MEE30003

Machine design report

Formula SAE: A-Arms

Syed Hassan Ali 2098571 ___% ________________________

Nathan Cook 7664958 ___% ________________________

Leonard Healy 7668030 ___% ________________________

Abstract

In 2016 Team Swinburne will again be competing in the Australasian leg of the FSAE (Formula Society of Automotive Engineers) competition. This is a competition where students build and race a Formula style racecar against other universities.

Our group was posed with the problem of designing, developing and optimising the a-arms (control arms) for the vehicle. Before the start of the design period a Brainstorming session took place to put out general ideas on what the design should look like. Designs had to be able to withstand theoretical forces calculated from the theoretical information received from previous years. Designs were created using SOLIDWORKS® and verified using ANSYS®. During the design process multiple verification stages took place such as an 80% and 100% design freeze during which other students, alumni and industry professionals were consulted.

Components were analysed under full competition load a number of times and the data received from this was used to refine weight and strength under load to optimize them to a set factor of safety.

While designs are completed actual results can only be received outside of simulation on a race track and by design judges, although at this point all set objectives have been achieved.

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Introduction

The following report is an in-depth design review that has been completed on the A-arm components of a Formula SAE race car. This report details how the A-arms interact with the different systems of the race car. This report was written to record the theory used, discoveries made and the processes that were taken by the SAE team for the design and creation of the A-arms so that this information could be imparted to those looking to design A-arms or those who seek a greater understanding of what the A-arms do and how they interact with the rest of the systems. The following will include an in depth literature review covering all relevant theory related to the A-arms and its surrounding systems. A close analysis of the design process to give an understanding of what considerations needed to be made during the creation of the design. A showcase of any adjustments made to the design over the course of the creation process. A look at creative solutions to problems posed by the constraints given and improvements made on previous years designs. Extensive calculations of suspension forces acting on the system to aid in the creation of the most effective designs. Extensive market research on designs created by other groups and how these can aid in improving our own design. Responses to reviews of our designs explaining how suggestions will or will not be implemented and why this is the case, documentation of our designs with drawings to AS1100 standard which will allow the reader to recreate the design if he so chooses, and reflection on the design and the steps taken to reach the final design stating any different approach that might have been more effective for the design. After reading this report the reader should have gained an understanding of how the theory is implemented in the design process of an a-arm and should be capable of reproducing the same or a similar design to the one showcased in this report.

Literature Review

Suspension System

Role:

Over the years different types of suspension systems have been used in race cars, the one used in the SAE car is the double wishbone independent suspension system. This system has two wishbone shaped arms which connect the wheels to the chassis.

The designer can arrange this design in any way he wishes such as equal or unequal, parallel or nonparallel. The suspension system has many functions but the most basic of these is to connect the wheels to the chassis. Other functions include providing the driver with a smooth driving experience, a good handling of the vehicle and provide good control of the steering system while rounding a corner. An important reason for using the double wishbone suspension system over the other available options is to provide the car with four-wheel independence. Four-wheel independence is wanted since if an upset is experienced by a certain corner of the car, only the wheel and the tire at this specific corner will be affected by this upset. The suspension system needs to be able to withstand any changes in toe. If this is not possible the change in toe should be minimised as much as possible. Another requirement for the suspension system is for it to provide isolation from high frequency vibrations from tire excitation. A very important feature is also to have as little compliance in the suspension system as possible (preferably none at all) therefore a material with a high modulus of elasticity should be used to prevent as much bending as possible.

A-arms (control arm)

Role:

The A-arms (or control arms) are a part of the suspension system of the sae race car that have the simple but important task of connecting the uprights to the chassis. There are a total of four sets of A-arms on the FSAE race car: Two sets at the front and two sets at the rear. These as well have two different arms an upper and a lower arm. The lower arm is connected to the suspension spring and damper (shock absorber) and therefore all forces experienced due to this have to be taken into consideration during the calculations for the design process. The a-arms are connected to the chassis and uprights through rod ends and spherical bearings, both of these have different sizes and the most suitable one has to be selected to avoid load failure during the maximum load cases. When designing an A-arm it has to be kept in mind that as little compliance as possible is wanted, preferably none at all, and therefore the material used has to have a high stiffness to prevent any excessive deformation from taking place.

Rod ends & spherical bearings:

Rod ends are spherical bearings that consist of an eye-shaped head with integral shanks forming housings and standard spherical plain bearings, or a spherical plain bearing inner rings, or spherical plain bearing inner ring and a sliding layer between the bore of the head and the inner ring. They may have either a male or female thread for mounting, which may be right or left handed threaded. There are different sizes of rod ends and spherical bearings which have different properties, all of the bearings looked at are from the Aurora bearing company.

Threaded rod ends are spherical bearings that have a threaded shaft attached. They are not designed to be used in a lateral loading case since this tends to cause the threaded shank to bend which applies a shear load across the threaded roots. They are designed to be loaded in compression or tension. The eye tends to get distorted under bending loads causing the spherical element to seize.

For rod ends we have looked at the ¼”.

The ¼” rod ends (AM-4) has a bore diameter of 0.25 inch, a ball width of 0.375 inch, a head diameter of 0.75 inch and an ultimate static load of 5260 lb.

Spherical Plain Bearings are spherical bearings that have an inner ring with a spherical convex outside surface and an outer ring with a correspondingly spherical but concave inside surface. Due to their design they are particularly suitable for bearing arrangements where alignment movements between shaft and housing have to be accommodated, or where oscillating or recurrent tilting movements must be permitted at relatively slow sliding speeds.

The two sizes of spherical bearings we’ve been looking at are ¼’’ and 5/16’’

The ¼” spherical (COM-4) has a ball bore diameter of 0.25 inch, a ball width of 0.3430 inch, an outside diameter of 0.6525 inch, a misalignment angle of 13.5 degree and an ultimate static load of 4950 lb.

The 5/16” spherical (COM-5) has a ball bore diameter of 0.3125 inch, a ball width of 0.3750 inch, an outside diameter of 0.75 inch, a misalignment angle of 12 degree and an ultimate static load of 6475 lb.

Circlips & stakings:

To secure a spherical bearing in its housing, multiple options of fasteners are given such as circlips and stakings. To understand which of these options would be preferable to the design a thorough look needs to be taken at both fastening methods to see which would give the most advantageous result to the final product. Factors such as weight, ease of use and installation and costs have to be taken into consideration when making the choice on which of these two options to use.

Circlips also known as c-clips are a type of fastener that fits into a machined groove on a part to prevent lateral movement but at the same time still permit rotation. For a clip to function correctly it has to be installed into a pre cut groove with the shoulder of the circlip protruding from the groove which retains the component.

Staking is a process that connects two components by creating an interference fit between the pieces. When using this process to secure a spherical in a housing a thin piece of sheet metal is fit in around the component to create the interference.

When comparing which of these two fastening methods would be best for securing a spherical in its housing the circlips would be better. When comparing the two it is obvious since the circlips are easy to install. They are lightweight which is a big benefit when as little mass as possible is wanted such as in this case while the use of staking is relatively expensive, with the tool to do this costing around $400. As demonstrated from ts_14 staking is relatively difficult to do and very time consuming and if done incorrectly failure will occur under high loads.

Uprights

Role:

The role of the uprights is to connect all the parts of the suspension to the wheel. When designing the uprights a high amount of stiffness is wanted due to the high loads experienced during use which will cause unwanted plastic deformation. The uprights are an important part to consider when designing the a-arms since the mounting points at which the a-arms will connect to the uprights are pre-determined. The other important reason for basing the design of the a-arms on the position of the mounting points is that all the forces that will pass through the a-arms will pass through the uprights first.

Suspension Geometry[LW2]

Role:

The suspension geometry was pre-set by the team when starting this project by using data collected from previous year’s race cars. As previously mentioned the suspension system on this car is a double wishbone suspension system and therefore an understanding of how all the forces throughout this system interact with each other and more specifically how these forces affect the a-arms. The double wishbone suspension system is a setup where upsets in one wheel will not have any effects on the other wheels. The tires must also have enough vertical wheel movement as to not affect toe.

Bump & Droop

Bump and droop are some important factors when looking at suspension geometry that will impact the a-arms and must therefore be taken into consideration in the design process. Bump is the upward movement of the wheel and droop is the downward movement of the wheel. Paying attention to these factors is necessary as if not accounted for the car can bottom out and become unstable.

Springs & dampers

The springs and dampers while not all too important to the system must still be considered since an understanding of the loads which pass through them to the a-arms must be taken into consideration. The role of the spring and damper is to give the driver a more comfortable and controllable ride by reducing the forces received from the road, and to keep the contact patch from the weels constant to provide the maximum traction available.

Mass

When designing the sae car it is requirement for the car to have as small a mass as possible to increase the acceleration of the car as much as possible. This can be seen by looking at the rule F=ma with F being force, m being mass and a being acceleration. If F is considered to be a constant it can be seen that any decrease in m will cause an increase in a and vice versa. This proves that all parts of the car should be as light as possible to get the maximum amount of acceleration possible.

Sprung & unsprung mass

Sprung mass

Sprung mass is the mass that is on the chassis side of the suspension. This means that as weight transfer occurs, it is supported and therefore is better for vehicle dynamics and handling.

Unsprung mass

Unsprung mass is the mass that is connected to the bottom of the suspension. It is made up of the wheels, tires, hubs, hub carriers, and brakes plus approximately fifty percent of the mass of the suspension links, drive shafts and springs and shocks. The unsprung mass is what the shock absorbers have to control in a bump situation to keep the tires in contact with the ground. The less unsprung mass there is the better since it will amount to more control of the vehicle.

Suspension forces

One of the most important steps before starting to design the A-Arms is to determine the suspension forces that will act on them. This will give an idea of what dimensions and materials will be used produce the most effective A-Arms for the situation. The forces will be applied in analysis software such as Solidworks and Ansys to determine if there are any improvements that would need to be made on the design. Specifications given to us at the start of the project such as track widths, wheel base and centre of gravity which were all decided on when the suspension geometry was designed, will give further boundaries for the design of the A-Arms.

To determine the forces placed on each wheel during different scenarios we need to calculate the longitudinal and lateral load transfer. Longitudinal load transfer determines the load transfer on each wheel which can be used to determine the suspension forces on each corner for specific load cases. The latitudinal load transfer looks at acceleration and deceleration with the load being transferred to the front and to the rear respectively.

Formula for the longitudinal load transfer

Longitudinal load transfer(kg)= Longitudinal Acceleration (g) Weight (kg)centre of gravity height (m)Wheelbase (m)

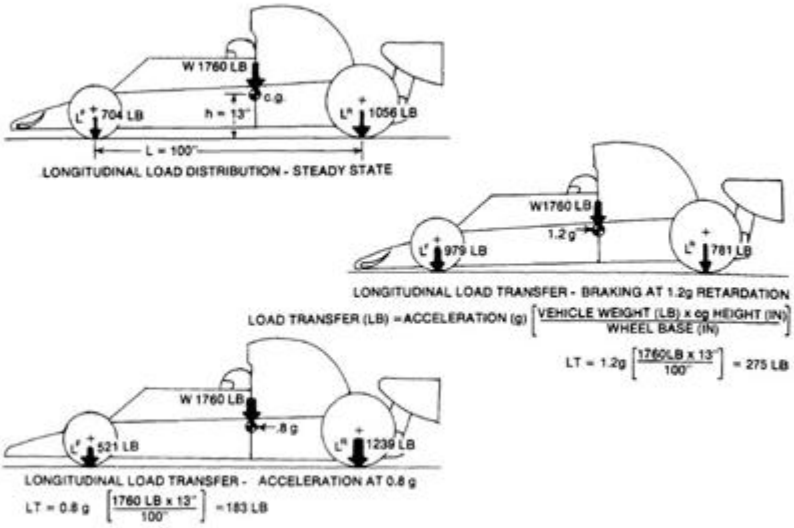


Figure (15): Logitudinal load distribution and transfer due to linear acceleration.

Formula for the lateral load transfer

Lateral load transfer(kg)=Lateral Acceleration (g) Weight (kg)centre of gravity height (m)Track width (m)

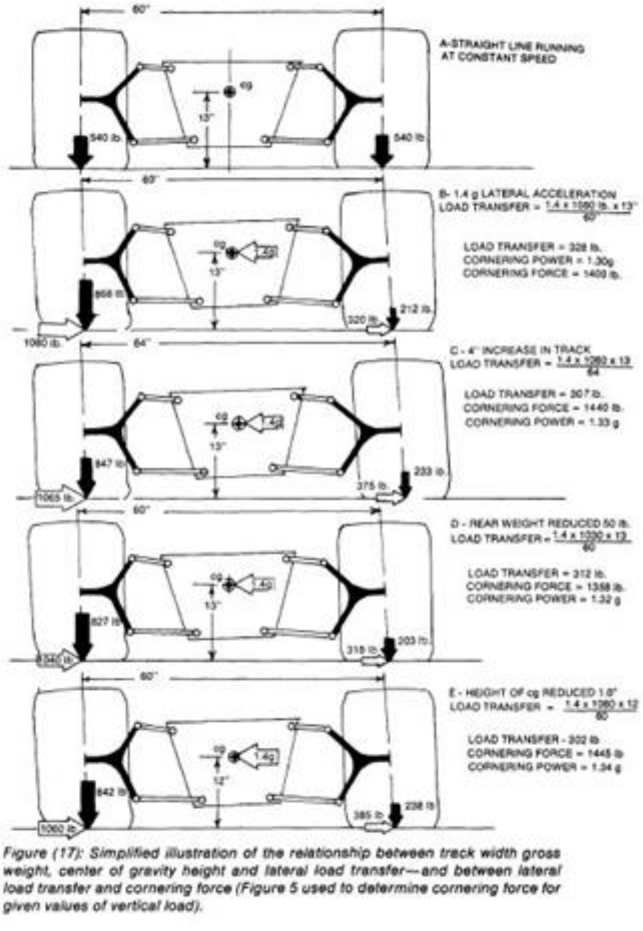


Figure (17): Simplified illustration of the relationship between track width gross weight, center of gravity height and lateral load transfer—and between lateral load transfer and cornering force (Figure 5 used to determine cornering force for given values of vertical load).

After determining the longitudinal and lateral load transfer for each load scenario, determining the forces on the contact patch will be possible. The equation used to determine these forces at the maximum potential acceleration g force is:

Force at contact patch (N) = Acceleration Force (g) Load on rear wheels (kg)

This equation can also be used to determine the forces on the contact patch during the other load scenarios by replacing the maximum potential acceleration g force with the maximum possible braking or cornering g forces acting on the contact patch:

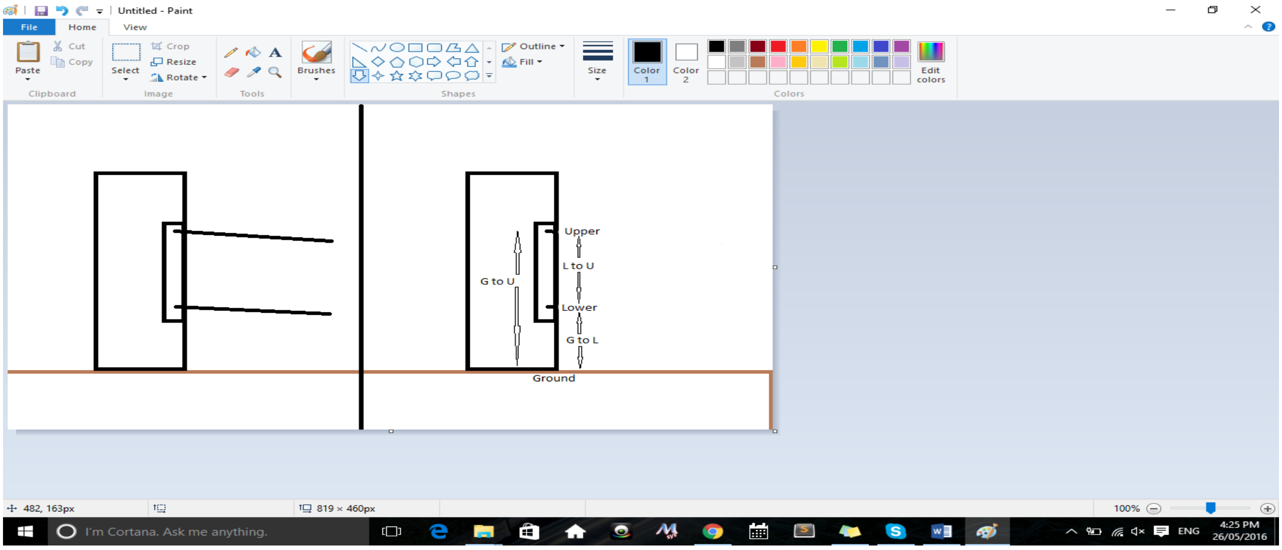
Force at contact patch (N) = Braking Force (g) Load on front wheels (kg)

Force at contact patch (N) = Cornering Force (g) Load on rear wheels (kg)

Knowing these forces allows us to determine the forces acting on the upper and lower parts of the front and rear uprights. With these forces we can calculate all the forces acting through each of the a-arms which then allows us to determine the forces on the chassis. We can determine the forces acting on the Uprights through these equations:

Force at Lower Upright Point (N) = Force at contact patch (N) Lg to uL g to l

Force at Lower Upright Point (N) = Force at contact patch (N) Ll to gL l to u



After determining these forces it is just a simple matter to calculate the forces passing through the A-Arms by using trigonometry and in turn the forces on the chassis can be determined by using these forces.

