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**Dissertation**

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**Design and Optimisation of an  
MCR Front Upright**

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## Abstract

The purpose of this dissertation is to optimise an existing MCR upright. Highlighting the importance of reducing the unsprung weight to increase performance, endearing to the design specifications such as ball joint and steering arm locations all whilst the design being stiff enough to withstanding forces that the upright will be subjected to including a factor of safety.

Soldworks was implemented to design and analyze the existing MCR upright using analysis programs such as Finite element analysis (FEA) and Truform, thus giving reference points on which to base the prototype design upon. The prototype design was 79.29% lighter than the existing design and deflected 0.0365mm less under a lateral loading when compared to existing upright. The optimal design also proved quite significantly less over engineered when compared to the existing design

Manufacturing processes and material selection are considered to comply with other specifications such as longevity and cost, disclosing how various motorsport teams confront these issues.

After reading this dissertation the reader should gain an insight of how the motorsport industry contemplates the different stages of design thro optimization, design and manufacturing.

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## 1.0 Introduction

The objective of this dissertation is to optimise an existing front upright of an MCR to compete within the regulations of the Sports 2000 championship.

*"If any item can be called more important than others in the suspension, the upright has a good claim. At its most advanced it is a marvelous piece of design having to cope with a singularly complicated mixture of load, stresses and strains of a very high order."*

(Staniforth, 2006)

The function of the upright is to provide a connection between the control arms and hub thus connecting the assembly to the wheel. Uprights comprise of mountings for a steering arm which allows steering of the wheels and brake caliper allowing the vehicle to stop. The hub is a direct link to the wheel and rotates in tandem. On the other hand the upright remains parallel with the chassis. This achieved with a bearing between the assemblies. The upright also allows components such as the top and bottom mounts and steering link to be located and mapped on programs such as WinGeo3.

Figure 1.1 demonstrates a non-driven independent wishbone suspension.

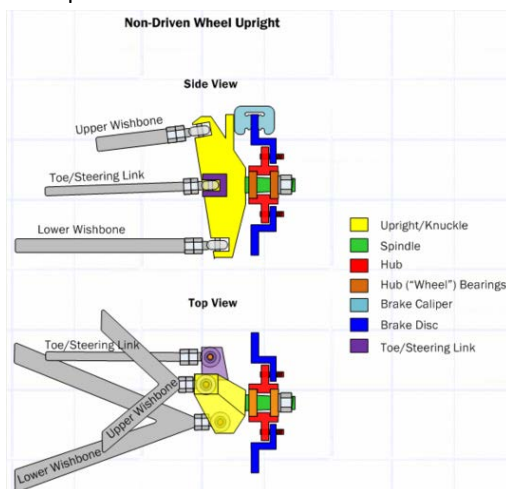


Figure 1.1 Typical upright (Car suspension basics., n.d.)

Highlighted in yellow is the upright connected via the top and bottom rod ends, allowing translation and rotational movement about the king pin axis. The steering link shown in purple manages the toe angle of the upright and this too is connected by rod ends. A spindle (green) locks the upright into place whilst the hub (red) is allowed to freely rotate around the spindle with use of a bearing (red).

### 1.1 Motivation

The dissertation presented was chosen as an insight of how engineers engage the design process from a concept to a final design whilst endearing to the constraints set. It shows good practice to familiarise myself with what would be expected of an engineer in the workplace before leaving university. Utilising and gaining skills within, time management, Computer aided design (CAD) software and the analytical software evaluating the prototype, materials and their properties and manufacturing processes of said materials.

Another factor that played a role was developing an assembly with the aim of increasing the performance of the MCR and seeing visible results.

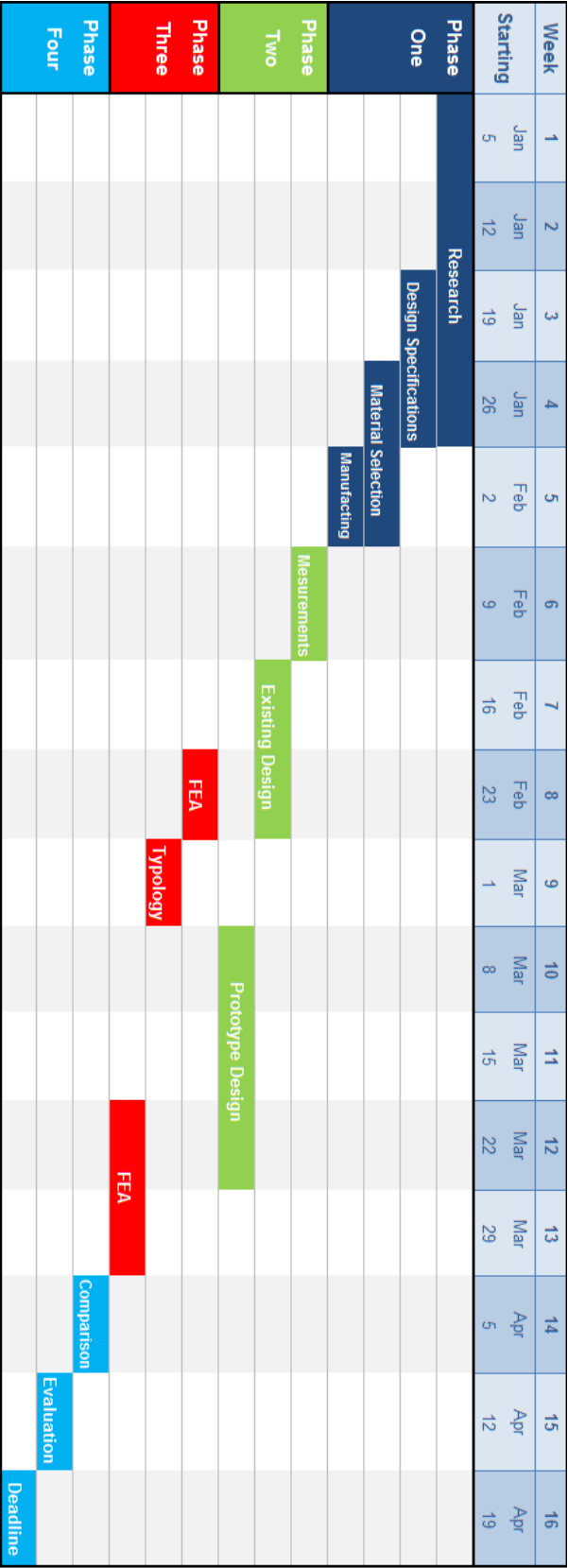
### 1.2 Objectives

The main purpose of this dissertation is to create a suitable prototype which adheres to all the constraints set whilst increasing performance. However other objectives should be set to structure the dissertation and are as follows;

- Research Uprights and roles from relevant resources.
- Create a Gantt chart to visualise the project.
- Consider all the design specifications.
- Analyse existing and prototype uprights.
- Determine a suitable manufacturing process.
- If time and costs permits build and test design.



2.3 Gantt Chart



## 2.0 Literature Review

This review analyses suspension concepts with regards to upright designs rather than uprights and the roles they must fulfil as this will be mentioned in detail within the design specification. Many amateur forums discuss the effects of geometry online and give various explanations and personal views. However, there are peer viewed authors such as, Milliken, Saniforth and Olley, who revolutionised this field and are still referenced to this day as a creditable source.

### 2.1 Suspension Characteristics and Geometry

Upright designs along with rod ends adjustments directly affect suspension characteristics and geometry. The following criteria allude to parameters engineers can optimise.

#### 2.1.1 Kingpin Inclination & Scrub

Kingpin Inclination or Steering axis inclination is the angle that is produced from the top and bottom mounts relative to the centre of the tyre tread, continuing this theoretical line to the road surface creates an offset and this is known as the scrub radius. Scrub occurs as the tyre is not rotating around its centre but rotates around the kingpin axis. This can be shown visually with Figure 2.1 below.

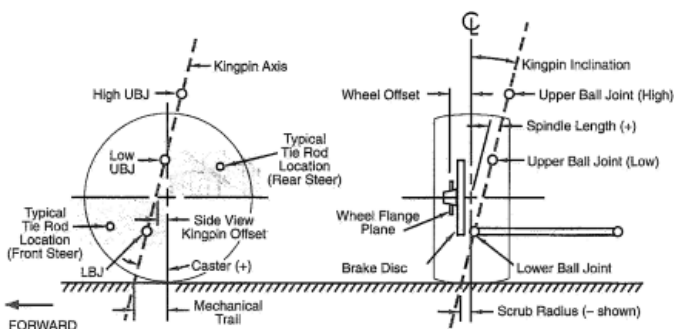


Figure 2.1 Kingpin and Scrub Geometry, (Milliken. 1995)

A benefit of having an inclined kingpin axis is that it forces the rotation of the tyre downwards, as the ground is stable it induces lift. This produces a self-aligning torque as the tyre will follow the path of least resistance this keeps the vehicle straight without any input.

(Milliken W. M., Steering Geometry, 1995)

#### 2.1.2 Caster Angle and Trail

Observing Figure 2.1 again the side view shows the tyre trailing the steering axis (Upper ball joint lying behind the lower ball joint measuring this gives caster angle). To envision this, imagine a shopping trolley, when moved in any direction the wheel always aligns itself behind the pivot point. More trail indicates a larger moment to act upon the kingpin axis requiring a higher steering force. This also produces a self-centring effect about the kingpin axis but at higher speeds.

