

The redesign of the UWTSD formula student chassis to meet current regulations, optimising strength, and rigidity.

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Major project

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Abstract

The redesign of the current UWTSD formula student chassis in the workshop is the chosen chassis for this project, the intention is to bring the chassis up to date with the current 2019 regulations and the draft 2020 regulations whilst also improving the strength and rigidity and, in the next phase, to improve the overall weight by applying a new set of materials. The current UWTSD formula student chassis was believed to be designed circa; 2011. To start with the formula student racecar was placed on the flatbed in the university workshop for measuring, this then leads to the designing/modelling phase using the Solidworks computer-aided design system to model the current chassis and apply real-world tests, a front impact test, a side impact test (due to this being the area with most of the redesign) and a twist test (torsional test). To validate this computer-aided design system some real-world testing had to be conducted, to validate the testing two separate welded components were attached to the Denison tensile tester, the strength of the welds were therefore re-simulated to the tests carried out in Solidworks confirming software validation. A comparison of all the static results has been carried out to determine if refinement was required in the design. Once all the static analysis has been completed testing will be carried out on the chassis with suspension set-up to include measures on 'real world' loads, displacements and stresses etc.

Table of abbreviation's

CAD	Computer Aided Design
MPa	Mega Pascals
Pa	Pascals
Kg	Kilograms
M	Metres
Mm	Millimetres
kN	Kilonewtons
N	Newtons
N/m ²	Newton metres squared
Kg/m ³	Kilograms metres cubed

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1 Introduction

In this project, the goal is to improve the strength and rigidity of the current University of Wales Trinity Saint David formula student chassis. To do this the current chassis, believed to be a design from 2011, shall be brought up to standard and redesign to meet the current regulations. Using the Solidworks computer-aided design system to model the current chassis and redesign it into a new model with the latest regulations met, then running tests and making comparisons by applying repeatable tests. Real-world testing shall be conducted to validate the results from Solidworks. Once the testing on the new design is completed a look into different materials shall be conducted, results shall be compared to the tests with the old materials. The testing is to be done in three phases, firstly the original chassis shall be tested, following this any re-designing shall be carried out leading on to updated materials on the redesign, the aim is to keep a high level of validation throughout testing only changing minimal variables at each stage.

2 Formula student

Formula Student is a race series which is largely contested by universities across the world, it is used to give young engineers the opportunity to be involved in a racing environment and advance their skills along the way. Formula student is the world's leading educational engineering competition and recently celebrated its 20th anniversary in 2018, it is an organisation which is backed by leading engineers such as its Patron, Ross Brawn 'OBE', formula student aims to develop the knowledge and understanding that young engineers have and give them an opportunity to innovate their skills and also encourage young people into the world of motorsport, (Institution of Mechanical Engineers, 2020). Companies view this as a good way to give the student the transferable skills to take from university to the workplace in any motorsport industry, formula student is viewed as the standard of engineering students are encouraged to meet as part of a degree level project, it is also a recommended way for the students to transfer into the motorsport industry, encourage students to hone in on their engineering skills and give them essential real-world experience, (Institution of Mechanical Engineers, 2020).

Teams gather to compete in a series of challenges to show off their engineering skills, innovative designs and engineering ideas to companies & businesses within the industry, in the UK this competition is held at the home of British motorsport - Silverstone race circuit local to Milton Keynes, Buckinghamshire. Each year over 100 universities from around the globe aim to compete in various static and dynamic tests, (Institution of Mechanical Engineers, 2020).

The Silverstone website gives further insight into what the event at Silverstone includes, taking part in Formula Student gives students the chance to demonstrate their technical, engineering, design, and manufacturing skills. Each team is asked to design and manufacture a single seat racecar for 'sprint' or 'autocross' racing and present their designs, hypothetically, to manufacturing firms, (Silverstone, 2020), the rules also state that the car must be low cost, easy to maintain, be of high reliability and have high performance in acceleration, handling and braking. During the competition the teams must show fully their logic behind the designs and show their ability that it can support the business model proposed by each team, (Silverstone, 2020).

The event at Silverstone includes:

“Each team also goes through a rigorous testing process with the car:

Static Events:

- *Design, Cost and Sustainability, and Business Presentation Judging*
- *Technical and Safety Scrutineering*
- *Tilt Test*
- *Brake and Noise Test*

Dynamic Events:

- *Skid Pad*
- *Sprint*
- *Acceleration*
- *Endurance*
- *Fuel Economy”* (Silverstone, 2020).

2.1 The 2019 regulation

In the 2019 regulations which were released in late 2018, section T3 is the general chassis requirements section in which all the regulations regarding spaceframe chassis requirements have been added into the appendices. The draft 2020 regulations were released in late November 2019 but with no regulation change in the chassis department therefore no further investigation into this section of the 2020 regulations was required. Below is an example of one of the regulations the original chassis will have to meet during the redesigning.

“The side impact structure must consist of at least three steel tubes, see T3.2, on each side of the cockpit, see figure 8. • The upper member must connect the main hoop and the front hoop. It must be at a height between 240mm and 320mm above the lowest inside chassis point between the front and main hoop. • The lower member must connect the bottom of the main hoop and the bottom of the front hoop. • The diagonal member must triangulate the upper and lower member between the roll hoops node-to-node.” (IMECHE, 2018).

same at the midplane in a top view with a wireframe sketch and measure all the lengths and widths.

With the sketches and measurements completed a cad wireframe was modelled in Solidworks using 2D sketches and geometric planes, the 3D sketch function was then utilised to connect the 2D sketches, below is a series of figures showing how the measurements were transferred from paper into Solidworks.

Sketch one in figure 3, is the base of the front end, this was used as a starting point to add a geometry plane to enable the sketch of the next section.

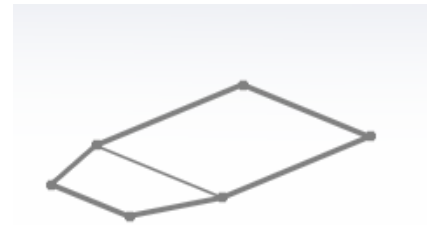


Figure 3 - Sketch One UWTSD Formula Student Chassis - Oliver Gordon

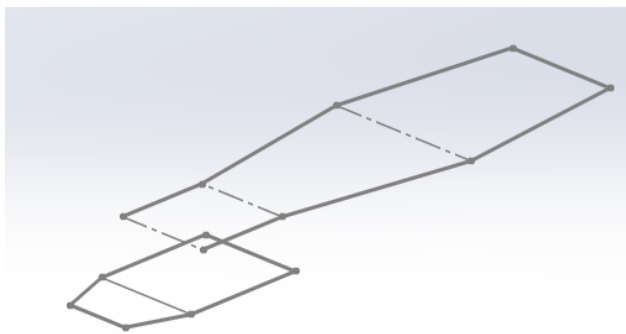


Figure 4 - Sketch Two Mid-section of Chassis - Oliver Gordon

Sketch two in Figure 4, using the geometry plane tool when designing enabled us to offset to the flat mid-section of the chassis and sketch from the front end round to the rear end utilising the construction lines to allow all the required reference points during the sketching process.

Sketch five in Figure 5 on the right is a few more steps down the sketch process, here more construction lines have been included for the angle of the front roll bar and also added the front bulkhead along with a centre construction line to go past the length of the sketch.

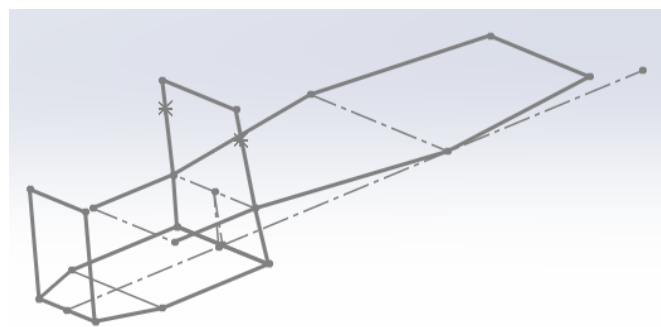


Figure 5 - Sketch Five Front end sketch - Oliver Gordon

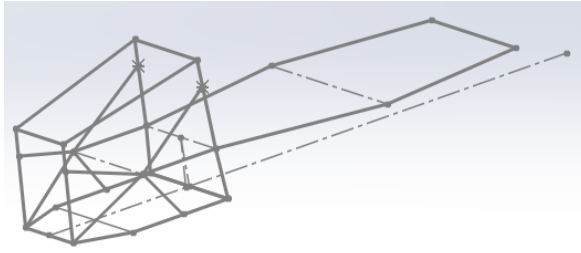


Figure 6 - Sketch 12 complete front end of chassis sketch – Oliver Gordon

Sketch 12 Figure 6 on the left, is a completion of the position of the front end, to compile this series of sketches the 3D sketch tool was used to join the centre point to all the different sides of the front end, this was achieved by using the mid-point as shown in the previous sketches and drawing a line to the edge and dimensioning into position. This sketch also adds the structural members for the front suspension.

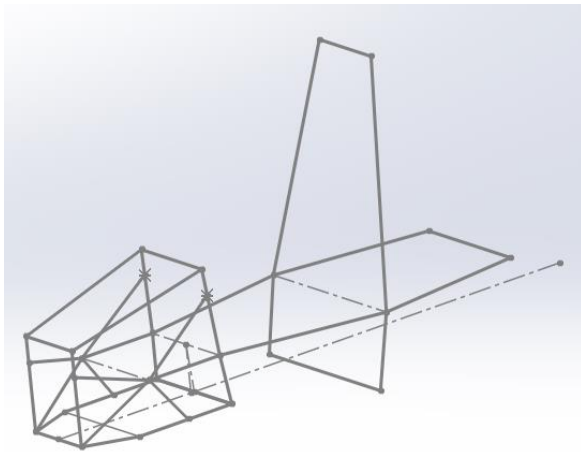


Figure 7 - Sketch 17 Adding main roll bar - Oliver Gordon

In Sketch 17 Figure 7 the main roll bar was added, this inclusion allowed the design to pull together as a number of points can now be connected.

The 3D Sketch in Figure 8 on the right shows where the 3D sketch tool has been used to connect the main and front roll bars at the bottom and midpoints.

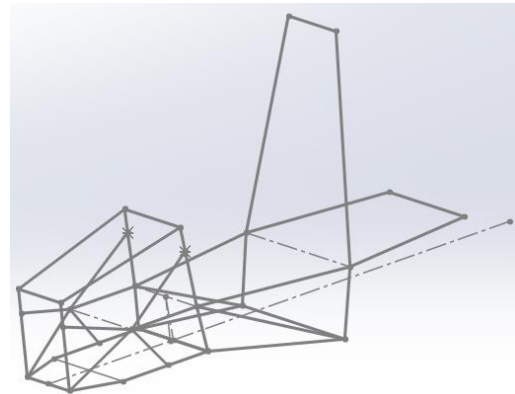


Figure 8 - 3D Sketch Connecting the roll bars - Oliver Gordon

The 3D Sketch in Figure 9 on the left is a further 5 steps down the sketching process, in this completing the bottom part of the rear end, adding all the structural members for where the rear suspension would mount.

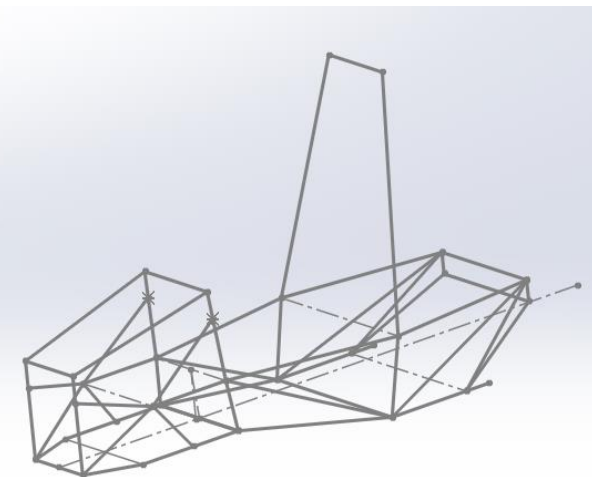


Figure 9 - 3D sketch three Rear end bottom section completion - Oliver Gordon

The next few steps were to add the structural supports for the roll bars. For example, the angled support bars for the main roll bar to the rear of the car, and then the bars between the front and main rolls bar connecting near the top of the front roll bar.

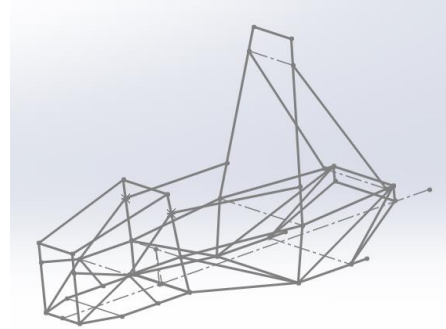


Figure 10 - Sketch 70 Adding roll bar supports - Oliver Gordon

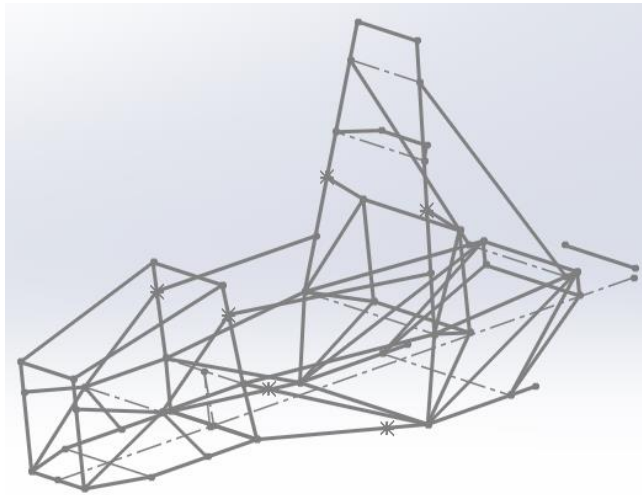


Figure 11 - Final sketch preview - Oliver Gordon

The final sketch in Figure 11 shows the sketching phase of the model of the chassis completed, added here are the rear jacking point and the main roll bar seat belt mounting bar, the engine mounting bars have also been added in this part of the sketching phase.