Tire

Role:

The tires are the only part of the car that are in contact with the ground. This means that almost all forces acting on the car come from the tires which therefore are a really important part when determining the forces for designing the A-Arms.

Camber

The camber angle is the measure in degrees of the wheels vertical and perpendicular alignment. Camber can be described in two different ways: negative or positive. Negative camber occurs when the top of the wheel’s tilt towards the chassis while positive camber occur when the top of the wheels point away from the chassis. Each type of camber has its advantages with the negative camber being the most popular option due to its better grip during cornering, however the disadvantage is that during acceleration on a straight road the contact between the tire and the road will be reduced. Zero camber will give more even tire wear and a greater contact patch on the straights, but it will reduce the performance during cornering. In the end, the best camber depends on the driver's style and the conditions the vehicle is being exposed to.

Toe in & toe out

Toe is the angle of the tires pointing inward or outward when looking from a top down view. There are two types of toe, toe in and toe out. Toe in occurs when the front edges of tires point towards each other while toe out occurs when the front edges of tires point away from each other, both having their advantages and disadvantages. Toe in allows both wheels to constantly generate force against each other which decreases turning ability but will increase the vehicle's ability to move in a straight line. Toe out will increase the cornering ability of the car but will decrease straight line mobility.

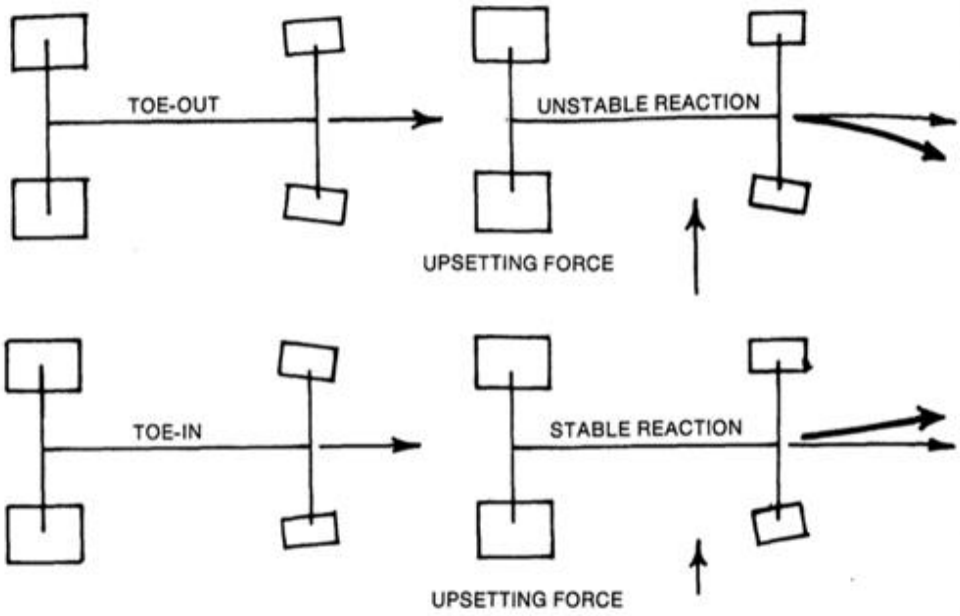
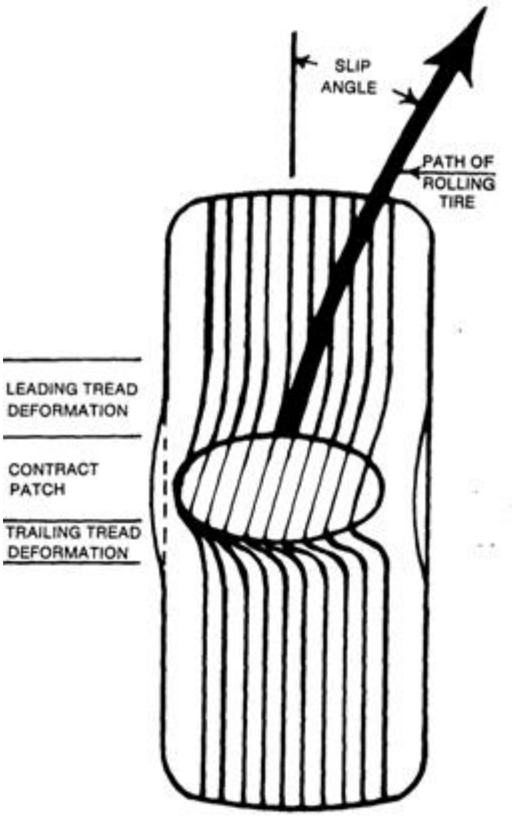


Figure (31): Effects of toe-in and toe-out on directional stability in response to upsets.

Contact patch

The contact patch is the area of the racing tire that is in contact with the ground at all times. This means that any of the forces acting on the tire and any of the forces of the tire applies to the ground pass through the contact patch. The greater the area of the contact patch the greater the load that can be transferred through the patch.

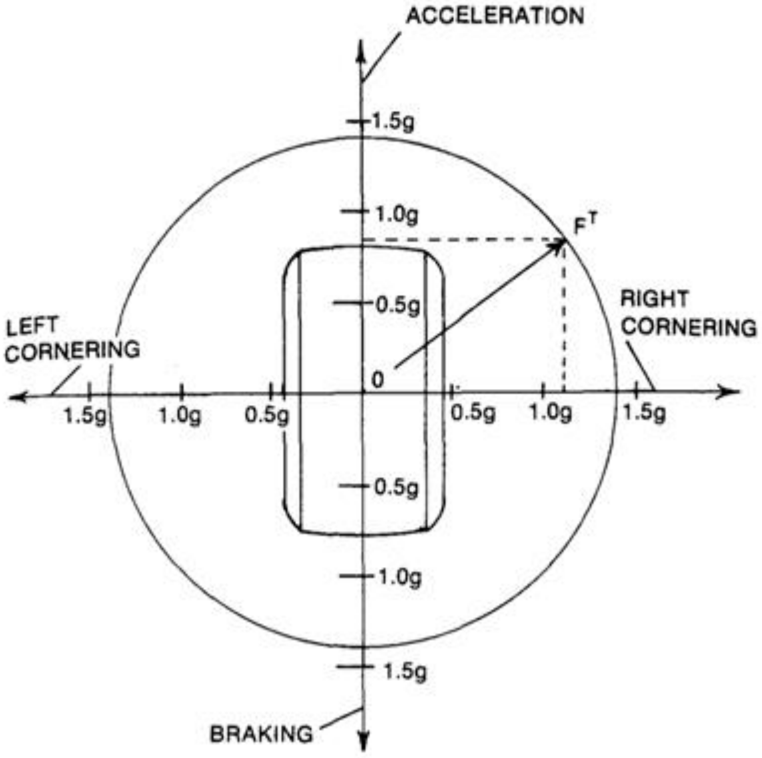


Normal Load

The normal load is the amount of force applied to an individual tire perpendicular to the road surface (Tune to win). It is the instantaneous sum of that portion of the vehicle's total weight and the aerodynamic down force, which is acting on the individual tire at any given moment (Tune to win).

Traction circle

The traction circle is used to simulate how a tire will react under combinations of lateral and longitudinal forces. The traction circle is a very important element during the calculation of the suspension forces as it will give an indication of how much load is being transferred in any given scenario such as accelerating, braking, turning, turning while accelerating and turning while breaking. The traction circle also tells us that maximum turning can’t take place during maximum acceleration, therefore it can be used to determine what the ideal accelerating/turning and braking/turning ratio is.



Materials

Choosing the correct material when designing a product is vital since choosing the incorrect materials may lead to a catastrophic result. When choosing a material factors such as tensile strength, yield strength and young's modulus have to be taken into consideration because if one of these factors is not sufficient the material can't be used for the intended design. Materials selected for the a-arms should be as stiff as possible since no plastic deformation and as little deflection as possible is wanted.

Aluminium and its alloys

Aluminium and its alloys are characterized by a relatively low density (2.7g/cm^3 as compared to 7.9g/cm^3 for steel), high electrical and thermal conductivity and a resistance to corrosion in some common environments. The mechanical strength of aluminium may be enhanced by cold working and by alloying. However both processes tend to diminish resistance to corrosion. The chief limitation of aluminium is its low melting temperature (660°) which restricts the maximum temperature it can be used. Aluminium is regarded as a low weight metal with high tensile and yield strengths commonly used in the aviation and automotive industry. The cost of aluminium whilst higher than plain carbon steel is much lower than titanium and comparable exotic materials. (Calister Page 374-375)

(Table from calister Page 376)

Aluminium Number	Condition (temper Designation)	Tensile Strength (MPa)	Yield Strength (MPa)
1100	Annealed	90	35
3003	Annealed	110	40

5052	Strain Hardened (H32)	230	195
2024	Heat treated (T4)	470	325
6061	Heat treated (T4)	240	145
7075	Heat treated (T6)	570	505
295.0	Heat Treated (T4)	221	110
356.0	Heat Treated (T6)	228	164
2090	Heat Treated Cold worked (T83)	455	455
8090	Heat Treated, Cold worked (T651)	465	360

Carbon Fibre

Carbon is a high performance fibre material that is the most commonly used reinforcement in advanced (ie non fiberglass) polymer matrix composites. The reasons for this are as follows:

1. Carbon fibres have the highest specific modulus and specific strengths of all reinforcing fibre materials
2. They retain their high tensile modulus and high strength at elevated temperatures; high temperature oxidation, however may be a problem.
3. At room temperature, carbon fibres are not affected by moisture or a wide variety of solvents, acid and basics.
4. These fibres exhibit a diversity of physical and mechanical characteristics, allowing composited incorporation these fibres to have specific engineered properties
5. Fibre and composite manufacturing processes have been developed that are relatively inexpensive and cost effective (Calister Page 598)

Carbon-reinforced polymer composites are currently being utilized extensively in sports and recreation equipment (fishing rods, golf clubs), filament-wound rocket motor cases, pressure vessels and aircraft structures. (Calister Page 599)

(Table from calister Page 600)

Property	lass	arbon	evlar
Specific gravity	2.1	1.6	1.4
<u>Tensile Modulus</u>			
Longitudinal GPa	45	145	76
Transverse GPa	12	10	5.5

Tensile strength			
Longitudinal GPa	1020	1240	1380
Transverse GPa	40	41	30

Titanium

Titanium and its alloys are relatively new engineering materials that possess an extraordinary combination of properties, The pure metal has a relatively low density ($4.5g/C^3$), a high melting point (1668°C) and an elastic modulus of 107 GPA titanium alloys are extremely strong; room temperature tensile strengths are as high as 1400 MPa are attainable, the alloys are highly ductile and easily forged and machined. The major limitation of titanium is its chemical reactivity with other materials at elevated room temperatures. This property has necessitated the development of non conventional refining, melting and casting techniques; consequently titanium alloys are extremely expensive. In spite of high temperature reactivity the corrosion resistance of titanium at normal temperatures is unusually high they are virtually immune to air marine and a variety of industrial environments. Titanium is often used in aircraft structures space vehicles, surgical implants and in the petroleum and chemical industries, most famously titanium alloys were used in the construction of the sr71 blackbird where temperatures of the skin of the aircraft would reach over 500°C and whilst titanium can be seen as the ultimate material for many applications its high price must be factored in. (Calister Page 377)

(Table from calister Page 379)

Common Name	Condition	Tensile strength (MPa)	Yield Strength (MPa)
Unalloyed (R50500)	Annealed	484	414
Ti-5Al-2.5Sn (R54520)	Annealed	826	784
Ti-8Al-1Mo-1V (R54810)	Annealed (duplex)	950	890
Ti-6Al-4V (R56400)	Annealed	947	877
Ti-6Al-6V-2Sn	Annealed	1050	985
Ti-10V-2Fe-3Al	Solution + aging	1223	1150

Steel

Steels are iron-carbon alloys that may contain appreciable concentration of other alloying elements; there are thousands of alloys that have different composition and or heat treatments. The mechanical properties are sensitive to the content of carbon which is normally less than 1.0wt%. Some of the more common steels are classified according to carbon concentration-namely into low, medium and high carbon types. Subclasses also exist within each group according to the concentration of other alloying elements. Plain carbon steels contain only residual concentration of impurities other than carbon and a little manganese’s. Steel can be favourable due to its relative low cost and its ability to be welded easily on what can be seen as rudimentary equipment, unlike aluminium steel does not require a tig welder to be used and a mig or arc welder can be used with high effectiveness. (Calister Page 377)

(Table from calister Page 361)

Astm Number	Tensile strength	Yeild Strength
1010	325	180
1020	380	205
A36	400	220
A516	485	260
A440	435	290
A663	520	380
A656	655	552

Mild steel

Mild steel (AISI 1018) is a low carbon steel that has a density of 7.85 g/cm^3. It has a very balanced set of material properties making it a widely used material. Mild steel has a ultimate tensile strength of 440 Mpa and a yield strength of 370 Mpa. It is widely available since it is the most common form of steel.

Chromoly 4130

AISI 4130 is a steel alloy that contains chromium and molybdenum as strengthening agents. It can be easily welded since it has a low carbon content. It has a density of 7.85 g/cm^3 which is equal to other types of steel such as mild steel. Though when comparing other properties such as ultimate tensile strength (670 Mpa) and yield strength (435 Mpa) it exceeds other steels by far.

Manufacturing

3 axis machining

3 axis CNC milling machines are used to subtractively machine materials to dimensions specified by the operator through computer models. The cutting is done by employing a variety of cutting tools which can be exchanged depending on the type of cut that needs to be made. Accurate cuts are achieved by moving the rotating cutting tool along three linear axis of motion. Many of these machines are available in house at Swinburne and is therefore the best and cheapest option for manufacturing the spherical housing.

5 axis machining

5 axis machining is a very similar process to 3 axis machining, the difference being that it has an additional two axis' which allow for rotational movement. This allows for 5 sides of a workpiece to be machined without any direct interaction with the workpiece from the operator. There are two different types of 5 axis machining: 5 axis simultaneous machining and 3+2 axis machining. The difference between these two is that 5 axis simultaneous machining can move in all 5 axis' simultaneously which allows for more complex parts to be machined while 3+2 axis machining moves the part to a particular position and then clamping the rotary axis in place with hydraulic clamps, machining the workpiece in that position and then moving the piece to another position which continues until the process is complete.