(Milliken W. M., Steering Geometry, 1995)

#### 2.1.3 Ackerman Angle

Ackerman provides a reduction of sideways scrub as the steering linkage encourages the inner wheel to follow tighter radii than the outside wheel these phenomena is more pronounced at lower speeds and tighter corners such as hairpins.

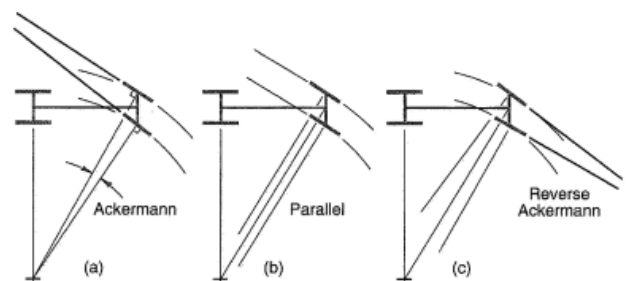


Figure 2.2 Different Ackermann. (Milliken. 1995)

The amount of Ackerman steer can be altered with the position of the steering rack in conjunction with a hypothetical line at the front axle starting at the kingpin axis. Moving the steering rack forward will induce more parallel eventually reverse Ackermann, moving towards the rear increase the toe out on turning.

(Milliken W. M., 1995)

#### 2.1.4 Camber Angle

The SAE describe camber angle as the inclination of the wheel plane to the vertical. It is considered positive when the wheel leans outwards and negative when it leans inwards.

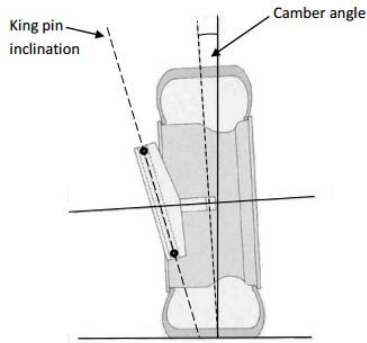


Figure 2.3 Camber Angle (Stanforth, 2006)

Positive camber aids cornering at low speeds as it decreasing the scrub radius which in turn decreases driver effort as less tyre is in contact with the road surface this also reduces tyre wear. These two properties proportional, and is achieved with unequal length wishbones, as when cornering weight transfer forces compression increasing camber, the wishbones allows the wheel to maintain much of the same contact patch it had when straight.

#### 2.1.5 Toe Angle

SAE describes toe angle as the static toe angle of wheel, at a specified wheel load or relative position of the wheel centre with respect to the sprung mass, is the angle between the longitudinal axis of the vehicle and the line of intersection of the wheel plane and the road surface. The wheel is toed in if the forward portion of the wheel is turned towards the central longitudinal axis of the vehicle and toed out if away.

Toe in promotes stability as during cornering the tyre will straighten in the direction of turning due to the kingpin inclination.

Toe out tends to produce unstable behaviour during these conditions as the slip angle increases on the outside wheel, on the other hand it's also improves turn in.

### 3.0 Design Specification

*"It is important to recognise the relationship between available resources and expectation"*

(Milliken M. &, 1995)

To expand on what Milliken & Milliken alluded to designing a component must incorporate, objectives, considerations and constraints while working within the resources provided such as time, budget, facilities and experience.

This section of the dissertation intendeds to discuss all the parameters mentioned above to produce a prototype that can meet the expectations required.

#### 3.1 Design Considerations

Before the design of the upright considerations about the functionality must be considered.

- Contributes to unsprung mass.
- Sufficient wall thickness to make the component ridged to forces encountered.
- Provide sufficient thickness to brake caliper mounts to endure sudden torque from the disk rotor.
- House the top and bottom suspension pick-up points based on the kingpin inclination.
- Steering arm mount to be placed on the opposite side of the brake caliper mount.
- House the Wilwood brake calliper.
- Press fitting tolerance to be provided in the central bore diameter to press fit the stub axel.

Another specification to consider is adjustability of camber, caster and toe in / toe out. Take in consideration the Porsche 962 where the top mount and steering arm were located on a separate part that is bolted to the upright this reduces the complexity of design, however it increases the unsprung mass and increases manufacturing time and costs. This was not part of the existing design and due to the disadvantages mentioned will not be a part of the new design.

To expand further on some of the points noted such as uprights contributing to unsprung mass. Unsprung mass can have profound effects on handling and performance.

Uneven road surfaces cause the wheel assembly to compress transferring forces to the chassis and in extreme conditions can cause the tyre to briefly leave the track. The amount of movement, for short bumps, is inversely proportional to the weight. Reducing unsprung weight, therefore, decreases inertia and this in turn reduces the work of the shock absorbers and springs to keep the tyres in contact with the surface. Heavier assemblies would oscillate more taking longer to disperse the energy absorbed from a bump so is consequently unstable for longer.

Weight reduction in general can improve performance and can be shown using Newton's second law of motion as mass = force x acceleration, as reducing the mass of a given force will increase the acceleration.

## 3.2 Design Constraints

Every component should obey limitations imposed on the design, be it, physical, monetary or governing organisations constraints. The prototype in question faces the following constraints.

### 3.2.1 Physical Constraints

As the upright is the only part being optimised it must fit within the pre-arranged assembly. This includes such things as the same dimension centre bore as the previous design to accommodate the hub, spindle and bearing. Use existing geometries such as kingpin inclination, caster and steering arm location, house the same calliper mountings and exist within the finite design space with no extreme tolerances. As the previous the design only used a single bolt hole for the steering mounts to counteract the shear forces exerted so will the prototype design. Finally, sufficient fillet radius throughout the design to minimise notch sensitivity.

### 3.2.2 Monetary Constraints

Costs effect every project and this is no different. It has been stated to design an upright that can withstand cyclic load for the duration of the Sports 2000 Championship without hindering the performance too much, as designing an upright that would last one race but give max performance is just unfeasible. Costs must also be minimised where possible so manufacturing and material costs should also be analysed.

### 3.2.3 Governing Organisation Constraints

After reading the SRCC Sports 2000 Championship sporting and technical regulations 2017 with regards to section 5.8 Suspension the only constraints applicable would be under heading 5.8.6 Prohibited Modifications

- a) Titanium is prohibited
- b) Chromium plating of any steel suspension part is forbidden

## 3.3 Design Objectives

As the considerations and constraints have been accounted for practical objectives can stated with the aim of fulfilling and are listed as followed.

- Reduce the unsprung mass.
- Maintain or if possible increase stiffness.
- Longevity of assembly to withstand a year's worth of cyclic loading without extensive fatigue that could hinder performance.
- Easy of manufacture
- Relatively low costs for manufacturing and materials.
- Must incorporate previous assembly parts.
- Maintain previous geometry settings.
- Fit within design space allocated.
- Not to be overly engineered as compromise the weight reduction.

These goals can be verified by through FEA.

### 3.4 Resources

There will be several resources needed throughout the dedicated time to complete the project and are as listed below;

- Access to engineering based books to deepen knowledge of subject. Loaning books from university library. However, books could be already loaned out.
- A computer with design programmes such as Solidworks and the analysis software built into the programme such as Finite Element Analysis (FEA) and Topology, to help with the design process. I intend to gain access via the university in free time slots when computer rooms are not in use. However, this can lead to limitations as only being able design within certain time could put strain on time that project can be finished.
- Access to the workshop and the MCR itself to make measurements on which the designs will be based upon. Measurements will also be taken from a program called WinGeo3 where the MCR suspension geometry can be found.

### 3.5 Material Properties

As mentioned above in design objectives, weight consideration will dictate the choice of material as the design prefers a low unsprung mass that can also negate the forces induced during motion as well as remaining lightweight. The chosen material for the upright, should excel within the parameters set, costs of said material must be taken into consideration. Suitable parameters are;

**Specific Strength** – Measures a materials strength (per unit area at failure) divided by its density (Strength to weight ratio). As the optimisation focuses on a lightweight design this will be a key parameter.

**Yield Strength** – Using the calculations shown in the methodology, the uprights will need to endure a maximum of 13204N laterally, without physically deforming too much, which will lead to changes in camber angle hindering performance. However, deformation is a precursor of failure and ductility can allow for pre-emptive removal of the failing part.

**Fatigue Strength** - Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading, as the upright will go through many cycles throughout their lives, the material would need to excel in this area.

**Young's Modulus** – Predicting how much a material sample extends under tension or shortens under compression. Some calculations also require the use of other material properties, such as the shear modulus, density, or Poisson's ratio. It will help in selection of materials for structural applications. Previously shown that uprights will experience both compression and tension so again the material will need to excel.

**Tensile Strength** - Measures the force required to pull something such as rope, wire, or a structural beam to the point where it breaks. The tensile strength of a material is the maximum amount of tensile stress that it can take before failure a key element when constructing an upright.

Components subjected to fluctuating loading fail at much loads than their service loads due to fatigue. When not considered in the design stage it can lead to catastrophic failures. Factors leading to fatigue include a large number of sufficient cycles of stress applied and a large variation of loads induced which is very common for a upright to experience.