Once the sketches are complete, we turn this wireframe version of the chassis into an actual solid spaceframe chassis to run FEA tests on. So, using the weldments tool and

Group1
Group2
Group3
Group4
Group5
Group6
Group7
Group8
Group9
Group10

Figure 13 - Structural member groups - Oliver Gordon

selecting the structural member tool, as seen circled on the right, we then select all the lines in the wireframe version of the chassis splitting them into different groups, splitting into groups is done because when you would get to a turn in the pipe you can order the groups so the joints in the pipes are what desired by the designer, and so that the pipes don't intersect.



Figure 12 - Weldments menu Solidworks - Oliver Gordon

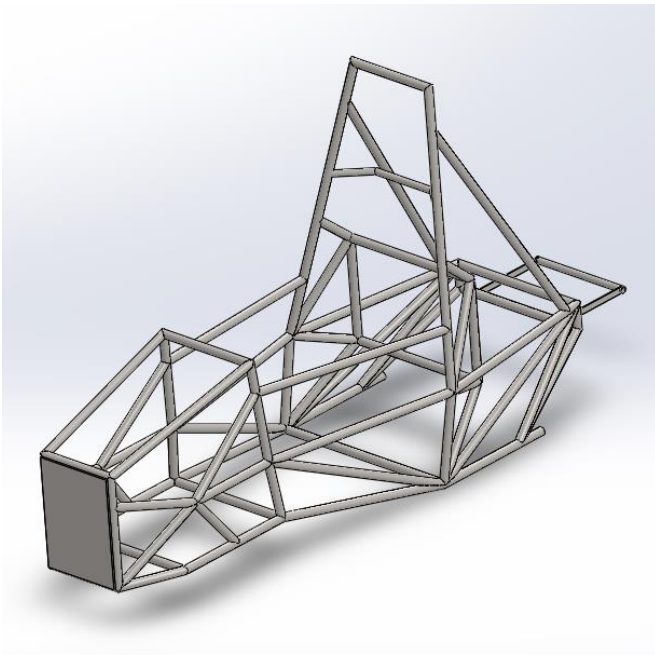


Figure 14 on the left is a completed model of the UWTSD formula student chassis, with an added front bulkhead plate as is on the car in a previous photo. The whole chassis in Solidworks is made as a 26.9mm pipe with a thickness of 3.2mm because this is the closest to the real world size that can be achieved with Solidworks, the original being 25mm pipe with a thickness of 3mm.

Figure 14 - Complete Uwtss formula student chassis - Oliver Gordon

4 Solid works

As previously mentioned in the report Solidworks CAD software has been used, in the research section there will be a bigger insight into what the abilities of Solidworks are including its advantages and disadvantages. We then move on to FEA and what it is utilised for, looking into how to set-up some static analysis, what meshing is and the different types there are.

The use of CAD software is having more of an impact on the engineering as a whole every year, *“The increasing use of CAD software in the packaging industry and automotive industry is one of the key factors positively impacting the growth of the market.”* (DIYguru, 2020).

4.1 Solid works cad software

The Solidworks 2019 computer-aided design system is the CAD software being used to design the Formula student chassis from the UWTSD workshop. A system where using the 2D sketch on different planes or 3D sketching to model any component. Solidworks



Figure 15 - Solidworks logo - <http://www.josh-ua.co/blog/2019/2/13/solidworks-vs-siemens-nx-vs-onshape>

themselves are very proud of their software and in a statement on their website, they said *“Dramatically improve the way you develop and manufacture products. As the foundation for the entire SOLIDWORKS suite of product development solutions, SOLIDWORKS CAD packages cover design, simulation, cost estimation, manufacturability checks, CAM, sustainable design, and data management.”* (Solidworks, 2020).

4.2 FEA (finite element analysis)

Finite element analysis is utilised by engineers and scientists to recreate real-world tests, by implementing the same measures and parameters as the real-world tests such as forces, vibrations and heat thus allowing predictions of future work that minimises the need for physical testing of new designs. (AutoDesk, 2020) & (Comsol, 2016). This is then used to determine whether the component design works, fails, breaks or wears out, this is used during the designing process before anything is manufactured and also used to see how an existing part can be improved/optimized, *“Finite element analysis shows whether a product will break, wear out or work the way it was designed. It is called analysis, but in the product development process, it is used to predict what's going to happen when the product is used.”* (AutoDesk, 2020).

The process done by turning the part into thousands and thousands of small triangular pyramids or cubes dependant on the software or settings. *“EA works by breaking down a real object into a large number (thousands to hundreds of thousands) of finite elements, such as little cubes.*

Mathematical equations help predict the behaviour of each element. A computer then adds up all the individual behaviours to predict the behaviour of the actual object.” (AutoDesk, 2020).

Meshing is used to refine the results of testing with different scaling of meshing being able to be utilised to do so, for example in solid works there’s a scale of coarse to fine mesh. *“The accuracy that can be obtained from any FEA model is directly related to the finite element mesh that is used.”* (Comsol, 2016).

With a coarse mesh, the results will be vague and just show a general area of stress, displacement, and strain. *“The initial mesh looks a little coarse, but okay for a first analysis.”* (Akin, 2009). This is only ever used on a component on an initial basis, but with fine meshing, this will show how a component reacts in a simulation of a real-world test to a high spec. *“Since the mesh is so crude the beam stress is probably the most accurate and the plane stress value will match it as a reasonably fine mesh is introduced.”* (Akin, 2009). This would, in turn, give a high quality of results and allow for extensive analysis.

3D meshing and 2D meshing are two common ways of meshing that get confused very often. 2D meshing being used to conduct FEA analysis on a single face of an object whereas the 3D meshing is used to conduct FEA analysis on an entire part. In a recent paper that has come to light 3D meshing gets overused, to summarise the author said this *“I would never argue against using 3D elements when they are needed. But I know that those elements are greatly overused! In my Breakthrough FEA course, I refer to 3D mesh as the “mesh of last resort”. What I mean is, that you should use 3D elements, only if using 2D elements is not possible.”* (Skotny, 2020).

When using a standard mesh, this will allow the user to specify the global element size and tolerance values, it uses Voronoi-Delaunay triangulation, this tries to maximise the smallest angles of all the triangles, all while the geometry is to be split into tetrahedrons and triangles. (HAWK RIDGE SYSTEMS ENGINEERING TEAM , 2013).

Static analysis is what shall be conducted during the validation and analysis of the formula student chassis. Static analysis is a testing simulation where time has no effect and doesn’t influence the loading, fixture or the results of the test. *“Static FEA is based on the assumption that time does not play important role and its influence on the results can be ignored. In case of solid mechanics simulations, it means that load and response to it are static.”* (Michalski, 2019).

With static analysis fixtures are required to be added, this can be done by adding a fixed geometry, roller/ slider, fixed hinge, bearing or foundation, as seen in the menu on the right. There are also advanced fixtures including joints which are being utilised later in the report with the analysis of the UWTSD formula student chassis.

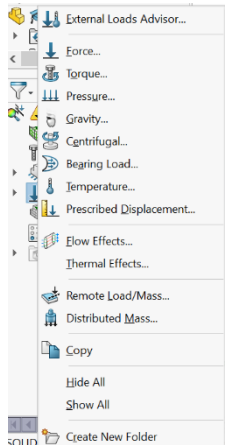


Figure 17 - Solidworks 2019 - Oliver Gordon - External loads menu

On to the external loads, which include adding loads such as force, torque, pressure, gravity, centrifugal, bearing load, temperature, and prescribed displacement, other elements can be added like flow effects and thermal effects, even remote load/mass and distributed mass.

With the analysis that will be conducted on the chassis further into this report, the stresses shall be analysed

by reviewing positive and negative stress, this is due to the direction of the load applied therefore any negative stress is stress acting in the opposite direction to the load being applied. "The axial stress is calculated at both ends. In general, the value at each end is different (for instance if the beam is curved). When

viewing stresses, the software shows one value for each element, this value is the algebraic highest or lowest stress calculated (whichever is furthest from zero)." (Solidworks, 2006).

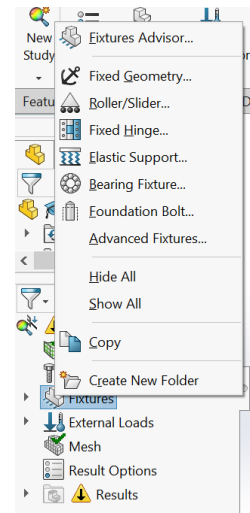


Figure 16 - Solidworks 2019 - Oliver Gordon - fixtures menu in static analysis

5 Solid works validation

To validate solidworks testing a test in real life and copying it exactly to the CAD software will show how the results compared. For this testing the idea was to combine multiple tests into one, example, by testing two types of welding joints a 45° cut weld and a 90° butt weld. The test is to put these

components onto the Denison and pull the component apart, over a period of time pulling the component apart until it breaks to find the max load it would be able to withstand.

“Principle of Tensile Testing

A materials test subjecting a specimen of a material or component to a mechanical force, either: -

- Tensile or pulling
- Compressive or crushing
- Bending and flexing
- Torsion or twisting
- Frictional and dragging
- Peel resistance



Figure 18 - Denison Tensile Tester -
<https://www.prweb.com/releases/2006/11/prweb478461.htm>

The testing machine automatically measures the forces applied and the resulting behaviour of the material. Mechanical properties can be derived from the load/displacement or Stress / Strain curves:

- Yield Strength / Yield Point (YS)
- Ultimate Tensile Strength (UTS)
- Youngs Modulus for elasticity
- Modulus of Toughness - Brittleness
- Elongation and reduction in area – Ductility” (Els, 2020).

All data collect was in notepad, so it was copied and pasted into excel for ease working with and creating tables and graphs if required.

5.1 45 degree cut welds

The first test was the 45° cut weld as seen on the right. The test duration was 135 seconds with the breaking point came at 74.4 seconds during the test at a load of 7.555 kN, the test had a stress value of 68.17 MPa and a displacement of 25.154 mm.

Due to the data output from the machine giving only the duration of time, displacement of the part and the stresses of the component, the strain and young's modulus will have to be calculated separately.

$$\sigma = \frac{F}{A} = \text{stress equation}$$

$$\epsilon = \frac{e}{l} = \text{strain equation}$$

$$E = \frac{\sigma}{\epsilon} = \text{Young's Modulus}$$

Figure 20 - Useful equations for stress, strain and young's modulus.
- <https://www.s-cool.co.uk/a-level/physics/stress-and-strain/remember-it/s-cool-revision-summary>

The strain formula in an alternative way of writing it is the (change in length – the original length)/ original length. Or (displacement)/ original length. The displacement

formula being used to calculate the strain from this group of testing giving a value of 0.069872222. Once this has been calculated a formula can then be put into the excel datasheet to calculate Young's Modulus, this being stress/strain, in excel this is =stress cell/strain cell. At the breaking point of the part, the value for Young's Modules is 976.39659696271Pa.



Figure 19 - Angle cut weld test screenshot - Oliver Gordon

As seen on the right with the load-displacement graph it's clear where the breaking point is, but also as the stress goes up and gets to around 6.2kN it drops slightly and then continues to increase, this is the yield point or elastic limit. So, for real-world racing you would never want your chassis getting close to/past the yield point. Replacing a chassis is usually not an option for teams on a tight budget!

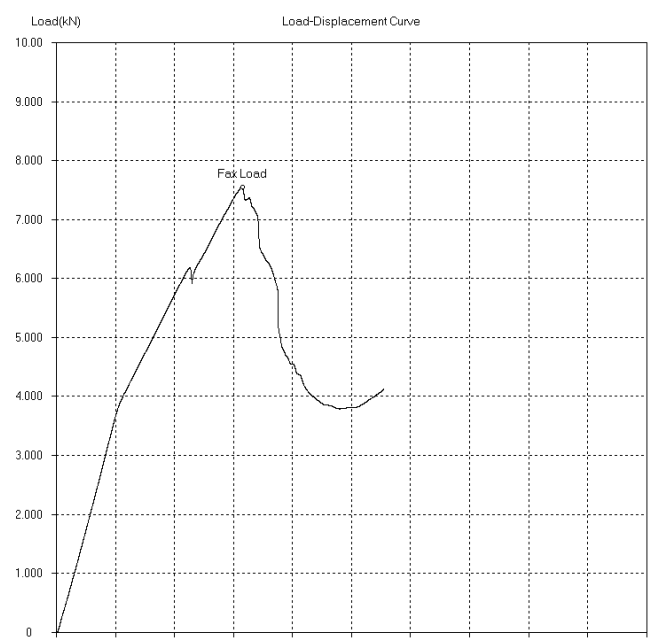


Figure 21 - 45 angle cut weld test load/displacement graph

For the test to be conducted in Solidworks all the measurements were taken and the model completed. Information and compounds of the material was unknown to the members of staff in the workshop at the time, and due to closure of the university workshops (Covid-19) parts could not be taken there to gather further information. However, after doing some research on Solidworks into the selection of materials available on their system, alloy steel was a close match with the figures before the test, so this was used for the selected material for validation testing.

On the right is the cad model, with the green arrows showing where the part is fixed and where the weld is in real life, the purple arrows, however, are showing the position and direction of the load being pulled on the part, which is, half the max load the part withstands at breaking point during the test, so, half the max load will be applied in both directions equalling the total max load of 7.555 kN.