Waterjet cutting

Water jet cutting is the process of using highly pressurised water or a mixture of water and an abrasive substance to shape materials into the desired form. It is usually used for manufacturing products from materials which cannot tolerate high temperatures. There are two main steps involved in the water cutting process

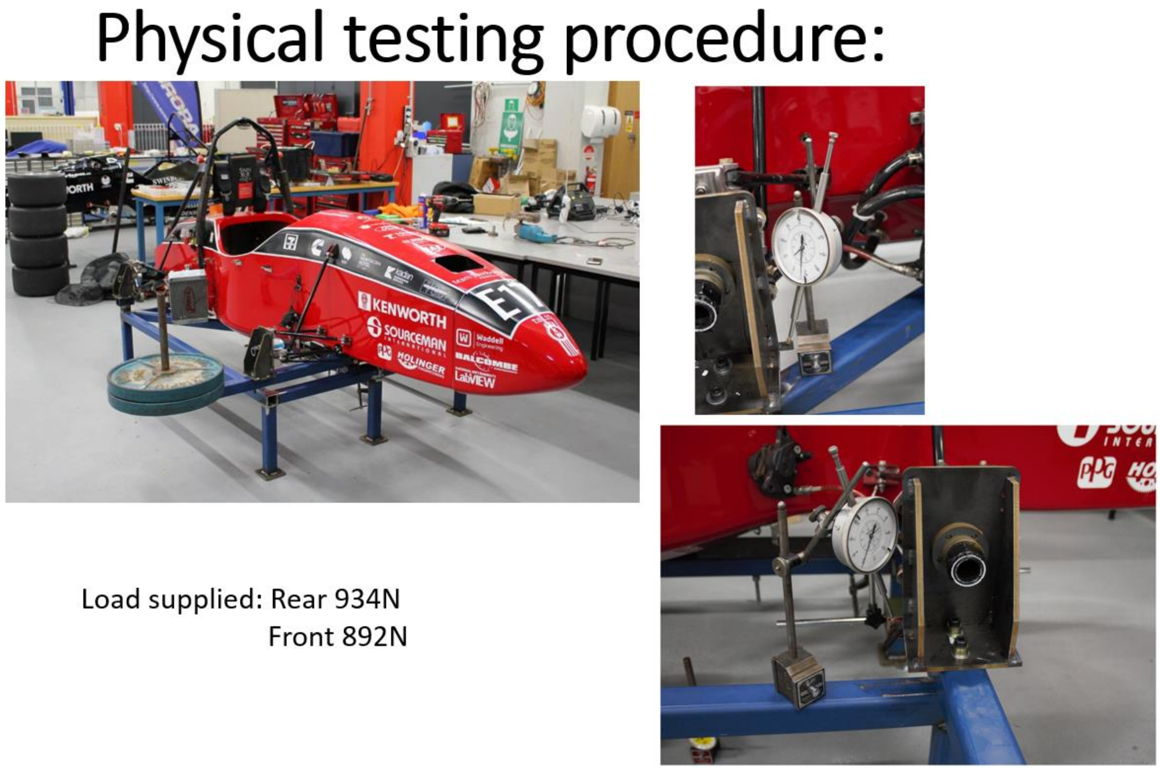
1. Water is pressurised by an electric servo pump to pressures above 50 kpsi to achieve the required amount of energy to make the cut.
2. An intense cutting stream is formed by focusing the water through a small orifice usually made out of gemstone. This stream can reach a velocity of up to 3 times the speed of sound depending on how the pressure is exerted.

Sandblasting

Sandblasting is a surface treatment process that involves using compressed air to blast fine sand or other abrasive material at a surface removing the surface impurities and leaving a clean surface. Sandblasting is similar to using sand paper with the advantages being that a more even finish is achieved and it is less time consuming.

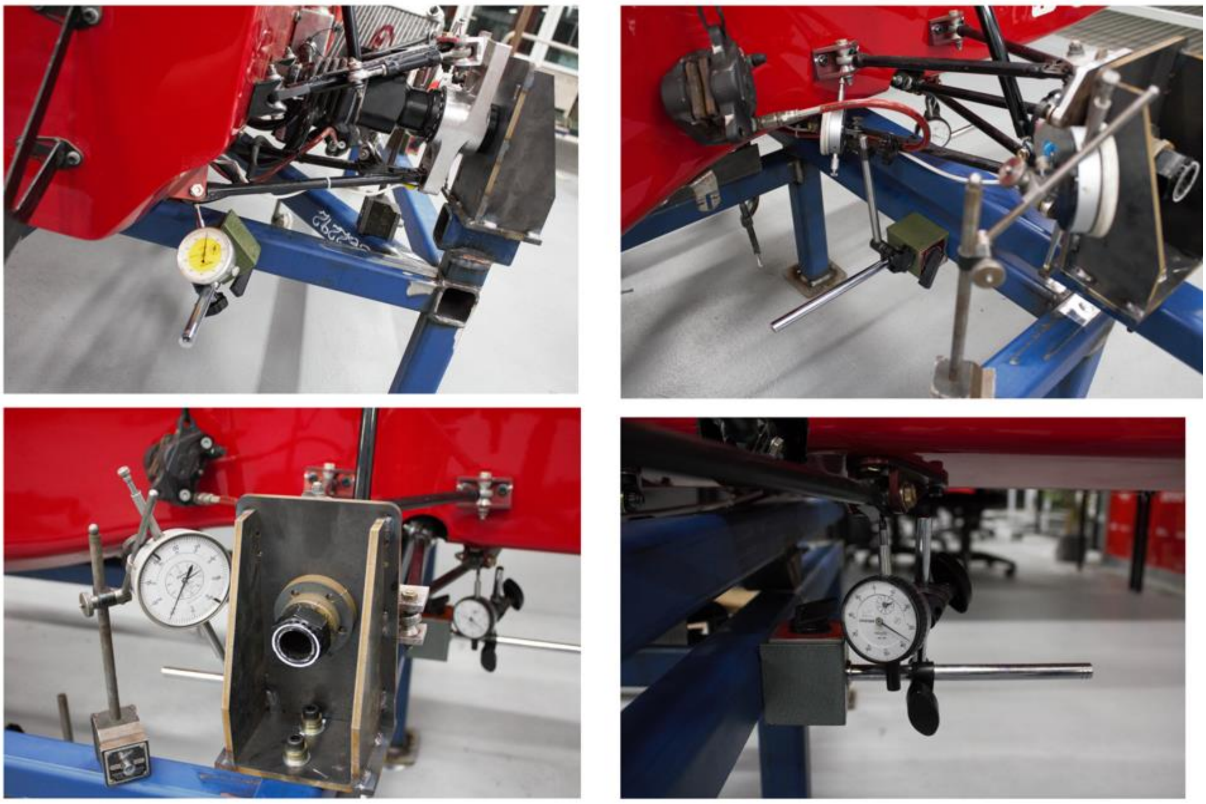
Gaps in material

Calculations are not always necessarily reliable. Similar materials from different companies may have different properties as well as different tolerances. In these cases it is usually better to perform physical tests on the components and manually calculate any required factors. A physical test was performed in order to provide us with the necessary information which we could not confirm.



Physical testing procedure:

Load supplied: Rear 934N
Front 892N



Physical test of the corner assembly measured from hub to hub with dial gages. Although not exactly accurate due to the unwanted deflection in the chassis, it still provided a resourceful comparison for the calculations and FEA's to the real world applications.

Design Approach/Strategy

Our group was given a design brief which contained a set of constraints which we had to abide by. The implementation of these constraints was vital since we were working as part of a large project, therefore if our design did not follow these constraints it would have prevented the part from integrating into the system.
Our approach to the problem was to split into 3 sections for which each person was responsible. These sections were:

Research: involved looking through theory, past projects and any other resources available to gain an understanding of what type of design would be the best to use for the A-Arms and of what type of materials this design would be made of.

Calculations: involved determining the forces which would impact the A-Arms. This would be used to adjust the design throughout the process to end up with the optimal design.

Design: involved the modelling and simulating of the part with the calculations and information received from the research and calculation sections to optimise the design.

When first starting the design process we began by brainstorming ideas of how we wanted the design to look and what type of materials we wanted it to be made of. Our initial idea was to bend the A-Arms since there were clearance issues with previous design which we planned to avoid with this idea. As for the initial material choices we considered the use of carbon fibre weave tubing due to its properties which would make it a suitable material to use. After our initial session of brainstorming we went off and started to work on our individual sections.

To understand how the A-Arms worked we started by researching various things such as the suspension system, the suspension geometry and any other parts of the overall system that might affect the A-Arms in any way. As can be seen by the literature review, this gave us a good understanding on how our part will interact with others in the suspension system and what type of forces would pass through the A-Arms.

After the conduction of the initial research, we took a closer look at the design that was implemented in last years ts_15 car. The suspension system implemented in ts_15 was the double wishbone suspension which is beneficial since it will prevent any upset experienced from a single wheel from affecting the other wheels as discussed in the literature review. After obtaining this information we chose to also implement this system into ts_16.

Our next task was to consult the suspension geometry provided by the team for this years car. This was a very important step since the suspension geometry will provide crucial details for designing the A-Arms such as the layout of the A-Arms, the expected bump and droop angles, the track width for both the front and rear and the wheelbase of the car. The location of the mounting point of the push rods, where the a-arms will connect to the uprights, and the chassis will also be provided by the suspension geometry.

After looking at the suspension geometry, we found that we did not have much freedom in designing the actual shape of the a-arms due to the constraints provided by this, and therefore had to scrap our initial idea of bending the A-Arms. Though this was a major reason for scrapping the initial idea it wasn't the only one other reasons such as additional manufacturing time and cost and the fact that bending the A-Arms introduces a bending moment which changes the forces in the A-Arms to non linear forces and adds an extra force in the arms. The rod ends we chose can not support a bending moment. All these factors contributed to discarding the idea.

Following this, discussion with team members who designed the A-Arms in the previous year took place to find out how they went about designing their A-Arms and what type of improvements they would make on their design. At this stage we created a very basic model in Solidworks which we gradually changed as we expanded on our knowledge, finished calculations and ran analysis' for the different load cases in Ansys.
After this, the calculation of the suspension forces took place using the information received from the suspension geometry and free body diagrams. With the completion of the calculation we gained a better understanding of the different load scenarios our design would be affected by.

At this point, modifications were made to the design to incorporate the data found through the calculations. From here we began looking into the material which we would use for our design. Initially we considered using a carbon fibre composite due to their high strength but decided against this since it is relatively difficult to work with. The next series of materials we considered were aluminium and its alloys due to the low density of the series but these were rejected since the other material properties such as the tensile strength was not high enough and the deflection was too great for the load cases that our part would experience. Finally we looked at different types of steel, most of these we deemed unsuitable for our purpose but then we looked at 4130 steel. The material properties of 4130 steel far exceed those of other steels such as mild steel as was covered in the literature review while still maintaining the same density. 4130 steel is also far easier to work with than some of the other materials we looked at. While 4130 steel has a higher density than the other choices, we felt that it would still be the best material choice for our design as its material properties would make up for this drawback.

For the dimension of the tubes we decided to go with a diameter of 12.7mm and a wall thickness of 0.889mm to start taking the design to the next stage of the design process.
The next stage of the design process was the selection of the bearings. All the bearings we looked at came from the Aurora bearing company. We chose the 5/16" spherical bearings over the ¼' spherical bearings for the connections to the uprights due to the 5/16" spherical bearings having a much higher static load bearing capacity then the ¼" spherical bearings which was discussed in the literature review. For the rod ends we selected the ¼" since these were able to handle the loads experienced at the chassis.
After deciding on the material, the tube dimensions and the bearings, we began designing the spherical housings that would be mounted on the uprights. Since we decided to use the 5/16" spherical bearings we had to design the spherical housings to be able to support this size of spherical bearing. To fasten the spherical bearings in their housing we looked at two different options. The advantages and disadvantages of circlips and stakings have been covered in the literature review. We decided to go with circlips since they are compact and relatively easy to install. Circlips were chosen in particularly for its ease of installation since when staking was used in a prior design difficulties were experienced when trying to push the staking into the housing. However the use of circlips adds some more constraints into the design such as groove diameter, groove width and shoulder length which were all covered in the literature review. After all these considerations were made we decided on the manufacturing process which was 3 axis machining. The major reason for this was that it is available in house at Swinburne.

When designing the A-Arms we ensured that they would be the first mode of failure on the car. This is because it is much better to bend an arm as they are cheaply made and can be replaced easily. You would much rather an A-Arm fail than a $5000 upright or a clevi mounted and moulded into the chassis to break. It is also a very visual mode of failure which means it can be detected easily just by looking at it, then actions can be taken to fix it without having to take apart a majority of the car.

At this point, we had reached a level at which we could start performing more in depth analysis using two types of simulation software; Solidworks and ANSYS to perform FEA's (Finite Element Analysis). We used solidworks for basic analysis to see if any obvious problems would occur before we started using ANSYS for more advanced analysis of all the possible situations that our design could be in when in use.

At this point we had reached 80% completion with our design. Our part only required minor tweaks to optimize it as much as possible. Soon after reaching this stage the 80% design freeze took place. This is an internal design review performed by the team to test our understanding of the design problem that we have received and to review our design to ensure no crucial aspect has been overlooked.

For the creation of our part, this was the best approach due to the fact that the problem has been looked at from every possible angle. With this approach we also received a lot of feedback from the other team members and other groups designing parts due to our frequent interactions with them during the initial research and design phase. This way we had the advantage of having no major setbacks since we were able to foresee any problems that might have occurred which we would have otherwise missed and thereby helped us to avoid any major time loss.

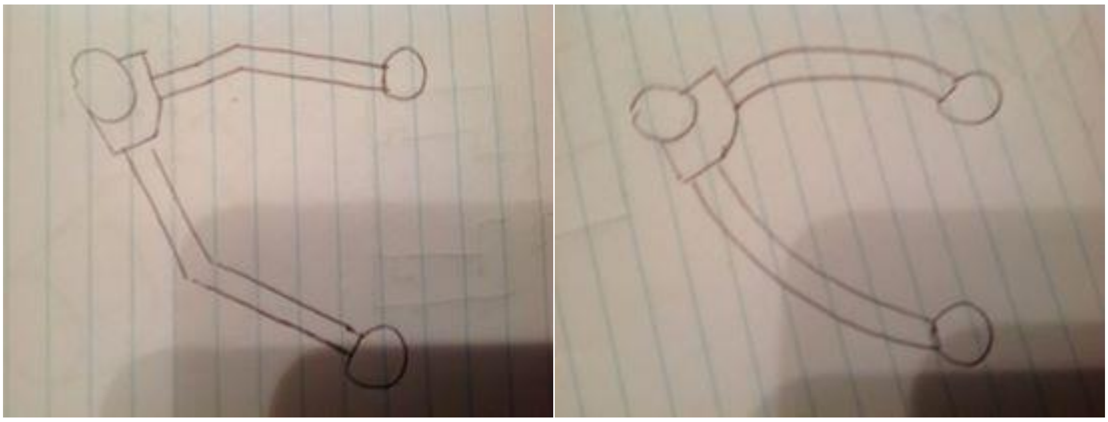
The decision to have each member work on their own section for the entire process helped us greatly, since each part of the process was able to be completed much more quickly and efficiently, rather than if we all had worked on the same section which might have caused some disagreements concerning certain aspects.

To ensure that each member of the group was on track with their assigned sections regular meetings were held once a week throughout the semester. This was done to get regular updates from each group member to see at what stage of their task they were in, who had fallen behind or who needed help with something in their task due to having fallen behind or needing a different perspective on their problem.
Regular dynamics section meetings were held during the semester where we could bring up any problems or challenges we faced with regard to our section. These meetings also ensured that the entire team was on track to completing their tasks on time.
Meetings involving the entire team took place each Monday night during which each section updated on the progress that was made during the week since the previous meeting. During these meetings any issues in regards to design process could be brought up to address what needed to happen to solve them. This kept the team working together as one and not just as individual sections that would come together and combine their completed projects at the end.

