REV 2011 Formula SAE Electric – Suspension Design

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Project Summary

This thesis covers the suspension and steering design process for REV's entirely new 2011 Formula SAE electric race vehicle. The team intends to utilise four wheel-hub motors endowing the vehicle with all-wheel-drive and extraordinary control over torque vectoring. The design objectives were to create a cost-effective, easy to manufacture and simple race suspension that would act as a predictable development base for the pioneering power train. The ubiquitous unequal-length, double-wishbone suspension with pull-rod spring damper actuation was chosen as the underlying set up.

Much of the design took place during low technical knowledge as none of the team members or supervisors had pervious experience in FSAE. As a result a great portion of the design was based on UWAM's 2001-2003 vehicles as these were subject to similar resource constraints and preceded the complex Kinetics suspension system.

The kinematic design of the wishbones and steering was completed on graph paper while design of the components including FE analysis was carried out in SolidWorks. The spring and dampers where set up for pure roll, steady state conditions. The major hurdle during design was overcoming the conflicting dimension of the electric wheel-hub motor and pull-rod. Most of the suspension components are to be made from Chrome Molybdenum steel (AISI 4130).

Acknowledgements

This project has led me along a difficult and at times frustrating journey, but in the end rewarded me the value of hard work. It elegantly concludes the many enjoyable years I have spent at university, pricelessly contributing to my educational and personal development. It could not have been possible without the support of the people around me. I would like to sincerely thank,

Dr. Adam Wittek for supervising my project and providing me with guidance.

The REV team for their determination and hard work spent on the project; the countless hours spent in discussion, idea generation and organisation.

UWA Motorsport for access to their extensive knowledge and experience, particularly Matthew Day for his expert advice on vehicle dynamics.

Dr. Angus Tavner for helping me overcome issues that were bewildering me.

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Finally and most importantly my family for providing the financial, psychological and motivational support I needed during my years at university.

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1 Introduction

1.1 Background information

1.1.1 FSAE competition

Formula SAE (FSAE) is a competition run by the Society of Automotive Engineers to engage university students to build a small formula style, open wheel race car. Students are given a notional situation of being employed by a design firm to develop a prototype vehicle for the non-professional competition market (FSAE 2011). The competition is held in many locations around the world each year including Australia. Students are to build as much of the vehicle as possible themselves to learn crucial engineering and team skills. The competition compromises of static events where students present details of the design, cost and manufacturing processes and dynamic events that test the vehicles acceleration, braking and handling. A breakdown of the point scoring structure is listed in Appendix 13.2. Originally the competition was only for petrol powered vehicle's of engine capacity less than 600cc but has been expanded to include electric vehicles in the Australasia competition. A special electric only competition 'Formula Student Electric' has been recently introduced in Germany.

1.1.2 REV team

Dr. Thomas Braunl relaunched the Renewable Energy Vehicle (REV) project in 2008 to addresses issues of emissions in transportation. The project has focused on electric vehicles supported by batteries and has thus far converted a Hyundai Getz, Lotus Elise and a Ford Focus to electric propulsion (REV 2011). With motorsport a good ground for technological development, the team in 2010 converted UWA Motorsport's 2001 FSAE vehicle to be powered by two electric motors mounted in the rear suspension cradle but could not participate in competition as the chassis did not comply with current rules. For 2011 the REV team plans to design their first all all-new electric vehicle to compete in FSAE. The revolutionary aim is to employ four wheel-hub motors, endowing the vehicle with all-wheel-drive and unprecedented control over torque vectoring. Currently plans are to use Plettenberg Predator 37 motors that are very compact, very light weight (1.9kg) and produce 15kw of power (Plettenberg 2010). They are to be mounted to a fixed 5:1 reduction gearbox in the orientation shown in Appendix 13.1.

1.1.3 UWA motorsport

UWA Motorsport (UWAM) team was created in 2001 and has since competed in FSAE each year with a new vehicle. It has proven to be at a class leading level winning the Australasia competition twice and the world championship in Detroit, Michigan in 2008 (UWAM 2011). Being an established team with a decade of experience it will be a great source of information appreciably open to the REV team, invaluable considering most outside teams keep their knowledge private. These sources will include past honours thesis papers, SAE papers and personnel expertise.

1.2 Objectives

While providing many benefits, the wheel-hub mounted motors will cause issues for the suspension design because their dimensions and location will conflict with suspension components (mainly the pull-rod) while the extra unsprung mass may be detrimental to handling on an uneven surface. These issues along with those discovered with the 2001 UWAM chassis provide the basis for some of the primary objectives addressed in the thesis.

The primary objectives will be:

- Designing the entire suspension and steering system for REV's 2011 FSAE vehicle,
- Overcoming conflicts of motor dimensions with suspension components,
- Producing a predictable handling car that will be a good development base for testing the electronic drive train system and
- Keeping resource requirements to a minimum.

The final point is particularly critical as the REV team's extremely limited budget of $10,000 plus a $5,000 credit at Altronics is severely short of the estimated $25,000 cost, therefore many design decisions will be cost driven. The team's lack of experience and numbers (only 3 mechanical engineering students and 4 electrical/mechatronics students) will cause difficulties in getting design and construction completed on time, therefore a simple and cost effective manufacturing process must be an important consideration of the design. These difficulties were already realized when the original target of competing in Germany mid-2011 had to be abandoned due to time and financial constraints. Current targets are for the design to be finished by end of June 2011 and the vehicle ready to compete in the Australasia competition in Melbourne, December 2011. Secondary objectives include;

- shortening of wheelbase and narrowing of track (relative to 2001 chassis) to reduce weight and improve manoeuvrability,
- locating rear suspension pivots on the main roll hoop to reduce chassis weight,
- adjustability of toe and camber,
- relocation of the steering rack from in front of the knees (to comply with rules) and
- moving the suspension rockers higher to protect them from damage on the track.

It is important to recognise that the team is building a brand new vehicle from scratch without any team member or supervisor being previously involved in FSAE. Much of the design took place while the team members were learning their respective arts. Hayward (2001) recommends teams in such a position should resort to established methodologies and existing vehicles to base their preliminary design upon during early stages of low knowledge.