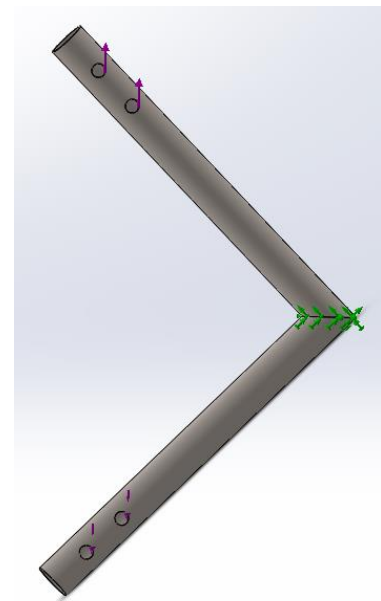


Figure 22 - 45 degree cut weld alloy steel validation testing component - Oliver Gordon

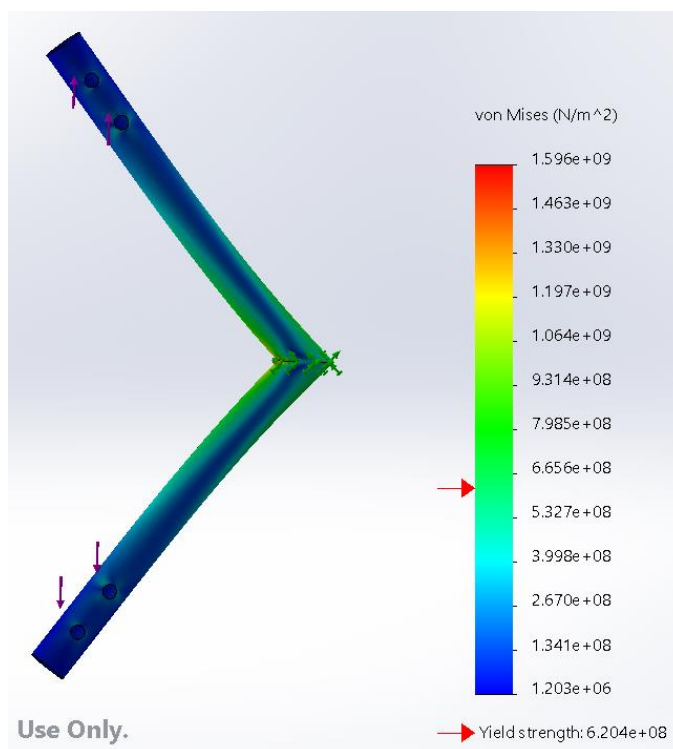


Figure 23 - Validation test 1 45 degree cut weld - Oliver Gordon

After

running the test with alloy steel selected as the material, the results of the showed the part had failed as expected as did the component in real life with a yield strength of $6.204 \times 10^8 \text{ N/m}^2$, which converts to 6.2 kN in load. This is, in fact, perfect and proves that the material chosen in Solidworks for validation has therefore validated the system and helped identify the material of the real-life component. With the results being within such small margins there isn't a lot of room for any error, so we can take this as correct validation of the software.

5.2 90 degree butt welds

With the second test being the 90° butt weld, the exact same test was run and knowing the materials were the same, all the same tests could be conducted on the Denison and in Solidworks, the test durations were 130.5 seconds, with the breaking point coming at 69.4 seconds during the test at a load of 5.055 kN with a stress value of 53.594 MPa and a displacement of 23.384 mm, at the breaking point, the value of strain is 0.064955556. Now, this has been calculated, the same formula can then be put into the excel datasheet to calculate Young's Modulus, at the breaking point of the part, the value for Young's Modulus is 825.0872391Pa.

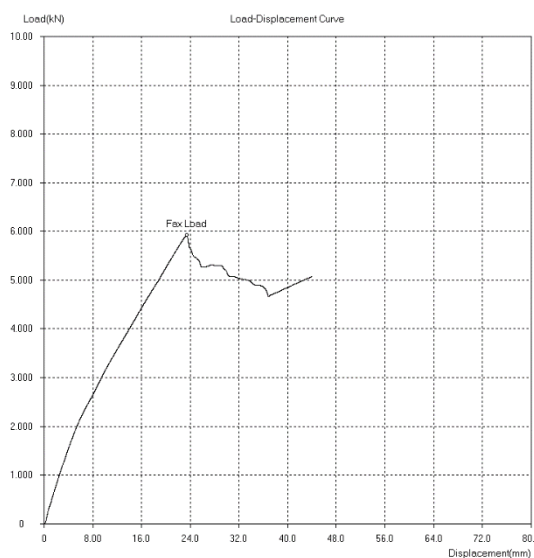


Figure 25 - Load displacement graph 90-degree butt weld - Oliver Gordon

Unlike the first test, this component seems to fail before reaching the yield point load of the last test which could be due to the different type of weld. The max load the part experiences is 5.955 kN.

Just like the first test the green arrows are showing where the part is fixed and where the weld is in real life, the purple arrows however, show the position and direction of the load being pulled on the part which is half the max load the part withstands at breaking point during the test, so half the max load will be applied in both directions equalling the total max load of 5.95kN with a yield strength of $2.206 \times 10^{-8} \text{ N/m}^2$ which converts to 2.2 kN in load.



Figure 24 - Butt weld test screenshot - Oliver Gordon

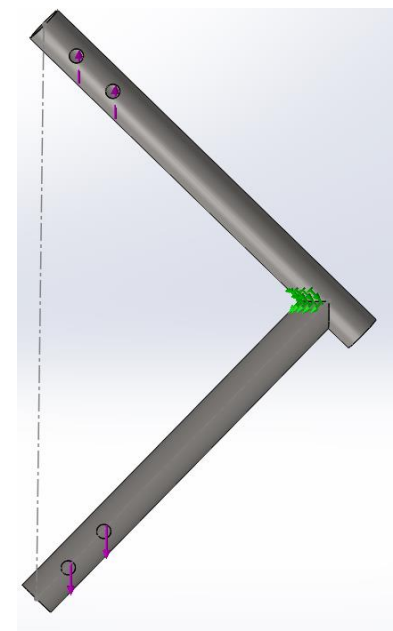


Figure 26 - 90-degree butt weld cad model - Oliver Gordon

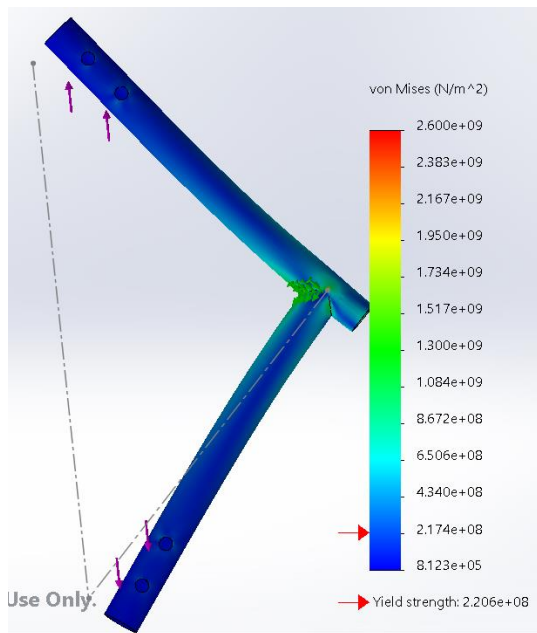


Figure 27 - Test 2 butt weld component - Oliver Gordon

So, we can assume this is the yield point on the load-displacement graph from the previous page, like the first test, the part has failed just like in real-life, which again is the second lot of validation which proves the software can give results within a small margin of the real results.

6 FEA on the current UWTSD formula student chassis

The following includes three tests conducted on the model of the current formula student chassis in the workshop, these include;

Test 1 simulating a head-on collision by fixing to the front end and applying a load in newtons equalling the weight of the vehicle and driver together.

The second test is a side impact to test the old system so, in future analysis, it can be compared to the new design.

Test 3 is a twist test, fixing the left-hand side and applying the load to the right-hand side in an upward direction.

6.1 Test 1 – forward strength test

With the first test being the head-on collision or “*A static analysis of a front impact situation*” (Kerkhoven, 2008) the test is carried out by applying the weight of the vehicle plus a driver to the engine mount points with the front end fixed, thus using the four joints at the engine mounts and spreading the load across the four points and the six-points at the front bulkhead the loads applied at the engine bay are 708.555N per point (x4).

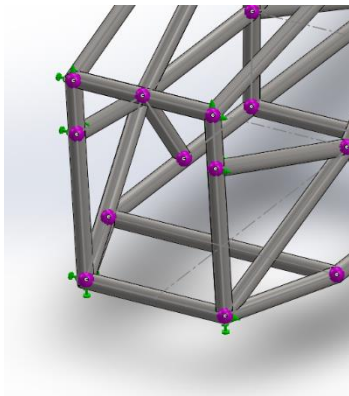


Figure 29 - Test 1 fixtures - Oliver Gordon

on the left is the fixing points at the bulkhead and on the right is the representation of the location of the loads being applied at the engine bay.

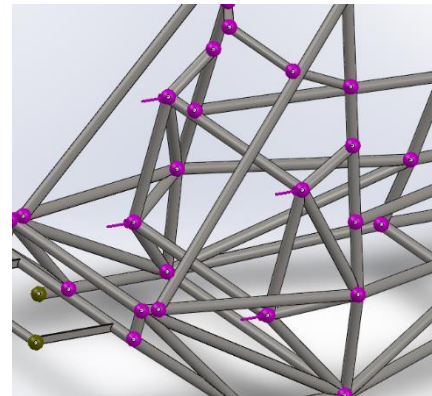


Figure 28 - Test 1 loading points - Oliver Gordon

When the sets of results are available, they are listed in an order of stress, displacement and then strain, that is how'll they will be addressed below.

The first test results - stress, as shown on the right the min. and max. stresses experienced at the front end are as expected with one of the support structures to the bulkhead experiencing negative stress of $-3.948 \times 10^6 \text{ N/m}^2$, - 3.948MPa. After extensive research into how there is negative stress, it has been determined that it is

stress acting in the opposite direction to the load being applied, even with this the stress value is extremely low. With these results some changes could be made to the front end to improve.

With the max stress being at the front roll hoop showing a max value of $8.073 \times 10^5 \text{ N/m}^2$ which is 0.8074MPa which is not regarded as a high stress value, leading to believe, that some components on the chassis could be removed and possible material changes made. This shown in Test 1 Figure 30 as the whole model is blue and very low on the right-hand scale, this backs up the possibility of material change or even component changes.

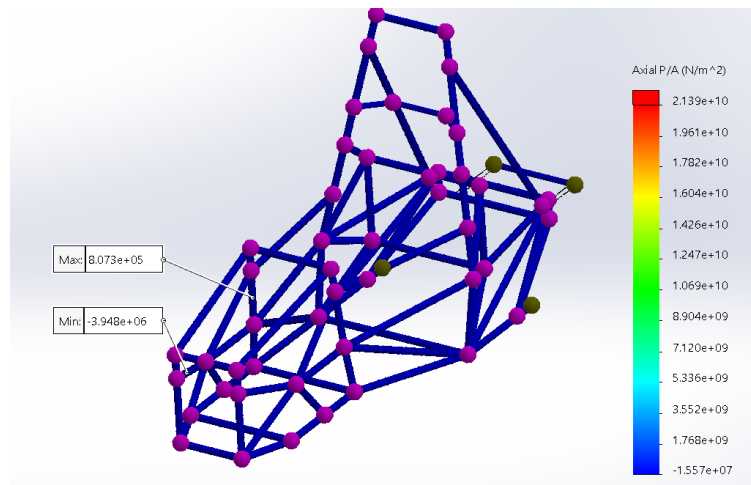


Figure 30 - Test 1 stress results – Oliver Gordon

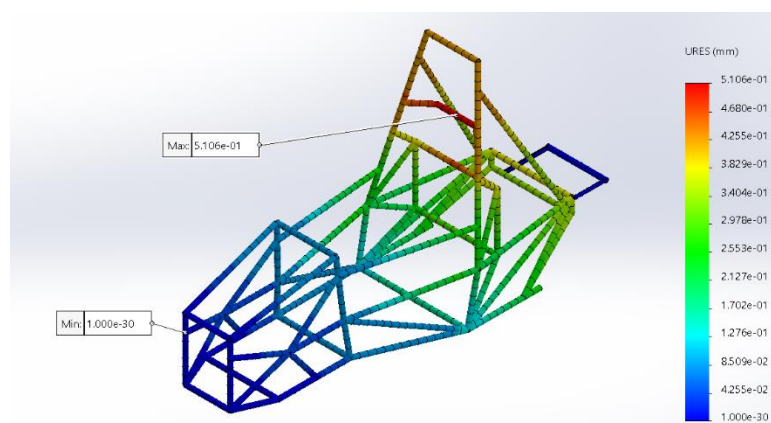


Figure 31 – Test 1 displacement results - Oliver Gordon

The second set of results is the displacement, with the colours being exaggerated to give easy identification of results, the test shows a minimum value of $1.0 \times 10^{-30} \text{ mm}$ at the front end and rear jacking point, this not an issue and quite expected, the bottom rear of the chassis experiencing the average value of

displacement of around 0.25 mm and then the max displacement noted around the mounting points for the seatbelts, as shown with the max value in “figure 32” with the value being 0.5106mm. The

view on this is that with some radius's added to the main roll hoop they could absorb the forces experienced and the displacement is spread through the surrounding areas.

The third results are the strain results which we show in two versions, because very little is experienced a deformed image, figure 31 and a non-deformed image, figure 33 are shown as results of an example of the strain.

As seen on the right figure 33 shows 0 strain values, this could be the reality or a fault in the simulation, with the loads being just the weight of the vehicle and a driver adding speed to the mass on impact it would be presumed to be a correct value and 0 strain is experienced. This, however, is an incorrect assumption as there will be strain experienced if there are stresses and displacements experienced.

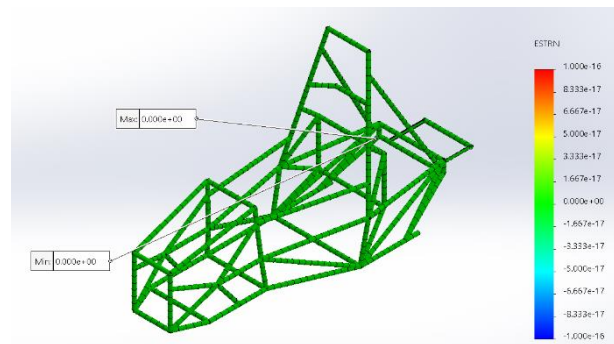


Figure 32 - Test 1 Strain result nondeformed - Oliver Gordon

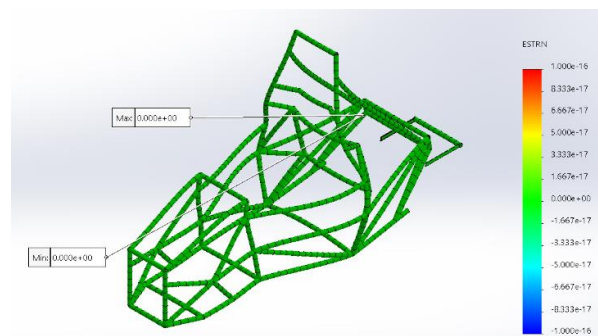


Figure 33 - Test 1 strain result deformed - Oliver Gordon

6.2 Test 2 – side impact test

For this test the idea was to simulate a side impact as this is the main area that requires a redesign, the test was to apply the weight of the vehicle and driver as a force across one side of the side impact structures and fix the opposite side and see how the chassis reacts. The load of the weight of a formula student car and driver together equals 2834.22N, this split across six points for the side structure is 472.37N.