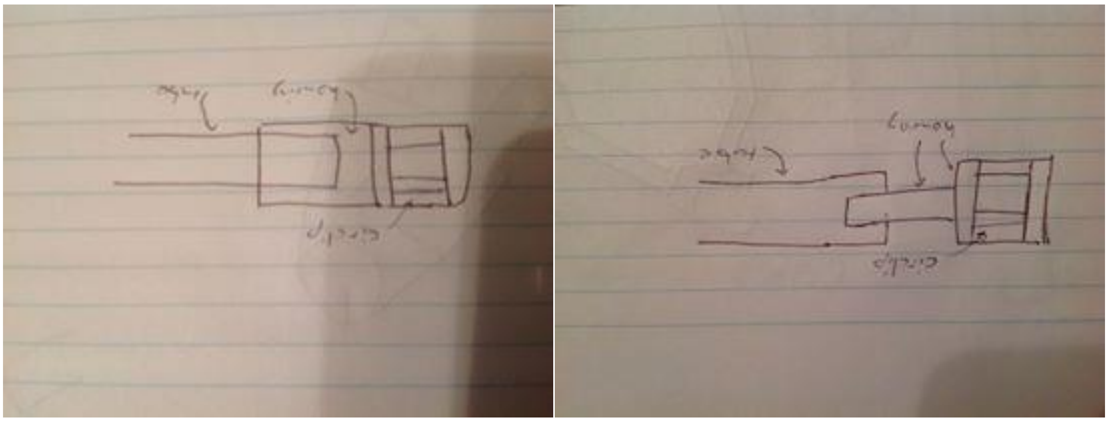
Went through many iterations. For the fronts there were 7 design iterations, for the rears there were 11 iterations. These iterations included:
-Changed rod ends to sphericals on rear top A-Arms to be more uniform and to support extra loading
-The chassis side sphericals were changed to be machined down so the weld could be in shear and so there was enough weld area to meet our forces from the weld calcs.
-Tube thicknesses were increased to provide extra strength where needed
-Lower spherical housing was broken up into 2 separate pieces to be welded together so one section was horizontal so it didn't interfere with the Uprights, also so the spherical had its full range of movement and would not interfere when in full bump or droop.
-Sections were cut out of the unstressed areas of the spherical housing to save weight.
-The spherical housings were changed so it had 1 flat surface so it didn't have to be re clamped when machining and so it could have the most accuracy possible.
-The bottom sphericals were changed to 5/16” to support the larger loads the bottom would be under.
-Self jigging area of the plate and damper tabs were added for much easier and more accurate assembly.
-Lower spherical housings were flipped upside down so the machined section of the housing could take the majority of the load rather than the circlip.
-Changed the design of the spherical housings to be longer and thinner so it would not interfere with the Uprights.
-Rear suspension point was changed multiple times. There was a flat plate with perpendicular tabs which was scrapped immediately due to it being weak. Small tabs were added underneath the suspension point and the tabs were in parallel. This was scrapped due to its hard assembly process and it didn't solve the problem. Then a sheet metal plate was designed so it could take the load but it was scrapped due to it being very hard the make and it being heavier than the final design. The final design was a flat plate with self jigging holes the had a rod underneath it which supported the applied load from the suspension and it also added extra support the the whole rear bottom A-Arm system. The damper tabs also jigged into the spherical housing itself which helped spread out the load and have easy assembly.

Creativity

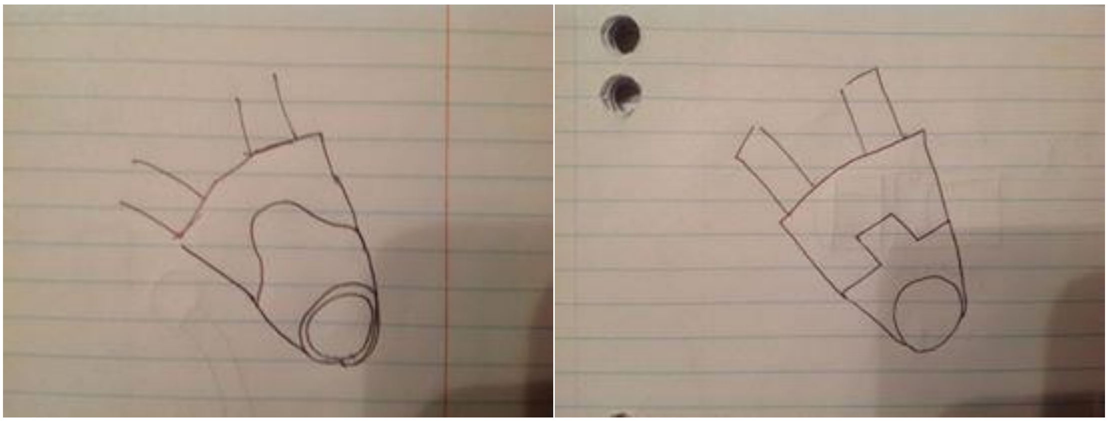
During the creation process a couple of problems arose which required a creative solution to solve these effectively.



These were the initial A-Arm designs for the rear lowers that we thought up. At the very beginning we knew that there was a huge clearance issue with that years rear lowers and we went about this idea to try and solve this issue. The one on the left is a cut and then angle welded tube while the one on the right is a bent tube aimed to go round the tripod housing. We realised this wouldn't be necessary seeing as the new suspension geometry for this year already covered that previous flaw.



Creating the connection between the spherical housings and the tubes was also quite troublesome because we initially didn't understand the reasoning behind the housing to tube connections. After presenting our first idea to the dynamics group, it became apparent that there must be enough space on the tube and housing for the weld. The figure on the left was our original design and since it inserts into the housing, the weld would be in compression and tension which is bad for the weld. Our final idea on the right is so that the weld can be in shear since the tube and the housing have plenty of space on the top and bottom.



The step on the lower A-Arms was created to allow extra clearance between the A-Arms and the chassis. The first image shows our first design which we made with smooth edges so there were no stress concentrations. The second design is last years model which was made as such for ease of manufacturing. The last image is out last design, which is shaped as such for the same reason as the housings-tubes. It's so that forces perpendicular to the A-Arms will also act in the direction of the welds putting them under shear stress and increasing their maximum load.

Issues:

Clearance:

One of the issues that was pointed out right at the beginning of the project was the matter of clearance issues in the previous year's design. This issue required a closer look as to avoid it from reoccurring in our design. The solution we came up with for this issue was to bend the A-Arms to allow more clearance and therefore effectively solving this problem. This idea was discarded for a variety of reasons. The initial reason that caused us to rethink this solution was the fact that this solution would add to the manufacturing time and to the manufacturing cost. The next reason for discarding this idea and admittedly the major reason for this was the fact that this problem had already been taken into consideration by the designer of this year's suspension geometry and therefore needed no further exploration. This fact was only mentioned to us after we began our consideration of this problem as it was one of the points outlined in the design brief that we had received at the start of this project.

Clearance:

Another clearance issue occurred later on in the design phase between the spherical housing and the Uprights due to alignment problems. We fixed this problem by making the spherical housing horizontal. Which was done by dividing the lower spherical housing into two separate parts which will be welded together. By doing this we had to make the angles parallel to achieve the highest contact surface. This was an important decision since this parallel alignment will cause any extra point loads to be removed and no extra forces on the weld which would otherwise be present if the angles were not parallel.

Material selection:

The next issue we encountered was the selection of the material that needed to be used to create our part. This issue required a more in depth look as the incorrect choice could lead to catastrophic results if not considered carefully. Multiple materials were taken into consideration as each of their properties were desirable for our design.

The first material series taken into consideration was the aluminium series. This was due to the fact of its low density which was a desired factor since one of our goals in designing our part was for it to have as low of a weight as possible. While the density was the best suited for our use it turned out after performing calculations that other material properties such as tensile strength, young's modulus and deflection were not high enough and would lead to failure and increased deformation if used.

The next materials considered were carbon fibre compounds specifically carbon fibre weaves. We considered this due to the fact that carbon fibre had superior material properties compared to other materials. After some further consideration we decided that this was not the option we wanted to go with since it was far more difficult to work with than the other materials adding more time to the manufacturing process.

Finally we looked at different types of steels which ended with our 4130 steel selection. 4130 steel was chosen because of a variety of reasons such as it having the same density as other types of steel while its other material properties are far greater. It is readily available since it is widely used for other application and is relatively easy to work with. The only drawback of this choice was that it had a density that was higher than most of the other choices but we chose to use this material anyway since we believe that the other material properties are worth this trade off.

Machining:

When designing the spherical housings we tried to design with efficiency in mind while still maintaining the properties that were required. To do this we designed them to be flat on one side. Doing this has multiple benefits such as removing the need for reclamping multiple times which allows for maximum accuracy, time and money to be saved and easy machining therefore, there will be no drawbacks.

Rear suspension points:

An issue that occurred later on in the design stage was problems fitting the rear suspension points and providing enough support. We fixed this by making all the damper tabs and spherical housing jig into each other so there is little room for error to occur when mounting these points. Also a support bar was added underneath the plate that would be welded to the other A-Arm tubes. This helps support both the suspension point and provide strength and support to the bottom A-Arms.

Modelling and Calculating

Theory

To make our design we had to calculate all of the theoretical loads that were applied to the suspension. These forces were calculated from the theory from the ‘tune to win’ book and design parameters from 2016 design and analysis performed to gain data on ts_15 car. The same calculations were used for Uprights and A-Arms with slightly more in depth equations applied for A-Arms as the force in the individual members was required. From the calculations there were 5 major scenarios that were analysed. Braking, accelerating, turning, braking and turning and accelerating and turning. These calculations (below) were calculated from the base data given in ‘tune to win’ and by using the reference geometry and the suspension geometry given to calculate all distances and angles, also physical measurements of ts_15 were taken as well to find unknown measurements and to double check the measurements that were taken. For each situation the reaction forces, loads, moment around chassis clevi, force in each of the members were calculated, the bending moment in each member, bolt torques for individual bolts and the weld calculations for the length of weld required to join our parts together. All these calculations then went into a corner simulation for both front and rear corners in ANSYS.

Formulas used:

Lateral Load Transfer(kg)=Lateral acceleration(g)weight(kg)centre of gravity height(m)Track width(m)

Longitudinal Load Transfer(kg)=acceleration(g)weight(kg)centre of gravity height(m)Wheelbase(m)

MO=0

Fx=0

Fy=0

Given data:

Constants	
Braking (g's)	1.5
Accelerating (g's)	1.3
Turning (g's)	1.6
Gravity (m/s^2)	9.81
Wheel Base (m)	1.53
Track Width (m) (front)	1.1
Track Width (m) (back)	1.1
CG (height, m)	0.27
CG (Length from front, m)	0.918
Car Weight (kg)	300
CG Ratio (back)	0.6
CG Ratio (front)	0.4
Wheel Height (m)	0.44
Height of Front Lower A-Arm	0.172
Height of Front Upper A-Arm	0.292
Angle of bottom front A-Arm	5.62
Angle of top front A-Arm	9.4
Height of Back Lower A-Arm	0.158
Height of Back Upper A-Arm	0.307
Angle of bottom back A-Arm	3.88
Angle of top back A-Arm	14.83

Data from Suspension Geometry

Height of Front Lower A-Arm	0.172
Height of Front Upper A-Arm	0.292
Angle of bottom front A-Arm	5.62
Angle of top front A-Arm	9.4
Height of Back Lower A-Arm	0.158
Height of Back Upper A-Arm	0.307
Angle of bottom back A-Arm	3.88
Angle of top back A-Arm	14.83
Looking At Front of Car	--
Angle on front front Upper	25.06
Angle on back front Upper	11.32
Angle on front front Lower	19.22

Angle on back front Lower	11.09
Angle on front back Upper	12.3
Angle on back back Upper	11.2
Angle on front back Lower	22.34
Angle on back back Lower	24.23
Angle on front back Upper	28.71
Angle on back back Upper	41.23
Angle on front back toe	14.99
Angle on back back toe	23.38

Braking

Longitudinal Load Transfer(kg)=acceleration(g)weight(kg)centre of gravity height(m)Wheelbase(m)
(1.5*300*0.27)/1.53 = 39.7kg

Front wheels
((300/2)*0.4+39.7)*9.81=978.11N
978.11*1.6= 1467.17N

Rear wheels
((300/2)*0.6-39.7)*9.81=493.39N
493.39*1.6=740.1N

Moment in z direction

Front upper
(1467.17*0.172)/(0.292-0.172)=2102.95Nm

Front Lower
(1467.17*0.292)/(0.292-0.172)=3570.12Nm

Rear upper
(740.1*0.158)/(0.307-0.158)=784.78Nm

Rear lower
(740.1*0.307)/(0.307-0.158)=1524.86Nm

Front Front upper
(2102.95*cos(0.197))/sin(0.437-0.197)=3476.49N

Back Front upper
(2102.95*sin(0.197))+3476.49*cos(0.437+0.197)=3211.71N

Front Front lower
(3570.12*cos(0.194))/sin(0.335-0.194)=6941.95N

Back Front lower
(3570.12*sin(0.194))+6941.95*cos(0.335+0.194)=6679.75N

Front Back lower
(1524.86*cos(0.423))/sin(0.423+0.390)=1914.76N

Back Back lower
1524.86*sin(0.423)+1914.76*cos(0.390+0.423)=1942.14N

Front Back upper
(784.78*cos(0.72))/sin(0.72+0.50)=628.33N

Back Back upper
784.78*sin(0.72)+628.33*cos(0.72+0.50)=732.75N

Find sum of forces

Accelerating

Longitudinal Load Transfer(kg)=acceleration(g)weight(kg)centre of gravity height(m)Wheelbase(m)
(1.3*(300/2)*0.27)/1.53=34.41kg

Front Left & Right
(300/2)*0.4-34.41*9.81=251.02N

Back Left & Right
((300/2)*0.6+34.41)*9.81=1220.48N

Force gravity in z Front
251.02*1.3=326.37N

force in rear z direction
1220.48*1.3=1586.62N

Front upper
326.33*0.172/(0.292-0.172)=467.74N
Front Lower
326.33*0.292/(0.292-0.172)=794.06N

Rear upper
1586.62*0.158/(0.307-0.158)=1682.46N

Rear lower
1586.62*0.307/(0.307-0.158)=3269.08N

Front front upper
467.74*sin(0.437)+714.344*cos(0.437+0.197)=773.23N

Back front upper
467.74*cos(0.437)/(sin(0.437+0.197))=714.34N

Front Front Lower
794.06*sin(0.335)+1485.70*cos(0.335+0.193)=1544.02N

Back Front upper
794.06*cos(0.335)/(sin(0.335+0.194))=1485.70N

Front Back Lower
3269.08*sin(0.39)+4163.67*cos(0.39+0.422)=4104.98N

Back Back lower
3269.08*cos(0.39)/sin(0.422+0.39)=4163.67N

Front back upper
1682.46*sin(0.50)+1570.92*cos(0.5+0.72)=1347.04N

Back back upper
1682.46*cos(0.50)/(sin(0.50+0.72))=1570.92N

Turning Left/Right

Lateral load transfer=Acceleration (g) Weight (kg)centre of gravity height (m)Wheelbase (m)
Force = mass * gravity
Moment = force * distance
Convert degrees to radians

Turning and braking

Lateral Load Transfer(kg)=Lateral acceleration(g)weight(kg)centre of gravity height(m)Track width(m)
(300*1.13*0.27)/1.1=83.31kg

Longitudinal Load Transfer(kg)=acceleration(g)weight(kg)centre of gravity height(m)Wheelbase(m)
(300*1.06*0.27)/1.53=28.08kg