2 Literature Review

2.1 Important parameters / Definitions

2.1.1 Vehicle motions

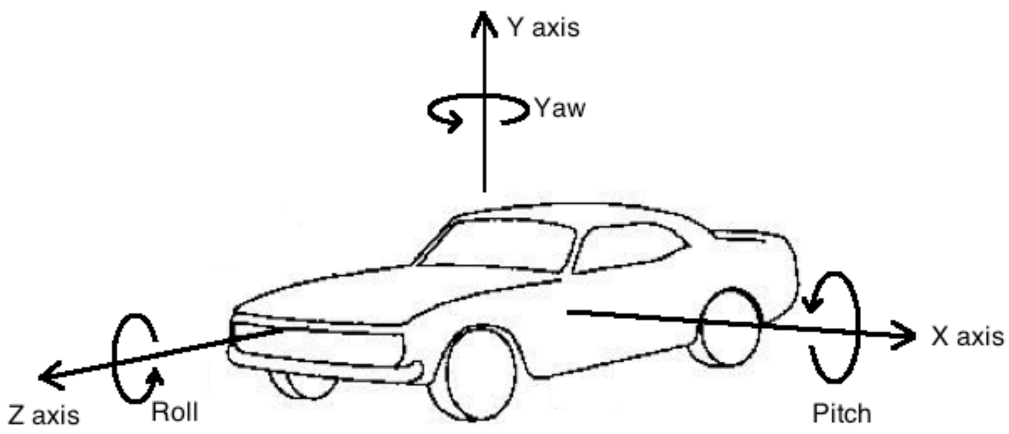

Figure 2-1: Vehicle axis (Milliken & Milliken 1995, p.116)

The following x, y and z axes have been selected for lateral, vertical and longitudinal velocities respectively. These are different to the orientation conventionally used in describing vehicle motions but have been chosen to coincide with the axes used in SolidWorks to thus reduce confusion. Roll is the rotation of the vehicle’s sprung mass about the vehicle’s longitudinal axis usually during cornering. Yaw is the rotation about the vehicle’s vertical axis as a result of the vehicle’s change of direction and pitch is the rotation about the lateral axis usually a result of braking or acceleration.

2.1.2 Tyre contact patch

Portion of the tyre surface in contact with the ground.

2.1.3 Wheelbase & track

Wheelbase is the distance between the front and rear tyre contact patches. Track is the distance between left and right tyre contact patches (Smith 1978).

2.1.4 Ackerman steering

Ackerman is used to describe the principle of the steered wheels requiring to be turned at different angles to follow the same turning radius of the vehicle. As displayed in Figure 2-2 the inside tyre is required to turn a larger angle (STEER 1) than the outside tyre to prevent the tyres from skidding. When the rolling axis of the steered wheels meet on the axis of the non–steered wheels this is termed 100% Ackerman.

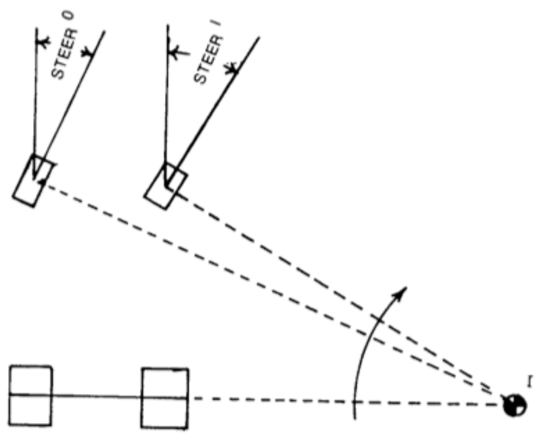

Figure 2-2: Ackerman steering (Smith 1978, p.60)

2.1.5 Camber angle

Camber is the angle between the wheel centreline and a perpendicular line projected from the ground surface when viewed from the front of the vehicle. Camber is considered positive when the top of the tyre tilts away from the vehicle centre as seen in Figure 2-3.

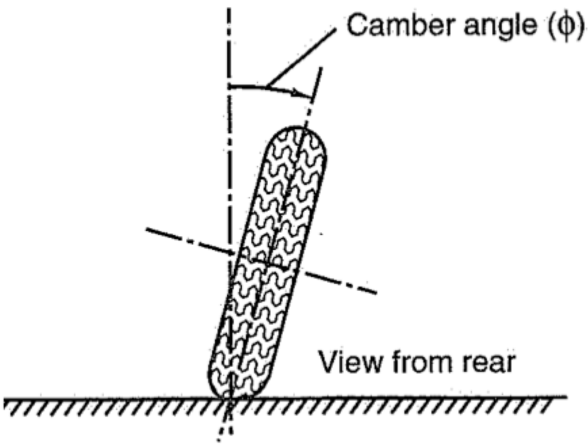

Figure 2-3: Camber angle (Milliken & Milliken 1995, p.47)

2.1.6 Castor angle

Castor is the angle between the steering axis and the wheel centreline (perpendicular to the ground) when viewed from the side. Positive orientation can be seen on the left of Figure 2-4.

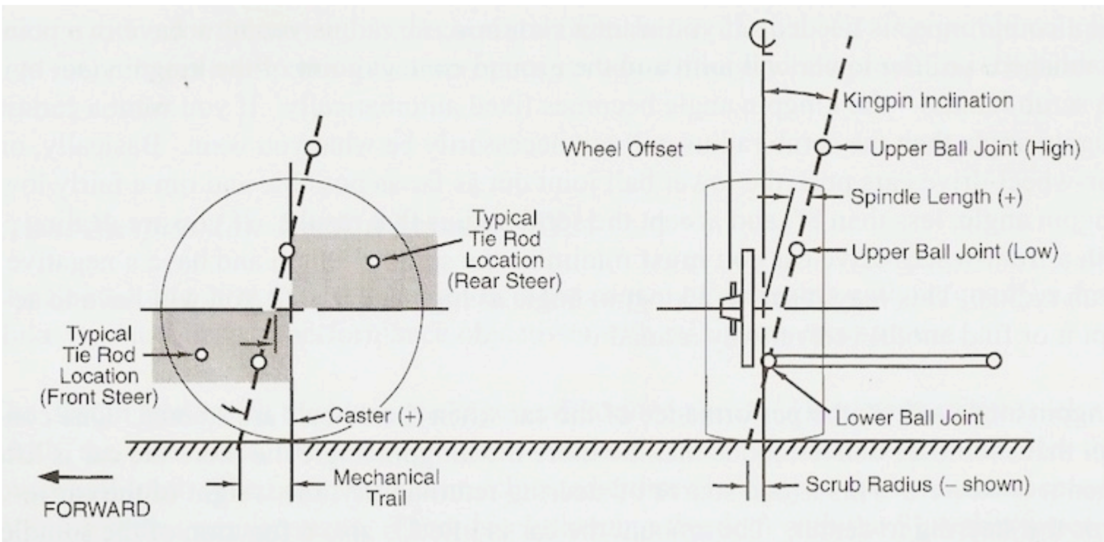

Figure 2-4: Front suspension (Milliken & Milliken, p.625)

2.1.7 King pin inclination /angle

King pin inclination, sometimes referred to as king pin axis is the angle between the steering axis and the wheel centreline (perpendicular to the ground) when viewed from the front (Figure 2-4).

2.1.8 Scrub radius / king pin offset

The scrub radius, sometimes referred to as king pin offset is the horizontal distance between the where the king pin axis intersects the ground and the tyre centre line, when viewed from the front. Scrub radius is negative when the kingpin axis intersects the ground level outboard of the tyre centreline as shown in Figure 2-4.

2.1.9 Toe angle

Toe is the angle between the tyre centreline and the vehicle longitudinal axis when viewed from the top. Positive toe is when the front of the tyres are tilted towards the vehicle longitudinal centre as shown in Figure 2-5.