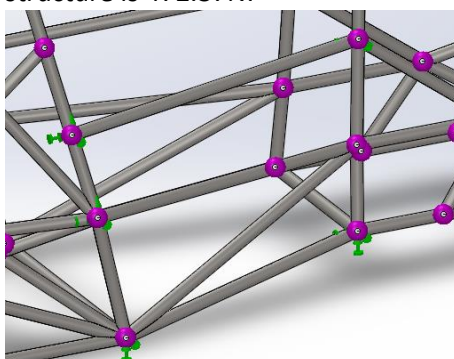


Figure 35 - Test 2 original chassis side impact fixtures - Oliver Gordon

On the left is the fixture positions and on the right is the load application positions

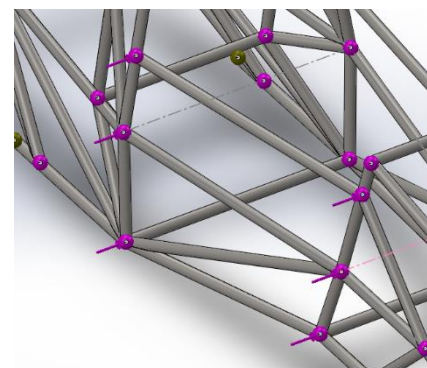


Figure 34 - Test 2 original chassis side impact load application - Oliver Gordon

Same with the previous test with the order of results. The stress experienced ranges from -

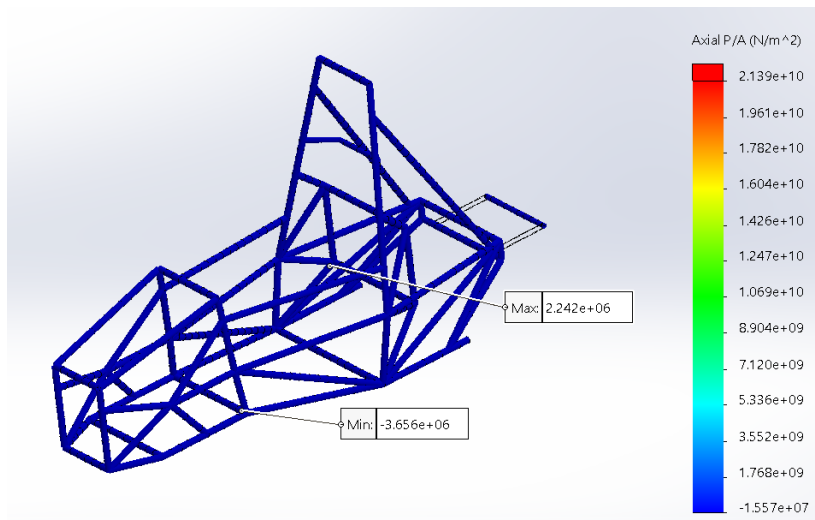


Figure 36 - Test 2 original chassis stress results - Oliver Gordon

stress than the load itself. The max value is a very low value due to the force applied not being very high, this is something that will be looked at in further testing. The max stress being located in the engine bay means that it is well supported, weight saving could be done in this area but only ensuring the engine mounting is not compromised. Engine mount failure would be catastrophic to a team during a race weekend.

Displacement results have a minimum displacement at the fixture area which is what would be

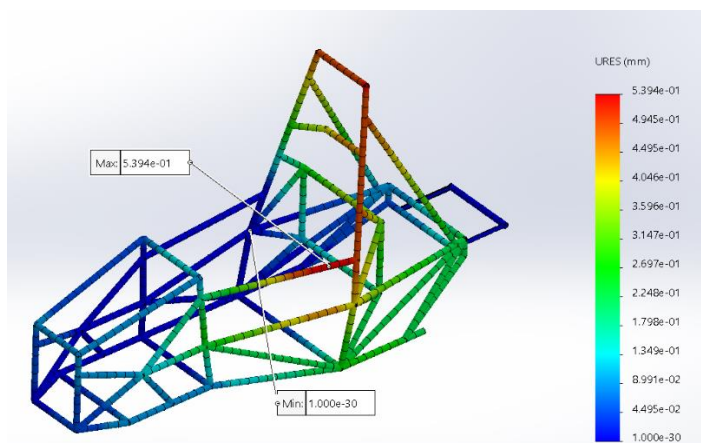


Figure 37 - Test 2 displacement results - Oliver Gordon

3.656x10⁶ N/m² to 2.242x10⁶ N/m² which convert to -3.656MPa to 2.242MPa, with the negative stress being the higher value than the positive value, this means there is more stress acting in the opposite direction to the loads, this result could mean the fixture of the chassis causes more

expected, getting a value of 1x10⁻³⁰ mm is a good result and repairable. The max displacement of 0.5394mm at a side impact structural member connecting the two roll bars is a poor result, should a crash occur this would ultimately be damaged and require replacing as would the main roll bar, this supports the idea of adding some radius's so that they can absorb some of the forces.

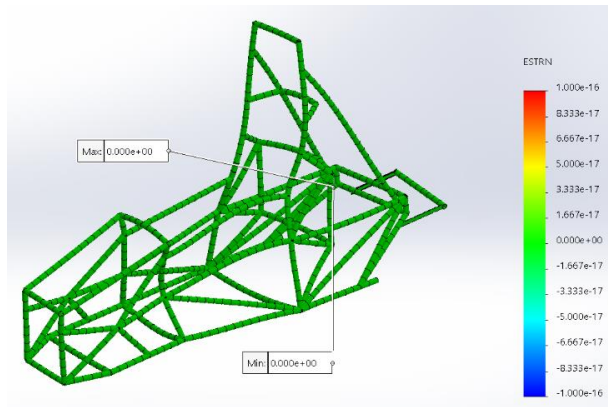


Figure 38 - Test 2 original chassis strain deformed - Oliver Gordon

Third results - strain results, this includes a deformed and non-deformed view of the results.

On the left being the deformed view of the results which again show a maximum and minimum value of 0 strain which is inconclusive so the assumption is that there is a problem in the simulation, with minimal computing power and no access to the university computers (as of lockdown Covid-19), the weird glitch with strain results cannot be resolved.

On the right is the same set of strain results without the deformation results, with the figure above showing deformation you would presume there is strain again the values aren't being shown, a resolution cannot be achieved due to computing power, access to university etc.

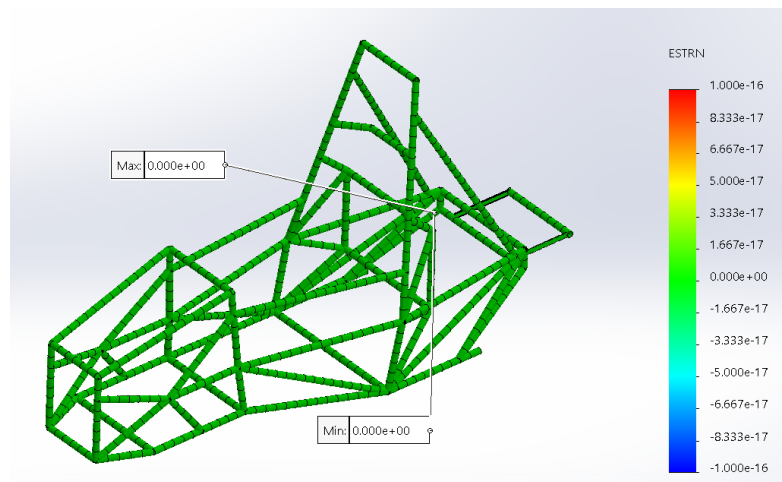


Figure 39 - Test 2 original chassis strain nondeformed - Oliver Gordon

6.3 Test 3 – twist test

The twist test is something that during a supervisor meeting it was recommended that doing a static twist test would give a basic set of results to show what would happen if the formula student car was to go over a curb on a race track. This test is to look at the torsional stiffness, after looking at examples of this test in the Design, Analysis and Testing of a Formula SAE Car Chassis technical paper this is what was found, “Torsional loads, induced by an undulating road surface or cornering forces are one of the most important and highest magnitude loads transmitted through the frame.” (George, 2002). As suggested, this is a good test to conduct to show how the chassis would react in a real-world scenario, the test

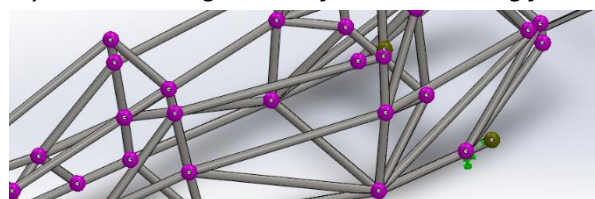


Figure 40 - Test 3 twist test fixture points - Oliver Gordon

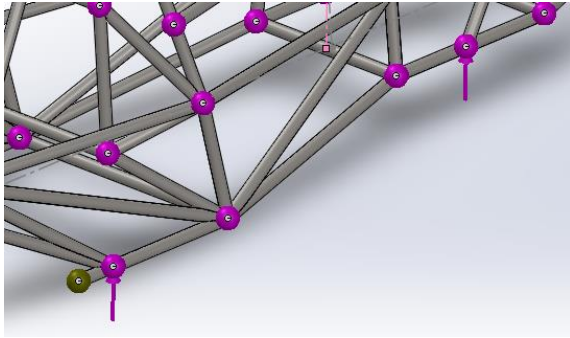


Figure 41 - Test 3 twist test loading application - Oliver Gordon

shall be conducted at first on a basic static analysis with the view to take the test further with some motion involved.

With this now being the aim of the 3rd test, fixing the chassis on the left-hand side and applying the load on the opposite side in an upward direction towards the sky for example. On the left is the fixture points as seen with the green arrows.

Above are the load application points and the direction shown by the purple arrows, the load is applied to the two points is 1417.11N.

Stress results for the twist test, the chassis experiences extremely little stress on the maximum value, which is $1.122 \times 10^7 \text{ N/m}^2$ this converts into 11.22MPa, which leads us to believe refinement of the chassis can take place in this area, the negative stress is a lot but not higher than the

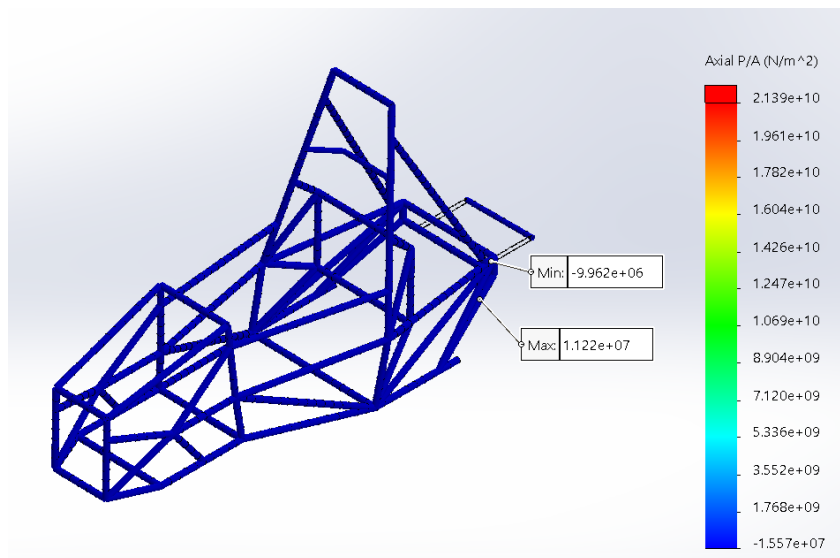


Figure 42 - Test 3 twist test original chassis stress results - Oliver Gordon

positive stress just in the opposite direction, this is being caused by a twist in the chassis from going over a theoretical bump, an interesting analysis is the max and min stress is acting next to each other showing a significant amount of twist but with the values being so low this isn't enough to damage the chassis itself. The minimum or negative stress value being $-9.962 \times 10^6 \text{ N/m}^2$ which converts into -9.962MPa.

The displacement results show that the max displacement of 26.89 mm on the main roll hoop is

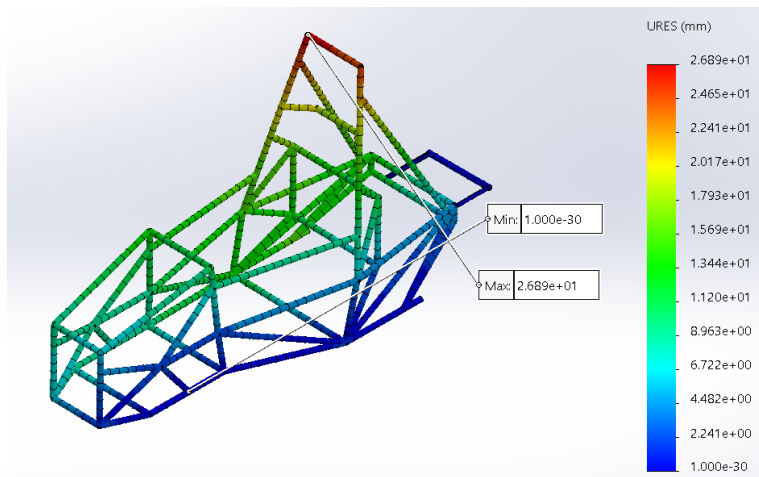


Figure 42 - Test 3 twist test displacement results - Oliver Gordon

old chassis #1			
test no.	stress +	stress -	dispalcement
1	0.8074	3.948	0.5106
2	2.242	3.656	0.5394
3	11.22	9.962	26.89
	Mpa	Mpa	mm

Figure 43 - Full set of results for original chassis - Oliver Gordon

Strain results similar to the previous two tests & don't give a value. Therefore, these are shown to give an example of the results but with no significant

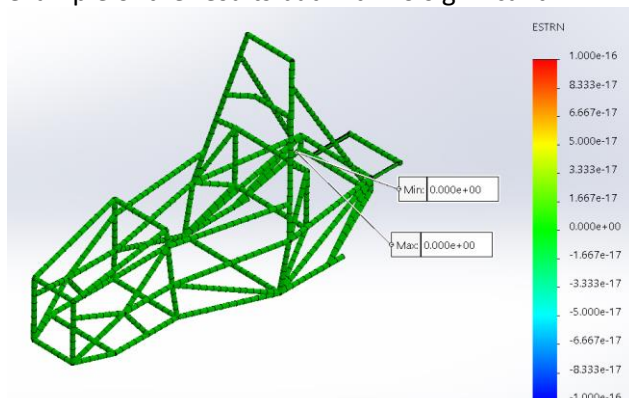
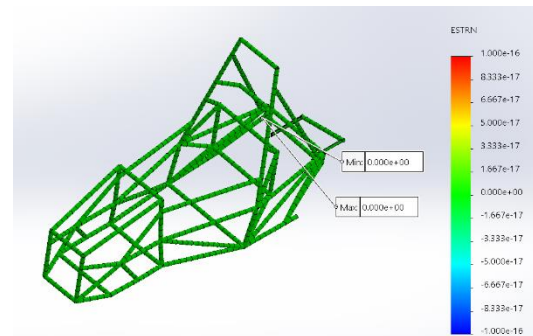


Figure 44 - Test 3 twist test original chassis strain results without deformation - Oliver Gordon

exactly on the sharp corner which gives added reason to alter the design on the roll hoops. The minimum value being 1.0×10^{-30} mm, which again is absolutely nothing and therefore does not need to be taken into consideration for the redesign.