Front Left
((((300/2)*0.4)+28.08)+83.31)*9.81=1681.3N

Back left
((((300/2)*0.6)+41.65)-28.08)*9.81=1016.11N

Front Right
((((300/2)*0.4)+28.08)-83.31)*9.81=226.20N

Back Right
((((300/2)*0.6)-41.65)-28.08)*9.81=428.02N

Force through front a-arm in x direction

Gravity force
1.13*1272.66=1439.86N

Front upper
(0.172*1439.86)/(0.292-0.172(cos(0.164)))=2091.88N

Front lower

Use simultaneous equations to find force in lower and force in suspension
Force in lower=3395.63N
Force in suspension=1269.64N

Force through rear a-arm in x direction

Gravity force
1.06*1016.10=1149.6N

Rear upper
(1149.6*0.158)/((0.307-0.158)*cos(0.259))=1261.04N

Rear lower
Use simultaneous equations to find force in lower and rear suspension
Force lower=2316.28N
Force in suspension=852.03N

Force through front a-arm in z direction

Gravity force
1272.66*-1.0.6=-1349.86N

Front upper
-1349.86*0.172/(0.292-0.172)=-1934.81N

Front Lower
-1349.86*0.292/(0.292-0.172)=-3284.67N

Force through rear a-arm in z direction

Gravity force
1016.11*-1.06=-1077.74N

Rear upper
-1077.74*0.158/(0.307-0.158)=-1142.84N

Rear lower
-1077.74*0.307/(0.307-0.158)=-2220.59N

Magnitude of forces in front upper
sqrt(2091.88^2 + 1934.81^2)=2849.46N

Angle of forces in front upper
tan^-1(2091.88/1934.8)=0.824 radians

Front front upper
2849.46*cos(0.824+0.198)/(sin(0.437+0.198))=2506.25N Tension

Back front upper
2506.25*cos(0.437+0.198)+2849.46*sin(0.824+0.198)=4448.75N Compression

Magnitude of forces in front lower
sqrt(3395.63^2+3284.67^2)=4724.34N

Angle of forces in front lower
tan^-1(3395.63/3284.67)=0.80 radians

Front front lower
4724.34*cos(0.8+0.193)/sin(0.193+0.335)=5092.71N Tension

Back front Lower
5092.71*cos(0.913+0.335)+4724.34*sin(0.8+0.194)=8360.61N Compression

Magnitude of forces in rear lower
sqrt(2316.28^2+2220.56^2)=3208.76N

Angle of forces in front lower
tan^-1(2316.28/2220.56)=0.81 radians

Front back lower
3208.76*cos(0.81+0.423)/sin(0.423+0.39)=1479.40N Tension

Back back Lower
1479.40*cos(0.43+0.39)+3208.76*sin(0.81+0.423)=4040.60N Compression

Magnitude of forces in rear upper
sqrt(1261.04^2+1142.84^2)=1701.85N

Angle of forces in front lower
tan^-1(1261.04/1142.84)=0.83 radians

Front back upper
1701.85*cos(0.83+0.72)/sin(0.5+0.72)=30.20N compression

Back back upper
1701.85*cos(0.83+0.72)+30.20*cos(0.5+0.72)=1711.98N Compression

Toe
(0.03*1701.85*sin(0.83))/(cos(0.41)*(0.03+0.04))=588.79N Tension

Turning and Accelerating

Lateral Load Transfer(kg)=Lateral acceleration(g)weight(kg)centre of gravity height(m)Track width(m)
(300*1.13*0.27)/1.1=83.31kg

Longitudinal Load Transfer(kg)=acceleration(g)weight(kg)centre of gravity height(m)Wheelbase(m)
(300*0.919*0.27)/1.53=24.33kg

Front Left
((((300/2)*0.4)+24.33)-83.31)*9.81=758.53N

Back left
((((300/2)*0.6)+41.65)+24.33)*9.81=1530.24N

Front Right
((((300/2)*0.4)-24.33)-83.31)*9.81=58.74N

Back Right
((((300/2)*0.6)+41.65)-24.33)*9.81=646.68N

Force through front a-arm in x direction

Gravity force
1.13*758.53=858.18N

Front upper
(0.172*858.18)/(0.292-0.172(cos(0.164)))=1246.8N

Front lower
Use simultaneous equations to find force in lower and force in suspension
Force in lower=2023.86N
Force in suspension=756.73N

Force through rear a-arm in x direction

Gravity force
1.13*1530.24=1731.27N

Rear upper
(1731.27*0.158)/((0.307-0.158)*cos(0.259))=1899.10N

Rear lower
Use simultaneous equations to find force in lower and rear suspension
Force lower=3488.28N
Force in suspension=1283.14N

Force through front a-arm in z direction

Gravity force
758.53*0.919=697.27N

Front upper
697.27*0.172/(0.292-0.172)=999.42N

Front Lower
697.27*0.292/(0.292-0.172)=1696.69N

Force through rear a-arm in z direction

Gravity force
1530.24*0.919=1406.66N

Rear upper
1406.66*0.158/(0.307-0.158)=1491.62N

Rear lower

1406.66*0.307/(0.307-0.158)=2898.28N

Magnitude of forces in front upper
sqrt(1246.78^2 + 999.42^2)=1597.92N

Angle of forces in front upper
tan^-1(1246.78/999.42)=0.895 radians

Front front upper
626.00*cos(0.895+0.438)+1597.92*(sin(0.856+0.438))=2064.80N Compression

Back front upper
1597.92*cos(0.895+0.438)/sin(0.438+0.198)= 636.00 Tension

Magnitude of forces in front lower
sqrt(2023.86^2+1696.7^2)=2640.98N

Angle of forces in front lower
tan^-1(2023.86/1696.7)=0.873 radians

Back front lower
2640.98*cos(0.873+0.193)/sin(0.193+0.335)=1854.4N Tension

Front front Lower
1854.4*cos(0.913+0.335)+2640.98*sin(0.873+0.193)=4074.51N Compression

Magnitude of forces in rear lower
sqrt(3488.28^2+2898.28^2)=4535.21N

Angle of forces in front lower
tan^-1(3488.28/2898.28)=0.878 radians

Back back lower
4535.21*cos(0.878+0.423)/sin(0.423+0.39)=3990.94N Tension

Front back Lower
3990.94*cos(0.43+0.39)+4535.21*sin(0.878+0.423)=7908.4N Compression

Magnitude of forces in rear upper
sqrt(1899.10^2+1491.62^2)=2414.86N

Angle of forces in front lower
tan^-1(1899.10/1491.62)=0.905 radians

Back back upper
2414.86*cos(0.905+0.72)/sin(0.5+0.72)=749.93N compression

Front back upper
749.93*cos(0.905+0.72)+749.92*cos(0.5+0.72)=2124.94N Compression

Toe
(0.03*2414.85*sin(0.83))/(cos(0.41)*(0.03+0.04))=886.71N Tension

Data For tubing

Diameter	12.7	12.7	12.7	15.875	15.875	9.525	7.9
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	0.889	0
UTS	670000000	670000000	670000000	670000000	670000000	670000000	670000
E	200000000000	200000000000	200000000000	200000000000	200000000000	200000000000	200000000
Yield	435000000	435000000	435000000	435000000	435000000	435000000	435000
FFU	0.21539	0.21539	0.21539	0.21539	0.21539	0.21539	0.21
BFU	0.19335	0.19335	0.19335	0.19335	0.19335	0.19335	0.19
FFL	0.25778	0.25778	0.25778	0.25778	0.25778	0.25778	0.25
BFL	0.24524	0.24524	0.24524	0.24524	0.24524	0.24524	0.24
FBU	0.13453	0.13453	0.13453	0.13453	0.13453	0.13453	0.13
BBU	0.16913	0.16913	0.16913	0.16913	0.16913	0.16913	0.16
FBL	0.2053	0.2053	0.2053	0.2053	0.2053	0.2053	0.2
BBL	0.20902	0.20902	0.20902	0.20902	0.20902	0.20902	0.20
Toe	0.09	0.09	0.09	0.09	0.09	0.09	

Area of desired tubing

One calculation will be demonstrated for the tube with a diameter of 12.7mm and a wall thickness of 0.889mm which has to just be repeated to receive the results for the other sizes

Area =Outer Area - Inner Area10002
Outer Area =Pi*(Diameter2)2
Inner Area =Pi*(Diameter - Wall thickness2)2

Outer Area = Pi*(12.7/2)^2= 126.68mm^2
Inner Area = Pi*((12.7-0.889)/2)^2= 109.56mm^2
Area= (126.68-109.56)/1000^2= 0.00001711404511m^2

Diameter (mm)	12.7	12.7	12.7	15.875	15.875
Wall Thickness(mm)	0.889	1.2446	1.651	0.889	1.2446
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844

Moment of inertia of desired tubing

One calculation will be demonstrated for the tube with a diameter of 12.7mm and a wall thickness of 0.889mm which has to just be repeated to receive the results for the other sizes

Moment of Inertia = (Pi64*Diameter4-(Diameter-(2*Wall thickness))4)/10004
Moment of Inertia = (Pi64*12.74-(12.7-(2*0.889))4)/10004= 0.0000000005784624351

Diameter (mm)	12.7	12.7	12.7	15.
Wall Thickness(mm)	0.889	1.2446	1.651	0.
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.00000000117908З

Factor of safety for stress

One calculation will be demonstrated for the tube with a diameter of 12.7mm and a wall thickness of 0.889mm which has to just be repeated to receive the results for the other sizes

Braking & Turning
F.S.=Yield Strength(ForceArea)
Front Front upper
F.S.= 670000000/(2506.25/0.00001711404511)=4.575

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	2506.253452	2506.253452	2506.253452	2506.253452	2506.253452	250
F.S.	4.575119972	6.312243244	8.232506774	5.760384214	7.971613182	3.38

Back Front upper
F.S.= 670000000/(4448.75/0.00001711404511)=2.577

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	4448.75365	4448.75365	4448.75365	4448.75365	4448.75365	444
F.S.	2.577443285	3.556070456	4.63787167	3.245174706	4.490894445	1.90

Front Front lower

F.S.= 670000000/(5092.71/0.00001711404511)=2.252

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	5092.709888	5092.709888	5092.709888	5092.709888	5092.709888	509
F.S.	2.251534148	3.106417166	4.051428212	2.834833151	3773.354995	1.6

Back Front lower

F.S.= 670000000/(8360.61/0.00001711404511)=1.371

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	8360.606758	8360.606758	8360.606758	8360.606758	8360.606758	836
F.S.	1.37148063	1.892216902	2.467853006	1.726786492	2.38964511	1.0

Front Back lower

F.S.= 670000000/(1479.40/0.00001711404511)=7.751

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	1479.40048	1479.40048	1479.40048	1479.40048	1479.40048	147
F.S.	7.750714161	10.69357599	13.94669587	9.758671197	13.50471584	5.7

Back Back lower

F.S.= 670000000/(4040.60/0.00001711404511)=2.838

Diameter	12.7	12.7	12.7	15.875	15.875	

Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	4040.600081	4040.600081	4040.600081	4040.600081	4040.600081	404
F.S.	2.837798839	3.915280181	5.106357497	3.572979886	4.944533646	2.1

Front Back Upper
F.S.= 670000000/(30.201/0.00001711404511)=379.674

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	30.20064123	30.20064123	30.20064123	30.20064123	30.20064123	30.
F.S.	379.6743961	523.8326331	683.1890872	478.0356386	661.5383726	281

Back Back Upper
F.S.= 670000000/(1711.977/0.00001711404511)=6.698

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.00
Force	1711.976914	1711.976914	1711.976914	1711.976914	1711.976914	171
F.S.	6.697759839	9.240826372	12.0520016	8.432930781	11.67006569	4.9

Toe
F.S.= 670000000/(588.789/0.00001711404511)=19.475

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	

UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	588.7886162	588.7886162	588.7886162	588.7886162	588.7886162	588
F.S.	19.47457866	26.86886428	35.04270964	24.51980629	33.93218297	14.4

Acceleration & Turning

Front Front upper
F.S.= 670000000/(2064.801/0.00001711404511)=5.553

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	2064.801164	2064.801164	2064.801164	2064.801164	2064.801164	206
F.S.	5.553275745	7.661794121	9.992607946	6.991948218	9.675935583	4.1

Back Front upper
F.S.= 670000000/(636.004/0.00001711404511)=18.029

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	636.0035458	636.0035458	636.0035458	636.0035458	636.0035458	636
F.S.	18.02884638	24.87420318	32.44124762	22.69953196	31.413163	13.3

Front Front lower
F.S.= 670000000/(4070.514/0.00001711404511)=2.817

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	4070.514249	4070.514249	4070.514249	4070.514249	4070.514249	407
F.S.	2.816943885	3.886506827	5.068830928	3.546722093	4.908196319	2.08

Back Front lower
F.S.= 670000000/(1854.3995/0.00001711404511)=6.183

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	1854.399535	1854.399535	1854.399535	1854.399535	1854.399535	185
F.S.	6.183354776	8.531107305	11.1263771	7.785260159	10.77377484	4.58

Front Back lower
F.S.= 670000000/(7908.396/0.00001711404511)=1.4499

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	7908.39554	7908.39554	7908.39554	7908.39554	7908.39554	790

F.S.	1.449903481	2.000416056	2.608967699	1.825526144	2.526287785	1.07

Back Back lower

F.S.= 670000000/(3990.938/0.00001711404511)=2.873

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	3990.93841	3990.93841	3990.93841	3990.93841	3990.93841	399
F.S.	2.873111292	3.964000388	5.169899002	3.617440643	5.006061479	2.12

Front Back Upper

F.S.= 670000000/(2124.943/0.00001711404511)=5.396

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	2124.942947	2124.942947	2124.942947	2124.942947	2124.942947	212
F.S.	5.396102629	7.444944083	9.709789409	6.794056678	9.402079752	3.99

Back Back Upper

F.S.= 670000000/(-749.9299/0.00001711404511)=-15.28997

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000

Force	-749.9298607	-749.9298607	-749.9298607	-749.9298607	-749.9298607	-749
F.S.	-15.28997687	-21.09541471	-27.51290434	-19.25111077	-26.64100218	-11.