Additional factors which exaggerate fatigue failure are:

- Corrosion
- Residual Stress
- Stress Concentrations
- Temperature
- Surface Finish
- Stress Range
- Use of Welding

The mechanisms of fatigue failure can be separated into 2 stages, crack initiation and crack propagation to the point of static failure. This crack once formed begins to grow in each cycle of loading. Growth is accelerated by higher amplitude of loading of characterised by multiple cracks. Final catastrophic failure occurs when a crack has grown to a significant length such that the next application of load results in static failure due to reduced area in that region. Fatigue cracks may start at various places such as concentration of plastic strains, extrusions or intrusions of the surface and grain boundaries.

The below were graphs are cited and used in conjunction with CES Edupack 2013 using the parameters above to make analyse material properties in order to select a suitable candidate for an upright. The graphs contents are arranged in "Material Families" and are labelled within the graph.

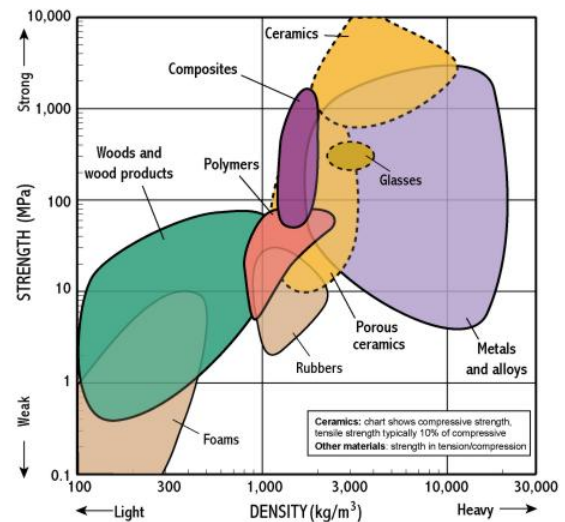


*Table 3.1 Specific Strength Graph (Cambridge, n.d.)*

The graph highlights specific strength and ideally a material that is in the top left quadrant would be a perfect candidate but as there is no such material and elimination process began. For obvious reasons foams, woods and rubbers were ruled out. The graph is slightly deceiving as ceramics and glass seem a suitable material however being brittle means a much lesser energy to failure and ductility. The same could be said for composites such as carbon fibre, which have a high young's modulus and high ultimate strength, another issue here would be difficulty in predicting the stresses making it difficult to design a laminate for a three-dimensional composite material capable of handling complex and multiple stress directions. This leaves the metals and alloys to research as they are more ductile than the above materials although having relatively lower strength, the higher toughness to fail leads to being the perfect candidate for the application in question. The graph below highlights the different metals and alloys.

## Metals and alloys

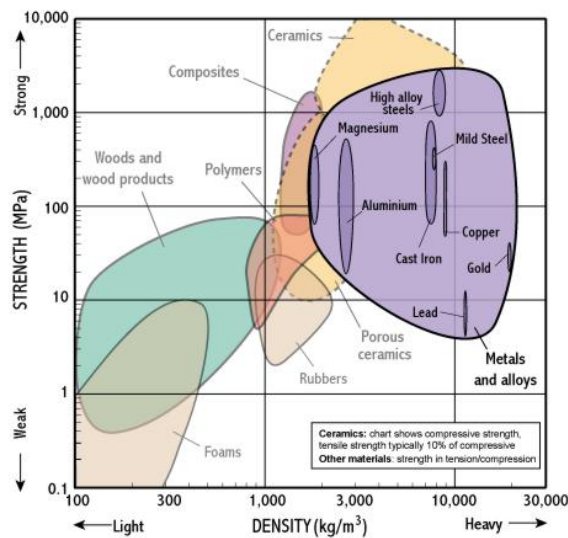


Table 3.2 Specific Strength of Metals and Alloys (Cambridge, n.d.)

The above graph breaks down the different metals and alloys which could be considered it shows the contrast in which density and strength can be traded off. As the optimisation is focusing more and a lightweight structure magnesium could be considered an easy choice as it is approximately 34% lighter than aluminium however its ductility and corrosive properties would be a major disadvantage the lightweight option that should be considered is an aluminium alloy due to its increased strength to weight ratio.

After consulting the above graphs, CES Edupack 2013 and taking the constraints into consideration the following suitable materials were analysed.

### 3.5.1 Aluminium 6061 T6

This high-strength aluminium alloy relies on age-hardening: a sequence of heat treatment steps that causes the precipitation of a nano-scale dispersion of intermetallics that impede dislocation motion and impart strength. This can be as high as 700 MPa giving them a strength-to-weight ratio exceeding even that of the strongest steels.

The 6000 series with Al with up to 1.2% Mg and 1.3% Si, are medium strength extrusions and forgings and are typically found for use in trucks, buildings, pipelines and other structural applications where strength, weldability and corrosion resistance are needed. And has good machinability. (Edupack 2013)

It has a density of between 2500-2900 Kg/m<sup>3</sup> and a price ranging between 1.45 – 1.60 GBP/Kg with price varying.

### 3.5.2 4130 Alloy Steel

AISI 4130 alloy steel is specified by the SAE. It contains chromium and molybdenum as strengthening agents and has low carbon content, and hence it can be welded easily.

The alloy mainly consists of approximately 97% iron, 1.1% Chromium, 0.9% Manganese, 0.33% Carbon, 0.25% Molybdenum, 0.35% Silicon, 0.04% Phosphorus and 0.04% Sulphur. Typical applications include general construction and general automotive applications.

It has a density of between 7800-7900 Kg/m<sup>3</sup> and a price ranging between 0.37 – 0.42 GBP/Kg with price varying.

### 3.5.3 Aluminium, 7075 T6

Aluminium 7075 T6 is an aluminium, with zinc as the primary alloying element. It is strong, with a strength comparable to many steels, and has good fatigue strength and average machinability. It has lower resistance to corrosion than many other Al alloys.

7075 aluminium alloy's composition roughly includes 6% zinc, 2.5% magnesium, 1.4% copper, and less than a half percent of silicon, iron, manganese, titanium, chromium, and other metals.

It has a density of approximately 2770-2830Kg/ m<sup>3</sup> and a price ranging between 1.56-1.71 GBP/Kg

The graph below shows the specific strength ratio of the materials above mentioned.

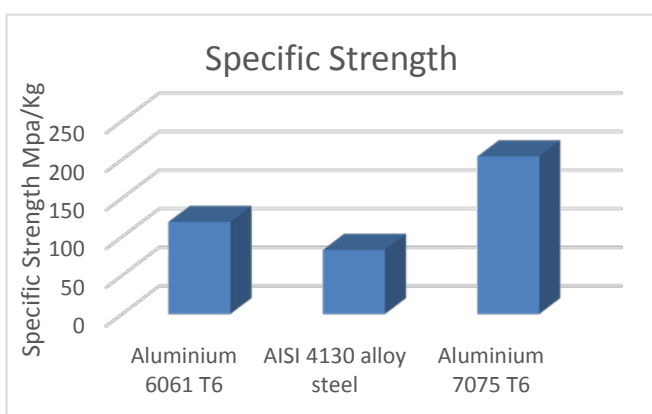


*Table 3.3 Specific Strength comparisons*

As seen in the above graph the Aluminium alloys outperform the steel counterpart. The obvious choice based on this parameter would be 7075 T6 with 204Mpa/Kg. However, this graph does not take into consideration cost and with this factor included 6061 T6 becomes the optimal material if it can sufficiently withstand the forces exerted it.

### 3.6 Manufacturability

Due to the nature of racing information about the specifics about manufacturing of uprights were hard to come by, however most FSAE teams chose between welding sheet metal to create the assembly or CNC machining the upright from a billet. As earlier discussed welding can lead to weak point within the design and increase fatigue CNC machining was chosen due to its precision. It also works well with the chosen material as it is said to have good machinability. Tool clearance must be incorporated into the design with no intricate detailing that could break the tool, increasing cost and time of the manufacturing so exaggerated fillet edging will be applied to the design.

### 4.0 Methodology

This segment now discusses the rationale for the techniques used to collect, map and analyse the existing and the prototype.

The existing upright was measured with a simple digital calliper that had a resolution of 0.01mm. Accuracy have been questioned of digital callipers and have a tolerance of approximately  $\pm 0.02$  mm. This was the preferred choice over a Vernier calliper as judgement can affect the measurements noted.

The geometries in which the Solidwork drawings were based on were accessed from a program called Wingeo3, where the MCR had been previously mapped. This suggested that the upper mount was 361.619 mm above ground and the lower mount was 141.879 mm above ground, extrapolating from this it was said that there was 219.74mm between the mounts. The kingpin inclination was noted as 5.8 degrees.

#### 4.1 Analytical Procedure

An upright is a crucial component as the majority of forces experienced within the suspension set up by passes it. Analyzing every effect of this complex, ever changing system can become complicated and daunting and many engineers tackle the problem in different ways, with varying results.

A favored technique within the FSAE community is to analyse loading of 3 translational axis that relate to lateral, longitudinal and bump loads from the contact patch and applied to the upright thro remote loading from the contact patch to the spindles bearing center plane, where the bearing is connected via a constraint to the upright. This adds moments into the static values inputted.

