300.0	0.0
58	1530.0	325.0	275.0
59	1530.0	-325.0	275.0
60	1530.0	289.5	465.0
61	1530.0	-289.5	465.0
62	1530.0	270.0	565.0
63	1530.0	-270.0	565.0
68	1530.0	0.0	1190.0
69	1630.0	182.0	960.0
70	1630.0	-182.0	960.0
71	1930.0	300.0	0.0
72	1930.0	-300.0	0.0
73	1930.0	300.0	275.0
74	1930.0	-300.0	275.0
75	2037.0	286.8	0.0
76	2037.0	-286.8	0.0
77	2058.7	285.6	275.0
78	2058.7	-285.6	275.0
80	2202.3	268.3	275.0
81	2202.3	-268.3	275.0
82	2370.6	245.5	0.0
83	2370.6	-245.5	0.0
84	2362.2	247.6	110.0
85	2362.2	-247.6	110.0
86	2349.7	250.6	275.0
87	2349.7	-250.6	275.0
88	2470.6	175.0	55.0
89	2470.6	-175.0	55.0
90	1930.0	262.0	0.0
91	1930.0	-262.0	0.0
92	1530.0	262.0	0.0
93	1530.0	-262.0	0.0
96	1330.0	279.0	0.0
97	1330.0	-279.0	0.0
105	687.0	153.0	525.0
106	687.0	-153.0	525.0
107	759.0	150.2	532.7
108	759.0	-150.2	532.7
109	1622.5	186.4	565.0
110	1622.5	-186.4	565.0
111	1530.0	171.1	1055.6
112	1530.0	-171.1	1055.6

Tube Number	Name	Coordinate Systems
1	Front Bulkhead Top	3 - 27
2	Front Bulkhead Left	27 - 26
3	Front Bulkhead Bottom	26 - 2
4	Front Bulkhead Right	2 - 3
5	Front Bulkhead Cross Member	27 - 2
6	Front Roll Hoop Support Right	3 - 107
7	Front Roll Hoop Support Left	27 - 108
8	Front Bulkhead Support Top Right	3 - 5
9	Front Bulkhead Support Top Left	27 - 29
10	Front Bulkhead Support Middle Right	2 - 5
11	Front Bulkhead Support Middle Left	26 - 29
12	Front Bulkhead Support Bottom Right	2 - 4
13	Front Bulkhead Support Bottom Left	26 - 28
14	Front Front Right Upright	4 - 5
15	Front Front Left Upright	28 - 29
16		4 - 28
17		6 - 30
18		4 - 6
19		28 - 30
20	Front Right Uprights Triangulation Member	4 - 7
21	Front Left Uprights Triangulation Member	28 - 31
22		5 - 7
23		29 - 31
24	Rear Front Right Upright	6 - 7
25	Rear Front Left Upright	30 - 31
26	Damper Bar	106 - 105
27	Front Roll Hoop	7 - 31
28	Side Impact Structure Lower Right	6 - 56
29	Side Impact Structure Lower Left	30 - 57
30	Side Impact Structure Cross Member Right	7 - 56
31	Side Impact Structure Cross Member Left	31 - 57
32	Side Impact Structure Upper Right	7 - 58
33	Side Impact Structure Upper Left	31 - 59
34	Firewall and Harness Floor Bar	96 - 97
35		56 - 57
36	Main Roll Hoop	56 - 57
37	Harness Bar	62 - 63
38	Harness Support Left	61 - 110
39	Harness Support Right	109 - 60
40	Battery Lower Right	56 - 71
41	Battery Lower Left	57 - 72
42	Battery Triangulation Right	56 - 73
43	Battery Traingulation Left	57 - 74
44	Battery Upper Right	58 - 73
45	Battery Upper Left	59 - 74
46	Battery Upright Right	71 - 73
47	Battery Upright Left	72 - 74
48		71 - 82
49		72 - 83
50		75 - 73
51		76 - 74
52	Front Rear Upright Right	75 - 77
53	Front Rear Upright Left	76 - 78
54		75 - 80
55		76 - 81
56		82 - 80
57		81 - 83
58	Rear Bulkhead Right	82 - 86
59	Rear Bulkhead Left	83 - 87
60	Rear Bulkhead Bottom	82 - 83
61	Rear Bulkhead Cross Member	83 - 86
62	Rear Bulkhead Top	86 - 87
63	Main Roll Hoop Support Right	86 - 111
64	Main Roll Hoop Support Left	87 - 112
65		73 - 74

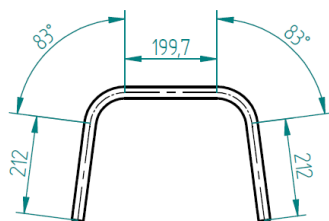
66		73 - 86
67		74 - 87
68		75 - 76
69		71 - 72
70	Upper Headrest Bar	69 - 70
71	Lower Right Jacking Bar Support	82 - 88
72	Upper Right Jacking Bar Support	84 - 88
73	Jacking Bar	88 - 89
74	Lower Left Jacking Bar Support	83 - 89
75	Upper Left Jacking Bar Support	85 - 89





SOLID EDGE ACADEMIC COPY

	NAME	DATE	Solid Edge		
DRAWN	mb00885	02/22/18	TITLE		
CHECKED					
ENG APPR					
MGR APPR					
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS ANGLES ±XX° 2 PL ±XXX 3 PL ±XXXX			SIZE A3	DWG NO	REV
			FILE NAME: Front bulkhead.dft		
			SCALE:	WEIGHT:	SHEET 1 OF 1



ALL BENDS 76.2MM RADIUS

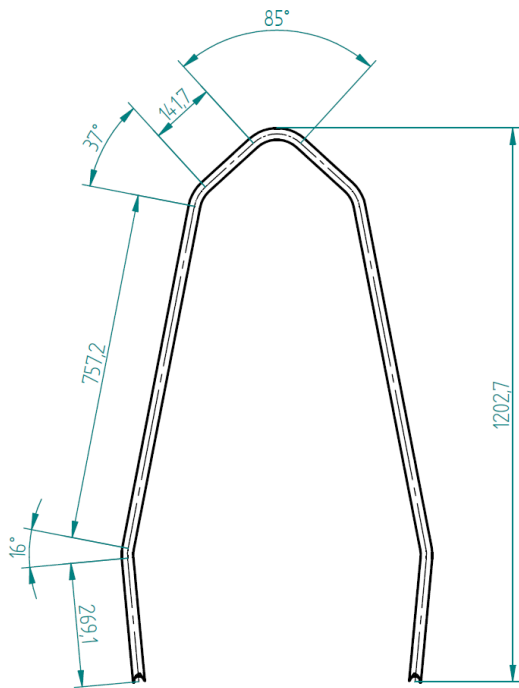
TUBE 25.4MM DIAMETER

2.4MM WALL THICKNESS

REVISION HISTORY			
REV	DESCRIPTION	DATE	APPROVED

SOLID EDGE ACADEMIC COPY

	NAME	DATE	Solid Edge		
DRAWN	mb00885	02/21/18	TITLE		
CHECKED					
ENG APPR					
MGR APPR					
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS ANGLES ±XX° 2 PL ±XXX 3 PL ±XXXX			SIZE A4	DWG NO	REV
			FILE NAME: Front Roll Hoop.dft		
			SCALE:	WEIGHT:	SHEET 1 OF 1



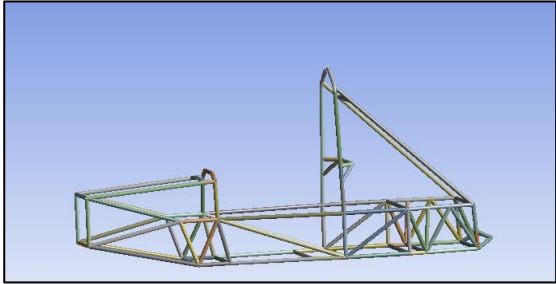
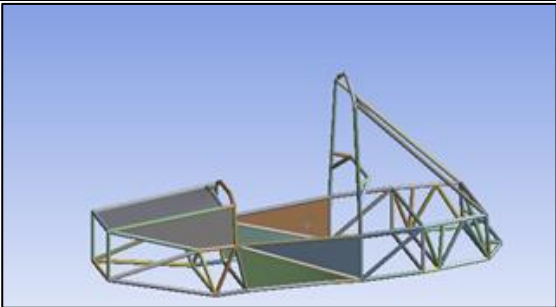
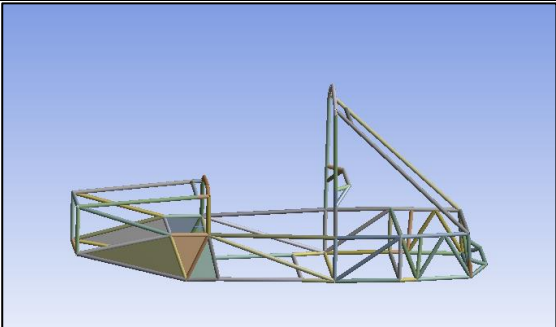
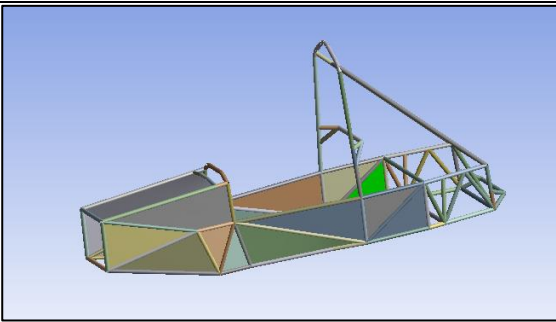
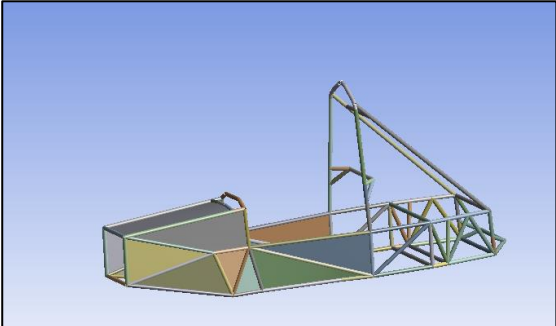
ALL BENDS 76.2MM RADIUS
TUBE 25.4MM DIAMETER
2.4MM WALL THICKNESS