Toe
F.S.= 670000000/(886.706/0.00001711404511)=12.931

Diameter	12.7	12.7	12.7	15.875	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	1.2446	
UTS	670000000	670000000	670000000	670000000	670000000	
Area	0.00001711404511	0.00002361206182	0.00003079514704	0.00002154773555	0.00002981922844	0.000
Force	886.7057595	886.7057595	886.7057595	886.7057595	886.7057595	886
F.S.	12.93147146	17.84141047	23.2689912	16.2815936	22.53158146	9.58

Factor of safety for buckling

One calculation will be demonstrated for the tube with a diameter of 12.7mm and a wall thickness of 0.889mm which has to just be repeated to receive the results for the other sizes

Breaking and Turning
F.S.=(Pi2*Modulus of elasicity*Moment of inertia)/(Rod Value/Compression Force)
Back Front Upper
F.S. = Pi^2*

Diameter	12.7	12.7	12.7	15.875	
Wall thickness	0.889	1.2446	1.651	0.889	
E	200000000000	200000000000	200000000000	200000000000	
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266	0
BFU	0.19335	0.19335	0.19335	0.19335	

Force	4448.75365	4448.75365	4448.75365	4448.75365
F.S.	6.865595181	8.823075509	10.61131314	13.99418165

Back Front Lower

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BFL	0.24524	0.24524	0.24524	0.24524
Force	8360.606758	8360.606758	8360.606758	8360.606758
F.S.	2.270829308	2.918275535	3.50974391	4.628644275

Back Back Lower

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BBL	0.20902	0.20902	0.20902	0.20902
Force	4040.600081	4040.600081	4040.600081	4040.600081
F.S.	6.468198621	8.31237545	9.997105743	13.18416598

Front Back Upper

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
FBU	0.13453	0.13453	0.13453	0.13453
Force	30.20064123	30.20064123	30.20064123	30.20064123
F.S.	2089.057789	2684.678331	3228.801842	4258.138355

Back Back Upper

Diameter	12.7	12.7	12.7	15.875

Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BBU	0.16913	0.16913	0.16913	0.16913
Force	1711.976914	1711.976914	1711.976914	1711.976914
F.S.	23.31662568	29.96453237	36.03766461	47.52641054

Acceleration and Turning

Back Front Upper

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BFU	0.19335	0.19335	0.19335	0.19335
Force	2064.801164	2064.801164	2064.801164	2064.801164
F.S.	11.91998184	15.31854079	18.42326216	24.29656667

Back Front Lower

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BFL	0.24524	0.24524	0.24524	0.24524
Force	4070.514249	4070.514249	4070.514249	4070.514249
F.S.	4.221406596	5.424990575	6.524513331	8.604517038

Back Back Lower

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266

BBL	0.20902	0.20902	0.20902	0.20902
Force	7908.39554	7908.39554	7908.39554	7908.39554
F.S.	2.172791638	2.79228591	3.35821904	4.428813535

Front Back Upper

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
FBU	0.13453	0.13453	0.13453	0.13453
Force	2124.942947	2124.942947	2124.942947	2124.942947
F.S.	29.69062529	38.15585131	45.88917843	60.51857015

Back Back Upper

Diameter	12.7	12.7	12.7	15.875
Wall thickness	0.889	1.2446	1.651	0.889
E	200000000000	200000000000	200000000000	200000000000
Moment of I	0.0000000005784624351	0.0000000007433904285	0.0000000008940588363	0.000000001179083266
BBU	0.16913	0.16913	0.16913	0.16913
Force	-749.9298607	-749.9298607	-749.9298607	-749.9298607
F.S.	-53.22834438	-68.404514	-82.26856016	-108.4956366

Weld
= Force0.707*l*h
h = 2mm
l =18mm
Sigma_(Ultimate Strength)= 300MPa
Sigma_(Yield strength)= 180MPa

Front Front Lower
Sigma = (5092.71/(0.707*0.002*0.018))/4 = 50022689.77

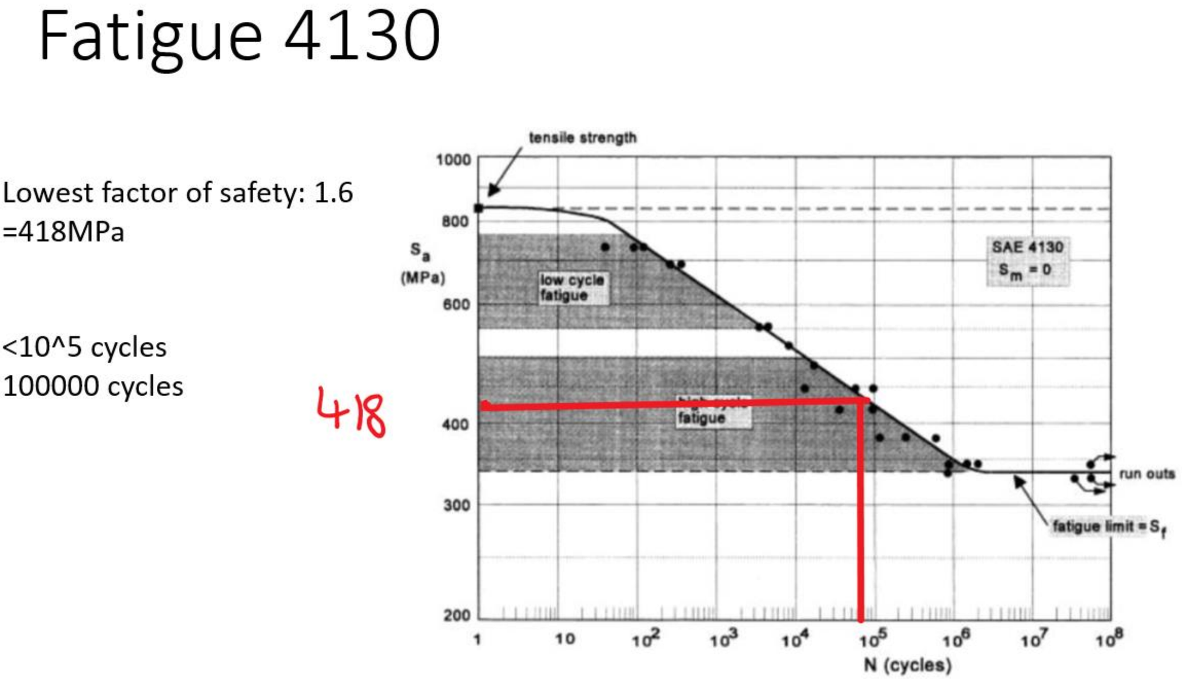
Bolt Torque

T=Fa12(P2*Pi+Raverage*Coefficient of fricitonCos+Rs*Coefficient of friction)=2107.17Nm

Torque	5
Lead of threads	1
R_av	2.5
R_s	4.5
Cos	0.866
Coefficient of Friction	0.3

Fatigue

The fatigue life for our lowest factor of safety in the A-Arm tubes was calculated to show how many cycles it will last without breaking.



Fatigue 4130

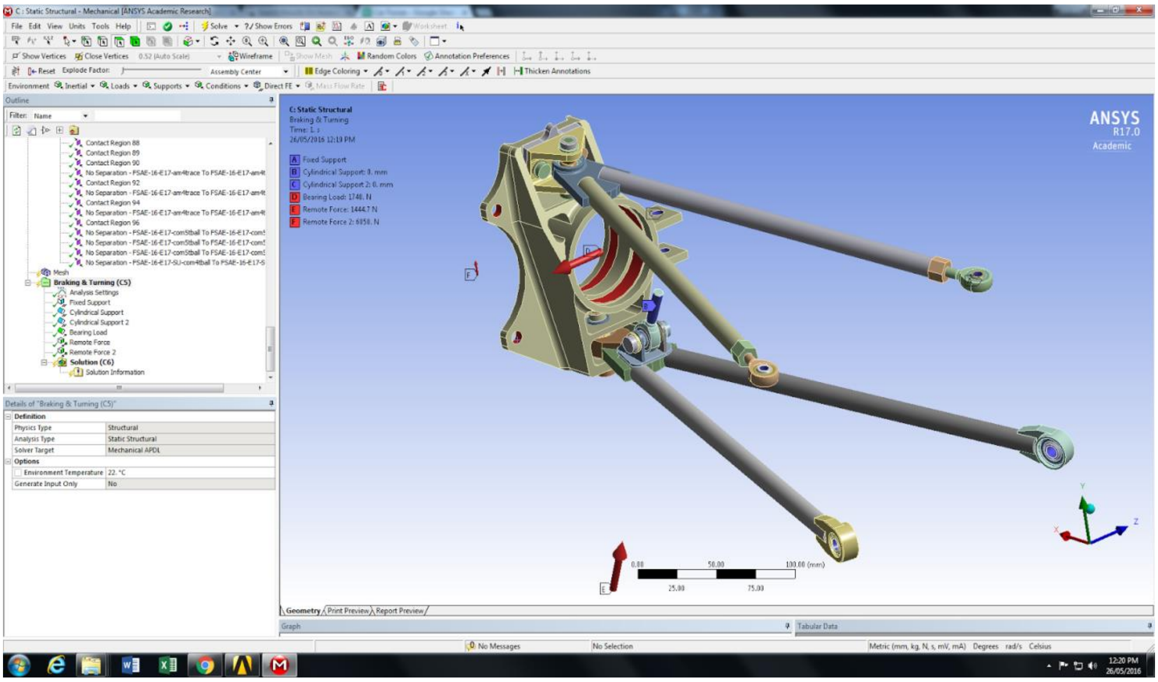
Lowest factor of safety: 1.6
=418MPa

<10^5 cycles
100000 cycles

418

Finite Element Analysis

Finite Element Analysis also knows as FEA’s are programs designed to confirm calculations using known forces to determine mainly the stress, strain and Factor of Safety of a chosen system. FEA’s should only be used to confirm calculations and to optimise the system to reduce stress as well as weight. Before preparing the model for analysis, it must have no interferences with any parts which also includes removing the clearances from any holes and removing any bolts and replacing them with cylinders perfectly depicting the bolts, washers and nuts.



C: Static Structural
Braking & Turning
Time: 1. s
26/05/2016 12:19 PM

- A Fixed Support
- B Cylindrical Support: 0. mm
- C Cylindrical Support 2: 0. mm
- D Bearing Load: 1748. N
- E Remote Force: 1444.7 N
- F Remote Force 2: 6858. N

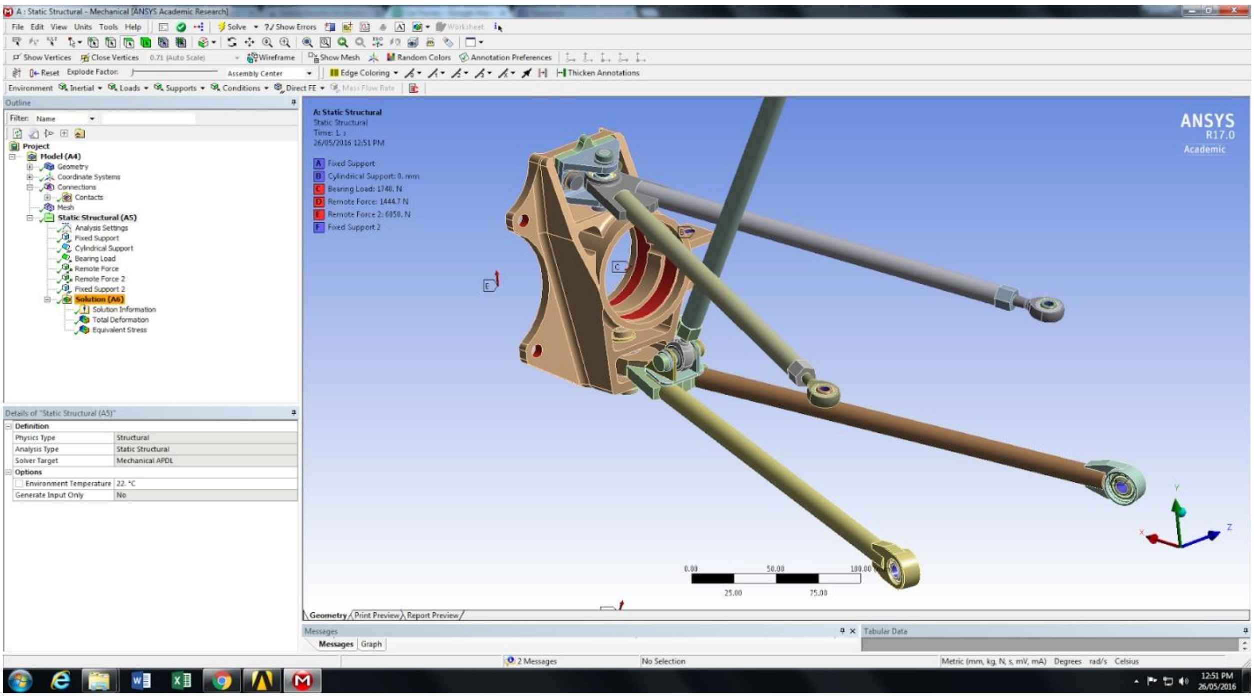
Details of "Braking & Turning (C5)"

Definition	
Physics Type	Structural
Analysis Type	Static Structural
Solver Target	Mechanical APDL
Options	
☐ Environment Temperature	22. °C
Generate Input Only	No

Figure 1: The following figure demonstrates the loads and support applied to the Front A-Arms under the braking and turning scenario.

To correctly determine the stresses through a system, it is crucial to include all relevant components, forces and fixtures. In this case, although the Upright stresses are not required to optimise the A-Arms, all the forces travel through the Upright causing a bending moment around the centre hub insert as well as causing the y displacement of the Upper and Lower A-Arm to be almost equal (under some loads the Upright will slightly deform). By making the spherical connections frictionless or no separation, as well as fixing the suspension rod and steering rod areas in certain directions, we can accurately determine the motion of the A-Arms under all the different loading scenarios. Furthermore, including a suspension rod with an infinite spring constant will further improve the results.

The spring constant is infinite because FEA's work as static models and not dynamic meaning that when the force is applied, the instant displacement will be zero meaning that the spring will have no time to deform.



A: Static Structural
Static Structural
Time: 1. s
26/05/2016 12:51 PM

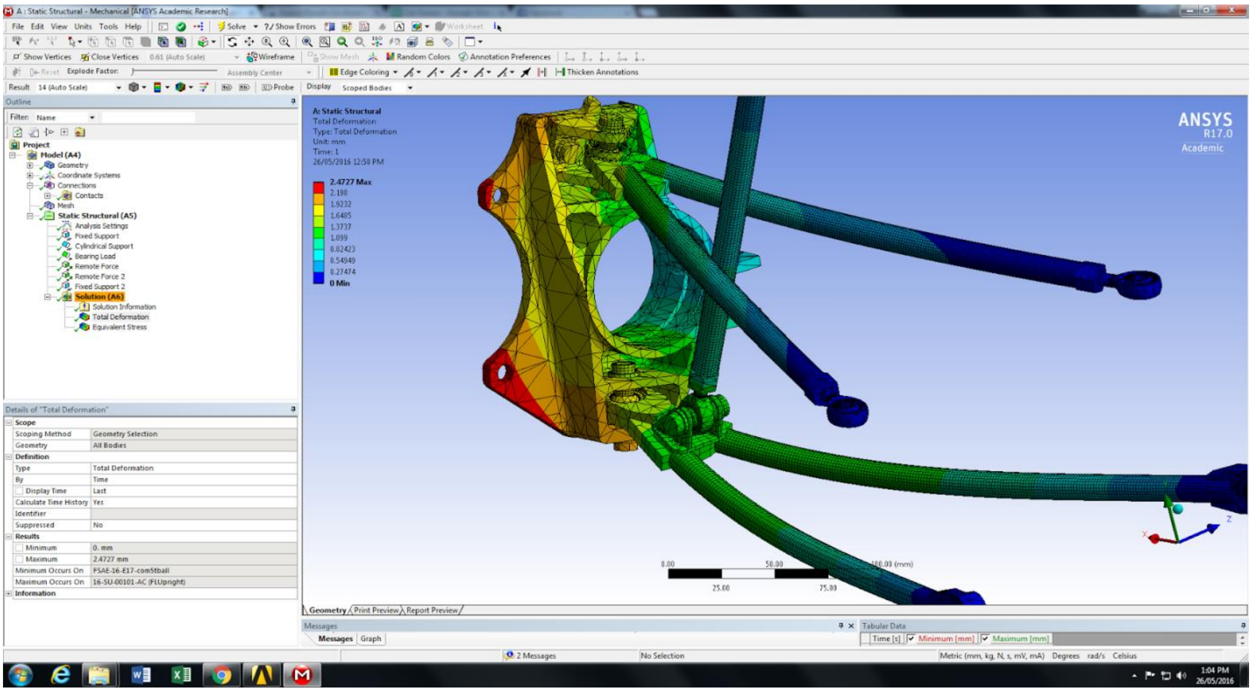
A Fixed Support
B Cylindrical Support: 0. mm
C Bearing Load: 1740. N
D Remote Force: 1444.7 N
E Remote Force 2: 6858. N
F Fixed Support 2

Details of "Static Structural (A5)"

Definition	
Physics Type	Structural
Analysis Type	Static Structural
Solver Target	Mechanical APDL
Options	
☐ Environment Temperature	22. °C
Generate Input Only	No

2 Messages

Figure 2: Front A-Arms with a suspension rod under Braking and Turning.