Remote loads and elastic constraints can be very useful when analysing an upright. Remote loads helps avoid endless calculations. Dummy parts representing hubs, bolts, bearings, a-arms etc, can increase the accuracy of results. However these load cases represent worst case scenarios with conservatism. This conservatism leads to very over engineered products which translates to an upright with excess weight and defies the main objective of this optimization. Another issue concerns the simultaneous loading as a tyre cannot experience 100% of the lateral and longitudinal loads at the same time. An example of this method can be seen below in Figure 4.1. Where the pink lines indicate the remote loads from the contact patch acting on the upright whilst the upper ball joint is constrained horizontally and lower ball joints constrained in all 3 translation axis indicating a push rod set up. The other constraint is applied horizontally to the tie link.

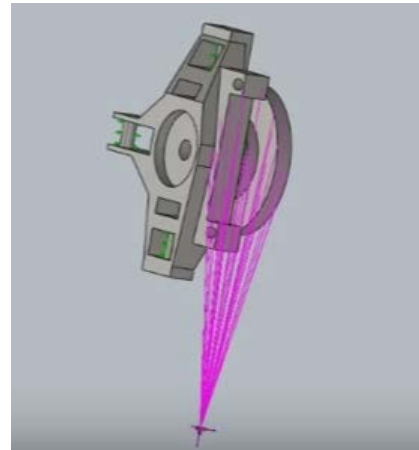


*Figure 4.1 Remote Loading of Dummy Hub*

To ease these complications and various vector forces experienced by the component, many engineers analyse the scenarios individually. Free body diagrams help visualize the forces and moments involved, these scenarios calculate the maximum force that could be endured, then added to the model in the relevant locations and constraints. This method allows from dynamic loads to be calculated which gives the analysis added validity when compared to the remote loading. With this in mind FEA for the upright will be conducted in this manner for 3 separate scenarios listed below.

- Brake Mounting Force
- Cornering Force
- Bump Loading

Testing data produced by the MCR at Pembrey was evaluated using an analysis program called Pi ToolBox. The program gathers sensory information from the vehicle and relays data back to the program for the user to analyze. And a relevant spreadsheet was produced in which loads can be calculated.

Each scenario mentioned can be analytically derived and are shown in the subsequent chapters.

#### 4.1.1 Cornering Force

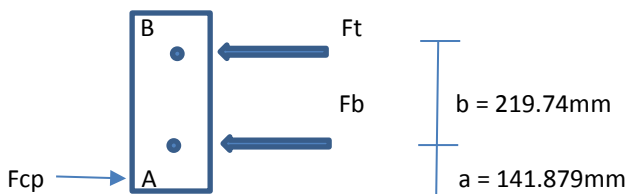
Before modelling on Solidworks it was necessary to calculate the forces exerted on an upright. Uprights are under the most stress during cornering as and overturning moment causes lateral weight transfer placing the assembly under compressive and tensional forces. After reviewing the track data, it was decided that 2.5 G would be a sufficient over estimation in which to calculate the forces produced. Calculations also assumed 100% Weight transfer so only the outside wheel is producing cornering force. And can be shown in the free body diagram below as Equation 4.1 and Equation 4.2

Vehicle mass : 798 Kg

Weight Split : 41 / 59

Maximum Cornering : 2.5 G

$$F = ma = (0.41 \times 798) \times (2.5 \times 9.81) \quad F_{cp} = 8024.1 \text{ N}$$



Taking Moments Around A

$$219.74 \times F_t + 141.879 \times F_{cp}$$

$$F_t = -141.879 \times F_{cp} / 219.74$$

$$F_t = -5180.9 \text{ N}$$

Negative highlights mount is in tension

*Equation 4.1 Top mount Tension*

Taking Moments Around A

$$219.74 \times F_b + 361.619 \times F_{cp}$$

$$F_b = 361.619 \times F_{cp} / 219.74$$

$$F_b = 13205 \text{ N}$$

Highlighting mount is in Compression

*Equation 4.2 Lower mount Compression*

#### 4.1.2 Braking Torque

Another high stress period is under braking as torsional forces produced from the caliper gripping the brake disk in conjunction with the longitudinal force produced by the decrease in speed can lead to catastrophic failures such as Sebastien Buemi dramatic crash during the 2010 Chinese Grand Prix practice.

This calculations also required taken the worst case scenario into consideration, hence the longitudinal weight transfer under braking is 70/30. The deceleration is over estimated as 2.5G. The calculation also require the tire diameter (r) in order to calculate the torque (T) required from the brakes. This is shown in Equation 4.3

Force applied due to De Acceleration

$$F = ma$$

$$= (0.7 \times 798) \times (9.81) \times 2.5 = 13699.7 \text{ N}$$

Torque produced at axle

$$T = F \times r$$

$$= 13699.7 \times 0.257 = 3520.8 \text{ N}$$

Therefore 1760.4 N at each front wheel axel

*Equation 4.3 Braking torque*

### 4.1.3 Bump Loading

Bump loading occurs when the wheel encounters a rough road surface or bump, this induces the wheel to leave the ground, as touching back down the whole corner mass including sprung and unsprung mass act upon and the suspension. This is a very load intensive cycle and happens repeatedly and can be a major factor in contributing to fatigue. The calculation below overestimates at 3g to give a factor of safety.

$$\begin{aligned} F &= ma \\ &= ((\text{Sprung corner mass} + \text{Unsprung mass}) \times 9.81) \times 3 \\ &= ((141 + 22.5) \times 9.81) \times 3 \\ &= 4811.805 \text{ N} \end{aligned}$$

*Equation 4.4 Bump Loading*

### 4.2 Existing Design

The design first took shape by drawing the center bore in which the hub lays within. Next with reference to front plane an angle of 5.8 degrees was created representing the kingpin inclination and measuring to the top and bottom mount.

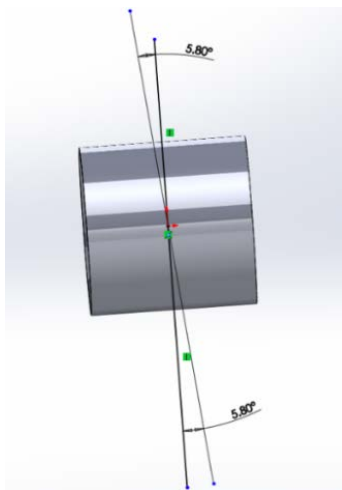


*Figure 4.2 Kingpin inclination*

Next was to extrude the upper and lower pillar joining the mounts the bore. After the pillars, the welded caliper plate was designed. As explained in the design specification the design must incorporate the Wilwood caliper and the specification was acquired via the website measuring the mounting points as 10mm and the distance between them 135mm. Finally the steering arm was extruded and placed into position. The final product resembles the existing upright with a close degree of accuracy. However instead of the weldment beads function being used to represent the welds, fillets of varying sizes were used to indicate the welds on joins. This was due to the Finite Element Analysis not being able to represent welds during the analysis. The existing model can be seen Figure 4.2.

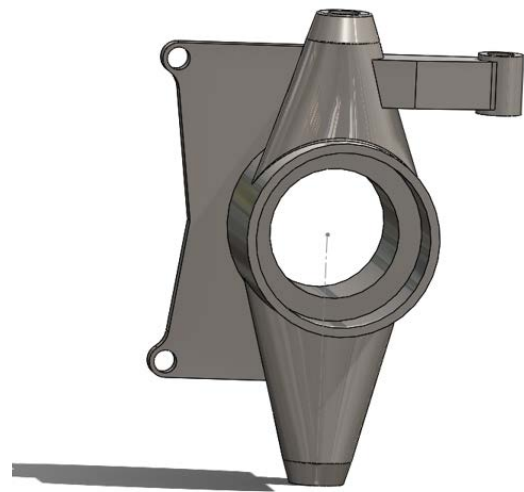


*Figure 4.3 Previous design*

The Steel 4130 properties was then added to the design through material selection and previous design was then prepared for simulation using Finite element analysis. This is discussed further in Chapter 5.4 Finite Element Analysis.

### 4.3 Topology

Topology defines load paths with a designated package space. The package space is assessed with previously discussed loads, then TruForm carves away material that is not heavily strained. The amount of space removed can be adjusted to find the most efficient design in terms of stiffness to weight ratio.

Figure 4.4 shows the designated package space that includes the centre bore upper and lower mounting points based on the kingpin inclination.

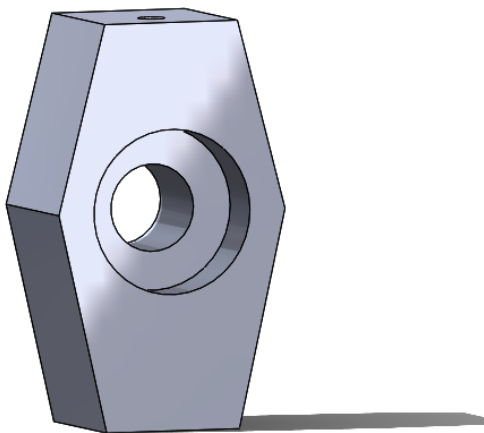


Figure 4.4 Package Space

The determined loads and constraints are applied to the design space these are discussed later within the Finite Element Analysis section. Within true form 40% of the design space was chosen. The results are as shown below.