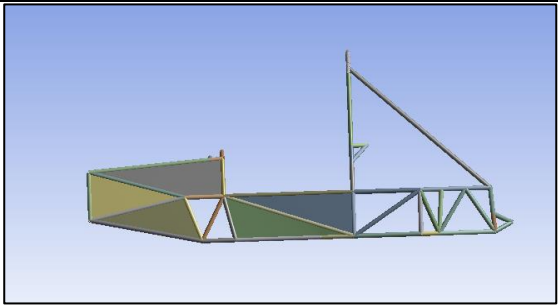
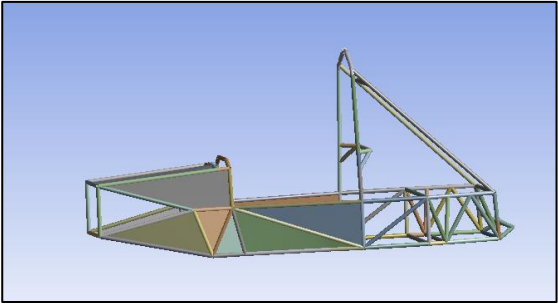
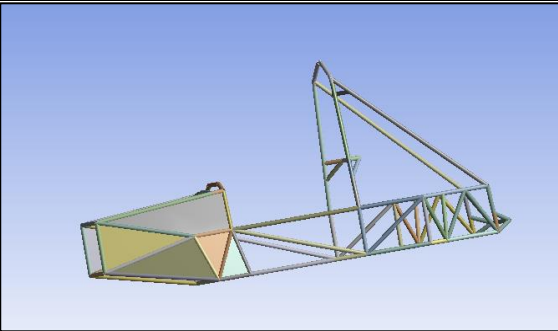
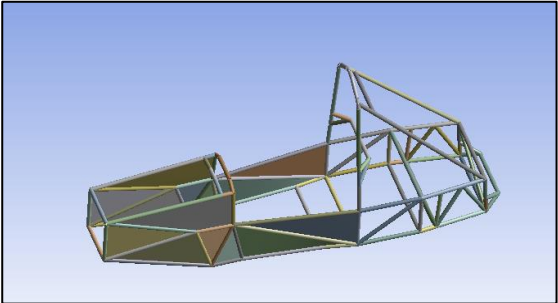
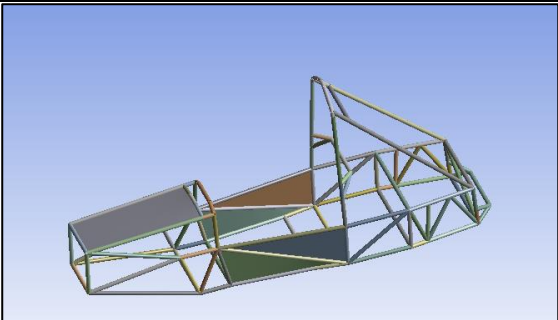
REVISION HISTORY			
REV	DESCRIPTION	DATE	APPROVED

SOLID EDGE ACADEMIC COPY

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DRAWN	mb00885	02/21/18		
CHECKED			TITLE	
ENG APPR			MAIN ROLL HOOP	
MGR APPR				
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS ANGLES ±XX° 2 PL ±XXX 3 PL ±XXXX			SIZE A4	REV
			FILE NAME: Main Roll Hoop.dft	
SCALE:		WEIGHT:	SHEET 1 OF 1	

Appendix 3 - ALTERNATE FRAME ITERATIONS

Iteration	Image	Mass (kg)	Torsional Stiffness (Nm/deg)
0		33.02	1231
1		34.26	1725
2		33.43	1247
3		35.23	1840
4		34.91	1813

5		34.74	1786
6		34.67	1762
7		34.33	1571
8		34.59	1464
9		33.93	1645

Appendix 4 – ENGINEERING DRAWINGS FOR ACCUMULATOR MOUNT

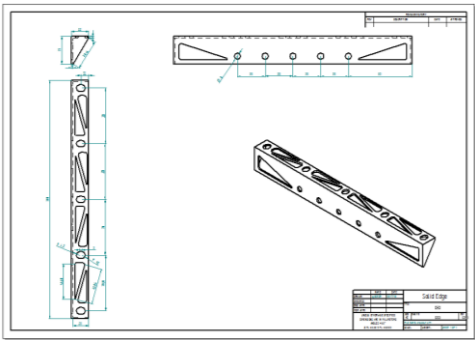


Figure 80: Engineering drawing of Altertek accumulator mount

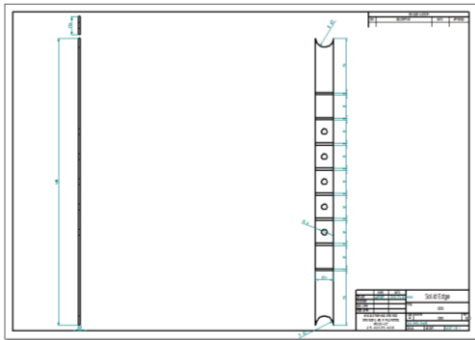


Figure 81: Engineering drawing of structural beams

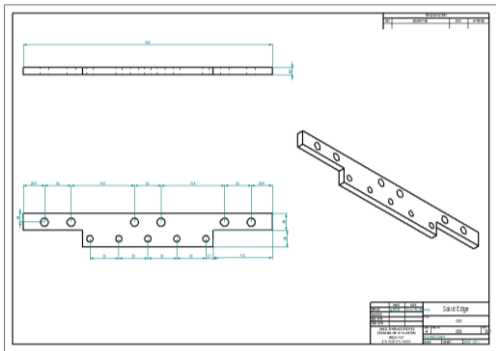


Figure 82: Engineering drawing of structural beams

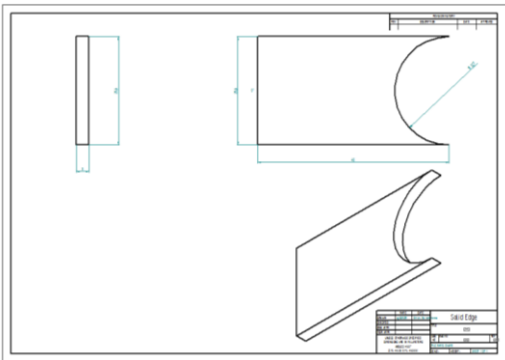


Figure 83: Engineering drawing of structural beams

Appendix 5 – BEAM THEORY CALCULATIONS FOR ACCUMULATOR MOUNT

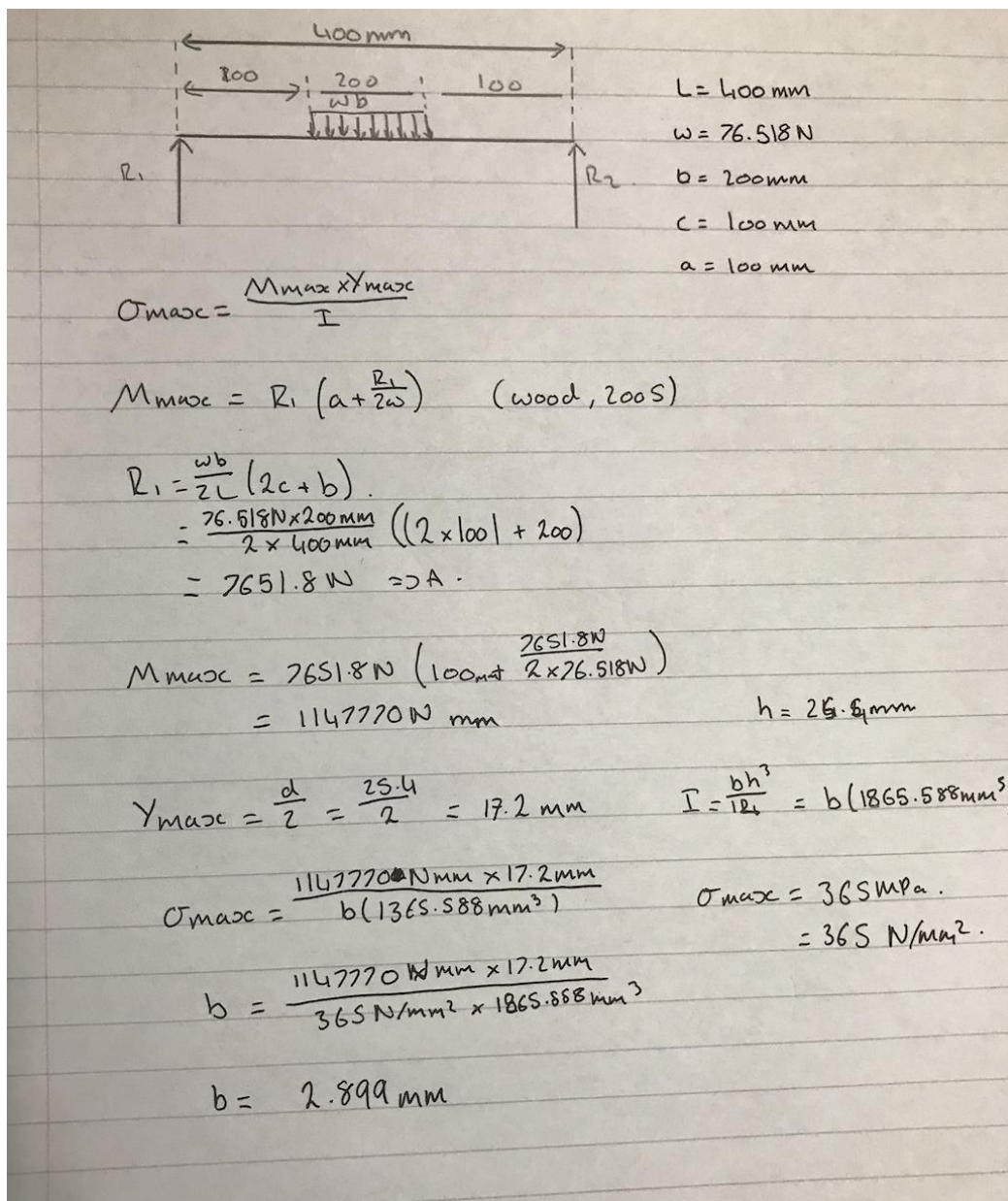


Figure 84: Beam theory calculations of the structural beams

Appendix 6 – BEAM THEORY FORMULAE USED FOR ACCUMULATOR MOUNT

- 10 bolts in shear
- 40g Acceleration
- 78kg Battery mass
- Bolt requirements: grade 8.8
 1. Proof strength = 580MPa
 2. Yield strength = 660MPa
 3. Tensile Strength = 830MPa

$$\tau = \frac{F}{A}$$
$$\tau_{10 \text{ bolt}} = \frac{78 * 40 * 9.81}{\pi \frac{8 * 10^{-3}}{4}}$$
$$\tau_{10 \text{ bolt}} = 608.91 \text{ MPa}$$
$$\tau_{1 \text{ bolt}} = 60.89 \text{ MPa}$$