A: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
26/05/2016 12:58 PM

2.4727 Max
2.198
1.9232
1.6485
1.3737
1.099
0.82423
0.54949
0.27474
0 Min

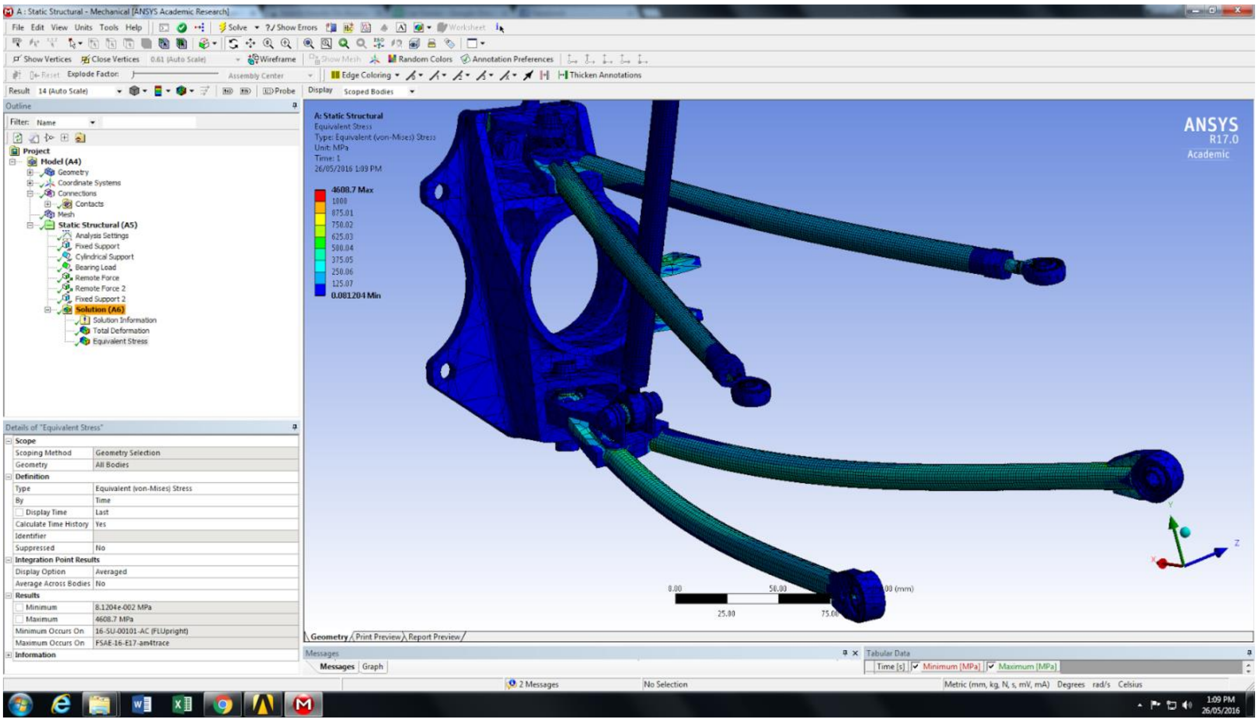
Details of "Total Deformation"

Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Total Deformation
By	Time
☐ Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Results	
☐ Minimum	0. mm
☐ Maximum	2.4727 mm
Minimum Occurs On	FSAE-16-E17-com5tball
Maximum Occurs On	16-SU-00101-AC (FLUpright)
Information	

2 Messages

Figure 3: The deformation of the Front A-Arms under Braking and Turning (14x Scale)

Determining the motion of the A-Arms under different load cases is a major step in optimising the part. It is evident that the Upright is forced upwards causing an upwards force in Upper and Lower A-Arms and a moment can also be seen generated on the spherical housing generated by the upwards motion and the radially fixed suspension rod.

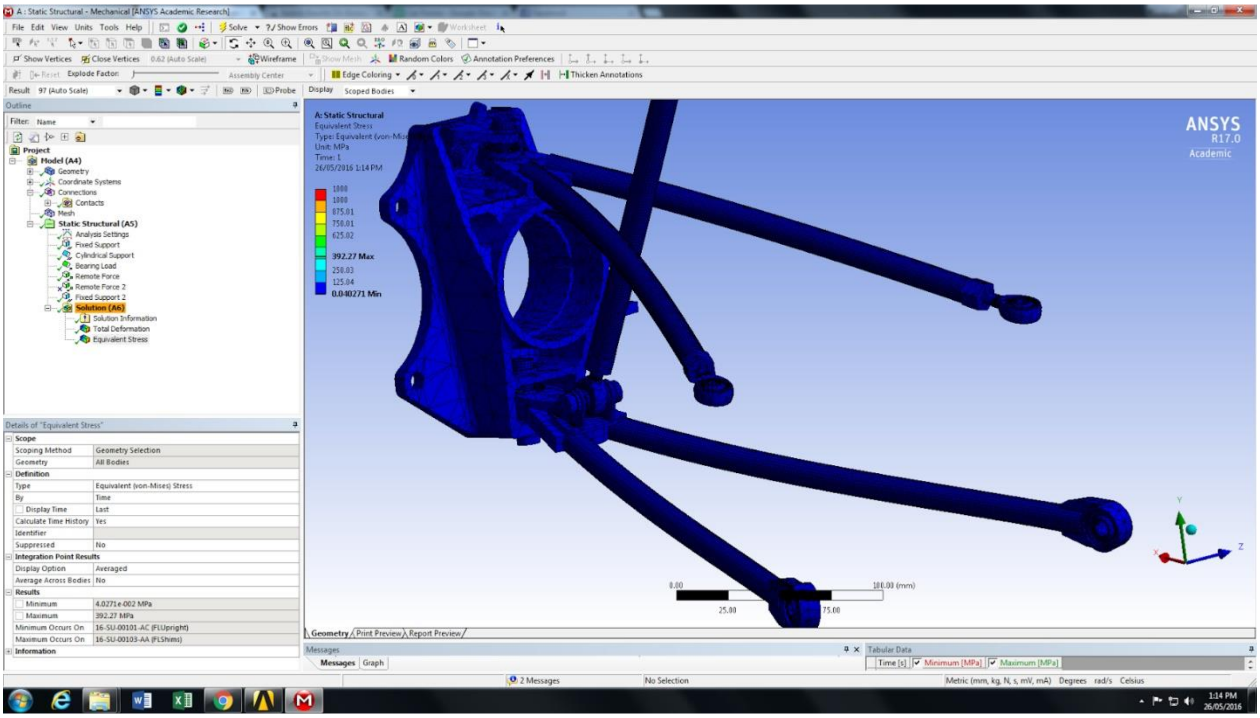


A: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
26/05/2016 1:09 PM

Details of "Equivalent Stress"

Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Equivalent (von-Mises) Stress
By	Time
☐ Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
☐ Minimum	8.1204e-002 MPa
☐ Maximum	4608.7 MPa
Minimum Occurs On	16-SU-00101-AC (FLUpright)
Maximum Occurs On	FSAE-16-E17-am4trace
Information	

Figure 4: The stress of the Front A-Arms under Braking and Turning.

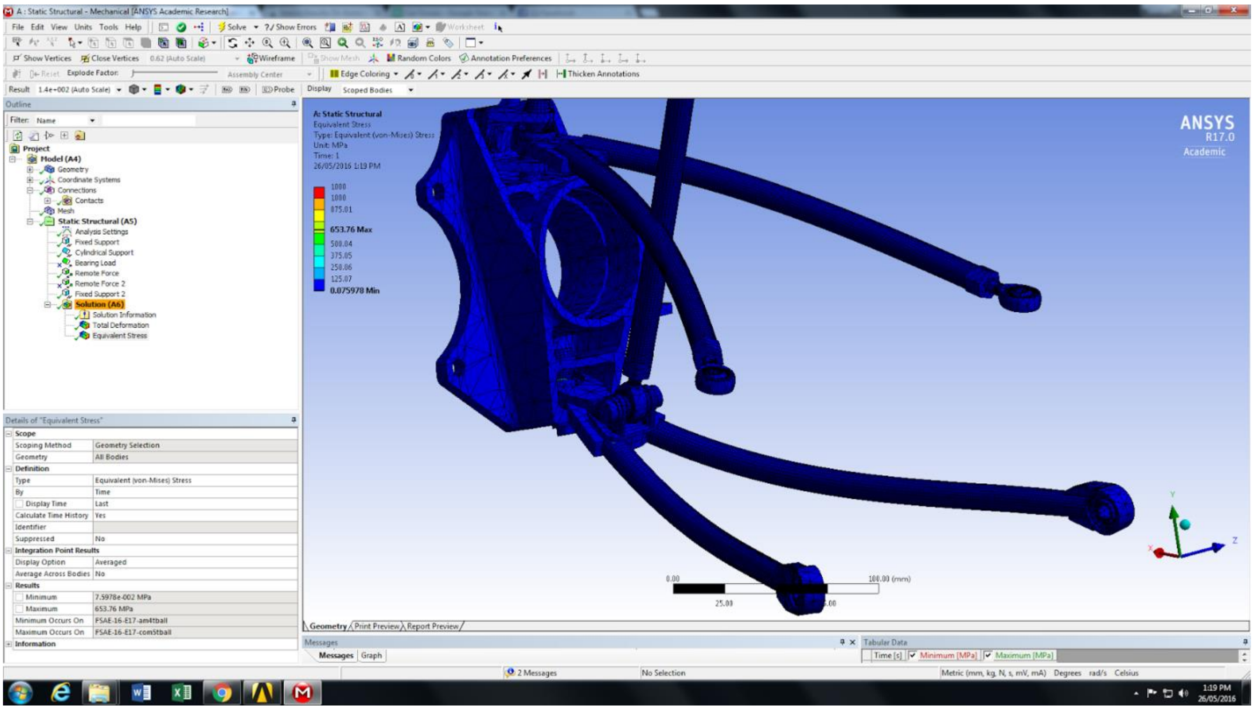


A: Static Structural
Equivalent Stress
Type: Equivalent (von-Mis
Unit: MPa
Time: 1
26/05/2016 1:14 PM

Details of "Equivalent Stress"

Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Equivalent (von-Mises) Stress
By	Time
☐ Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
☐ Minimum	4.0271e-002 MPa
☐ Maximum	392.27 MPa
Minimum Occurs On	16-SU-00101-AC (FLUpright)
Maximum Occurs On	16-SU-00103-AA (FLShims)
Information	

Figure 5: The stress of the Front A-Arms under Accelerating and Turning


A: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
26/05/2016 1:19 PM

1000
1000
875.01
653.76 Max
500.04
375.05
250.06
125.07
0.075978 Min

Details of "Equivalent Stress"

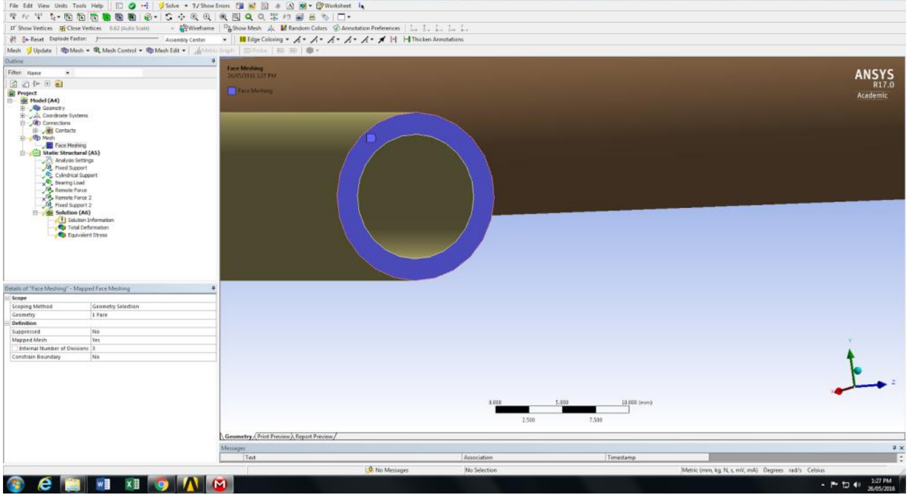
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Equivalent (von-Mises) Stress
By	Time
☐ Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
☐ Minimum	7.5978e-002 MPa
☐ Maximum	653.76 MPa
Minimum Occurs On	FSAE-16-E17-am4tball
Maximum Occurs On	FSAE-16-E17-com5tball
Information	

Figure 6: The stress of the Front A-Arms under Pure Cornering.

Comparing different cases together, low stress areas can be determined and new designs on those areas can be made. Removing mass from areas of low stress can really add up and plays a big part when building a racecar.

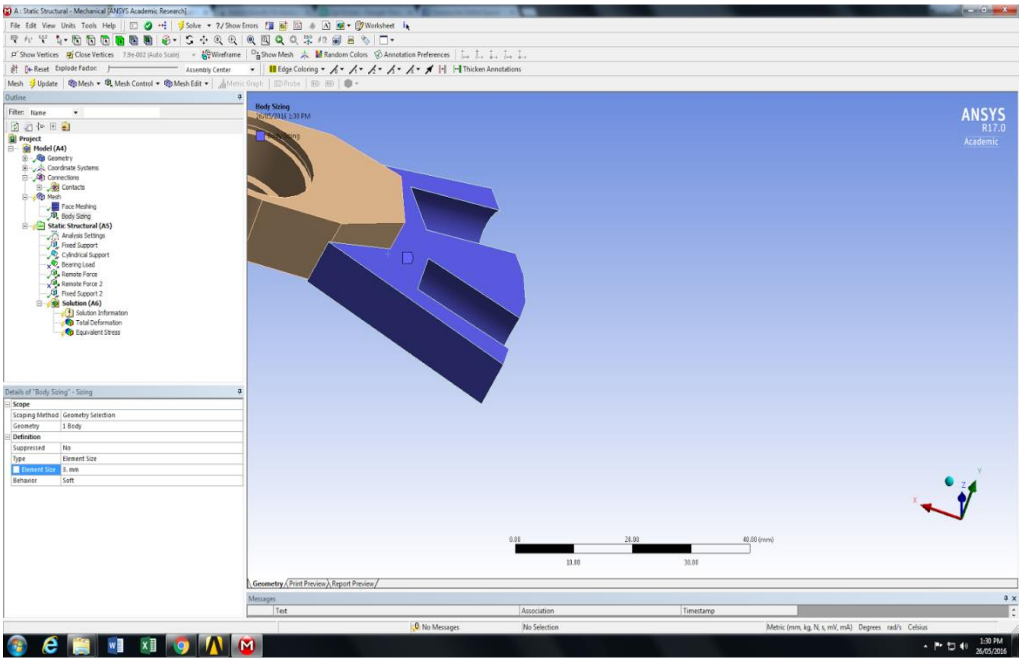
Likewise, areas of high stress will need to be adjusted in design to accommodate for these forces and the design will have to be tweaked in order to survive this stress. Factors of safety are extremely important on any component and using Chromoly with a Yield Strength of 435MPa, a design with a maximum stress of around 300MPa is ideal.

Many times insufficient mesh sizes will lead to poor results and higher stresses may be obtained. This is why it is common practice to apply a face meshing of three internal divisions to the front of the tubes as well as a body sizing of 3mm to any plates or blocks.


Details of "Face Meshing" - Mapped Face Meshing

Scope	
Scoping Method	Geometry Selection
Geometry	1 Face
Definition	
Suppressed	No
Mapped Mesh	Yes
☐ Internal Number of Divisions	3
Constrain Boundary	No

Figure 7: Face meshing applied to tube.



Details of "Body Sizing" - Sizing

Scope	
Scoping Method	Geometry Selection
Geometry	1 Body
Definition	
Suppressed	No
Type	Element Size
Element Size	3. mm
Behavior	Soft

Figure 8: Body sizing applied to spherical housing.

Even with extremely tight mesh sizes forces still remain which will require a remedial session in Solidworks to update and improve the part.