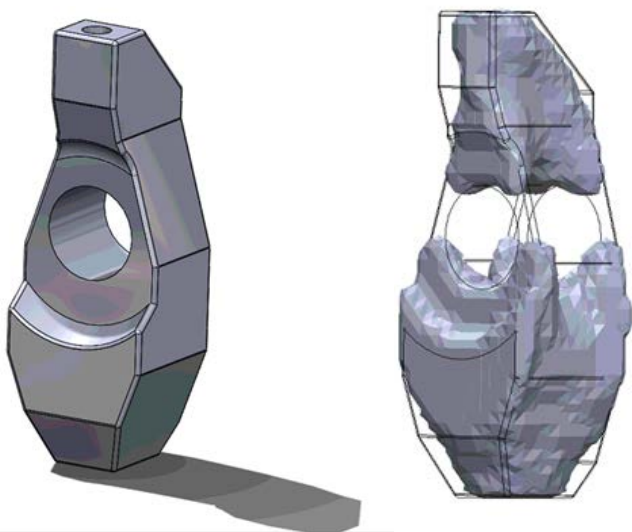


Figure 4.5 TruForm results Cornering Loading

Figure 4.5 shows the true form based from the cornering loads instead of bump loading as it gives a more honest depiction of the stress mainly attributed with the upright. The Truform however shows a weird result as the edges of the centre bore seem not an integral part of the design. It was deduced that either Truform was asked to remove too much material or a more likely explanation was the bearing constraint effecting the results. This was easily rectified as the material was left when manually cut extruding the design space.

Figure 4.6 below shows the Truform produced brake calliper mounting as discussed in Chapter 4.2 the dimensions for the Wilwood calliper remained the same. However it now became a design choice to which side the mount attached to in front of the upright or behind. Research suggest that caliper location has no effect on wheel loading. Having the callipers mass lower does have a minute effect as it lowers the COG however this could expose the break line and it considered not worth the risk. There is an effect on bearing loads though, as the disk tries to carry the disk upwards if trailing and downwards if the caliper is leading although these forces are contained within the closed loop of the assembly.

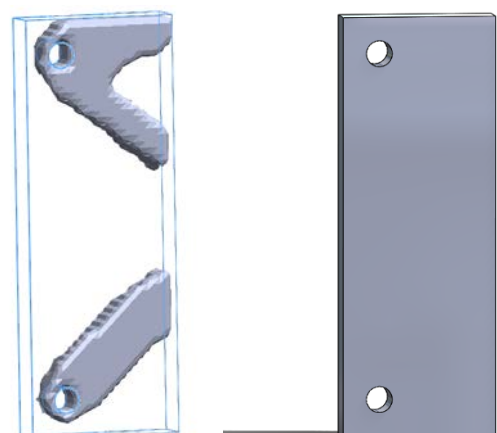


Figure 4.6 TruForm results Braking Mount

Tie rod point was then finally designed. As previously discussed the tie rod point will be located the opposite side of the brake caliper. The design differs from the previous design as dual bolt holes allow for the double sheer within the system. The design accommodates a SABP12S male tie rod end. The dimensions of this can be found in the appendices. The final design can be seen below in Figure 4.7.

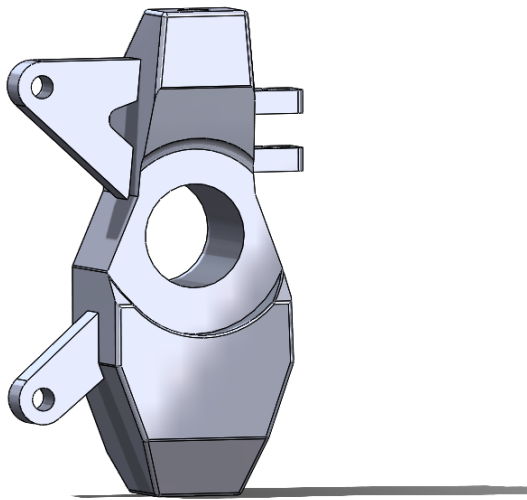


Figure 4.7 Optimised Design

#### 4.4 Finite Element Analysis

*"Efficiently optimizes and validate each design step using fast-solving, CAD integrated Solidworks Simulation to ensure quality, performance, and safety."*(Solidworks, n.d.)

The programme uses displacement formulas to calculate component displacement, strains and stresses under internal and external loads. This is calculated via the 1<sup>st</sup> and 2<sup>nd</sup> order tetrahedral meshing capabilities.

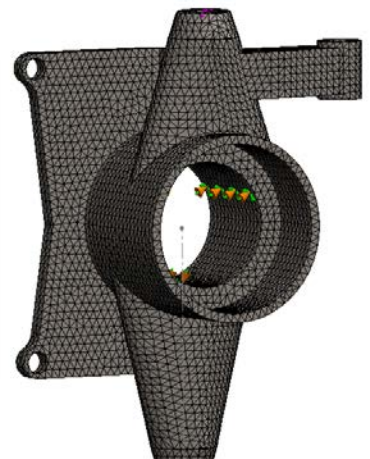
##### 4.4.1 Mesh

Meshing splits continuous mathematical models into finite elements. The type of elements created by this process depends on the type of geometry meshed. Solidworks offer 3 types of 3D elements: solid elements for meshing solid geometries, shell elements for meshing surface geometry and beam elements for meshing wire frame. (Kurowski, 2015)

The mesh must be as homogenous as possible. This would include minimising the difference in aspect ratio between elements, as well as element mapping quality. Ensuring that each mesh is identical for the different scenarios.

Below in Figures 4.8 the respective mesh applied to each model.

| Mesh Details                                  |                     |
|-----------------------------------------------|---------------------|
| Study name                                    | Static 41 (Default) |
| Mesh type                                     | Solid Mesh          |
| Meshes Used                                   | Standard mesh       |
| Automatic Transition                          | Off                 |
| Include Mesh Auto Loops                       | Off                 |
| Jacobian points                               | 4 points            |
| Element size                                  | 3.67483 mm          |
| Tolerance                                     | 0.163231 mm         |
| Mesh quality                                  | High                |
| Total nodes                                   | 50066               |
| Total elements                                | 58695               |
| Maximum Aspect Ratio                          | 79.014              |
| Percentage of elements with Aspect Ratio < 3  | 99.1                |
| Percentage of elements with Aspect Ratio > 10 | 0.114               |
| % of distorted elements (Jacobian)            | 0                   |
| Time to complete mesh(hh:mm:ss)               | 00:00:07            |
| Computer name                                 | S1429               |



| Mesh Details                                  |                     |
|-----------------------------------------------|---------------------|
| Study name                                    | Conering (-Default) |
| Mesh type                                     | Solid Mesh          |
| Meshes Used                                   | Standard mesh       |
| Automatic Transition                          | Off                 |
| Include Mesh Auto Loops                       | Off                 |
| Jacobian points                               | 4 points            |
| Element size                                  | 3.6 mm              |
| Tolerance                                     | 0.18 mm             |
| Mesh quality                                  | High                |
| Total nodes                                   | 127279              |
| Total elements                                | 86595               |
| Maximum Aspect Ratio                          | 12.583              |
| Percentage of elements with Aspect Ratio < 3  | 99.4                |
| Percentage of elements with Aspect Ratio > 10 | 0.00808             |
| % of distorted elements (Jacobian)            | 0                   |
| Time to complete mesh(hh:mm:ss)               | 00:00:07            |
| Computer name                                 | S1224               |

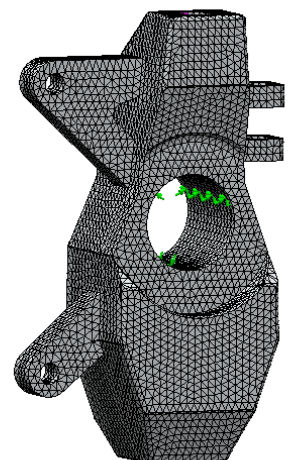


Figure 4.8 Existing and Optimised Design mesh details

For each design a fine mesh was chosen to increase the accuracy of the results given, with the previous design having an element size of 3.6mm and a total of 58695 elements with the optimised design having an element size of 3.6mm and a total of 86595 elements. With both above 99% of aspect ratio.

#### 4.4.2 Constraints

The Finite Element Analysis validity depends on the constraints placed upon the model. Each design was constrained with the goal of representing real world conditions. The toe link is constrained laterally whilst the top mount is constrained by the a-arms and prevent motion in the lateral and longitudinal directions, therefore these constraints were applied. Whilst the bottom mount was constrained in all 3 translational directions due to the pushrod suspension. To represent the constraints of the center bore in contact with the hub a cylindrical constraints in the vertical and longitudinal direction allowing for slip. These constraints are highlighted in green below in Figures 4.9. Although during the corner scenario the constraints could not be applied to the mounts as loading is acting upon the area

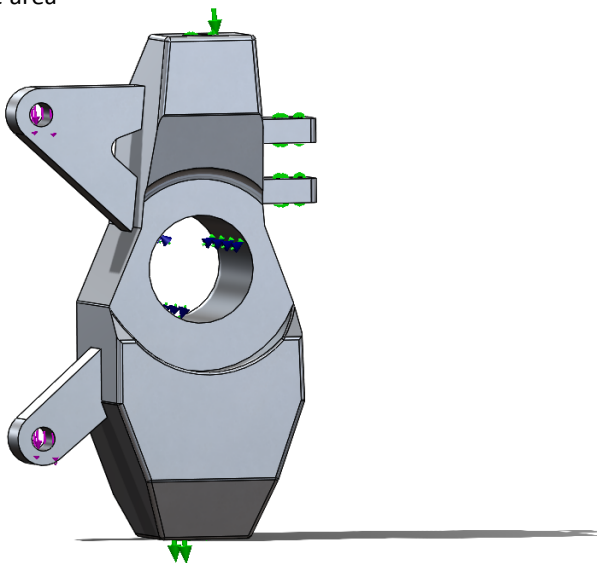


Figure 4.9 Constraints

#### 4.4.3 Loads

As discussed in the analytical procedures three load cases were analysed. Lateral compressive and tensional forces acting in shear upon the lower and upper mounts respectfully whilst cornering on one wheel at 2.5G. Torsional forces from the axle acting in shear upon the caliper bolts interacting with the bolt holes acting downwards as the mount is leading when under 2.5 G longitudinal force. Which can be seen in Figure 4.9. Finally, bump loading at 3G's acting from the contact patch and interacting with the upright through the bearing contact. These can be seen in purple in the Figures below.