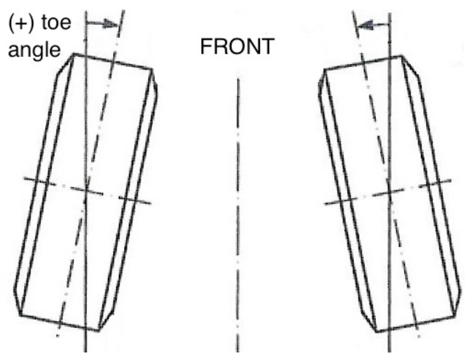

Figure 2-5: Top view of toe angle (Photo bucket 2011)

2.1.10 Anti-pitch geometry

Anti-pitch geometry can be used to reduce certain pitch motions by inclining the suspension link towards the centre of gravity in side view, as shown in Figure 2-6. Commonly anti-dive geometry is used on the front suspension and anti-squat geometry on the rear to offset the effects of braking and acceleration respectively.

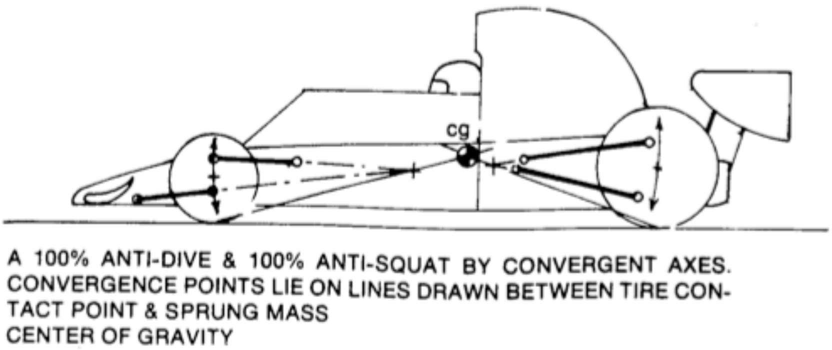

Figure 2-6: Anti-pitch geometry (Smith 1978, p.35)

2.2 Racing suspension

A suspension system's role is to maximise the grip generated from the tyres by keeping them in contact with the road at the optimal angles and forces (Milliken & Milliken 1995). The suspension must therfore allow for vertical travel of the tyres to absorb unevenness in the road while minimising the vehicle's excitation to the four main dynamic modes of roll (rotation about longitudinal axis), pitch (rotation about lateral axis), heave (uniform vertical wheel movement) and warp (non-uniform vertical wheel movement) (Kowalyck 2000).

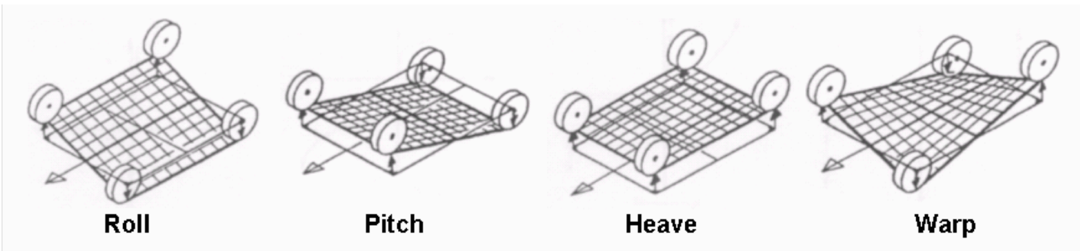

Figure 2-7: Main vehicle dynamic modes (Zapletal 2000)

Since the eighties, modern road racing vehicles have ubiquitously used unequal length, non-parallel double wishbones to connect the wheels to the vehicle body (Staniforth 1999). This set up allows for infinite variability on the theme to achieve any desirable camber curve (change of camber relative to suspension travel) for all conditions, unfortunately not at all the same (Staniforth 1999). Design of a suspension system is perpetual adjustment of conflicting parameters in search of an allusive all satisfying condition that ultimately concludes in the best achievable compromise. As there is no definitive solution to suspension geometry design, sometimes considered more art than science, guidelines have been devised based on empirical evidence (Staniforth 1999).

Totten (2004) deduces performance of a race vehicle can be summarised in one word, tyres. The objective of a race vehicle to transverse a course in the shortest possible time by maintaining the highest average speed (Milliken & Milliken 1995) is achieved by maximising the tyres acceleration, braking and cornering potential (Dradburn 2006). Steering in a vehicle forces the front wheels to rotate and change the direction of the contact patch travel relative to the wheel axis, thereby creating a slip angle that is the main method of generating a lateral force to turn a vehicle. Lateral force increases with increasing slip angle until the tyre's maximum co-efficient of friction is breached and the tire breaks loose. It is important for the slip angle to be communicated to the driver to allow him to know when the tires limit is being approached.

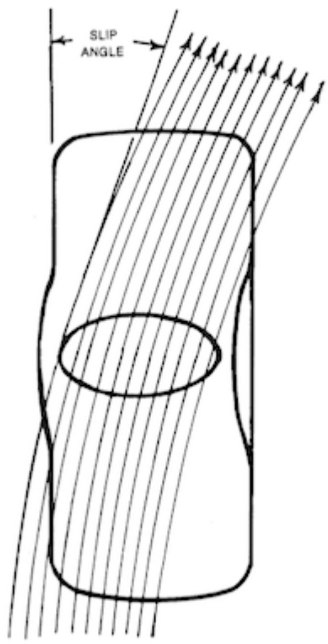

Figure 2-8: Tyre slip angle (Smith, p.5)

A pneumatic tyre's coefficient of friction decreases with added vertical load (Figure 2-9) therefore to maximise total lateral forces generated by the tyres, lateral weight transfer should optimally be minimised.

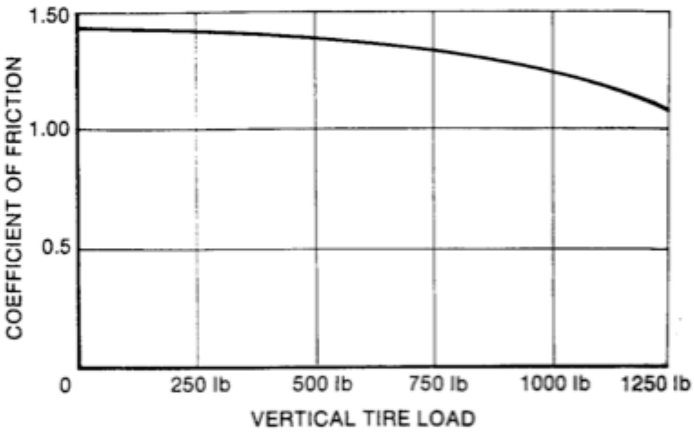

Figure 2-9: Decreasing tyre coefficient of friction (Smith, p.7)

When the lateral forces generated by the front tyres are smaller than at the rear the vehicle understeers (front pushes wide) and oversteers (rear pushes wide) when the front lateral forces are larger (Figure 2-10). Neutral steer occurs when the front and rear lateral forces are balanced, this is often considered the fastest handling around a turn.