On the left is a full set of results for the stress and displacement of the chassis, the strain is not included because of an error in calculation during the testing phase, this has been explained previously in the report.



information from the set of results.

Figure 43 - Test 3 twist test original chassis strain result with deformation - Oliver Gordon

7 Design process

The design process is a process used by multiple industries, including engineering, graphics, film & tv and more. The design process that was followed was an engineering design process found on “discover design’s website”.

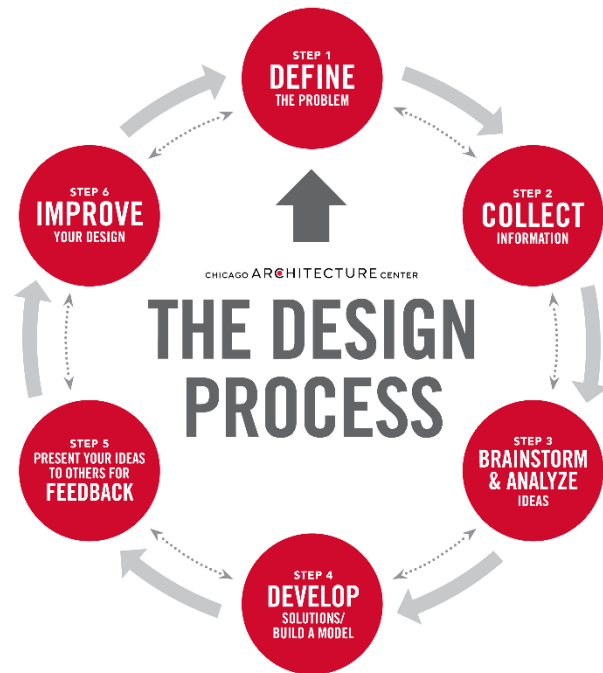


Figure 45 - Design process diagram - <https://discoverdesign.org/handbook>

Step 1: is to define the problem - find the regulations the current chassis doesn't meet.

Step 2: collect information - find the areas of the chassis in which the new regulations would require change.

Step 3: Brainstorm and analyse idea's - initial ideas for a new design implementing the new regulations

Step 4: develop solutions/ build and model - adapt a copy of the original chassis to create a new one.

Step 5: present your ideas to others for feedback - this doesn't apply to in this design process.

Step 6: improve your design - this is where the design shall be refined and/or materials will be changed.

7.1 The initial idea and creating it in Solidworks

After copying a file of the original chassis, each sketch was adapted where required and changed to meet the regulations.

The only regulation that required the chassis to be changed is regulation T3.15.1 involving the side impact structure.

The 2019 t3.15.1 regulation: *“The side impact structure must consist of at least three steel tubes, see T3.2, on each side of the cockpit, see figure 8. • The upper member must connect the main hoop and the front hoop. It must be at a height between 240mm and 320mm above the lowest inside chassis point between the front and main hoop. • The lower member must connect the bottom of the main hoop and the bottom of the front hoop. • The diagonal member must triangulate the upper and*

lower member between the roll hoops node-to-node.” (IMECHE, 2018). This regulation comes with an example image as Figure 46 – Side impact structure regulation.

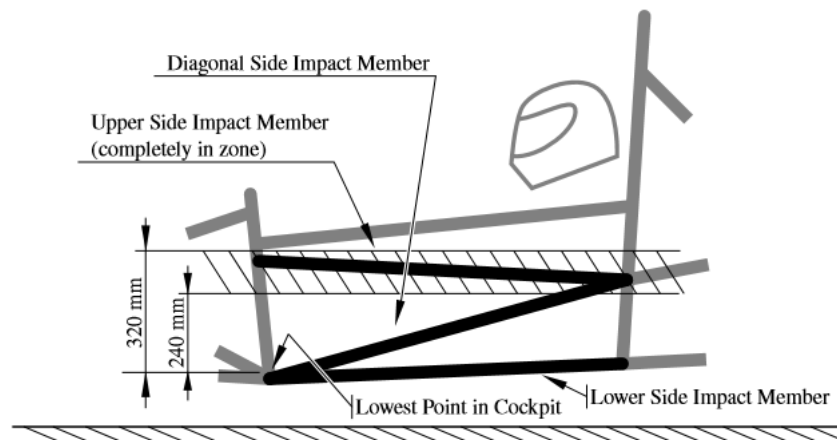


Figure 8: Side impact structure

Figure 46 - Side impact structure regulation - (IMECHE, 2018).

After reading about this regulation and not being able to find another that the old chassis doesn't already meet, this allows the redesign to stay very simple. When reviewing the results of the FEA analysis the idea came that the few points made during the analysis could be implemented as well. This includes the adding of radius's to the roll hoops and removing one of the support bars at the front end because it is overly defined, removing the same bar either side wouldn't remove too much strength in theory but the idea is to be able to remove it and keep the strength because in any motorsport series the goal is to have as much strength and as little weight as possible, this would be done by removing the bar as seen in the image of the initial idea below.

Onto the final sketch idea, with the removed bar circled on the right and with the new side-impact structure set-up it should improve the strength and experience less stress and displacement compared to the old version. The engine mount bracket had also to be moved slightly because of other structures moving.

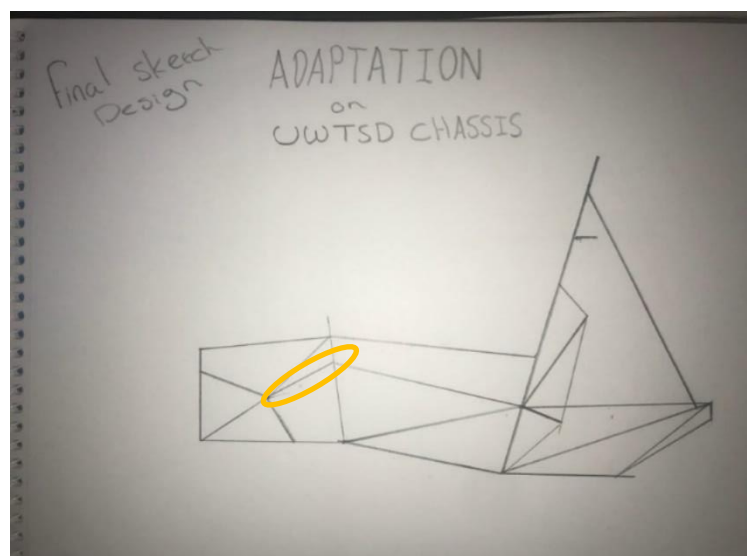


Figure 47 - Final sketch idea - Oliver Gordon

On the right is a cad model of the new design (side profile). As seen on the right the bar circled previously has been removed, this is due to it being far too close to the supporting bar above and therefore not adding any strength so it was decided during the design process to remove and check the

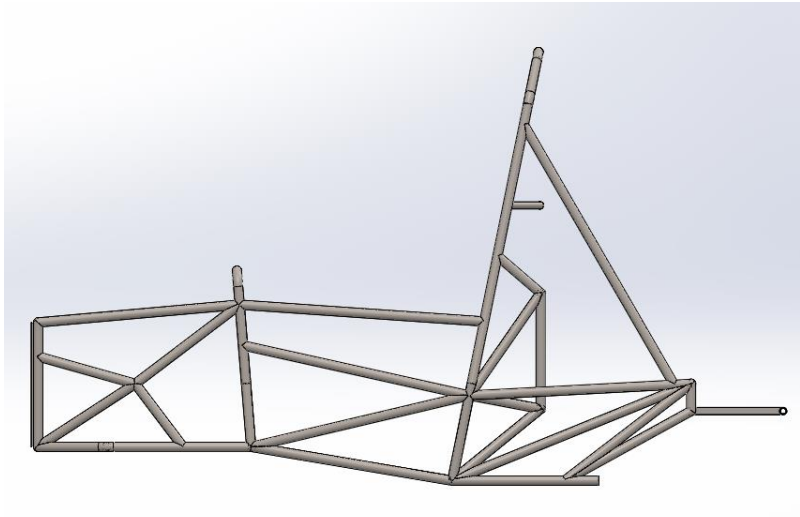


Figure 48 - New chassis design - Oliver Gordon

Also, the side impact structure has been altered to meet regulations as stated previously, the engine bracing has also been moved as stated in the initial idea's, in comparison to the old design there is no longer a mid-plane support structure in place and the front end bars have been removed, the side impact has been altered and the joint from the main roll hoop to the rear end has been lowered.

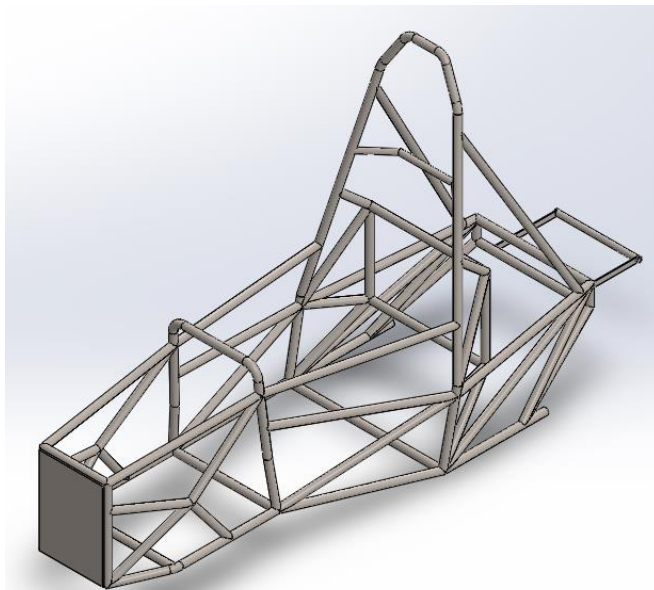


Figure 49 - Dynamic image of new chassis cad model - Oliver Gordon

On the left is a dynamic view of the new chassis design, the radius's discussed during the analysis of the existing chassis have been added to the floor of the front end and the front and main roll hoops.

8 The New design FEA results

Now to show how the new design compares to the old, the same three tests were conducted, this is to keep the validation of the results and to show if the new design has hindered its strength to meet the regulations. Predictions for the results are that the new design will see less stress and displacement in all three tests, therefore showing the new design is an improvement and adds strength and rigidity to the chassis.

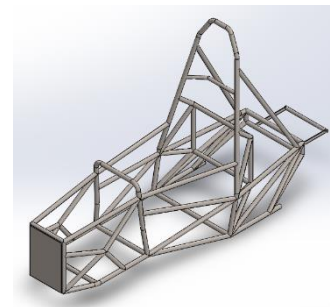


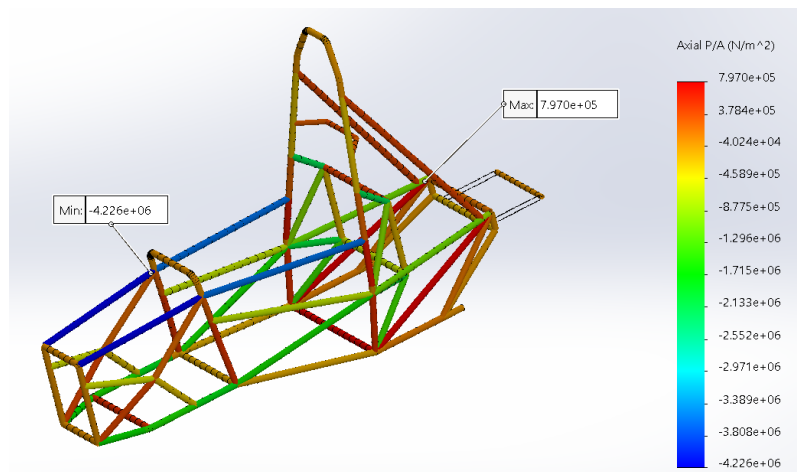
Figure 50 - New chassis CAD model - Oliver Gordon

8.1 Comparison to the old chassis

The first test is the forward strength test, the second test is again the side impact structure test and the final test the twist test.

8.1.1 Test 1

Results 1 – stress, for the test results the scaling was adjusted to show the colouring on the model by selecting the automatic high and low scale feature in the results menu. The max value being 0.797MPa, in comparison to the old chassis'



0.8074MPa, the improvement is Figure 51 - Test 1 new chassis stress results - Oliver Gordon

0.01MPa which is converts to 0.1 N/m², although a small improvement it agrees with the predictions made before the test, in comparison the negative stress has increased by 0.278MPa form 3.948MPa to 4.226MPa, this isn't a significant amount and is due to the changes made to the front end of the chassis, as expected it shows that strength wasn't compromised to save weight. This also shows the changes in regulations implemented by IMCHE are successful in making the chassis stronger.

Results 2 - displacement, the max value the new chassis experiences is 0.4516mm around the seat belt attachment area just like the original chassis but with less displacement and less average displacement, the original max displacement value recorded as 0.5106mm & the average value at around 0.25mm, these are both higher than the new chassis value differences of 0.06mm for the max and 0.025 for the average, this shows the new chassis' strength and rigidity increases in comparison.