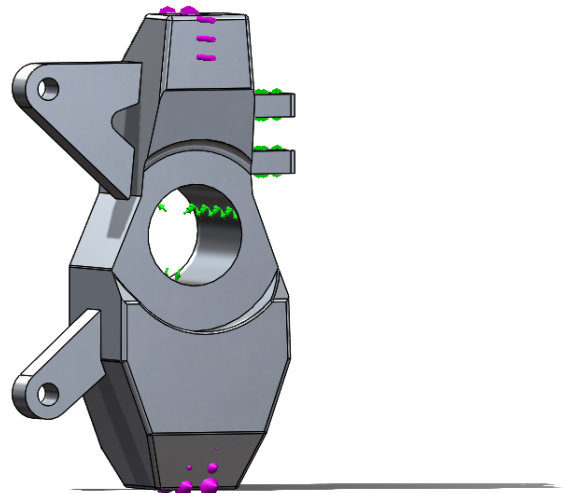


Figure 4.10 Lateral Loading

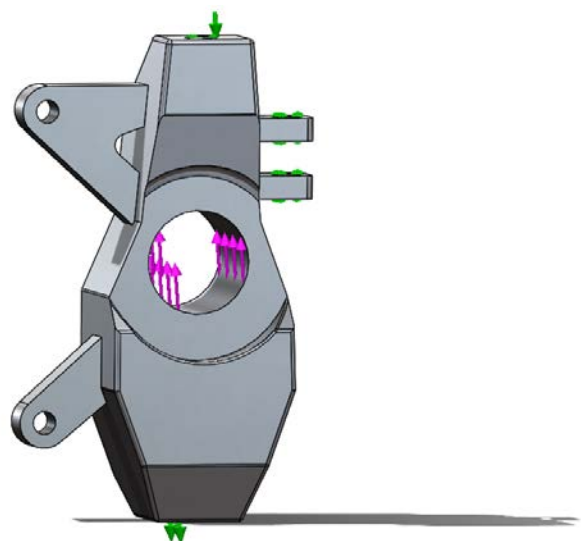


Figure 4.11 Bump Loading

## 5.0 Analysis

This chapter addresses the results produced from the methodology. Both the existing and prototype are evaluated and compared in the parameters explained within the design specification. The main parameter in which the analysis was based on is deflection as this leads to offset camber which can hinder performance. Other objectives to consider is stiffness to weight ratio. This can be achieved by viewing the material properties and noting the Von Mises stress values, which shows the material yield strength and areas which are likely to deform, these values are the basis for the factor of safety feature within Solidworks, and is most likely a stress concentration in which fatigue is likely to happen.

### 5.1 Material Properties

The material properties show that the existing design weighs approximately 3113.61g or 3.11 Kg. In comparison the optimised design weighs 644.18g or 0.64Kg, indicating a decrease of 2.466 Kg a 79.29% reduction in mass thus lowering the inertia of the sprung mass which will benefit handling, steering response and bump oscillation. These masses can be found in the appendices.

### 5.2 Cornering Force

The previous design was constrained and loaded as described in the methodology. The previous design being made from 4130 Steel has a yield strength of 415 Mpa the largest recorded stress was 243.1 Mpa. However this stress was located where welds lay and as the weldments were not applied to the models the real life counterpart would be robust under this loading. This is highlighted in Figure 5.1

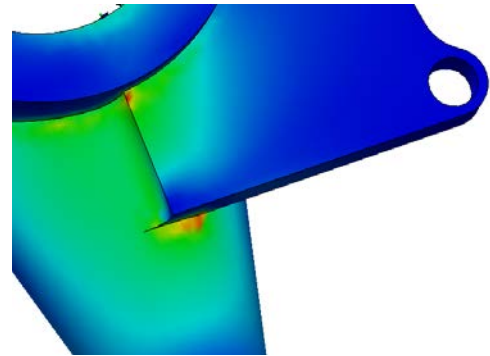


Figure 5.1 Max stress Cornering Force Existing Design

The high stress areas are indicated in red and can be seen on the lower mount pillar. The model other than this anomaly ranges from between 60.79 Mpa to 182.4 Mpa located on the lower pillar peaking under the centre bore again this has weldments on the upright and would experience a lesser stress under the compressive load. The tensional peak stress can be found on the caliper mount experiencing 141.8 Mpa again this was strengthened with welds. The analysis suggests that the model is within the elastic region and would retain its shape after loading. The full model can be found in the appendices.

The design has a factor of safety rating of 1.9. This can be seen in Figure 5.2 below.

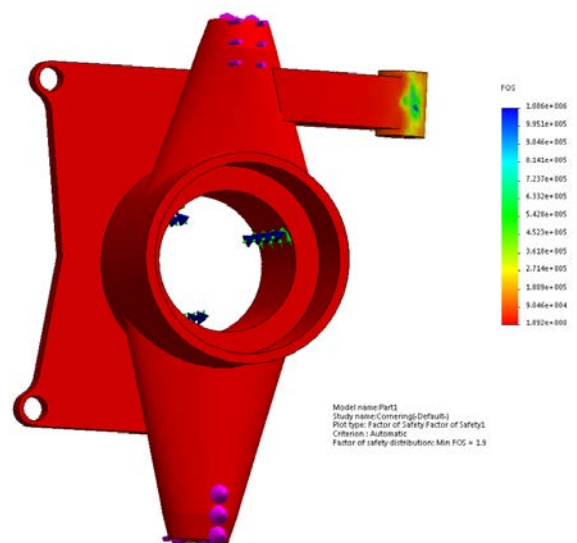


Figure 5.2 Factor of safety Cornering Force Existing Design

The stiffness of the upright was indicated by the model deflection rate under lateral loading. The existing model exhibited a maximum of 0.11mm deflection on the lower pillar due to the compressive force acting upon it. The deflection rate gradually petered out at 0.012mm approximately half way up the pillar deflecting the bottom of the mount bracket also. The tensional force produced a deflection rate of 0.027mm around the mounting point. These results can be seen in Figure 5.6 below.

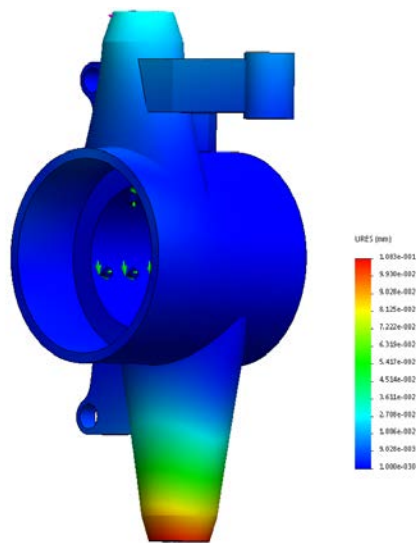


Figure 5.3 Deflection rate Cornering Force Existing Design

The optimised design was constrained and loaded as described in the methodology. The design was constructed from Aluminium 6061, T6 which has a yield strength of 275 Mpa the largest recorded stress was 76.19 Mpa located at the bottom edge of the center bore and can be seen in Figure 5.3 below.

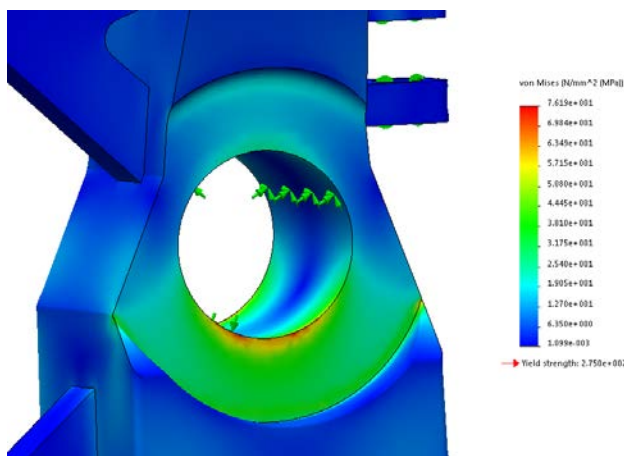


Figure 5.4 Max stress Cornering Force Optimised Design

The lower pillar under the compressive load shows stresses ranging from 0.001 Mpa to 50.8Mpa peaking under the centre bore. The tensional forces acting on the upper pillar range from 0.001 Mpa to 20.5 Mpa. The analysis suggests that the model is within the elastic region and would retain its shape after loading. The full model can be found in the appendices.

Producing a factor of safety ratio of 3.6 which is significantly higher than the original design.