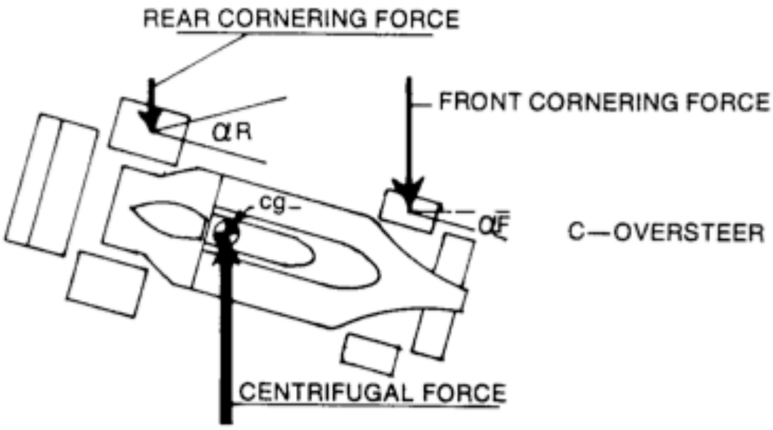

Figure 2-10: Lateral forces when oversteering (Smith, p.4)

The Magic Tyre Formula given in Pacjeka et al. (1993) is often used to calculate tyre forces and moments that can be generated to yield lateral and longitudinal acceleration. The formula requires empirically gathered coefficients that can be found in the 'Formula SAE Tire Test Consortium' from Milliken Research Associates.

Race Car Vehicle Dynamics (Milliken & Milliken 1995) is often considered the bible of race car suspension among the FSAE community. It covers an expansive array of topics ranging from tyre behaviour, race car design, chassis set-up, kinematics, wheel loads and ride and roll rates to name but a few. The vehicle dynamic equations provided in Milliken & Milliken are from the addition of tyre force models to equations developed for modelling the dynamics of high performance aircraft. The low speeds encountered in FSAE means aerodynamic forces can be ignored without major consequence. This design thesis will be primarily based about the design techniques covered in this book.

The design of UWAM's first few vehicles is particularly pertinent to REV's limited resource situation as they were subject to similar constraints. Hayward (2001) describes the design methodologies employed in the development of UWAM's 2001 first vehicle, with focus on reducing the time required to solve suspension variables by using a computational algorithm as opposed to traditional design methods. Reasoning behind basic parameter selection is well explained but does not extend to description of more intricate calculations.

Winzer (2002) provides details about many of UWAM's 2002 vehicle parameters and usefully highlights the improvements made upon the 2001 design. Particular attention is made to the design of the spring/damper actuation with detailed stress and kinematic analysis of the pull-rod and rockers.

Finalyson (2003) investigates the FOX Van RC damper unit that had been used on all UWAM vehicles up until 2003. Even though the FOX damper possessed damping adjustment in both compression and rebound, Finalyson concluded the low speed compression damping characteristic to be unsuitable for FSAE application. Compression damping is adjusted by altering the preload pressure on a spring that throttles flow into a reservoir. It appears the spring remains closed until a certain pressure is reached and than opens, explaining the knee in Figure 2-11. Hence only the knee position can be altered and not the slope. Finalyson suggested changing the bypass orifice to a rotating barrel with various sizes to allow for appropriate control. Being originally designed for mountain bike application the dampers have low compression damping relative to rebound damping Figure 2-12.

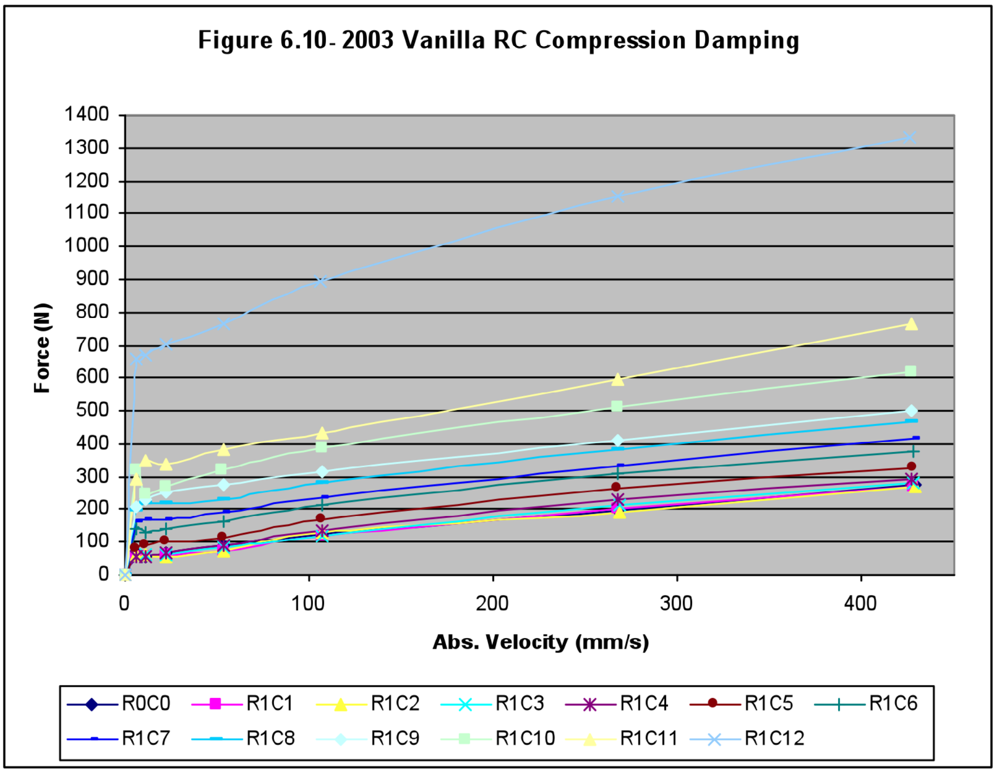

Figure 2-11: Compression damping of Fox Van RC under various settings (Finalyson 2003, p.48)