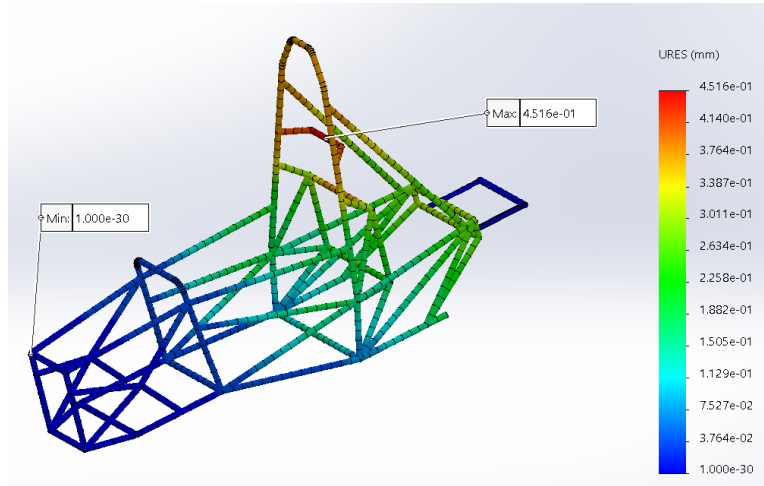


Figure 53 - Test 1 displacement new chassis - Oliver Gordon

test 1 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacement	displacement diff
1	0.8074	0	3.948	0	0.5106	0
2	0.797	0.0104	4.226	-0.278	0.4516	0.059

Figure 52 - Original chassis to redesign chassis result comparison - Oliver Gordon

As seen above in the table 'test 1 comparisons' the format for the positive stress, negative stress and the displacement comparisons between the original chassis and the redesigned one.

Results 3 - strain values, these are no different to the old chassis as proven on the right with the values still being 0 minimum and maximum strain, this proves the system was incapable of calculating the strain due to an error in the simulation and therefore we can now ignore the strain results in the further tests. As previously stated, ref Uni closed.

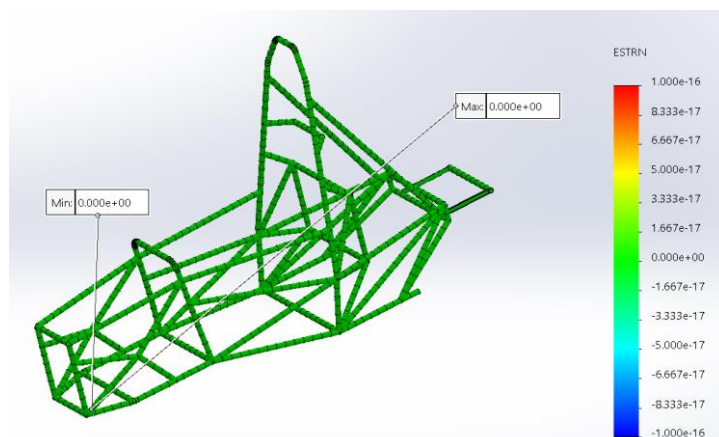


Figure 54 - Test 1 new chassis strain results - Oliver Gordon

8.1.2 Test 2

With test 2, the side impact structure, the max value on the results are now $2.430 \times 10^6 \text{ N/m}^2$ with the minimum recorded as $-3.195 \times 10^6 \text{ N/m}^2$, converting to 2.430MPa and -3.195MPa. The side impact test on the old chassis results are -3.656 MPa for the negative stress and the positive stress recorded as 2.242 MPa. A stress increase is noted in the engine bay at the mounting brackets, the increase of stress is 0.188MPa and the stress acting in an opposite direction to the load has had a decrease of 0.461MPa which overall gives a total of 0.273MPa less stress experienced, this shows an overall improvement in the side impact if changed for the new design.

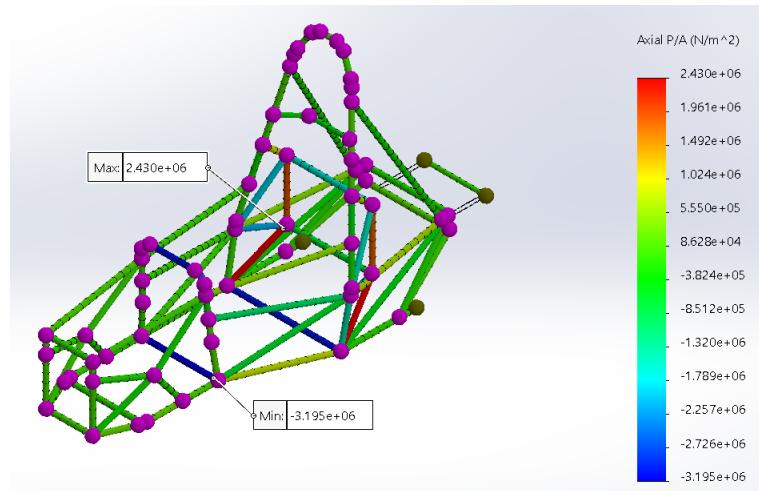


Figure 55 - Test 2 new design - Oliver Gordon

The displacement for the second

test on the new chassis design results came back with, 0.5194mm max displacement and a minimum

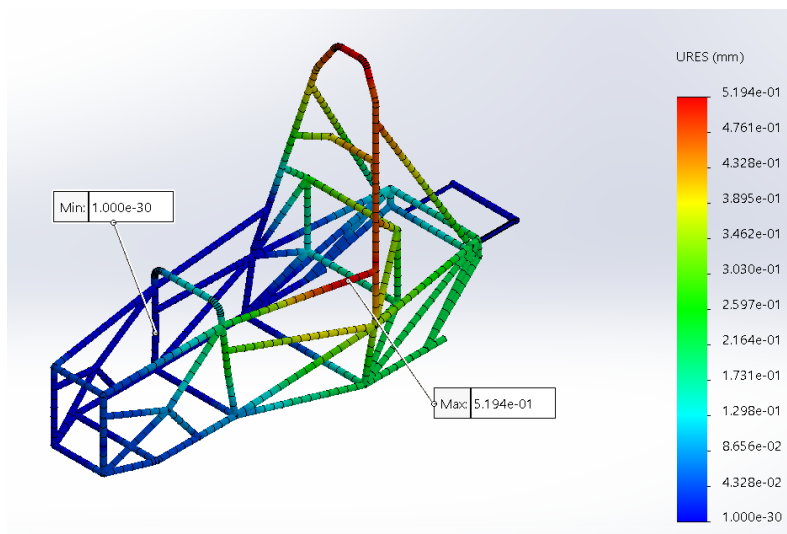


Figure 57 - Test 2 new chassis displacement results - Oliver Gordon

displacement of 1.0×10^{-30} mm, the max displacement of the old chassis was 0.5394mm so with an improvement in rigidity and displacement of 0.02mm which is a small improvement but will scale up as more load is applied, this could be shown in a test with huge loads applied or even in motion analysis study.

test 2 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacement	displacement diff
1	2.242	0	3.656	0	0.5394	0
2	2.43	-0.188	3.195	0.461	0.5194	0.02

Figure 56 - Test 2 original chassis to redesigned chassis results comparison - Oliver Gordon

Above is a table of the stresses and displacement from the original chassis in comparison to the newly redesigned chassis for test 2.

8.1.3 Test 3

The final test on the new chassis is the twist test, again being clamped on the left-hand side and load applied on the right-hand side, the load being applied towards the sky (in basic terms).

The max stress experienced on the new chassis in the twist test was $1.144 \times 10^7 \text{ N/m}^2$ which converts into 11.44MPa and the negative stress being - 1.069×10^7 which converts into -10.69MPa. the old chassis experiencing 11.22MPa in

positive stress, this shows a slight decrease in strength in a twisted scenario, this then would require refinement on the new design to stop this happening, something to be considered into the further work section. The negative stress being -9.962MPa, showing another failed attempt of improvement of 0.7MPa which shows again a refinement in the improvement section is required.

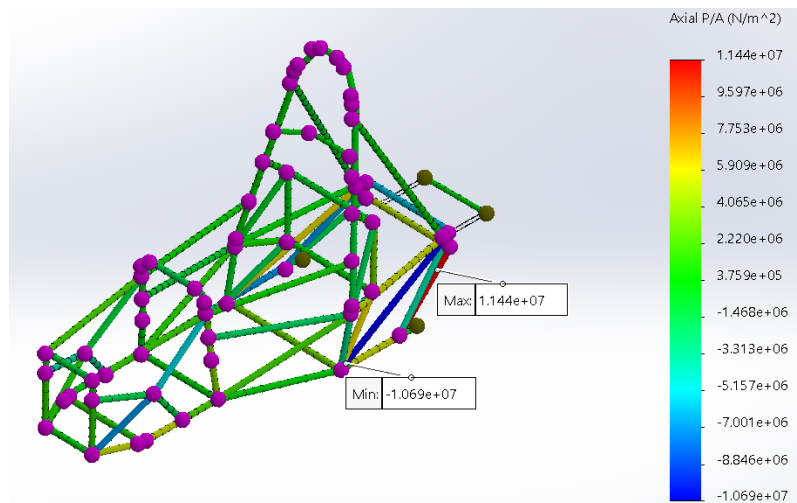


Figure 58 - Test 3 new chassis twist test stress - Oliver Gordon

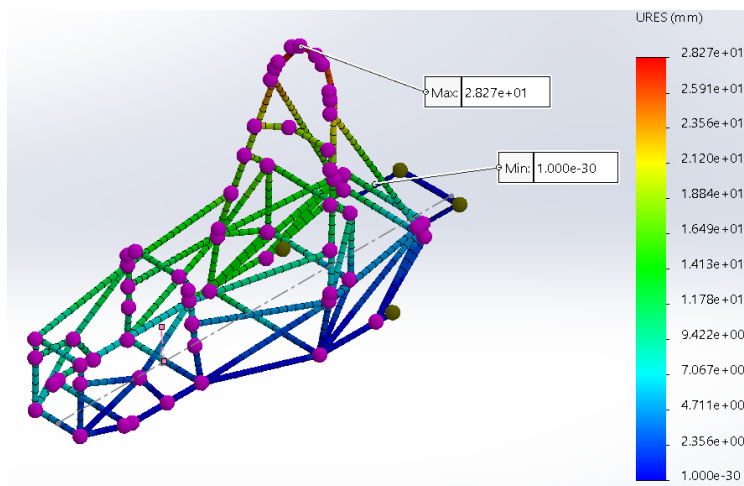


Figure 59 - Test 3 new chassis displacement results - Oliver Gordon

the displacement results show that there is another area that the new design doesn't improve upon with the old chassis having a max displacement of 26.89mm and the new chassis having a displacement of 28.27mm, only a 1.38mm difference which would make little difference in the real-world testing or crash scenario. This figure would scale up as the load increases but

only by a very little amount.

On the left is the full set of results for stress and displacement for the chassis redesign.

new design old materials #2			
test no.	stress +	stress -	dispalcement
1	0.797	4.226	0.4516
2	2.43	3.195	0.5194
3	11.44	10.69	28.27
	Mpa	Mpa	mm

Figure 60 - Full set of result for chassis redesign - Oliver Gordon

test 3 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacemnt	displacemnt diff
1	11.22	0	9.962	0	26.89	0
2	11.44	-0.22	10.69	-0.728	28.27	-1.38

Figure 61 - Test 3 original chassis to redesigned chassis results comparison - Oliver Gordon

As seen above is the comparison of the results from the new chassis and the original chassis in the workshop, as the results show the twist test does make the chassis experience near 100 times the amount of stress in some area's and an extremely high amount of displacement in comparison to the other tests, tests 1&2 were showing displacement between 0.3mm to 0.9mm whereas the twist test is producing 26mm to 29mm so far.

9 Implementing a change in materials

Once the three tests on the new chassis were completed in the FEA, a look into different materials was the next step, for example aluminium and steel react differently and the idea would be if the material on all nonessential structural members was aluminium would this have an adverse effect on the chassis, also in the same test scenario as test 1 on the new chassis how differently would the chassis experience stress and displacement.

Therefore, all the nonessential structural members were changed on the new chassis from Alloy steel to 2014 alloy aluminium, In the regulations it states that the “main and front hoops, shoulder harness mounting bar, side-impact structure,

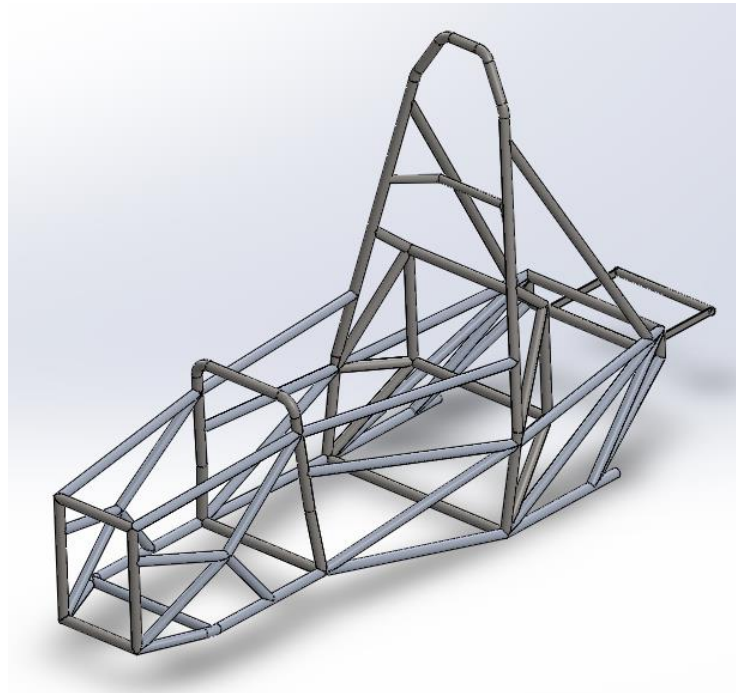


Figure 62 - New chassis with new materials - Oliver Gordon

front bulkhead, roll hoop bracing, driver's restraint harness attachment, front bulkhead support and main hoop bracing supports” (IMECHE, 2018), All, must be steel tubing and that this should be a form of unalloyed steel. “Steel tubing has to be made from unalloyed carbon steel with a maximum content of 0.3% carbon, 1.7% manganese and 0.6% of any other element. All other steel grades are considered alternative materials and require additional testing and documentation (see T3.3).” (IMECHE, 2018). This made a change to the steel used for the main structures, now using plain carbon steel.

The new material properties for the 2014 alloy aluminium and plain carbon steel are as seen below.

2010-T6 Insulated Mold Casting (SS)	Property	Value	Units
2010-T7 Insulated Mold Casting (SS)	Elastic Modulus	7.3e+10	N/m ²
2014 Alloy	Poisson's Ratio	0.33	N/A
2014-O	Shear Modulus	2.8e+10	N/m ²
2014-T4	Mass Density	2800	kg/m ³
2014-T6	Tensile Strength	165445000	N/m ²
2018 Alloy	Compressive Strength		N/m ²
2024 Alloy	Yield Strength	96509800	N/m ²
2024 Alloy (SN)	Thermal Expansion Coefficient	2.3e-05	/K
2024-O	Thermal Conductivity	160	W/(m·K)

2014 alloy
Aluminium.