Appendix 7 – COST ANALYSIS OF IMPACT ATTENUATOR

Component	Cost (£)	Material cost per m (£/m)	Material length required (m)	Quantity	Total cost
25.4mm frame with 1.6mm thickness	N/A	19.8	2.80	2	£110.88
25.4mm frame with 1.2mm thickness	N/A	20.5	£0.46	2	£0.92
IA	45	N/A	N/A	4	£180.00
AIP	61	N/A	N/A	3	£183.00
Total cost					£474.80

Appendix 8 – CALCULATIONS OF FORCES ON FIREWALL

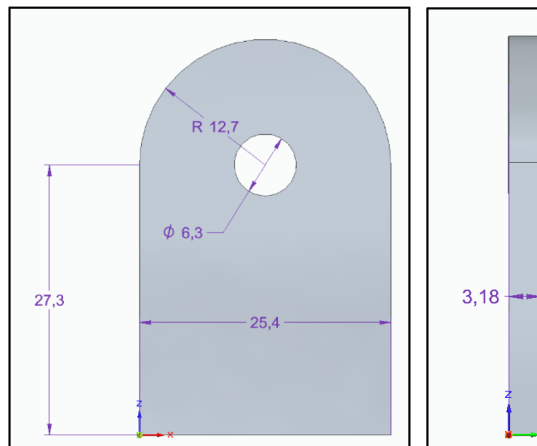
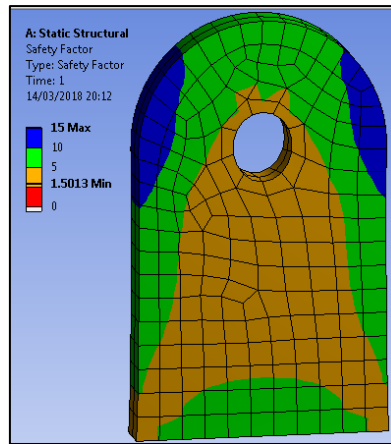
In the x- direction only the reaction force from the peal box will act on the firewall.

- $F_x = 2000 \cos (65) + 540 \cos (65)$

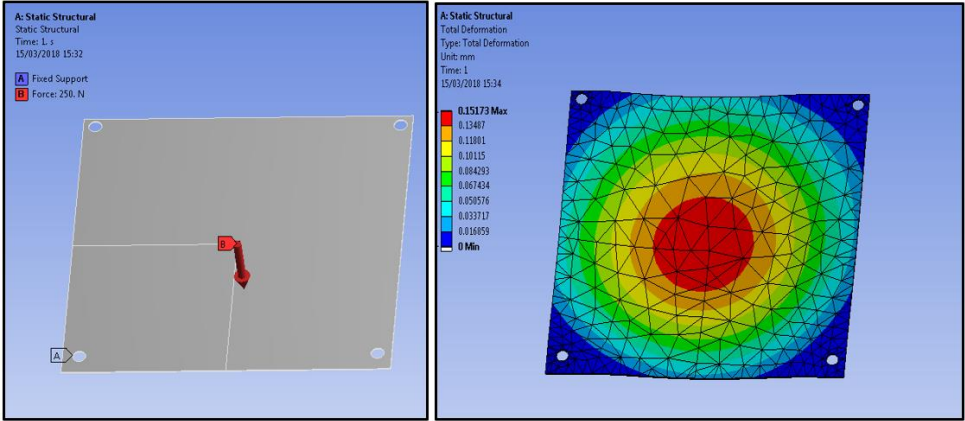
The maximum force in the y direction will be the bump load from the driver's mass and the firewall's mass.

- $F_y (\text{Bump Load}) = (54 \times 3g) + (2 \times 3g)$

- For bolt size $d = \sqrt{\frac{F}{0.58(\pi 180 \text{ MPa})}}$



Appendix 9 – FIREWALL TEST JIG AND FEA ANALYSIS



Appendix 10 – DRIVER HARNESS BOLT CALCULATIONS AND TABS

$$\text{Yield shear stress of bolt} = \frac{F}{\pi d^2}$$

$$0.58 (\text{Maximum Yield strength}) = \text{Yield shear stress of bolt}$$

Rearranging and combining above equations:

$$d = \sqrt{\frac{2F}{0.58(\pi 180 \text{ MPa})}}$$

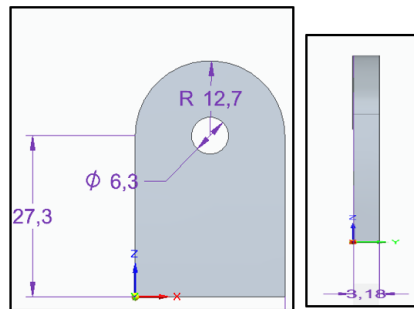


Figure 85: Shoulder Harness Tabs

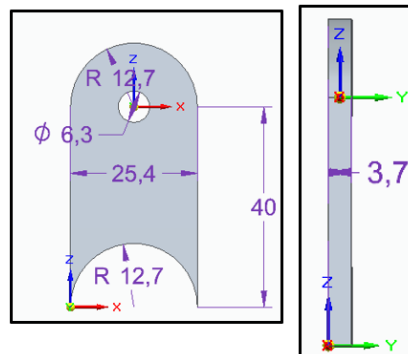
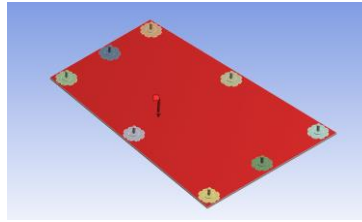


Figure 86: Submarine and lap belt final tabs

Appendix 11 - ANSYS SETUP AND DATA RECORDED FOR FLOOR

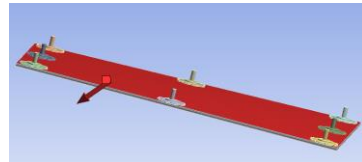
Front Section



-Expected to withstand a maximum of 2kN based on **FSG rules** for pedals (

Loads (N)	Total Deformation (mm)	Stress Eqv (MPa)	Normal Stress (MPa)
2000	0.55	22	26
4000	1.10	46	53
6000	1.60	69	79

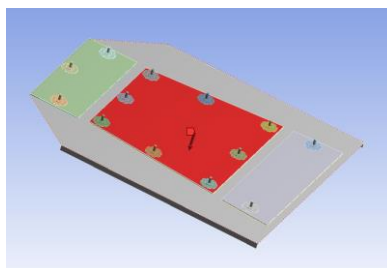
Front Section – Pedal Analysis



Pedal force analysis for the front panel only. Conducted using ANSYS FEA and component forces that were hand calculated and using the max angle the pedals would bend by and limitation on f

Net Loads (N)	Total Deformation (mm)	Stress Eqv (MPa)	Normal Stress (MPa)
2000 @ 60 deg	0.48	20.3	23
2000 @ 70 deg	0.52	22.1	25.2
2000 @ 50 deg	0.44	18.6	21.2

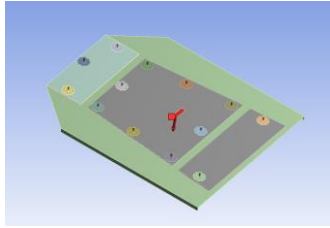
Mid-Section



-Expected to take driver weight of 750 N
 -Bump forces which means a safety factor of '3g' to then make a load of 2250 N
 -Forces added to mid-mid-section only to ensure

Loads (N)	Total Deformation (mm)	Stress Eqv (MPa)	Normal Stress (MPa)
750	0.0056	0.26	0.074
1500	0.011	0.53	0.15
2100	0.016	0.75	0.2
2850	0.022	1.05	0.28
3600	0.026	1.28	0.35
4350	0.032	1.55	0.42

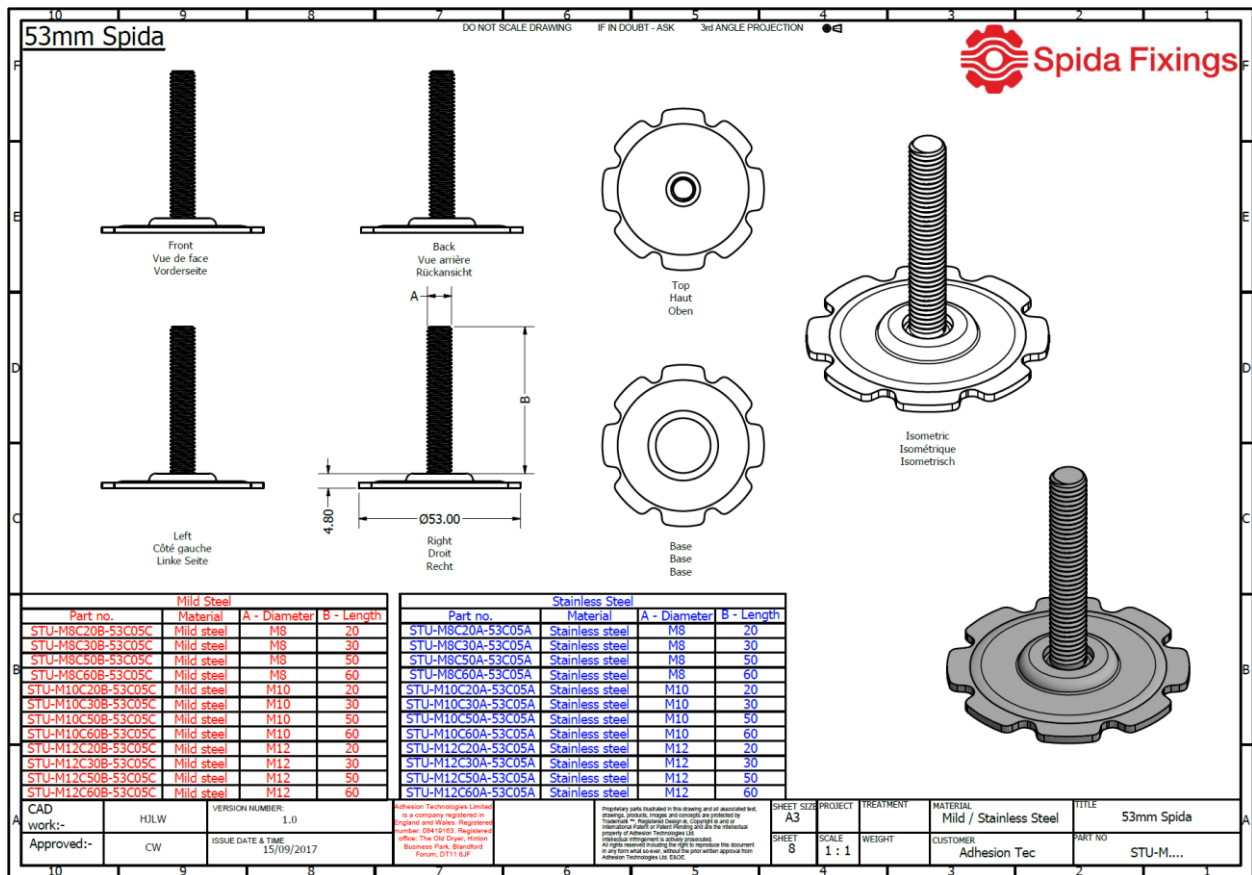
Mid-section with Shoe Heel test



- Force applied in the cross section with dimensions imitating a heel (70x20mm)
- Forces applied in increments of 750 N