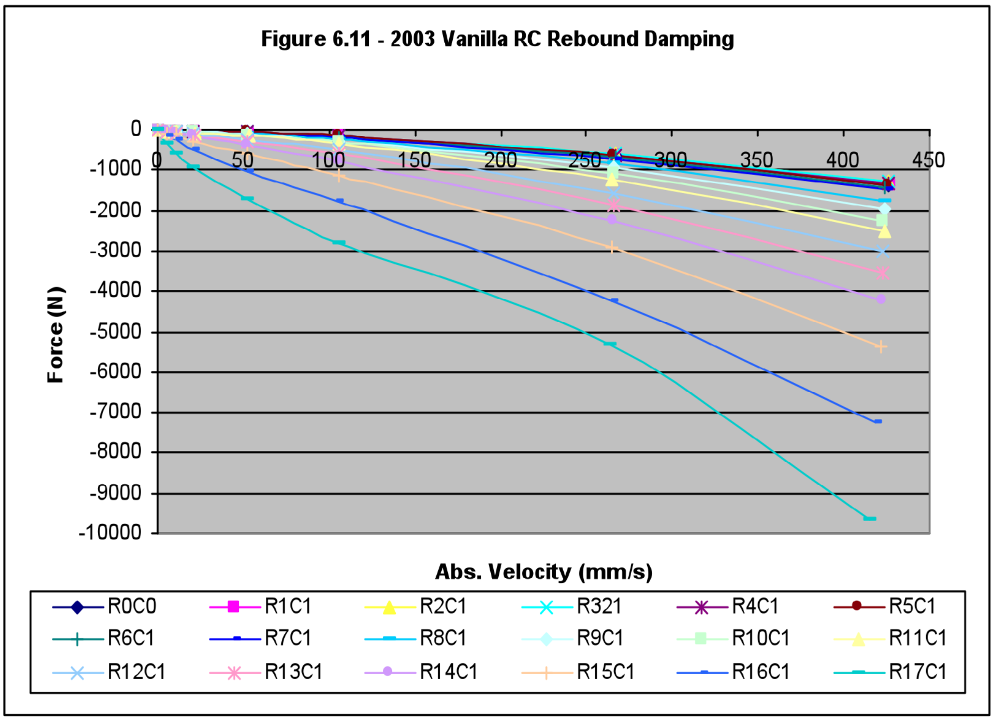

Figure 2-12: Rebound damping of FOX Van RC under various settings (Finalyson 2003 p.48)

The 2003 vehicle was the first time UWAM used Chrome Molybdenum high strength steel (AISI 4130) for its wishbones. Sands (2003) describes the force and strength analysis procedure used in determining the size of the wishbone tubing. Kazmirowicz (2004) further optimises the wishbone tube sizing for the 2004 UWAM vehicle to achieve maximum weight saving. He finds a 50 % weight reduction in unsprung mass will only have a 10% beneficial reduction in vertical wheel response. Kazmirowicz findings suggest the corollary of increased unsprung weight from the electric motors should not have serious adverse consequences on the vehicle dynamics.

2004 was the first year UWAM introduced the Kinetics H2 damping system that decouples the four dynamic modes and allows independent tuning (Guzzomi 2004). Third springs at the front and rear control heave and pitch modes allowing for reductions in wheel rates making the suspension more compliant to warp modes. A unique feature of the Kinetics system is the interconnection of the dampers that allows for control of damping compliance in warp and at the same providing a very large resistance to roll (Chiou 2005). Kinetics issued UWAM exclusive rights to use the H2 system and also assisted its development (Guzzomi 2004).

Another state of the art technology used on recent UWAM vehicles is the carbon fibre composite flexure that replaces the lower a-arms in a double wishbone system. The flexure is designed so that there is no roll moment distribution, improving handling predictability (Davies 2009).

All-wheel-drive (AWD) vehicles possess benefits of better traction in all road conditions, better acceleration in low gears, reduced torque steer effects and even tyre wear (Reimpell et al. 2001). Most FSAE vehicles have been rear wheel drive as the complexity and weight of AWD systems often outweighed the benefits. Most research into AWD vehicles focuses on the differentials used in distributing the power and therefore irrelevant to this thesis. The electronic systems that will be used on the REV vehicle to control power distribution and torque vectoring had not yet been designed at the time of writing.

Critical review of relevant literature is continued in the thesis body.

3 Process – Design Approach:

3.1 Methodology

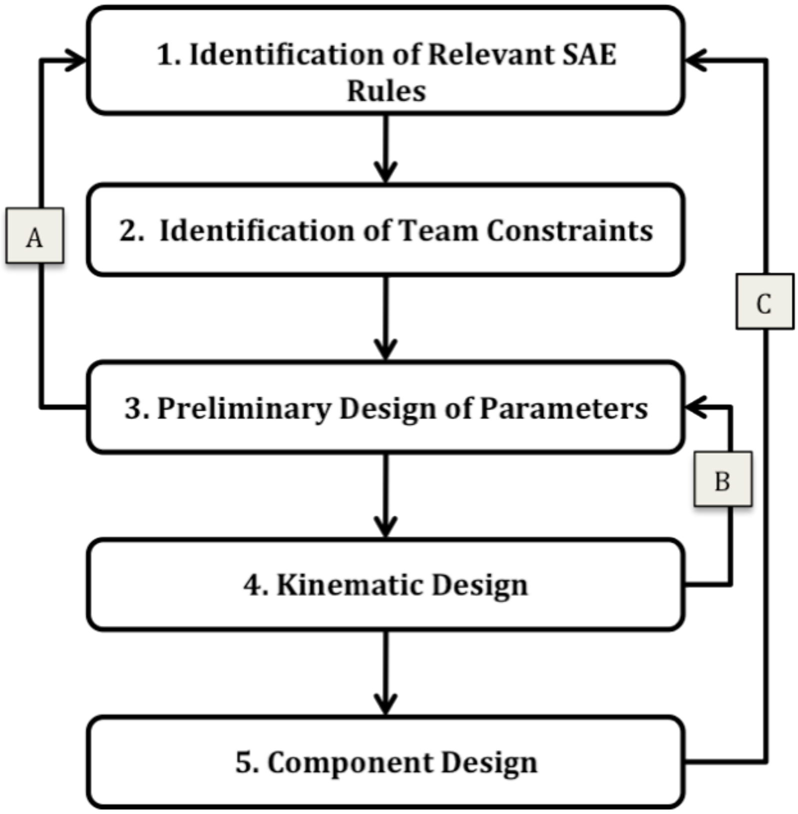

Figure 3-1: Design process flowchart

The design of the suspension system is primarily book based, centred about methodologies covered in 'Race Car Vehicle Dynamics' by Milliken & Milliken (1995). The design approach has been divided into process steps illustrated by the flowchart in Figure 3-1, the first being 'identification of relevant SAE rules' with subsequent steps flowing downwards. The reverse flow arrows on the side indicate a review of whether work to date complies with requirements of the previous step/s. The first three steps in the process are outlined in 'Chapter 10: Race Car Design' of Milliken & Milliken 1995.

3.2 Identification of relevant FSAE rules

Identification of the competition rules relevant to the suspension was the important first step as it dictated the restraints on every subsequent step in the design and ensured the vehicle will be eligible for competition; the ultimate purpose of the REV FSAE project. These rules will be outlined in a table that form a checklist for the review stages.

3.3 Identification of team constraints

It is important to correspond expectations to the available resources (Miliken & Miliken 1995), therefore the next step was determining the team's constraints concerning its financial resources, human resources, team member experience and time availability. These were generated from discussions with the REV team members. The first two steps produced many of the thesis objectives.

3.4 Preliminary design of parameters

The aim of 'preliminary design of parameters' was to establish the arrangement of components as to satisfy packaging requirements, performance targets and weight distribution for the desired dynamics (Milliken & Milliken 1995). This began with collecting information on the components to be used followed by estimating the weight of the vehicle and centre of gravity. The centre of gravity was calculated on an excel spreadsheet.