Figure 63 - 2014 alloy aluminium materials properties - Solidworks

Cast Alloy Steel	Property	Value	Units
Cast Carbon Steel	Elastic Modulus	2.1e+11	N/m ²
Cast Stainless Steel	Poisson's Ratio	0.28	N/A
Chrome Stainless Steel	Shear Modulus	7.9e+10	N/m ²
Galvanized Steel	Mass Density	7800	kg/m ³
Plain Carbon Steel	Tensile Strength	399826000	N/m ²
Stainless Steel (ferritic)	Compressive Strength		N/m ²
Wrought Stainless Steel	Yield Strength	220594000	N/m ²
Iron	Thermal Expansion Coefficient	1.3e-05	/K
Aluminium Alloy	Thermal Conductivity	43	W/(m·K)
	Specific Heat	440	J/(kg·K)

Plain carbon steel

Figure 65 - Plain carbon steel material properties - Solidworks

Repeating test 1 that was conducted on the previous two chassis was critical to keeping testing valid across the board, so the decision was made to repeat the test because then a direct comparison could be done to the new chassis that had been constructed from the old materials, this way only one variable was changed at any one time, firstly the design and then secondly the material.

The expected results are for the chassis to show more stress and displacement across the entire chassis due to aluminium being a weaker material than steel but if the chassis doesn't get near to regulated failure or breaking points then the test can be concluded as a success, although the strength will be less it is proven to be within the regulations therefore significant enough than to have the chassis remain entirely made out of steel. After all, this would be far too heavy and as previously stated if you can improve the weight of the vehicle without compromising the strength this is going to win your races/championships.

New chassis with old material max stress result was 0.797 MPa, with the max displacement value being 0.2827mm. the realistic goal is to get as close to these values as possible because, in theory, the chassis shouldn't be experiencing less stress or displacement. But also due to the steel also being changed to Plain carbon steel, this could see the results be very similar pending on how the stress reacts and where on the chassis the max and min stresses occur.

9.1 Test 1 with new materials

The results of the new chassis as shown on the right with the max positive stress value being $9.00 \times 10^5 \text{ N/m}^2$ which is

0.900MPa, this has a difference of 0.103MPa which is essentially such little change it doesn't need to be worried about, although if the team/university could afford to change the material of the chassis then it would be recommended to do so. The negative chassis or stress acting against the direction of load applied is $-4.196 \times 10^6 \text{ N/m}^2$

which converts into -4.196MPa, this is a lower stress value than the test results from test 1 on the new chassis with a value of -4.226MPa, a difference of 0.03MPa, this result further enhances the recommendation to change the material of all nonessential structural members to 2014 alloy aluminium due to the huge weight saving. The aluminium has a mass density of 2800 kg/m^3 and the plain carbon steel having a mass density of 7800 kg/m^3 which is what the entire chassis would have to be made of if you were to switch to an entire steel chassis. A saving of 5000 kg/m^3 .

The displacement on the updated materials chassis as shown on the right shows the max being 0.6173mm and the minimum being $1 \times 10^{-30} \text{ mm}$, so with the updated material the chassis only experiences a slight increase in displacement in the main roll bar section of the chassis, 0.4516mm is recorded as the displacement of the new design with old materials, a difference of 0.1657mm.

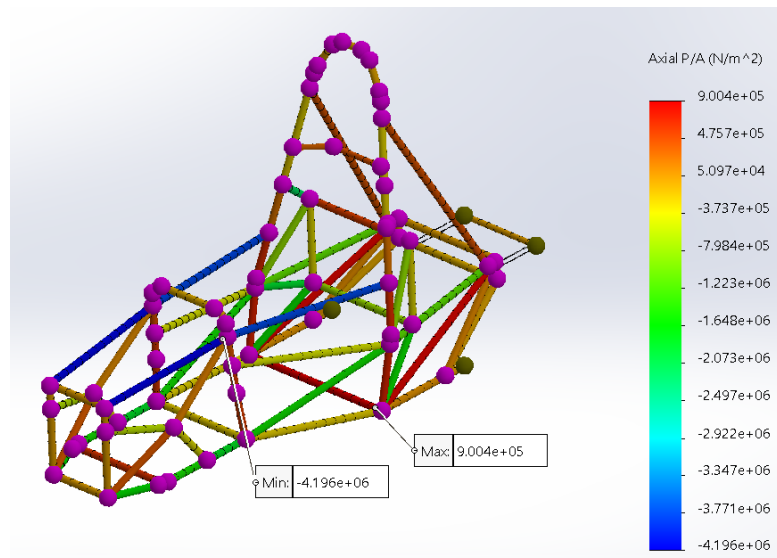


Figure 67 - New materials test 1 stress results - Oliver Gordon

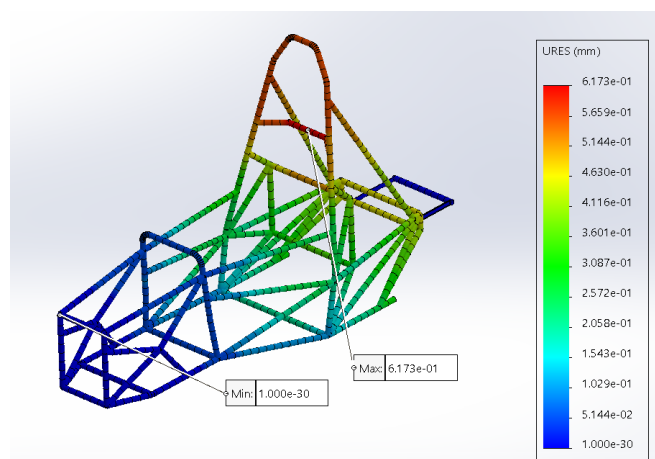


Figure 66 - New materials test 1 displacement results - Oliver Gordon

test 1 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacemnt	displacemnt diff
1	0.8074	0	3.948	0	0.5106	0
2	0.797	0.0104	4.226	-0.278	0.4516	0.059
3	0.9	-0.103	4.196	0.03	0.6173	-0.1657
	Mpa	Mpa	Mpa	Mpa	mm	mm

Figure 68 - Test 1 results comparison table for all 3 chassis' - Oliver Gordon

When looking at the results for all three chassis during & after the testing phase and putting results in excel the stress difference cells show whether there has been an improvement or not, also, if there hasn't been an improvement is it a significant decrease in strength where the chassis would require further redesigning? In the difference cells (pos stress diff, neg stress diff, displacement diff), a positive difference would be the improvement of strength and rigidity whereas as a negative difference would be a decrease in strength and rigidity. The same can be said with the displacement of positive and negative differences.

9.2 Test 2 with new materials

The results of the side impact test on the new chassis with the plain carbon steel support and aluminium frame were recorded as - max stress 2.502×10^6 N/m² which converts into 2.502MPa, the negative stress recorded as -3.178×10^6 converting into -3.178MPa, the old chassis results were 2.430MPa and -3.195MPa thus showing a difference of 0.072MPa in the max stress and 0.017MPa in the negative stress. This shows an average of 0.055MPa difference overall which is an excellent result considering the strength differences in steel and aluminium.

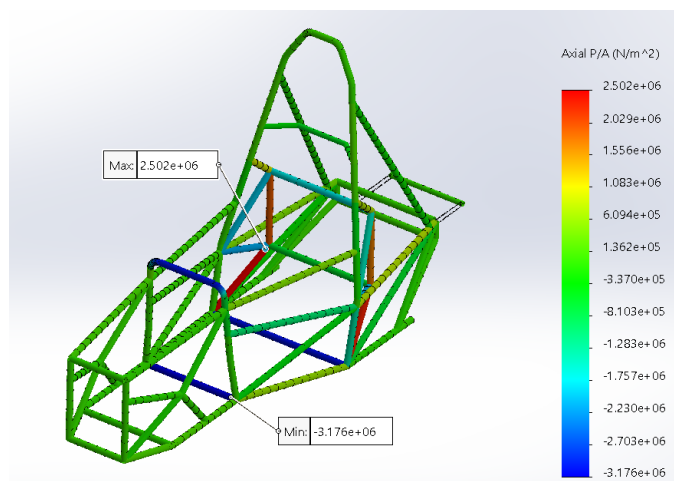


Figure 69 - New material test 2 stress results - Oliver Gordon

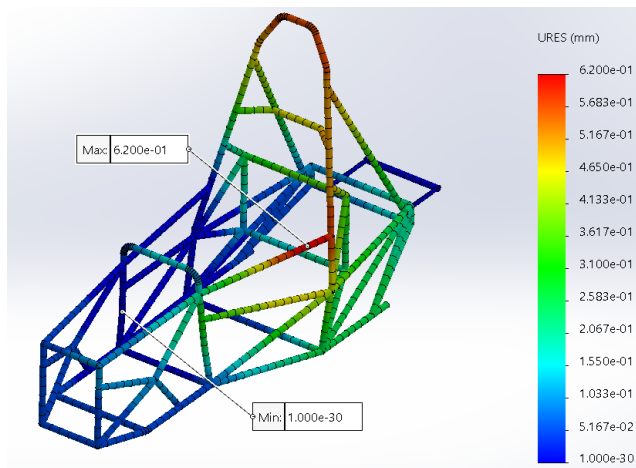


Figure 70 - New materials test 2 displacement results - Oliver Gordon

The displacement results recorded are 0.62mm, this is an increase of 0.1106 from the max displacement of the new design and 0.5194mm from the old materials, not significant enough to not change the material to aluminium which is now looking like the best way to progress the chassis and develop it further.

test 2 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacemnt	displacemnt diff
1	2.242	0	3.656	0	0.5394	0
2	2.43	-0.188	3.195	0.461	0.5194	0.02
3	2.502	-0.072	3.178	0.017	0.62	-0.1006
	Mpa	Mpa	Mpa	Mpa	mm	mm

Figure 71 - Test 2 full set of results comparison table - Oliver Gordon

As previously stated in the comparison of the previous results there are some improvements which show that the redesign of the chassis has been a success, this depends on the test and the direction of stress to show where the conclusion differs with the new materials, this is due to the stress being overall higher on the aluminium chassis in comparison to the complete steel chassis.

The displacements show that the aluminium chassis flexes more than the steel chassis a point that becomes clearer after the twist test.

9.3 Test 3 with new materials

The 3rd test with the new materials chassis results show, a very small increase in stress positive and negative, the positive results $1.219 \times 10^7 \text{ N/m}^2$ converting into 12.19MPa, an increase of 0.75MPa for the last set of test 2 results in the previous set of testing, a 0.75MPa from 11.44MPa with the negative stress being -1.173×10^7 which in megapascals is -11.73MPa which is a total of 1.04MPa increase in stress from the -10.69MPa experienced in test 2 on the new chassis without the updated materials.

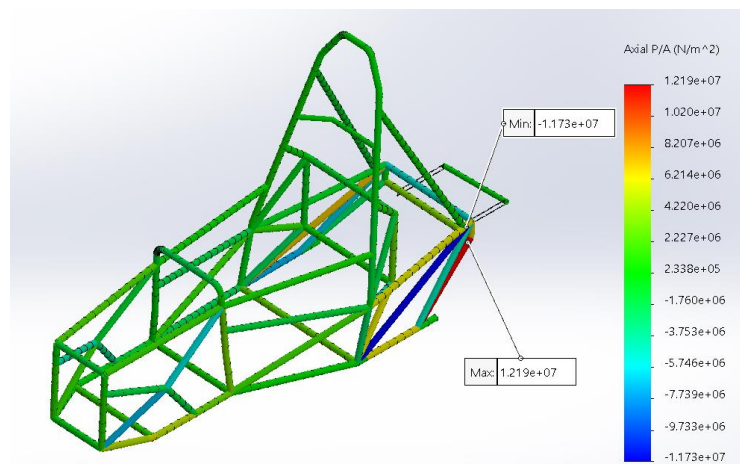


Figure 72 - New materials test 3 results stress - Oliver Gordon

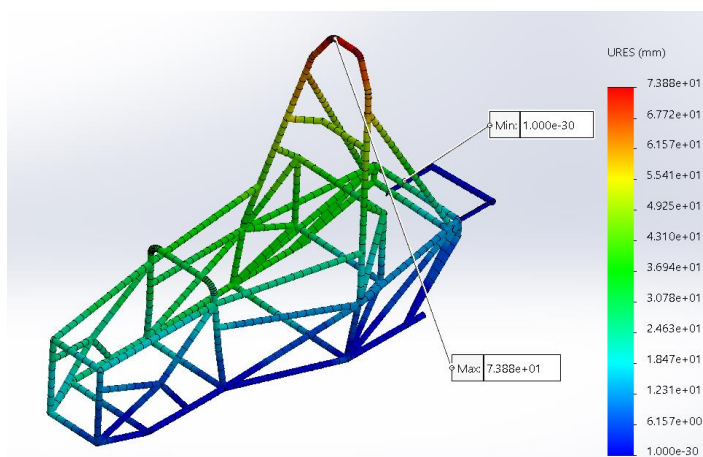


Figure 73 - New materials test 3 displacement results - Oliver Gordon

The displacement result is as follows, 73.88mm of max displacement at the top of the main roll bar, the old materials version of the chassis having a max displacement of 28.27mm, this means the new materials experiences around 2.5 times more displacement than the old materials which is a difference of 45.61mm, this is a huge increase of displacement in comparison to the old and new designs, showing in

a twist test scenario aluminium and plain carbon still struggle at spreading the displacement, so a refinement on the main roll bar would be recommended in this instance.

new design new materials #3			
test no.	stress +	stress -	dispalcement
1	0.9	4.196	0.6173
2	2.502	3.178	0.62
3	12.19	11.73	73.88
	Mpa	Mpa	mm

Figure 74 - Full set of results for the new materials chassis for all three tests - Oliver Gordon

On the left is a full set of results for the updated materials chassis.