Loads (N)	Total Deformation (mm)	Stress Eqv (MPa)	Normal Stress (MPa)
750	0.02	27	11
1500	0.05	55	23
2100	0.07	77	33
2850	0.1	105	44
4000	0.13	148	63
6000	0.21	222	94

Appendix 12 - SPIDA FIXINGS 53 MM M8 DRAFT



Appendix 13 - VARIOUS PRE-TREATMENTS DEFINED

Mechanical pre-treatment

Mechanical abrasion is the most widely applicable surface preparation technique, being suitable for most materials. Mechanical abrasion will remove weak boundary layers. It will also change the surface topography of the adherend, increasing the bondable surface area on a microscale. Furthermore, mechanical abrasion will enhance the adhesive's ability to 'wet' (when the adhesive readily and completely covers) the surface of the adherend.

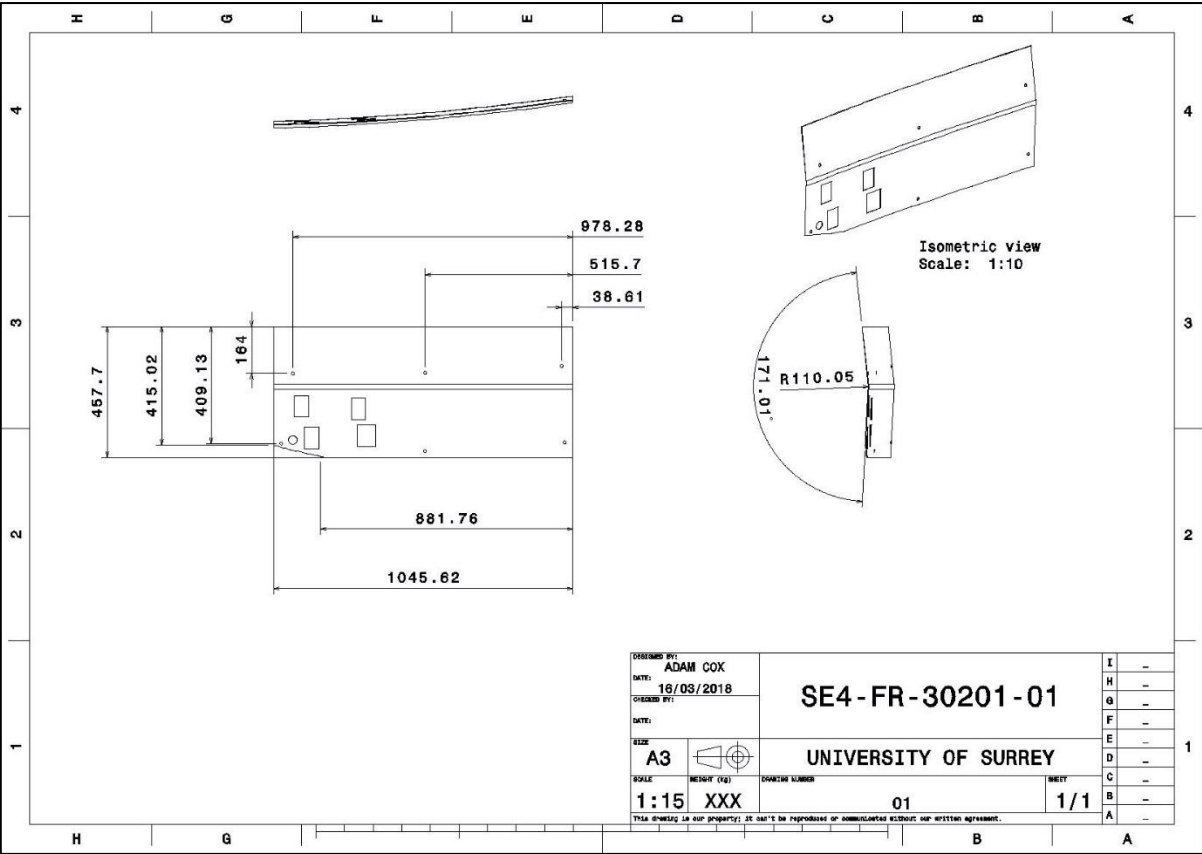
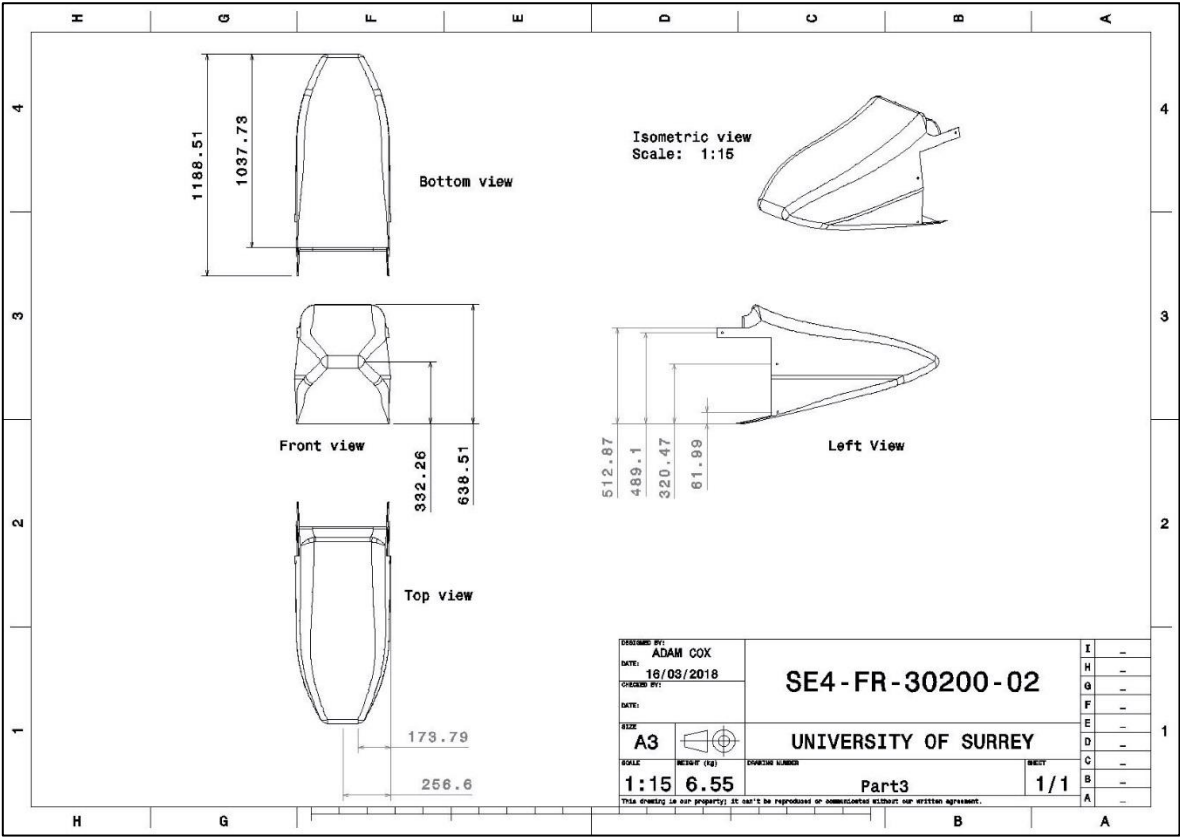
Chemical pre-treatment

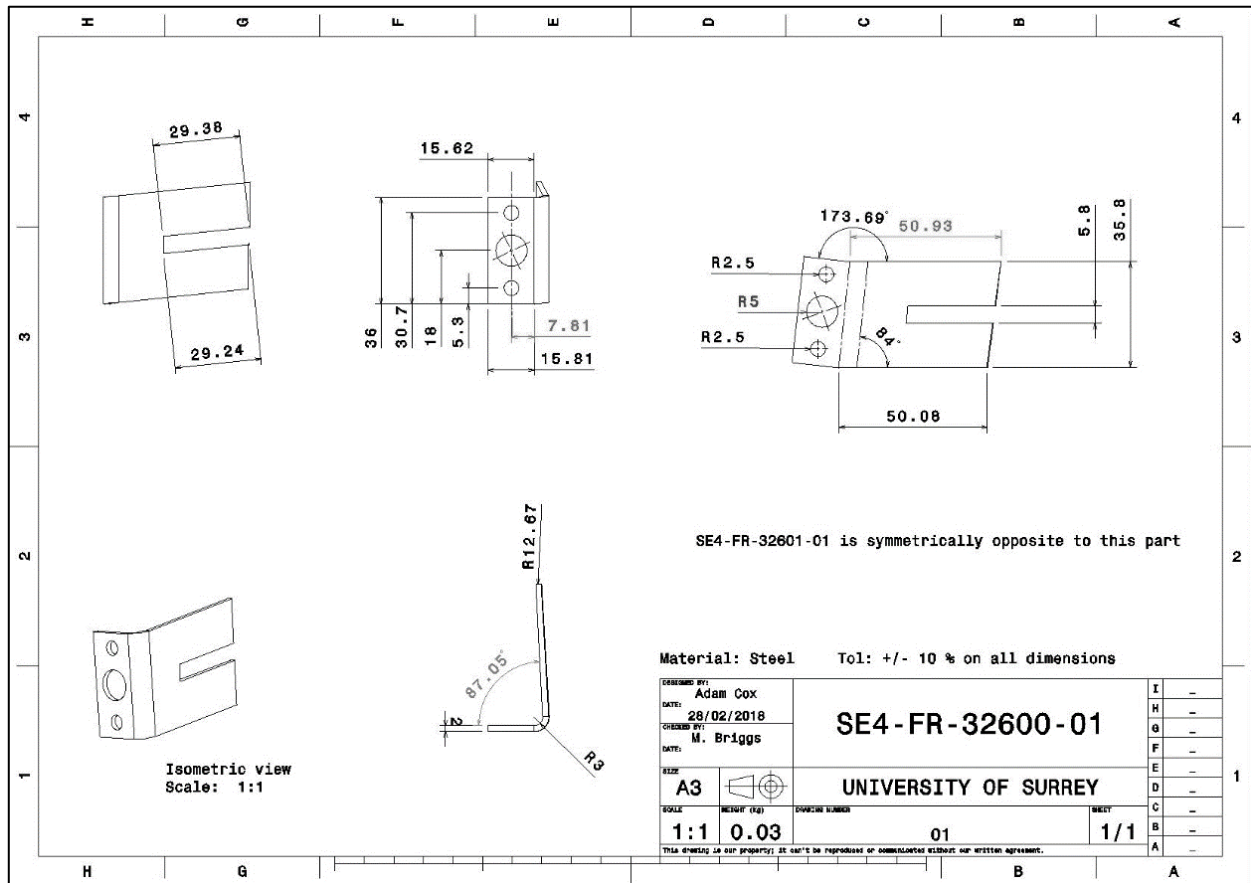
There are several different chemical methods by which adherends can be cleaned and prepared for bonding including: solvent cleaning, detergent wash, acid etch, anodising and primers.