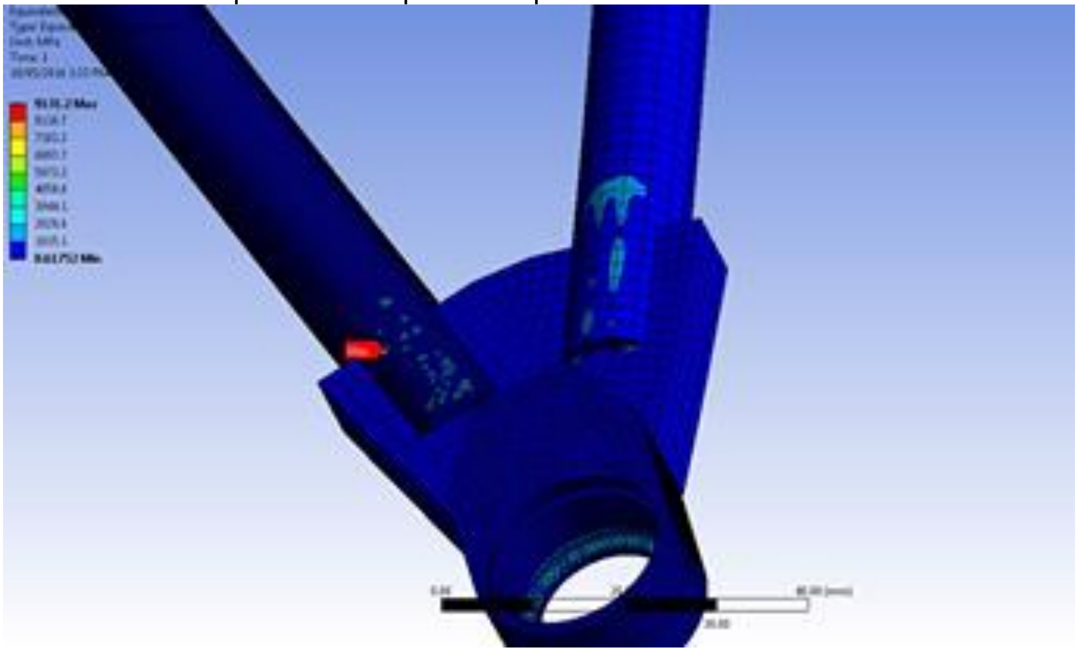


Figure 9: Extremely fine mesh size on the Front lower A-Arms.

Visualising connections is also necessary for perfect results. This means that only some connections such as the nut on the bolt and any welded parts should be bonded, all the rest sure either be no separation or rough. With this in mind, a pretension is required on the bolts in order to replicate them. From our bolt calculations and data gathered from websites, a pretension of 10 kN was applied to all NAS bolts and a pretension of 2.1kN was applied to the M6’s connections the Upright to the Clevi.

Design variables

Our design was guided by goals given to us at the start of the year. These were a weight goal of <600g for each corner A-Arm, to make the fronts backwards compatible with ts_15 and to fix the clearance issues that occurred during ts_15 in the rear A-Arms. All these goals had to be met to reach our whole team goal of 3 months testing. From these goals we discussed and chose what were the most important goals that we had to hit and then the design was taken from those criteria. The most important were the clearance issues, backwards compatibility and the weight goal. These components are critical for the running of the car where if even one it slightly wrong it will drastically change the performance of the car. The size of the A-Arms were defined by the suspension geometry and the size of the chassis clevi and the upright clevi. We had to make sure our parts integrated well with all interacting parts. The cost was a factor that had to be taken into account as well. As the material was not that expensive to buy, the main goal was to design all components so they could be manufactured in house by the technicians to keep the cost down. Therefore these designs had to be simplified for ease of manufacturing.

Modeling

All of the modeling for the A-Arm components was performed on Dassault systems ® SOLIDWORKS ®, using the suspension geometry. The A-Arms are compiled of over 30 individual parts not including all the given parts (sphericals and rod ends), ferrules, bolts washers and nuts. All parts are included in the documentation.
Our design had to be optimised so it was put through ANSYS to validate our designs choices and to see where the major forces were acting. So we could see how the design could change so it would be more optimal. The ANSYS results were used to optimise the whole A-Arms.

For the front we made the spherical housings thicker so the it could withstand the forces. The lower spherical housing was split into 2 parts so it would not interfere with the Upright when forces applied and not interfere with the ferrules.

For the rear we changed the force on the rear suspension point which had deformed the initial design (1). A design was put together so it had a sheet metal section that took the weight but it was not strong enough (2). This then had to be braced with a tube underneath and interlocking tabs and spherical housing so it would withstand the forces (2).

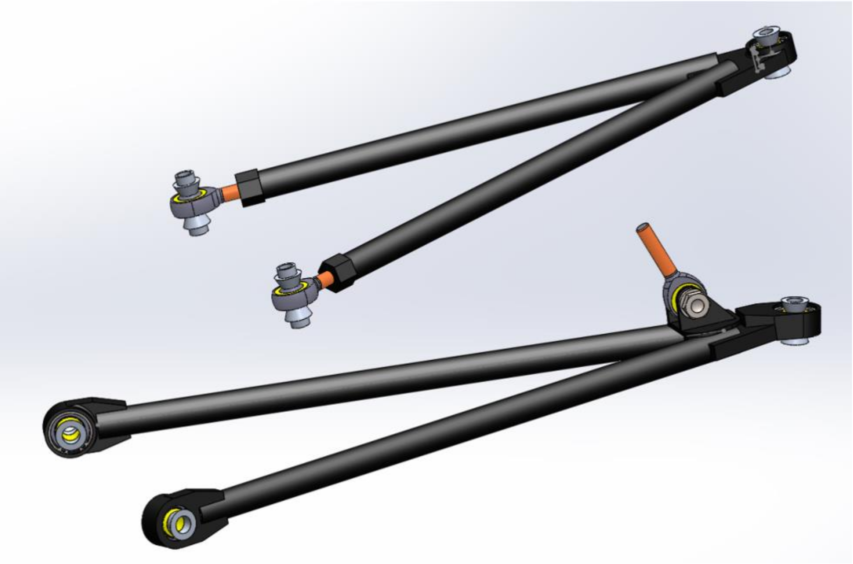
1.

2.

3.

Also the upright side spherical housing thickness was increased to withstand the forces. The lower spherical housing was again changed to 2 sections with one horizontal so it would not interfere with the Upright and the ferrules. (3)

Fronts: final

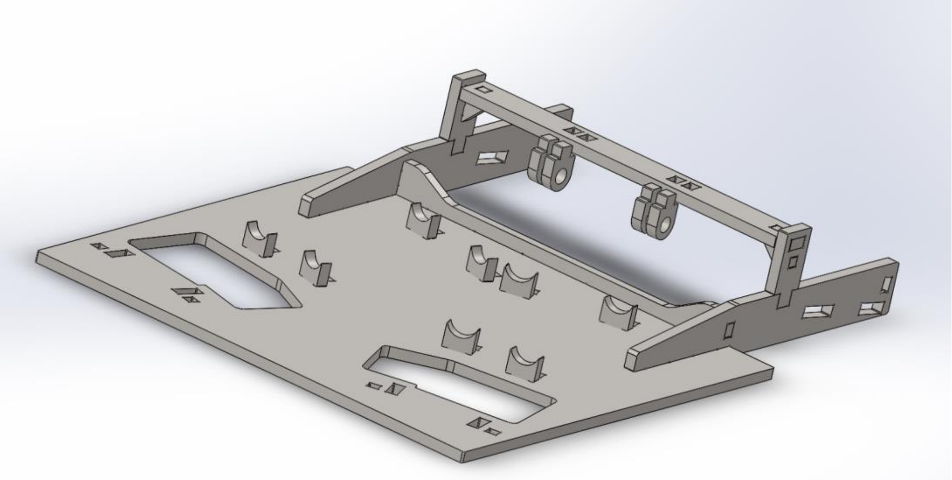


Rears: final

Jigging
We also had to design a manufacturing strategy so our parts could actually be made. A jig was designed so our parts could be put in the proper sections and orientation and then be welded in the correct position and multiple parts could be manufactured with ease. There are 2 separate jigs, one for the top A-Arms designed to they can be flipped over so both sides can be welded in the correct position. The 2nd jig is designed for the bottom A-Arms as the angle is most important, it is made up of a plate cut to the correct angle.

Top arms:

Bottom arms:



Market Research

For this section of the report we decided to complete several reviews on designs of parts relating to our part. We did this by all coming together at a set point where we extensively reviewed each other’s parts and by doing this we attempted to identify any areas that the other groups might have overlooked while designing their parts. In total we reviewed 4 different parts. The parts we reviewed were the Gearbox, the Uprights, the Wheel Centres and the Steering Housing.

Gearbox:

When reviewing the gearbox design, we found four corrections and considerations that should be made by the other group when progressing in their design. These were:

Work on serviceability:
A way should be implemented to that will allow easy access to the gearbox to allow for a faster and more efficient way to make repairs or adjustments to the gearbox if needed.
A way to check the oil:
Add something into the design that will allow for an easy way to check the oil.
Optimise the number of bolts:
Try to optimize the number of bolts to reduce the weight of as much as possible.
Jigging:
Start thinking about jigging.

The number of bolts required is very important as you don't want to many as they will drastically increase your weight and you don't want too few to hold the gearbox together. Also a way to check if the oil height is correct will be very helpful in actually running the car instead of just guessing if the right amount is in there.

This related to our part by the gearbox having to start thinking about the jigging of their parts. It is a completely different way in thinking to designing the part as all parts need to be constrained and it is very important as small errors can make the part incorrect.

Wheel centres:

When reviewing the wheel centres design we found four corrections and considerations that should be made by the other group when progressing in their design. These were:

- Lattice structure will be nice for 3D printing
- Confirm bolting or gluing
- Work on 3D titanium printed layup
- Aluminium spacer decreases friction but is there another material

As this is the first time that 3D printed titanium is used within the team it is very important to check if the layup is correct as you don't want to make a mistake. Gluing or bolting is also another thing that needs checking. As glue is weakest in peel and strongest in shear and bolting is weakest in shear and strongest in tension, it would be a good idea to use both glue and bolts as they cover each other's weaknesses. Also to check if there are any other materials that bond better to titanium other than aluminium to make the most out of the glue.

This related to our part by having to do research on different materials to see which will be optimal for our part.

Steering housing:

When reviewing the steering housing design we found four corrections and considerations that should be made by the other group when progressing in their design. These were:

- Not enough space on the front joint for welding
- May be too much deflection in front panel, maybe widen
- Move the rectangle hole
- Plan for welding

The problems for welding are a major part as if it can't we welded correctly it won't be made correctly and then it won't be as strong as it is designed. The rectangle holes on the front panel can not be there as they create major stress concentrators. They should be replaced with triangular holes. And the deflection can still be reduced as it was a major design consideration.

This related to our part by having to research the welding required and the proper preparation and techniques needed to do it correctly.

Uprights:

When reviewing the upright it was vital for as to pay close attention to see if anything was missed since this part is directly connected to our part. When reviewing the uprights design we found six corrections and considerations that should be made by the other group when progressing in their design. These were:

- Confirm space for brake calliper
- Think about the jigging when 3 axis machining
- Bearing shear forces
- Reduce weight
- Reduce deflection
- Think about bolt length for attaching clevises

The space around the brake caliper is very important as the brake caliper that is in CAD is not the correct size for the actual one that is on the car. This needs to be checked as if it is incorrect they need to design whole new uprights or buy 4 brand new brakes that will fit and it will be a very expensive exercise. The jigging is also very important as you need to know how many re clamps are needed and with each reclamp your accuracy drops dramatically. So the less re clamps the better. The bolt lengths had to be checked as well so they could make sure that you could fit a wrench into the uprights to be able to do up the bolts. An O Wrench is the best option but a C wrench is acceptable. Reducing weight and deflection is just to optimise the part. The shear forces in the bearing had to be checked as well as this could have a major affect on their design if it could not withhold the forces produced.

This related to our part as the upright and the A-Arms interact with each other the inherently affect each other as well. The bolts that are in question are the ones that bolt the uprights to the A-Arms and they need to be correct or otherwise our parts will not fit onto the car. The jigging also is related to our part as we need to be aware of how to jigg our part correctly so there are no changes or errors produced when making multiple of the same part. .

Response to external Review

This section contains a summary of the issues that were brought to light by the other groups during the external design review done on our initial design. Following this will be an explanation on why these concerns are unwarranted or how we will address the issues if deemed necessary.

- Change spherical housing
- Rotate spherical housing to align with chassis clevis
- Complete FEA’s of all loading scenarios
- Fillet the inner triangle as it will act as a stress concentrator
- Check if angle of the attachment to the chassis is concentric
- Ensure materials to be used are optimal for welding
- Make sure all documents and files are saved with correct naming convention
- Plan out all required jigging
- Potential clearance problems where a-arms connect to uprights. Ensure angle is large enough to avoid this
- Check all welds are in shear
- Check all welds have enough material to weld
- Check heat affected wrapping with whoever is welding it
- Do tests with connections not perfectly aligned to simulate inaccuracies in manufacture

Actions taken to solve these issues:

We took on all the feedback from all the other groups and we changed the spherical housings so they were adjusted to align with chassis clevis and further changes to the design have been changed to allow for much easier machining that only needs waterjet cutting and no reclamping for machining. Multiple FEA's for each load scenarios have been completed to ensure that all scenarios that the cor could possibly be under. The major cases were breaking and turning, pure braking, accelerating and turning and pure accelerating have been fully analysed. All of the spherical housings both the dynamic and master CAD have been checked so it fits perfectly and concentrically with all parts. We have performed weld calculations and double checked to see if all the welds are in shear and our heat affected zones will not diminish the strength of our parts and we researched our material 4130 to see if it is optimal to be welded. Also we have discussed the jigging and welding procedure with the technicians that will perform it and they have agreed that the A-Arms we have designed will be able to be welded with the current jigg. The problem with the angle of the A-Arms and uprights have been fixed so that now both the front and rear lower spherical housings are 2 parts welded together with one part horizontal to the uprights then the 2nd part on the correct angle for the rest of the A-Arms to be aligned properly. and cut outs (triangles) have been reviewed to have proper fillets and chamfers needed. The file names are all correct in the BOM and the master CAD.

Reflection

At this point our design is in the manufacturing stage and therefore has not been completed yet and with the testing of the car not being due until a later point, a performance evaluation can not be completed yet for our design. Due to this fact we have decided to perform an analysis on our design by looking at the dynamic CAD to see how it will interact with the system and on the FEA's that have been performed on our design.

Evaluation of the design approach:

The approach that we decided to use helped us greatly with the conservation of time during the entire design process. This is due to the fact that each person in the group had their own specific area to work on. This saved us time as each person could focus on their own task and if any member was ahead of the set schedule they could help another member on their task. Doing this ensured that nothing was missed in any of the three sections and therefore helped us avoid the majority of errors.

Advantages of this approach:

- Having the three sections research, calculations and design allowed for changes to be made each time a new discovery was made that would benefit the design.
- Following the schedule and tasks set in the weekly meetings ensured that time was conserved allowing for more in depth research.
- Having people be ahead of schedule helped other sections to advance more quickly in their task.

Possible improvements to the approach:

- Add a time for group members to consult about their findings and to clear up any confusions
- make sure that all weekly meetings are attended so the whole team knows what the group is up to and to make sure that the assigned tasks have been completed.
- More communication with current and previous team members to make sure our design was optimized to the fullest degree and more interaction with not only ts_15 but ts_14 and other groups that had done A-Arms before so they could help us avoid the major and easy mistakes made at the beginning of the project.

By ensuring that these changes are implemented the effectiveness and efficiency of the group would be increased by a great amount. These improvements would allow us to approach the issue even more effectively as issues that we faced throughout the design process might have been identified earlier and therefore have been fixed faster.

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Documentation

Appendix:

FMEA
External Reviews
Design checklist
Engineering drawings