As seen in the test 2 comparison's the aluminium is a downgrade in terms of strength, but what is interesting to see in test 3 is that the displacement has seen a huge rise, this is due to the lower elastic modulus and lower yield strength of the 2014 alloy aluminium in comparison the alloy steel.

test 3 comparisons						
chassis no.	pos stress	pos stress diff	neg stress	neg stress diff	displacemnt	displacemnt diff
1	11.22	0	9.962	0	26.89	0
2	11.44	-0.22	10.69	-0.728	28.27	-1.38
3	12.19	-0.75	11.73	-1.04	73.88	-45.61
	Mpa	Mpa	Mpa	Mpa	mm	mm

Figure 75 - Full set of results for test 3 comparison's - Oliver Gordon

10 Conclusions

To conclude this report, the chassis has been re-designed to bring it up to date with current FSAE & IMechE regulations, along with adding a twist to the design with subtle changes which help enhance the chassis ability to absorb stress in some areas as seen in the results for chassis one and chassis two in comparison tables. Once the first set of testing was complete the view was to update the side impact structures to meet regulations and then try and improve any area's that were experiencing high stress e.g. the main roll hoop, in theory adding in some radius to existing sharp corners allowing stresses to be more evenly distributed throughout the roll hoop, as seen with the testing results this was achieved. The same theory was applied to the redesign of the front roll hoop, adding a couple of radiuses to the top corners which allowed the front roll hoop to experience less stress and displacement. From just the redesign view the chassis experience an overall improvement across all three test scenarios, a few points require additional improvement which is a step for further work.

The plan was to take the chassis through a motion study phase to elaborate on the static tests already conducted, this had to be moved to further work due to closure of the university with no access to workshops, test facilities and project rooms.

11 Further work

This section shall be the plans that were in place during this project before the university closure due to the COVID-19 outbreak and any other objectives that could not be completed due to computer access away from the university after the closure.

The materials for the current chassis and the Solidworks validation components could not be confirmed, discussions were had that the material was alloy steel therefore was used during testing and this appears to be correct but there was never a confirmation from any senior staff in the workshop.

An objective of the project was to conduct a motion analysis of the chassis going around a cornering or over a bump, also hitting a wall or even have an object hit into the chassis from the side to see a real-world set of results in FEA to conduct analysis on, we would have also conducted a new static test on each version of the chassis and apply extreme loads to take the chassis close to breaking point and see how the results come back.

The motion study would include a full set of suspension front and rear and all four wheels and tyres.

None of the above could be concluded or tested due to the university closure.

12 Pandemic impact on dissertation

The global pandemic Covid-19 has had a lasting effect on the process and results of the project, by having the closure of the workshop in advance of the closure of the university hindered the ability of testing and gathering results.

1. Had to have a member of staff conduct validation testing because access was not granted to students, this meant a few assumptions had to be made about the Denison testing.
2. Through the university being closed there was not access to sufficient computing power to conduct advance testing for the dissertation in Solidworks.
3. No ability to grab any extra measurements that were required to do extra/more in-depth testing.
4. Having to move hundreds of miles away due to the lockdown rules enforced by the government and having no idea whether the university would be opening again.
5. The lack of communication leading up to the original deadline due to the staff holidays resulted in no feedback, any feedback was indeed accessible via email but a slow process, no email was sent out telling us that this was going to happen, just by talking to fellow students were some people able to find out.
6. No time extension could amount to the loss of access to the university facilities, no access to hands on lectures, workshops, face to face group study, computing facilities etc
7. The constant worry whether the course would or could be completed has had an impact on the concentration on the final stages of this dissertation.

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14 Appendices

2019 Regulation's

"T3 GENERAL CHASSIS DESIGN

T3.1 General Requirements T3.1.1 Among other requirements, the vehicle’s structure must include:

- Two roll hoops that are braced
- A front bulkhead with support system and IA
- Side-impact structures” (IMECHE, 2018)

“T3.2 Minimum Material Requirements T3.2.1 Table 4 shows the minimum requirements for the members of the primary structure if made from steel tubing. Item or application Minimum.

Minimum cross Minimum area wall thickness sectional area moment of inertia Main and front hoops, 2.0mm 175mm² 11320mm⁴ shoulder harness mounting bar Side impact structure, 1.2mm 119mm² 8509mm⁴ front bulkhead, roll hoop bracing, driver’s restraint harness attachment (except as noted above) Front bulkhead support, 1.2mm 91mm² 6695mm⁴ main hoop bracing supports

Item or application	Minimum wall thickness	Minimum cross sectional area	Minimum area moment of inertia
Main and front hoops, shoulder harness mounting bar	2.0 mm	175 mm ²	11 320 mm ⁴
Side impact structure, front bulkhead, roll hoop bracing, driver’s restraint harness attachment (except as noted above)	1.2 mm	119 mm ²	8509 mm ⁴
Front bulkhead support, main hoop bracing supports	1.2 mm	91 mm ²	6695 mm ⁴

Figure 76 - Table 4: Minimum Material Requirements - (IMECHE, 2018).

Table 4: Minimum Material Requirements” (IMECHE, 2018)

“T3.2.2 Steel tubing has to be made from unalloyed carbon steel with a maximum content of 0.3% carbon, 1.7% manganese and 0.6% of any other element. All other steel grades are considered alternative materials and require additional testing and documentation (see T3.3).” (IMECHE, 2018)

“T3.2.3 Except for inspection holes, see T3.7.6, any holes drilled in any tube which is a member of the primary structure must show proof of equivalency in the SES. T3.2.4 The steel properties used for the calculations in the SES must be:

Non-welded strength for continuous material calculations: • Young’s Modulus (E) = 200GPa • Yield Strength (Sy) = 305MPa • Ultimate Strength (Su) = 365MPa

Welded strength for discontinuous material such as joint calculations: • Yield Strength (Sy) = 180MPa” (IMECHE, 2018)

“T3 General Chassis Design

- Ultimate Strength (S_u) = 300MPa T3.2.5

Any tubing with a wall thickness less than 1.2mm or a minimum area moment of inertia less than 6695mm⁴ is considered non-structural and will be ignored when assessing compliance to any rule regarding the vehicle structure.” (IMECHE, 2018)

“T3.2.6 If a member of the primary structure (except for the roll hoops) is a bent tube or made from multiple tubes an additional tube must support it. This support tube must: • Have its attachment point at the position along the bend tube where it deviates farthest from a straight line connecting both ends. • Be of the same dimension as the supported tube(s). • Terminate at a node of the chassis. • Be angled no more than 30° from the plane of the supported tube(s).” (IMECHE, 2018)

“T3.2.7 Any welded seams shape must not be mechanically altered in any way.” (IMECHE, 2018)

“T3.3 Alternative Materials T3.3.1 Alternative materials may be used for all parts of the primary structure and the tractive system accumulator container with the following exceptions: • The main hoop and the main hoop bracing must be steel • The front hoop must be metal • Any welded structures of the primary structure must be steel • However, the front hoop may be an aluminium welded structure” (IMECHE, 2018)

“T3.3.2 If any other material than steel tubing are used in the primary structure or the tractive system accumulator container, physical testing is required to show equivalency to the minimum material properties for steel in T3.2.2” (IMECHE, 2018)

“T3.3.3 If alloyed steel as defined by T3.2.2 is used, the team has to include tests and documentation in the SES to show structural equivalency. This may include, but is not limited to: • Receipts and data sheets of the used tubing materials • Documentation about welding processes and filler materials • Documentation about heat treatments • Tests showing adequate strength and elongation at break in the welded condition.” (IMECHE, 2018)

“T3.7 Roll Hoops T3.7.1 Both roll hoops must be securely integrated to the primary structure using node-to-node triangulation or equivalent joining methods.” (IMECHE, 2018)

“T3.7.2 The minimum radius of any bend, measured at the tube centreline, must be at least three times the tube outside diameter. Bends must be smooth and continuous with no evidence of crimping or wall failure.” (IMECHE, 2018)

“T3.7.3 The roll hoops must extend from the lowest chassis member on one side of the chassis, up, over and down to the lowest chassis member on the other side.” (IMECHE, 2018)

“T3.7.4 Roll hoops attached to a composite primary structure must be mechanically attached at the top and bottom of both sides of the structure and at intermediate locations if needed to show equivalency.” (IMECHE, 2018)

“T3.7.5 Mounting plates welded to the roll hoops must be at least 2mm thick steel or 3mm thick aluminium, dependent of the roll hoop material.” (IMECHE, 2018)

“T3.7.6 Both roll hoops must have one 4.5mm hole in a non-critical straight location and its surface at this point must be unobstructed for at least 180°.” (IMECHE, 2018)

“T3.8 Main Hoop T3.8.1 The main hoop must be constructed of a single piece of uncut, continuous, closed section steel tubing.” (IMECHE, 2018)

“T3.8.2 Inside view the portion of the main hoop which is above its upper attachment point to the side impact structure must be inclined less than 10° from vertical.” (IMECHE, 2018)

“T3.8.3 Inside view any bends in the main hoop above its upper attachment point to the primary structure must be braced to a node of the main hoop bracing support structure with tubing meeting the requirements of main hoop bracing.” (IMECHE, 2018)

“T3.8.4 Inside view any portion lower than the upper attachment point to the side impact structure must be inclined either forward or not more than 10° rearward.” (IMECHE, 2018)

“T3.9 Front Hoop T3.9.1 The front hoop must be constructed of a continuous and closed section.” (IMECHE, 2018)

“T3.9.2 If the front hoop is made from more than one piece it must be supported by node-to-node triangulation or an equivalent construction.” (IMECHE, 2018)

“T3.9.3 Inside view, no part of the front hoop can be inclined more than 20° from vertical.” (IMECHE, 2018)

“T3.9.4 If the front hoop is a welded construction made from multiple aluminum profiles, the equivalent yield strength must be considered in the as-welded condition unless the team demonstrates and shows proof that it has been properly solution heat treated and artificially aged. The team must supply sufficient documentation proving the appropriate heat treatment process was performed.” (IMECHE, 2018)

“T3.10 Main Hoop Bracing T3.10.1 The main hoop must be supported to the front or the rear by bracing tubes on each side of the main hoop.” (IMECHE, 2018)

“T3.10.2 Inside view the main hoop and the main hoop braces must not lie on the same side of a vertical line coincident with the top of the main hoop.” (IMECHE, 2018)

“T3.10.3 The main hoop braces must be attached to the main hoop no lower than 160mm below the top-most surface of the main hoop. The included angle formed by the main hoop and the main hoop braces must be at least 30°.” (IMECHE, 2018)

“T3.10.4 The main hoop braces must be straight.” (IMECHE, 2018)

“T3.10.5 The lower ends of the main hoop braces must be supported back to the upper attachment point of the main hoop to the side impact structure and to the lower attachment point of the main hoop to the side impact structure by a node-to-node triangulated structure or equivalent composite structure.” (IMECHE, 2018)

“T3.10.6 If any item which extends outside of the primary structure is attached to the main hoop braces, additional bracing is required to prevent bending loads in a rollover situation.” (IMECHE, 2018)

“T3.11 Front Hoop Bracing T3.11.1 The front hoop bracing consists of two single, straight tubes (one on each side of the front hoop) which connects to the front hoop and the structure forward of the driver’s feet (e.g. front bulkhead).” (IMECHE, 2018)

“T3.11.2 The front hoop bracing structure must be attached no lower than 50mm below the top-most surface of the front hoop, see figure 5.” (IMECHE, 2018)

“T3.11.3 If the front hoop is inclined more than 10° to the rear, additional braces extending rearwards are required.” (IMECHE, 2018)

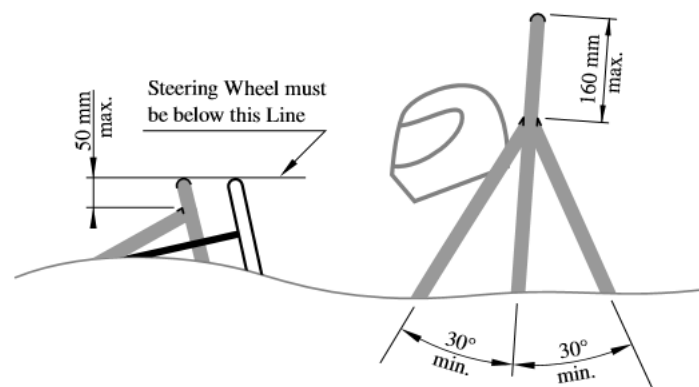


Figure 77 - Figure 5: Front hoop bracing, main hoop bracing and steering wheel requirements - (IMECHE, 2018).

“T3.14 Front Bulkhead Support

T3.14.1 The front bulkhead must be supported back to the front hoop by a minimum of three tubes on each side; an upper member, a lower member and diagonal bracing to provide triangulation. • The upper support member must be attached to the front bulkhead a maximum of 50mm below the top-most surface of the front bulkhead and attached to the front hoop a maximum of 50mm below the upper side impact member. If the attachment point of the upper member is greater than 100mm above the upper side impact member, node-to-node triangulated bracing is required to transfer load to the main hoop. • The lower support member must be attached to the base of the front bulkhead and the base of the front hoop. • The diagonal bracing must triangulate the upper and lower support members node-to-node.” (IMECHE, 2018)

“T3.15 Side Impact Structure T3.15.1 The side impact structure must consist of at least three steel tubes, see T3.2, on each side of the cockpit, see figure 8. • The upper member must connect the main hoop and the front hoop. It must be at a height between 240mm and 320mm above the lowest inside chassis point between the front and main hoop. • The lower member must connect the bottom of the main hoop and the bottom of the front hoop. • The diagonal member must triangulate the upper and lower member between the roll hoops node-to-node.” (IMECHE, 2018)

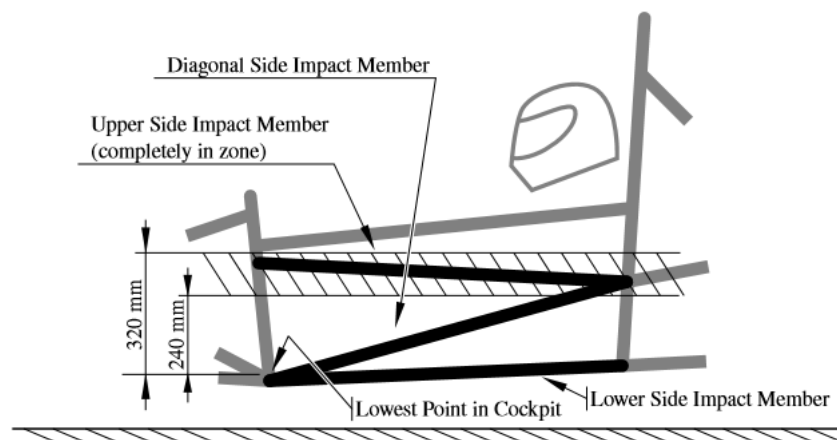


Figure 8: Side impact structure

Figure 78 - Figure 8: Side impact structure - (IMECHE, 2018)