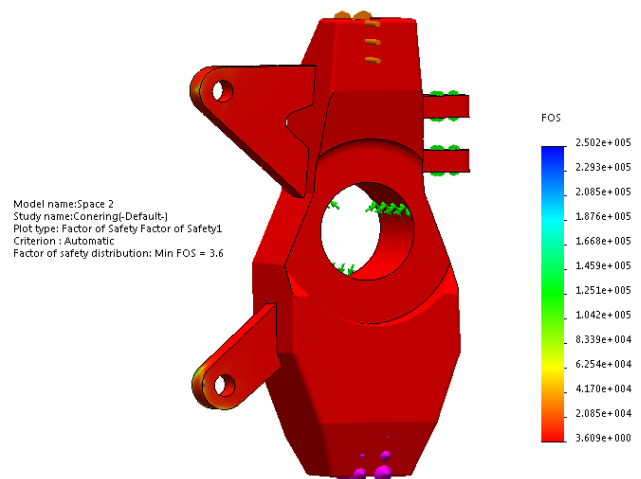


Figure 5.5 Factor of safety Cornering Force Optimised Design

The optimised design exhibited a maximum of 0.0745mm deflection on the lower pillar due to the compressive force acting upon it. This is 0.0365mm less than the existing design and can be said to be stiffer resulting less of a degree of camber change. The deflection rate gradually dispersed out to 0.003mm approximately half way up the pillar also deflecting the bottom of the mount bracket by 0.05mm. The tensional force produced a deflection rate of 0.037mm around the mounting point. These results can be seen in Figure 5.6 below.

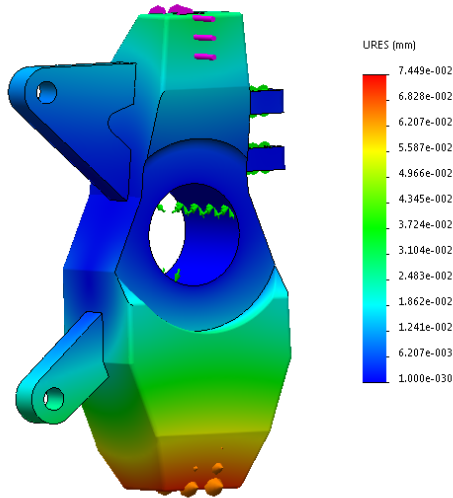


Figure 5.6 Deflection rate Cornering Force Optimised Design

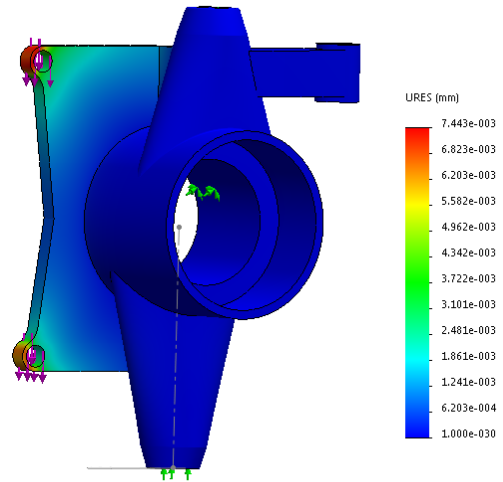


Figure 5.8 Deflection rate Braking Torque Existing Design

### 5.3 Braking Torque

The results proved as expected here with peak a force of 49.1 Mpa produced around the bolt hole as a results of the shear force produced from the bolt. Highlighted in Figure 5.7.

The factor of safety check confirms the belief that the mounting bracket of the previous design is over engineered with a safety factor of 9.4. With the results shown below in Figure 5.9

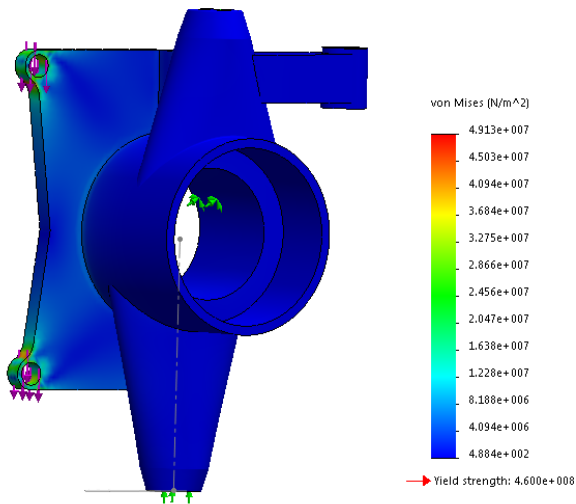


Figure 5.7 Max Stress Braking Torque Existing Design

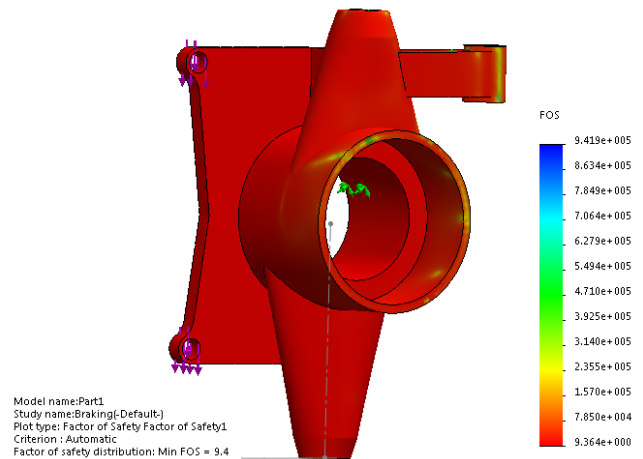


Figure 5.9 Factor of Safety Braking Torque Existing Design

Deflection was again located as expected with peak reaching a high of 0.007mm which is very minimal and could hint at the design being considerably over engineered. This is shown in Figure 5.8

Stress results for the optimised design were intriguing although the highest stress was 22.8 Mpa it was isolated where the bottom caliper mounting bracket. This in its self is not a surprise but stress concentrations such as this can easily lead to fatigue quicker. This can be seen in Figure 5.10

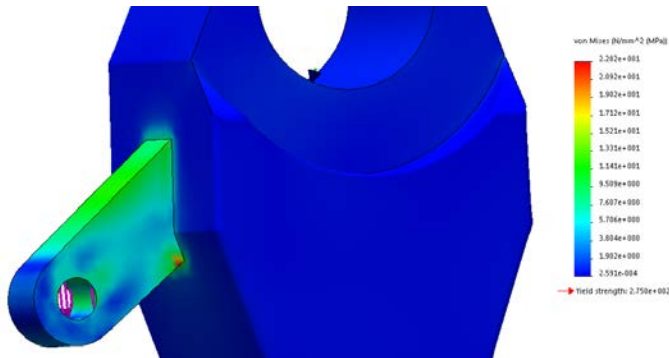


Figure 5.10 Max Stress Braking Torque Optimised Design

Deflection was at its greatest again at the bottom bracket 0.018mm which is a valid indication that the analysis is replicating real world situations as the inertia for the rotor disk would be greater due to moments the further away from the axels centre point. Such deflection rates indicate the optimised mount being over engineered maybe even less space could be taken from the design space. However the design would have to take into consideration the stress concentration. Seen below in Figure 5.11

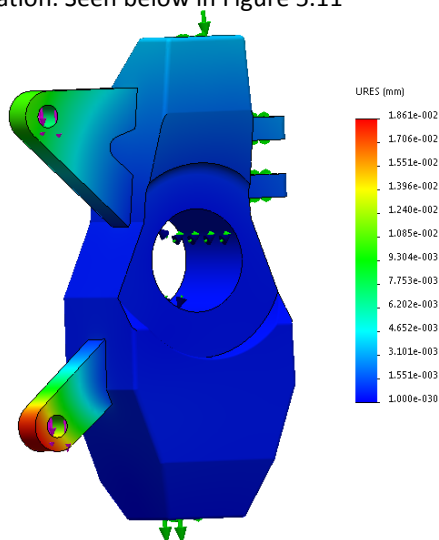


Figure 5.11 Deflection rate Braking Torque Optimised Design

The factor of safety check could describe the mounting bracket as over engineered with a rating of 12 and is shown below in Figure 5.12

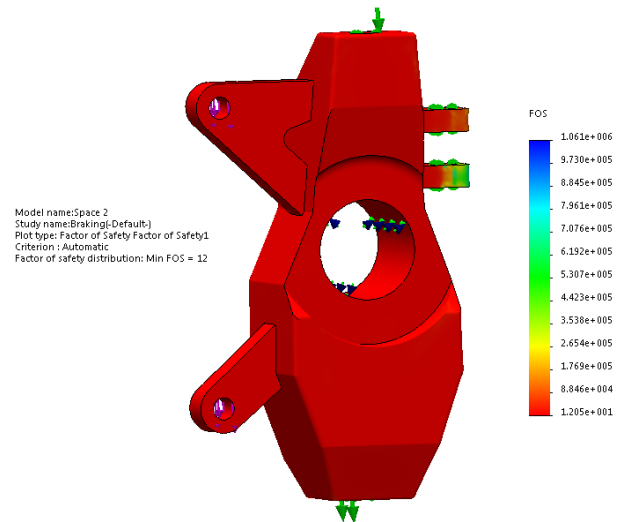


Figure 5.12 Factor of Safety Braking Torque Optimised Design

#### 5.4 Bump Loading

The final scenario analysed was bump loading. Max stress here indicated that both the upper and lower mounts would bare he brunt of the force with the lower mount exposed to highs of 13.1 Mpa and stress concentrations located within the bolt hole. As seen below in Figure 5.13