Energetic pre-treatment

Energetic surface pre-treatments which have been reported in the literature include flame, corona discharge and plasma (FCDP) and excimer laser. All of these procedures cause a change in the surface texture of the adherend, brought about by the interaction of highly energetic species with the adherend surface. These pre-treatment methods have been applied to metals and in particular composites and plastics.

Appendix 14 - FINAL BODYWORK DESIGN

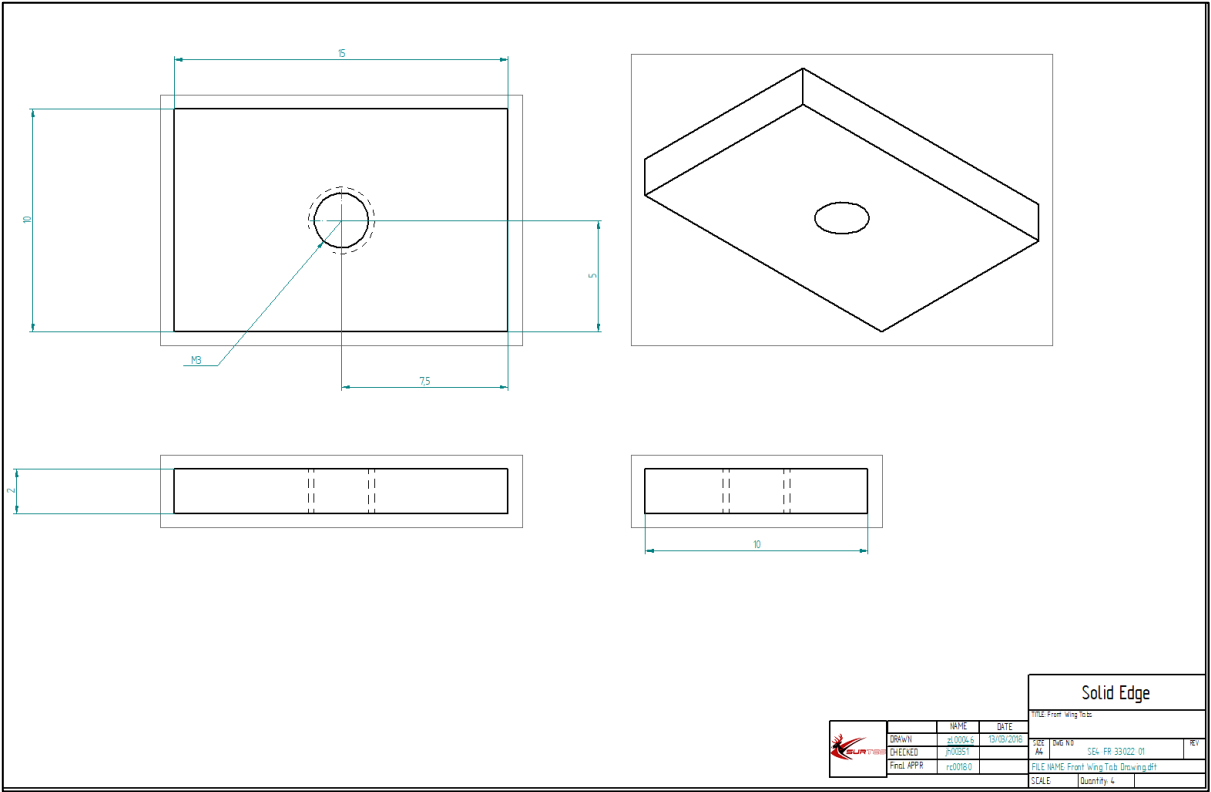




Appendix 15 - BODYWORK COST BREAKDOWN

DESCRIPTION	QUANTITY	UNIT COS	Line Cost
2mm Laser Grade MDF 900 x 600 mm	32	£2.95	£94.40
4mm Laser Grade MDF 1200 x 600 mm	6	£6.45	£38.70
Wood glue - evo stikwood adhesive interior 500ml	1	£5.99	£5.99
Priming Yellow Firestop Putty Aerosol, 500 ml	1	£10.49	£10.49
Biaxial Glass Cloth 320g 1.27m Wide	15	£3.14	£47.10
EL2 Epoxy Laminating Resin 1kg FAST	2	£16.75	£33.50
Lantor Coremat 1.4mm	4	£3.75	£15.00
EL2 Epoxy Laminating Resin 5kg FAST	2	£54.94	£109.88
Pitking Products Lightweight Aluminium 1/4 Turn Quick Release Self Eject Fasteners	22	£3.38	£74.36
DZUS FASTENER ATTACHMENT SPRING	22	£0.60	£13.20
Total Cost			£348.22

Appendix 16 - FRONT WING MOUNTING TAB ENGINEERING DRAWING



Appendix 17 - REAR WING MATERIAL COST ESTIMATE

Prepreg Carbon

Wing surface

$$1^{\text{st}} : 755.9 + 19.54 * 2 + 2 = 796$$

$$2^{\text{nd}} : 576.55 + 19.94 * 2 + 2 = 618$$

$$3^{\text{rd}} : 173.61 + 194.69 + 2 * 19.81 + 2 = 409$$

$$\text{Total} = 1823 \text{mm} = 1.823 \text{m}$$

$$\text{Area} = 1.78 \text{m}^2$$

$$\text{Volume} = 0.00178 \text{m}^3$$

Spare

$$57 + 57 + 114 + 42 + 60$$

$$53 + 43 + 86 + 31 + 67$$

$$30.5 + 28 + 66 + 15 + 26$$

$$775.5 \text{mm} = 0.7755 \text{m}$$

$$\text{Area} = 0.76 \text{m}^2$$

$$\text{Volume} = 0.00076 \text{m}^3$$

Total prepreg

$$0.00254 \text{m}^3 * 1750 \text{kg/m}^3$$

$$4.44 \text{kg}$$

End plates

$$0.52 \text{m}^2 * 2.71 \text{kg/m}^2 = 2.82 \text{kg}$$

$$7.26 \text{kg}$$

Appendix 18 - DETAILS OF EACH REAR WING

Design	Major Elements	Minor elements per major element	Chord lengths (m)
V1	2	1, 1	0.35, 0.35
V2	2	2, 2	0.13, 0.13, 0.13, 0.13
V3	2	3, 1	0.38, 0.20, 0.13, 0.20
V4	2	2, 2	0.3, 0.14, 0.14, 0.14
V5	2	2, 2	0.2, 0.2, 0.2, 0.2
V6	3	2, 2, 2	0.2, 0.2, 0.2, 0.2, 0.2, 0.2
V7	1	3	0.37, 0.2, 0.15,
V8	1	3	0.37, 0.2, 0.15,
V9	1	3	0.37, 0.2, 0.15,
V10	1	3	0.37, 0.2, 0.15,
V11	1	3	0.37, 0.2, 0.15,

Appendix 19 - ROTATION TRANSFORMATION

$$\begin{bmatrix} x' \\ y' \end{bmatrix} = \begin{bmatrix} \cos \theta & -\sin \theta \\ \sin \theta & \cos \theta \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix}$$

Excel Formulae (scaling, rotation, and translation)

X coord'=chord*(xcoord*cos(α)+ycoord*sin(α))+x distance

Y coord'= chord *(-xcoord*sin(α)+ycoord*cos(α))+y distance

Boundary conditions:

- Velocity inlet 13m/s
- 1.01Bar Pressure outlet
- Symmetry wall at half span
- Slip wall for top bottom and non-symmetry side face
- Non-slip wall for wing surface