Some of the major considerations at this stage were wheelbase/track lengths, ride height, type of suspension, roll stiffness, type and shape of overall structure and space for driver. The magic tyre formula (Pacejka 1993) discussed in the literature review could not be used in the design because of the lack of available tyre data. The 'Formula SAE Tire Test Consortium' from Milliken Research would cost the team an unjustifiable $USD500 and can not be shared with other teams. The vehicle will therefore be primarily set up for pure-roll, steady state conditions that do not require such tyre data. Conventional spring dampers will be used as opposed to the sophisticated, state of the art set ups like the Kinematics H2 as discussed in the literature review due to the limited resources of the REV team.

3.5 Kinematic design

Kinematics refers to the study of motion of interrelated parts without the consideration of forces acting on these members (Oxford 2011). In this thesis' context it specifically refers to the trajectory of the tyre corresponding to motions of the suspension members. There are special design programs that can simulate and analyse the kinematics of a suspension system. Two programs that were evaluated for their effectiveness under the constraints of the REV team and this thesis were Optimum K and ADAMS. Optimum K is produced by Optimum G consultants and allows the user to input various suspension types (including double wishbones) and their pivot points to than simulate roll, pitch, heave and/or steering angle, individually or simultaneously (Optimum G 2011). UWAM currently use this software but possess only one license and therefore can not be shared with the REV team. Optimum G offer the program at special student price of $US395. ADAMS produced by MSC Software is a more complete analysis software package being able to perform finite element analysis in addition to kinematic analysis. ADAMs however is significantly more expensive than Optimum K (Chau 2008). From discussions with UWAM's specialist on vehicle dynamics, such programs were judged to be too time consuming for a first year team, especially with only one person responsible for the entire suspension. It was recommended to focus on getting the general parameters correct following established methods as this would still yield a good result. The aforementioned tools are more effective for experienced teams to fine-tune variables like reduction of roll centre movement.

Matlab was investigated for its potential to simulate specific parameters and a program was written to simulate the camber curve. However it was later discovered that SolidWorks has the ability to graph many of the desired parameters using the Motion Analysis feature and was therefore used in verifying much of the kinematic design. SolidWorks is a 3D CAD program that will be used by the REV team for the design of the entire vehicle, including the forthcoming component detailed design of this thesis. Much time was saved and redirected to other areas by not having to input data into different systems or write new programs. The kinematic design will be further separated into wishbone and steering design.

3.5.1 Wishbones

The kinematic design or geometry design of the wishbones follows the procedure explained in chapter 17.5 Front Suspension SLA (Short-Long Arm) (Milliken & Milliken 1995) utilising instantaneous centres. The following procedure is related to Figure 3-2.

1. Establishing the ground level, wheel centre line and chassis centreline in front view.
2. Add a vertical line representing the virtual swing arm length (line A-A) of distance FVSA (Front View Swing Arm) from the tyre centre.

3. Establish the roll centre height (RCH) on the centre line and extend a line from the tyre contact through the roll centre to the line A-A. This intersection is now the instantaneous centre (IC).
4. Connect lines from the upper ball joint (UBJ) and lower ball joint (LBJ) to the instantaneous centre.
5. Choose wishbone arm lengths to get inner pivots.

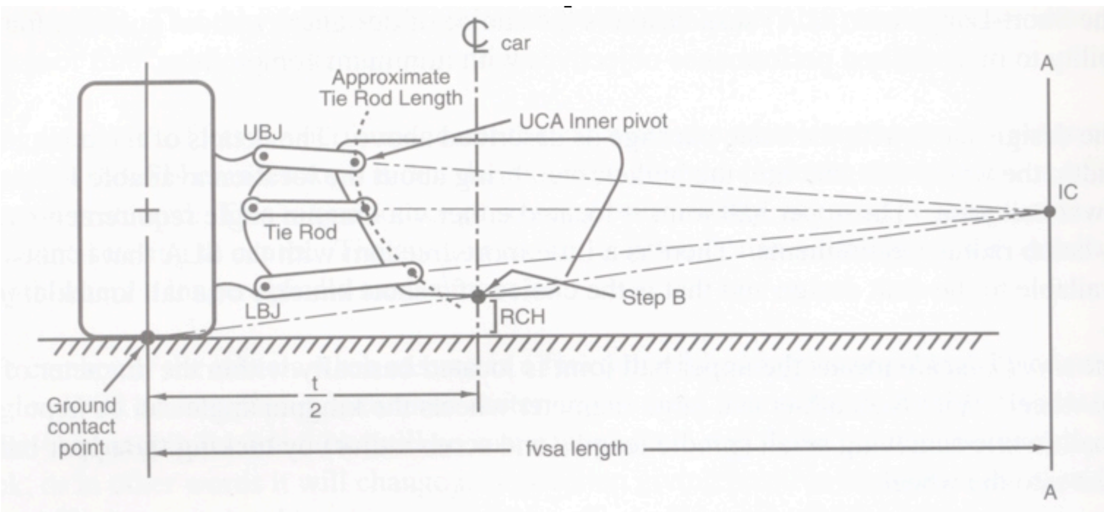

Figure 3-2: SLA Kinematic Design procedure (Milliken & Milliken 1995, p.628)

The camber curve was verified using the Motion Analysis feature in SolidWorks by graphing the roll of the wheel relative to its Y displacement.

3.5.2 Steering

The steering system pivot locations were determined using the procedure outlined in 'Chapter 19.2 Ackermann Steering Geometry' that produces a good approximation of 100% Ackermann. Firstly, lines are drawn from the kingpin axis to centre of the rear track (in top view) (Figure 3-3). The tie rod pivots are than located on these lines when the steering rack and tie rods are parallel to the front track. When the steering rack is behind the kingpin axis this is called rear steer, but can also be located in front of the kingpin axis. Tie rod length was determined by the ratio of the steering rack, rack travel and inside wheel radius.

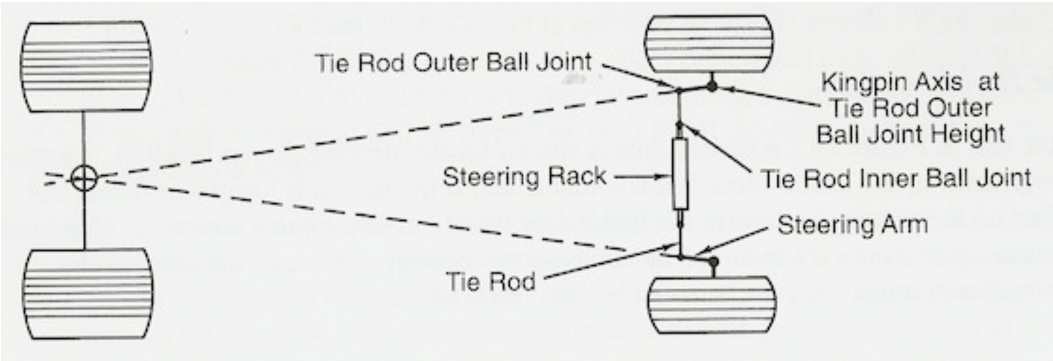

Figure 3-3: Ackerman steering geometry (Milliken & Milliken 1995, p.714)

Ackerman percentage can be altered by moving the rack backwards or forwards so the tie rods are no longer parallel with the steering rack (Figure 3-4). In the case of rear steer, moving the rack rearward will increase Ackermann. This procedure was deemed accurate enough for a first year vehicle although simulation software in the future would be useful in verifying and fine-tuning the result.