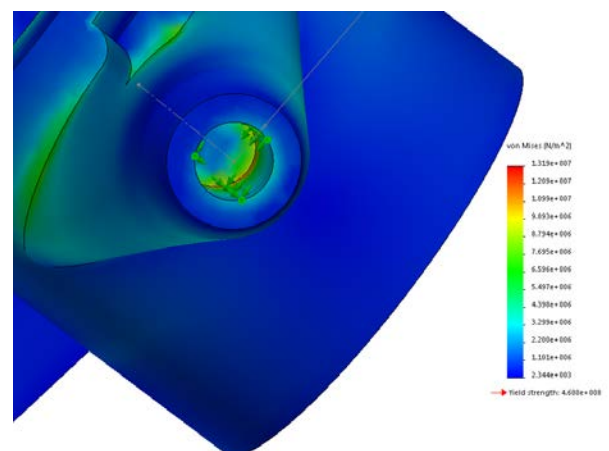


Figure 5.13 Max stress Bump Loading Existing Design

The deflection rate seems to congregate on the outer edges of the bore mainly around the excess material protruding from the front of the hub, where the maximum deflection is measured at 0.002mm. Seen below in Figure 5.14.

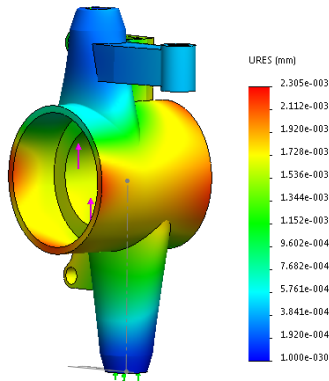


Figure 5.14 Deflection rate Bump Loading Existing Design

With such a low deflection rate again the previous design shows that it is significantly over engineered, with a factor of safety of 35 as seen below in Figure 5.15.

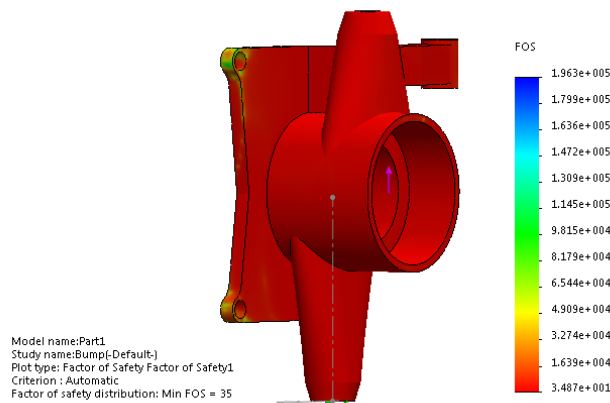


Figure 5.15 Factor of Safety Bump Loading Existing Design

Maximum stress for optimised design yielded similar results inducing a maximum of 11.3 Mpa on the bottom mount. Which is to be expected as the full weight of the corner is loaded where equal and opposite forces from the contact patch highlighted in Figure 5.16.

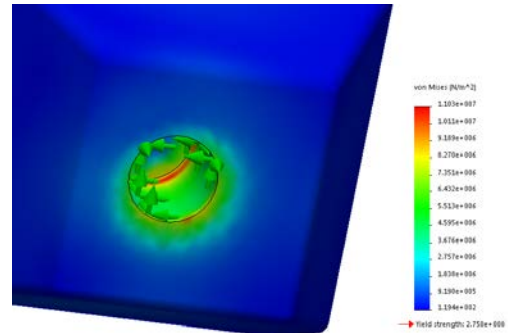


Figure 5.16 Max Stress Bump Loading Optimised Design

The optimized design shows that it is significantly less over engineered during the bump scenario with a peak deflection of 0.004mm and a factor for safety of 25. Highlighted below in Figures 5.17 and 5.18.

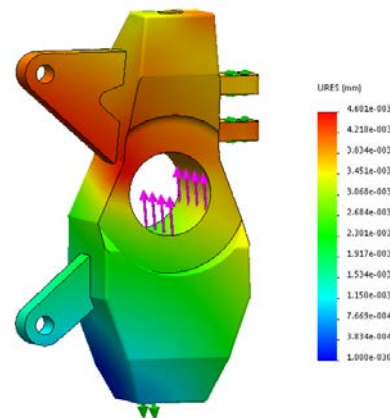


Figure 5.17 Deflection rate Bump Loading Optimised Design

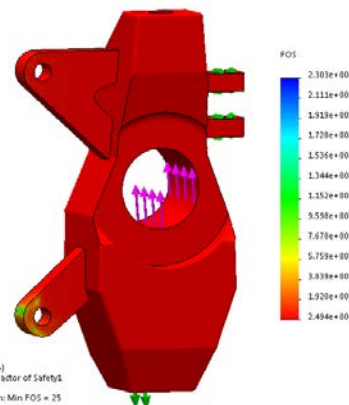


Figure 5.18 Factor of Safety Bump Loading Optimised Design

## 5.5 Fatigue

Modal analysis was completed on the optimised design to determine how the system behaves in its dynamic displacement response. This indicates how different frequencies naturally excite the system to a degree where resonance fluctuates through the component resulting in a dropping of life expectancy. For this study no external loads is applied to the component as it does not affect its natural frequencies. The component is on the other hand restrained.

Suspension components are under constant non uniform loading varying drastically with altered terrain. To simplify loading a fully reversed loading with a stress ratio ( $R$ ) = -1. Solidworks the derived a SN curve based on the elastic modulus for 601 T6, highlighted below in Figure 5.19.

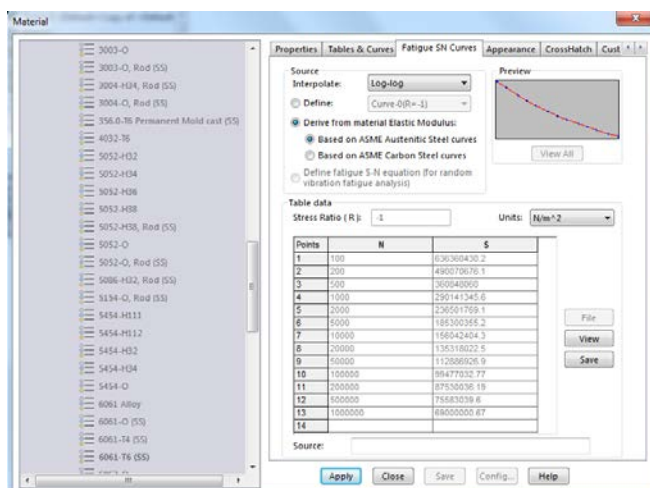


Figure 5.19 Solidworks 6061 T6 Defined SN curve

The fatigue analysis could be done for the individual scenarios however for a combined force situation it seems logical to optimise for all the forces. However as constraints and loads can't be applied to the same location this was not possible and renders the fatigue analysis validity in question. The results for cornering loads were still completed and can be shown in Figure 5.20.

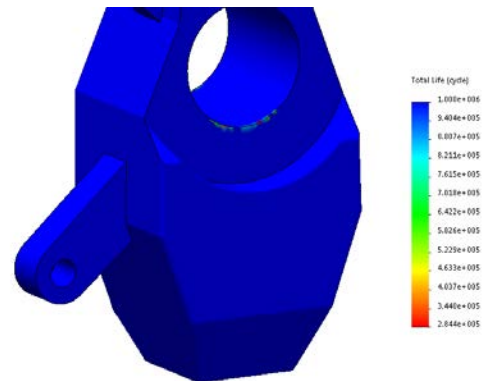


Figure 5.20 Fatigue Analysis

It can be seen that the optimised design is well defined. The Area that shows sign of reduced life is where the upright is attached to the spindle where failure is predicted after 284397 cycles. This only predicts fatigue due to cornering and the overall life of this design would be considerably less.

## 6.0 Conclusion

The main objective within this dissertation was to optimise the existing design in weight reduction and stiffness. The FEA results proved the design was highly successful decreasing the front unsprung weight by 2.46Kg with slightly increased stiffness. Although these results have not been replicated outside the simulation process and to this point are not validated. The motivation for this dissertation was to see a design come to fruition. This was unfortunately not achieved mainly due to the delay of the project due to unforeseen circumstances, on the other hand skills were still acquired in the preceding steps of design before the manufacturing.

Further work to enhance this dissertation would initially begin with re defining the torque loading experienced when breaking this would include measuring the distance from the brake pedal to the caliper piston so line pressure can be calculated, this would then allow clamping forces to be calculated which in turn would be used to define the torque of the brake pad on the effective area of the brake disk.

Other assembly parts such as the hub, caliper and disk brake could also be modelled. This would allow for remote loading from the contact patch. Where the results collected could be compared this time of loading scenario would also change the typology results and allow for a more valid fatigue study to be defined.

Creating an engineering drawing and a CNC program in which the model can be manufactured. This would result in tool diameters and clearances being researched this would then give a manufacturing cost which can be added to the material costs.

Finally physical testing could be done on the Denison universal testing machine of the prototype to validate the Solidworks model.

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