Appendix 20 - DRS CONCEPT SKETCHES

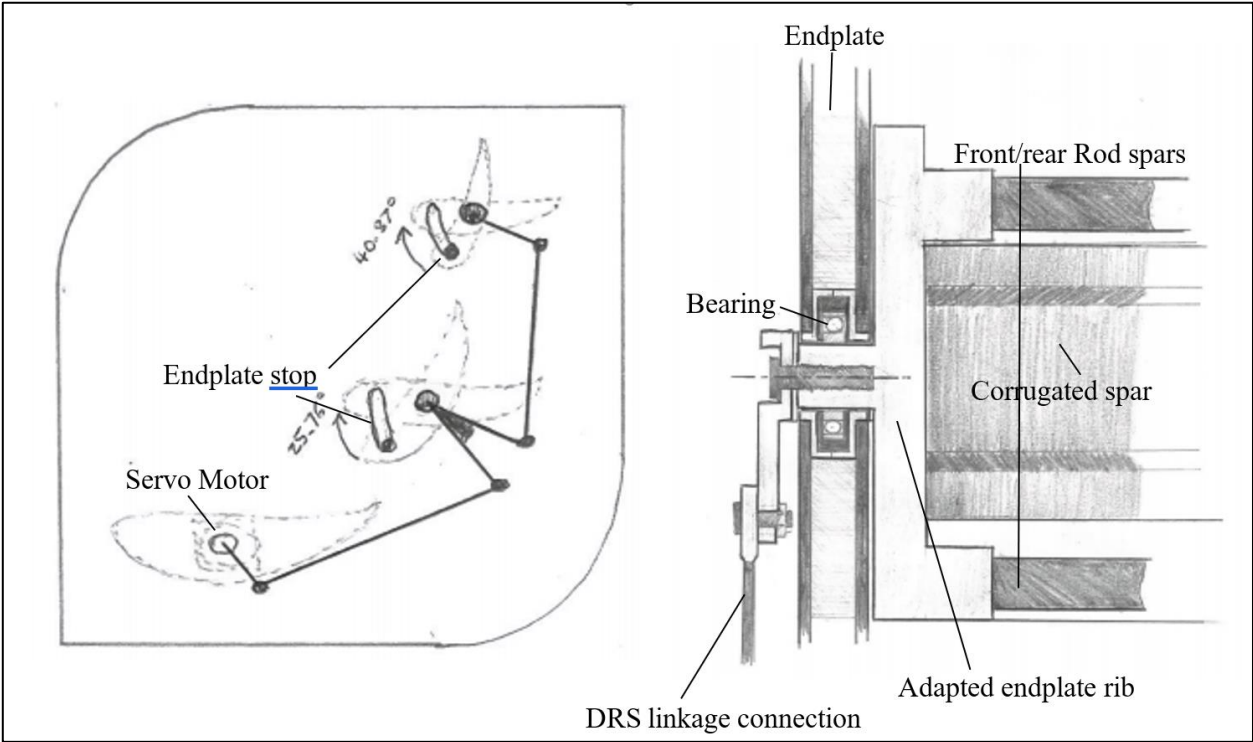


Figure 87: DRS concept sketches

Appendix 21 - ENGINEERING DRAWINGS

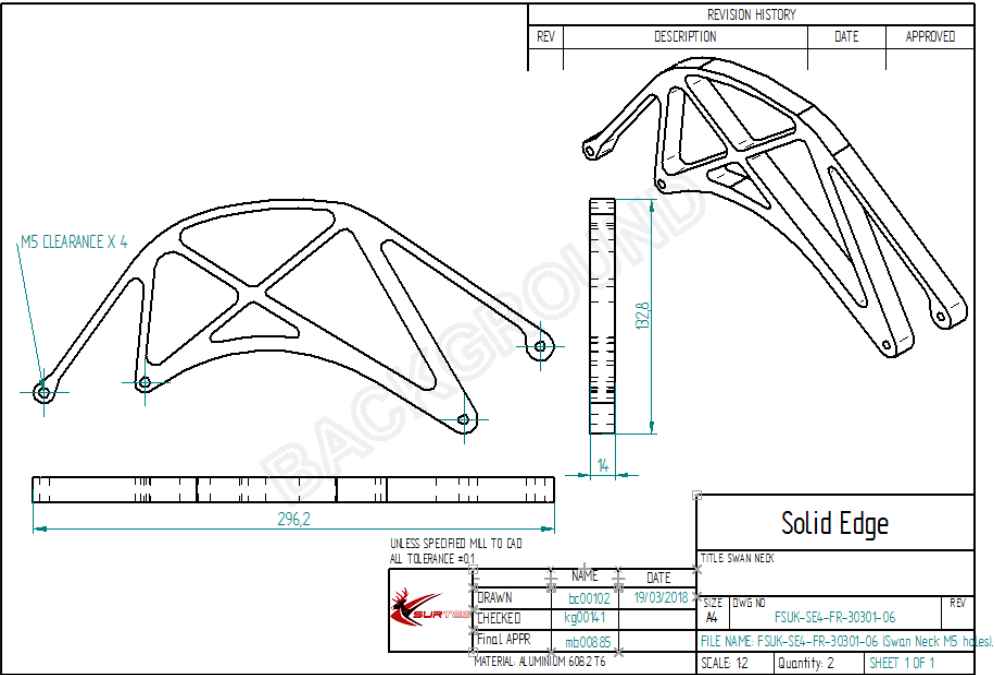


Figure 88: Final Swan Neck Design

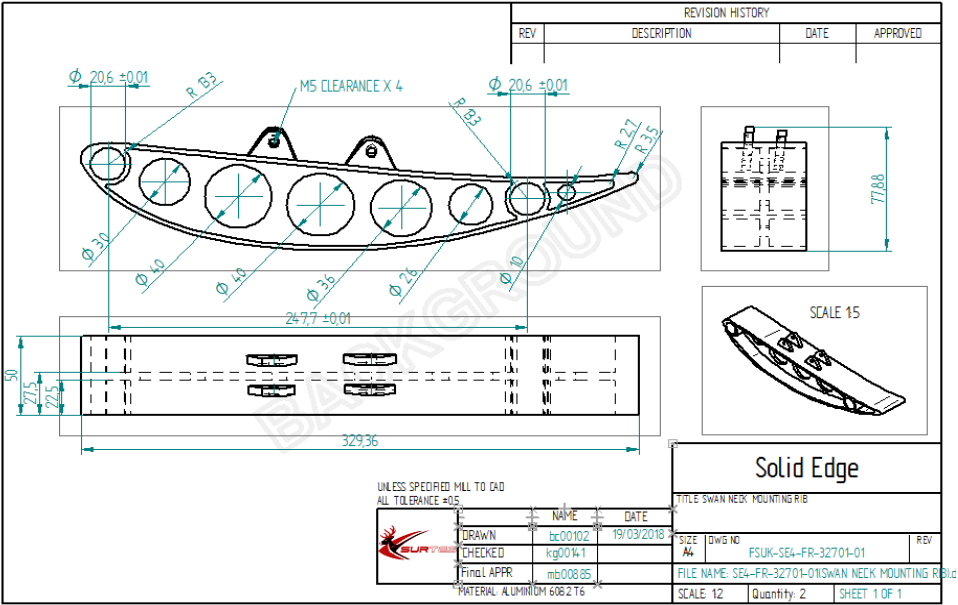


Figure 89: Final Swan Neck Mounting Rib Design

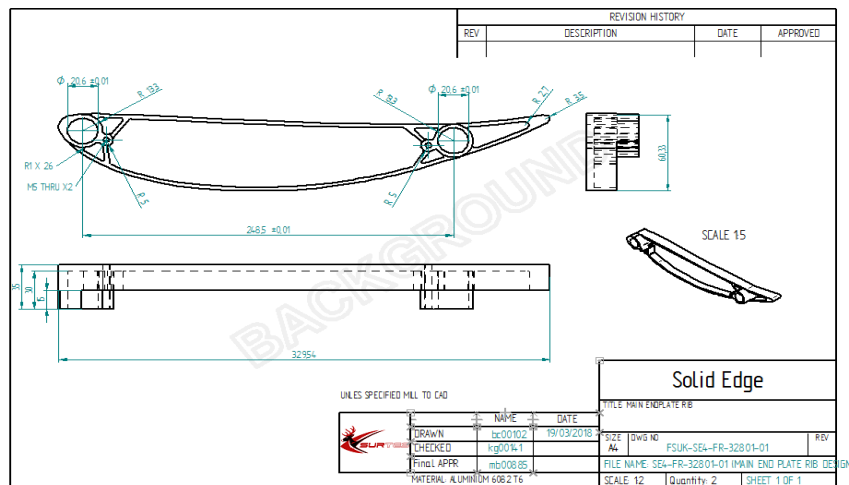


Figure 90: Main Element Wing Endplate

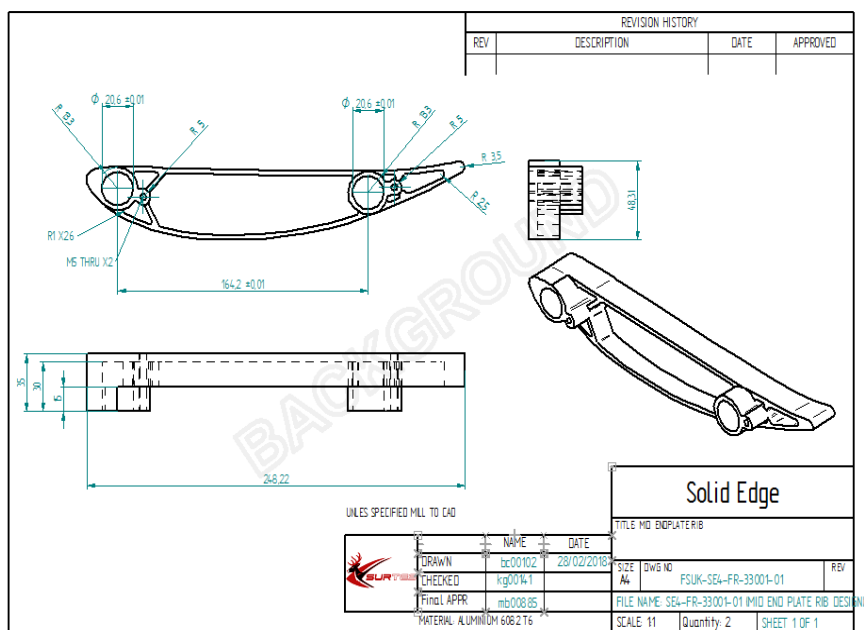


Figure 91: Top Element Wing Endplate

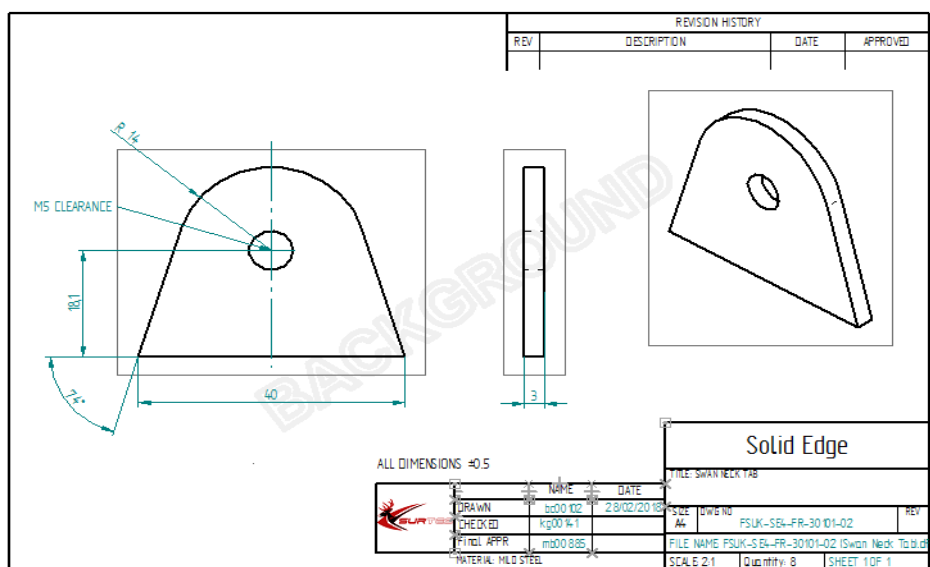


Figure 92: Final Swan Neck Tab Design

Appendix 22 – SWAN NECK FEA RESULTS

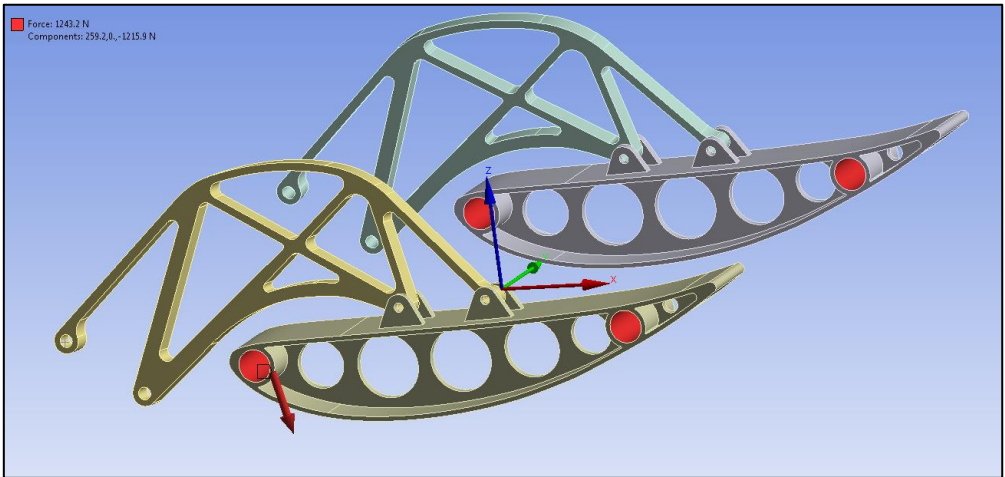


Figure 93: Loading case 1 demonstration on swan necks and the attaching internal ribs

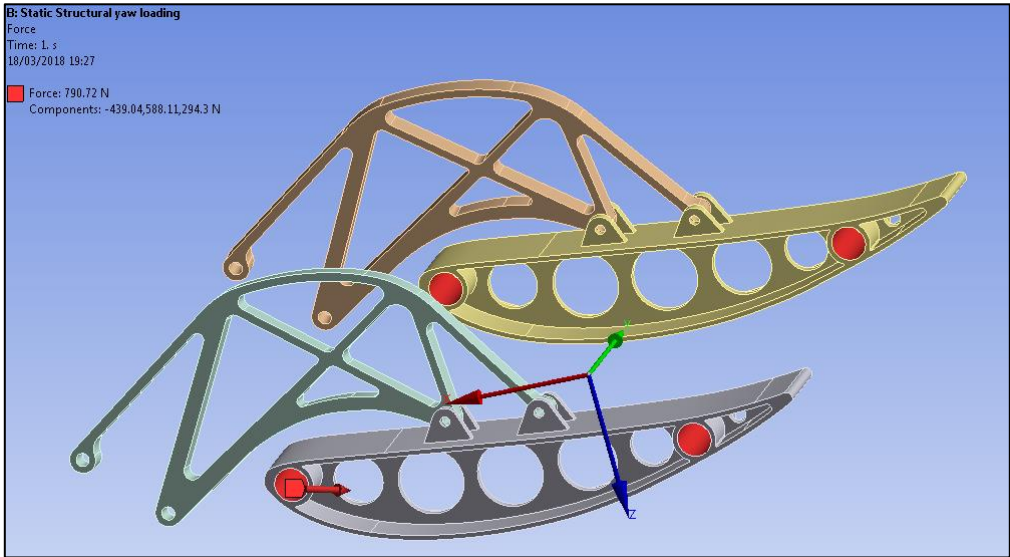


Figure 94: Loading case demonstrated on swan necks and attaching internal ribs

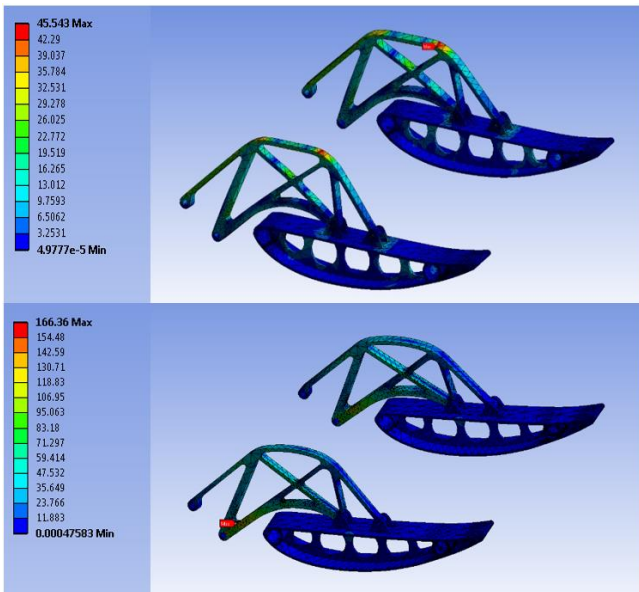


Figure 95: (Top) Equivalent Stress on swan necks and mounting ribs from loading case 1, (Bottom) Equivalent Stress from loading case 2

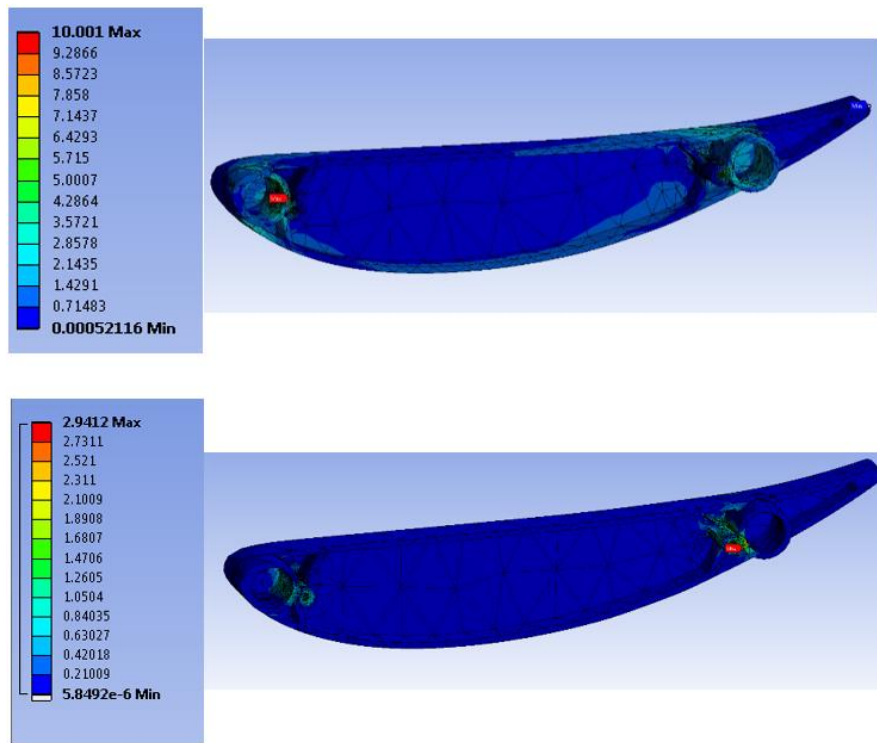


Figure 96: (Top) Equivalent Stress on End plate design from loading case 1, (Bottom) Equivalent Stress from loading case 2

Appendix 23 – SWAN NECK COST BREAKDOWN

Table 24: Cost break down of all materials used