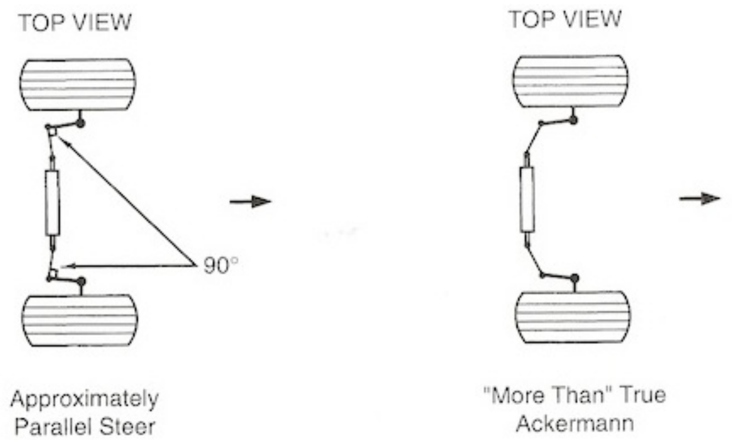

Figure 3-4: Alteration of Ackerman (Milliken & Milliken 1995, p.715)

3.6 Component detailed design

3.6.1 Material selection

This section began with selection of the material to be used by comparing the advantages and disadvantages (in respect to the team constraints) of the various materials currently used by FSAE teams for their suspension components. Material was selected for the wishbones, wishbone brackets, rockers and rocker actuators.

3.6.2 Bearing selection

The suspension pivot mechanisms were selected to satisfy the expected forces to act through the suspension. As the competition course will be predominantly flat, maximum design forces are based on the largest forces experienced during cornering and/or braking. A large safety factor of 4 is used to ensure reasonable heave and warp motions caused by running off the track or hitting curbs will not damage the suspension. Additionally, the wishbone’s expected angle range must not exceed the bearing’s maximum misalignment angle.

3.6.3 CAD design

The 3D CAD modelling began by inputting the suspension pivot locations determined during the Kinematic Design (Chapter 7) into SolidWorks using points on a 3D sketch. Using the points, lines representing the wishbones and bearing housings are drawn to form the suspension reference file. A weldment member feature generated the shape of the wishbones a-arms and bearing housings. The trim feature and fillet feature were used to correct the model. Each wishbone was saved as a separate part to later form the components of a moving assembly. The steering mechanism was similarly modelled. A part was created to represent the dimensions of the wheel and motor-hub assembly. Front right and rear right suspension corner assemblies were formed only as the suspension system is symmetrical along the longitudinal centreline. The Motion Analysis feature was than used to graph the camber, bump steer and steering.

Next the damper/spring mount, rocker mount and rocker actuator mount were added to the suspension reference file. Next various rocker dimensions were experimented with to attain the necessary installation ratio. A program was expected to be written in

Matlab to model the installation ratio but instead it was calculated from measurements taken in SolidWorks. The vertical distance of the wheel from the ground (W) (below ground is negative) and the length of the damper eye to eye (D) were recorded for eight instances of the suspension ranging from its full rebound to bump positions. The installation ratio was calculated using Equations 3.1 – 3.4 and than plotted.

$$W_{n+1} - W_n = \Delta W_n \qquad \text{Equation 3.1}$$

$$D_{n+1} - D_n = \Delta D_n \qquad \text{Equation 3.2}$$

$$IR_n = \frac{\Delta D_n}{\Delta W_n} \qquad \text{Equation 3.3}$$

$$W_{n(ave)} = \frac{W_{n+1} - W_n}{2} \qquad \text{Equation 3.4}$$

$$plot(W_{n(ave)}, IR_n)$$

Throughout the component design stage the components were checked for interference with other members. This included checking the wishbones for contact with the wheel rim under full bump and rebound conditions, under full steering lock and crucially the rocker actuator with the electric motor.

3.6.4 Strength analysis

FE Analysis would only be conducted on the rockers due to time constraints. The pull-rod could be analysed with simple equations as a tube under tension (Equation 3.5) whereas the wishbone's design would base tube diameter selection on past UWAM vehicles. FE Analysis of the rockers would be carried out in SolidWork's 'Express Simulation'. As the rocker is constructed of two sheets connected at the rocker's pivot (stresses are not expected to be major at the pivot), the analysis would be carried out on one of the sheets with half the forces. In the FE analysis the forces from the actuator and spring/damper would act on the hole faces and the rocker pivot would be a fixed constraint. Acknowledging, the rocker pivot should have been modelled as hinge but due to problems getting the analysis to the run as rigid body motion, the fixed constraint had to be used as a compromise. This would produce artefact stresses around the pivot that could be ignored as the largest stresses were not expected around this area.

$$\sigma = F / A \qquad \text{Equation 3.5}$$

4 Identification of Relevant FSAE Rules

The identification of relevant rules stage reviewed the '2011 Formula SAE Rules' (FSAE 2011), 'Formula SAE-A 2010 Addendum to Formula SAE 2010 Rules' (FSAE-A 2010) and '2010 Formula Student Electric Germany Rules' (Student Electric 2010) but only the first was found to cover matter pertinent to suspension design. The rules identified as relevant are listed in Appendix 13.2. The following section is divided into a summary of the technical requirements and discussion of the static and dynamic event objectives.

4.1 Technical requirements

Section A6 (FSAE 2011) stipulates competition vehicles are to be developed without the direct involvement of professional engineers and built as much as possible by the students. Section B outlines the technical requirements of the vehicles. The minimum wheelbase may be 1525mm while the narrower track of the vehicle cannot be less than 75% of the wider track. Non-crushable objects must be rearward of the front bulkhead. The vehicle must have an operational suspension system with at least 50.8mm of travel with all suspension pivots visible (covers may be removed). There is no minimum ground clearance but vehicles may be disqualified if their chassis touches and damages the track. Wheels must be at least 203.2mm (8 inches) in diameter and both dry and wet tyres are permitted. Dry tyres may be any size or type while wet tyres must have a tread of at least 2.4mm and must be grooved by the manufacture. The steering system must affect at least two wheels, is allowed to have a maximum of 7° of play and have positive stops to prevent the tyres or wheels from touching the suspension components. The vehicle must have adequate rollover stability and will be tested on a tilt bed at 60 ° with the tallest driver seated. Fastener grade requirements specify all fastener's in the driver cell must meet Metric Grade 8.8, SAE Grade 5 and/or AN/MS specifications, all critical fastener's on the suspension system must utilise positive locking mechanisms, all adjustable rod ends must be secured with a lock nut and lock nuts must have at least 2 full threads projecting from them. Spherical bearings must be in double shear or captured by a bolt/washer that has an outside diameter larger than the inside diameter of the bearing housing.