COMPONENT NAME	PART	COMPANY	DESCRIPTION	EQUIPMENT	COSHH	QUANTITY	UNIT COST (exc. VAT)	Line Cost
FSUK-SE4-FR-30301-05	SWAN NECK	Aluminium Warehouse	330mm x 160mm x 20mm 6082T6 Aluminium Plate	N	N	2	£20.59	£41.18
FSUK-SE4-FR-32701-01	SWAN NECK MOUNTING RIB	Aluminium Warehouse	375mm x 100mm x 70mm 6082T6 Aluminium Plate	N	N	2	£51.19	£102.38
FSUK-SE4-FR-32801-01	MAIN END PLATE	Aluminium Warehouse	375mm x 100mm x 30mm 6082T6 Aluminium Plate	N	N	2	£21.94	£43.88
FSUK-SE4-FR-33001-01	MID END PLATE	Aluminium Warehouse	300mm x 100mm x 30mm 6082T6 Aluminium Plate	N	N	2	£17.55	£35.10
FSUK-SE4-FR-33101-01	TOP END PLATE	Aluminium Warehouse	200mm x 100mm x 30mm 6082T6 Aluminium Plate	N	N	2	£11.70	£23.40
FSUK-SE4-FR-33201-01	20MM CARBON ROD 1M	PFT Profiles	20mm OD 17mm ID x 1m	N	N	4	30	120
FSUK-SE4-FR-33301-01	14MM CARBON ROD 1M	PFT Profiles	14mm OD 10mm ID x 1m	N	N	2	25	50
	Araldite 2015 adhesive	Farnell	2 part epoxy paste	N	Y	1	£16.68	£16.68
	M5 SHANKED BOLT	Magnor	M5 x 35 H/T Hex Bolt, Zinc Plated	N	N	8	£0.10	£0.80
	M5 BOLT	Magnor	M5 x 50 H/T Hex Setscrew, Zinc Plated	N	N	12	£0.11	£1.32

Appendix 24 – SWAN NECK MATERIAL PROPERTIES AND BENDING CALCULATIONS

Carbon 20x17		
Vezel volume	62,64	%
Totaal gewicht	137	g
Tensile strenght	2.128,12	Mpa
Tensile modulus	118,78	Gpa

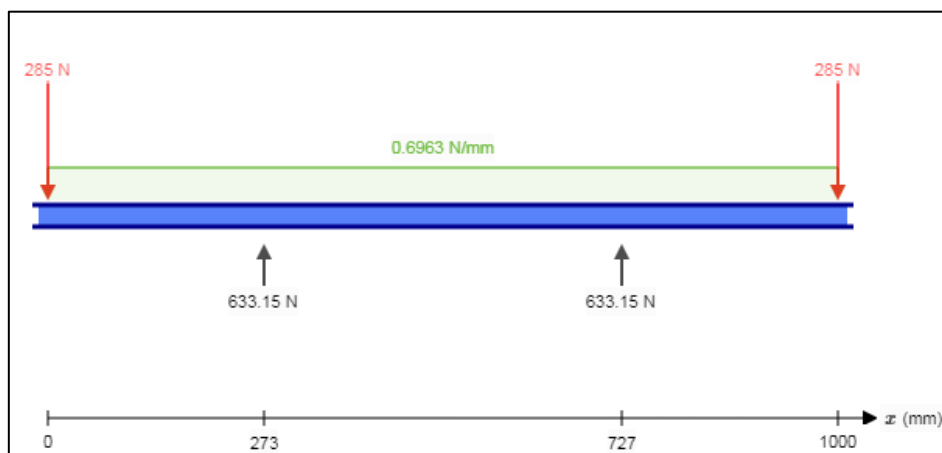


Figure 97: Bending load case on main plane

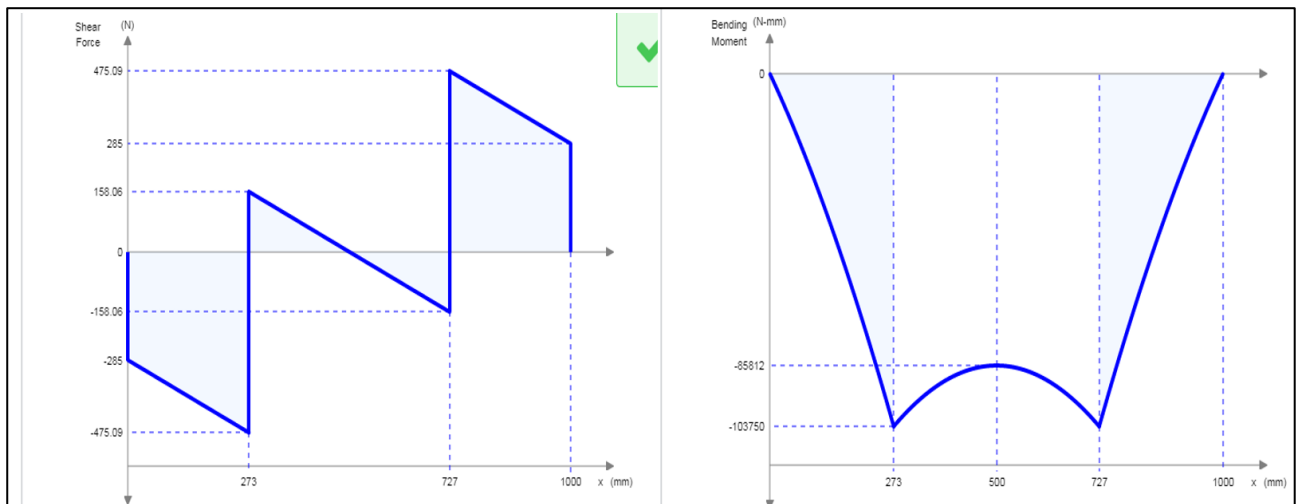


Figure 98: Resulting bending moment and shear force diagram (Skyciv, n.d.)

Using the above Figure 97 and the carbon rod properties the following symmetric bending calculations were undertaken:

Eqn 10 was used to calculate the second moment of area for the hollow section of rod, using 20 as D and 17 as d .

$$I = \frac{\pi D^4}{64} - \frac{\pi d^4}{64} = 3754.15 \text{ mm}^4$$

Eqn 10

This was then combined with the maximum bending moment about the x axis of 108750 Nmm and subbed into Eqn 11 to find the curvature of the beam

$$\begin{bmatrix} K_y \\ K_x \end{bmatrix} = \frac{1}{E} \begin{bmatrix} \frac{1}{I_{xx}} & 0 \\ 0 & \frac{1}{I_{yy}} \end{bmatrix} = \begin{bmatrix} M_x \\ M_y \end{bmatrix}$$

Eqn 11

The results from the equation can be subbed into Eqn 12 below to find the max tensile stress by using the max location from the centroid of the tub to be the outer diameter.

$$\sigma_z = xEK_x + yEK_y$$

Eqn 12

The result from this equation shows a max tensile stress of 552.72 MPa in the axial direction of the rod. This is far lower than the tensile strength of the carbon rod however does assume isotropic properties for the carbon.

Appendix 25 – SWAN NECK MATERIAL PROPERTIES AND BENDING

Topology tool was used to reduce mass by 60% and then using the results from the topology to best reduce material, with each iteration being FEA analysed against the two loading cases specified. An example of the iteration process between iterations 2 and 3 can be seen below in Figure 99.

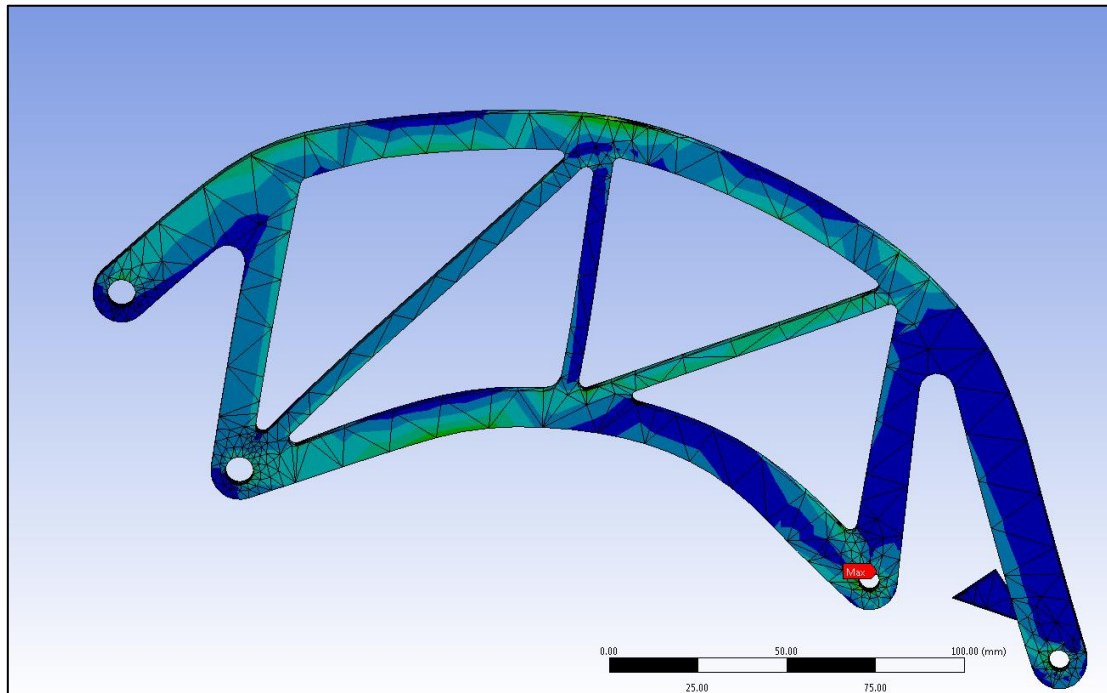


Figure 99: Swan neck iteration 2

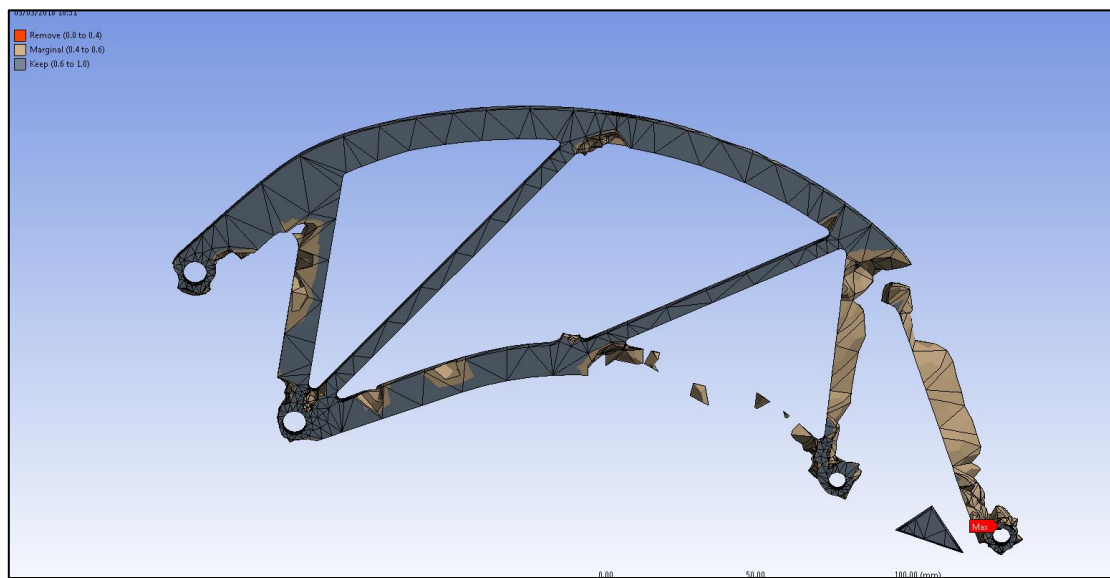


Figure 100: Swan Neck mass optimisation using 60% mass reduction

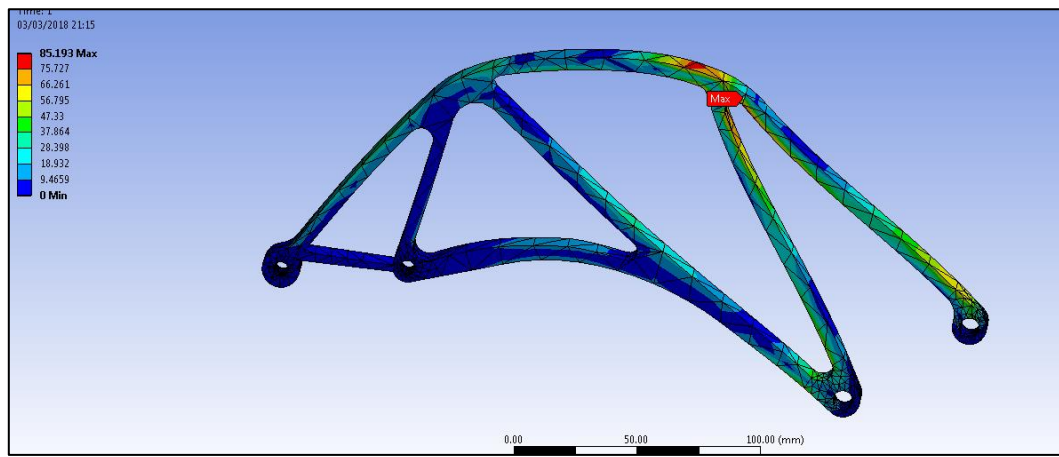


Figure 101: Swan Neck iteration 3 using mass optimisation analysis as a base for the new design