4.2 Static events

The static events consists of cost & manufacturing, presentation and design with the point allocation listed in Appendix 13.2 (Table 13-2). These events will reward teams that employ a lean manufacturing process, cost effectiveness, good engineering principles and show a good understanding of their vehicle.

4.3 Dynamic events

4.3.1 Acceleration

This event will require the vehicle to transverse 75m of flat pavement as quickly as possible from a standing start. Petrol vehicles generally reach a top speed of about 100kph, with traction off the line a governing factor of performance. Undesired suspension movement like squatting during acceleration can reduce traction on a vehicle with large camber compensation, however this will not be of major concern for the REV vehicle with AWD.

4.3.2 Skid-pad

The skid-pad event tests the vehicles cornering ability on a flat surface by completing a figure eight around two constant radius turns about two circles of inner diameter of 15.25m. This will largely place emphasis on the suspension system to provide optimal camber of the tyres to produce maximum grip during steady state cornering.

4.3.3 Autocross

The autocross event will test the vehicles maneuverability and handling while it individually negotiates a tight course requiring high performance of acceleration, braking and cornering. The average speeds will be between 40km/h to 48km/h with straights no longer than 60m, emphasizing the vehicle will spend most its time cornering. The tightest hairpins will have a minimum radius of 9m outside diameter, which can be considered a minimum turning circle for the vehicle.

4.3.4 Endurance

Unlike the autocross event, endurance will be run with other competitors on the track and will test the performance of the vehicle along with durability and reliability. Average speeds will range higher from 48kph to 57kph with a top speed of approximately 105kph. A predictable handling car that doesn't deteriorate in performance will be desirable to complete the 22km distance. Points will be awarded based on the shortest times but this will probably be at the expense of the economy score which is calculated from the fuel or energy used during the endurance.

4.3.5 Points allocation

As can be seen from Table 13-2 (Appendix 13.2), the endurance event combined with fuel economy makes up the nearly 60% of the dynamic event points, hence should be the main focus of the objectives. This means a reliable and durable vehicle is the first design priority. Secondly the vehicle suspension should be primarily designed for maximum cornering performance as most its time will be spent on a tight twisty circuit with traction being a more limiting factor in lateral acceleration than longitudinal acceleration for the AWD REV vehicle.

5 Identification of Team Constraints

Many of the team constraints have been discussed in the Objectives (1.2) section under the Introduction.

6 Preliminary Design of Parameters

6.1 General

6.1.1 Type of suspension

An unequal length, double-wishbone suspension system was chosen based on its ubiquitous use in FSAE competition as it provides very accurate control of the tyre camber during suspension travel while being lightweight.

6.1.2 Chassis overall shape

A steel space frame chassis was selected by other team members assigned to chassis design. This choice was based on the primary performance criterion of rigidity, cost effectiveness and ease of manufacturing plain steel. For a space frame structure it is recommended loads to be transferred into the nodes as these are strongest points (Costin & Phipps 1971). The inner suspension pivots and spring/damper unit mountings should be located on chassis nodes.

6.1.3 Wheelbase

The shortest wheelbase possible would be desirable due to the increased manoeuvrability it instils on a tight track like that experienced on autocross and endurance circuits at FSAE competition. Rudimentary estimates of fitting the driver within the wheelbase rule minimum of 1525mm suggested it could cause packaging complications (especially for a first year team without finalised component selection like the pedal box) so a slightly longer wheelbase of 1600mm was selected to introduce a margin for later design compatibility.

6.1.4 Track

Most FSAE teams employ rear wheel drive that reduces the rear tyres lateral grip under acceleration, hence its beneficial for these teams to have smaller weight transfer across the rear tyres. A narrower rear track resists a smaller portion of the roll (front and rear roll stiffness being equal) that is one of the reasons why teams have a 3.5% narrower rear track on average (Winzer 2002).

A track width of 1200mm was selected as a compromise between the benefits of reduced weight transfer from a wider track and the tighter travel path about a chicane a narrower track allows. Both front and rear track were made equal as AWD vehicles are recommended to be symmetrical (Milliken & Milliken 1995) and secondly the REV vehicle should not suffer from reduced lateral grip on the rear tyres like a RWD vehicle. A reduced rear to front track ratio was considered but not selected at this stage because; a narrower track at the rear would cause difficulties with the pull-rod coming into contact with the wheel-hub motor or a wider front track would increase the travel radius distance needed around a cone.

6.1.5 Ride height

An initial ride height of 50mm was chosen to provide sufficient ground clearance and prevent the bottom of the chassis from hitting the ground under full bump and maximum braking. Chassis design alterations that raised the lowest point of the chassis at the front allowed the ride height to be reduced to 30mm as now the concern of the front of the vehicle bottoming out during braking was mitigated.

6.1.6 Weight distribution/centre of gravity

Estimates of the all the significant component weights and their locations are made in Appendix 13.3. Z represents the longitudinal distance of the item from the front bulkhead and Y the vertical distance from the bottom of the chassis. These estimates yield a weight of 310kg for the vehicle including a 70kg driver, a centre of gravity height of 274mm from the ground (224mm from the chassis bottom with a 50mm original ride height) and a weight distribution of 45:55 front to rear. Both the weight and centre of gravity height are on par with 2001-2003 UWAM vehicles statistics (Table 6-1). It was feared the heavy battery pack and numerous motors would produce a much heavier vehicle than a petrol powered one, fortunately this is not the case. The rear heavy weight distribution is not the ideal 50:50 split (Milliken & Milliken 1995) although not outside the acceptable range for a first year vehicle. The centre of gravity estimate indicates the vehicle will past the tilt test, being able to be angled 68.34° before tipping over (Figure 13-3: Tilt test angle).

UWAM Vehicle	**Weight (kg)**	**Weight distribution (F:R)**	**Roll Stiffness (kg.m/rad)**	**Roll Stiffness Distribution (F:R)**
2001	315	45:55	3085	55:45
2002	315	50:50	2900	57:43
2003	300	43:57	2597	51.5:48.5
**Values from Winzer (2002) & Finalyson (2003)*				

Table 6-1: Roll Stiffness of 2001-2003 UWAM vehicles

6.1.7 Expected performance

Predictions about the vehicles performance are necessary to perform forthcoming calculations. Values were based upon performance achieved by inexperienced FSAE teams as it is unreasonable to expect REV's first vehicle to be able to match performance benchmark teams like UWAM. The strongest acceleration usually experienced in a vehicle is the braking as this is primarily limited by the grip of the tyres. Braking is expected to max at 1.5g (Hayward 2001). Under steady state cornering, 1.2g of lateral acceleration is expected which is smaller than braking partly due to the track being narrower than the wheelbase. Straight line acceleration will likely be limited by the torque of the electric motors even with AWD as there will only be one ratio. Based on the 5:1 step down ratio of the gearbox and maximum power of 15kw per motor at 5000 rpm, acceleration expected to max at 0.9g, but has been revised up to 1.1g in case the motors are able to produce short bursts of extra torque.

6.1.8 Wheels

Most FSAE teams use 10" or 13" outside diameter wheels. The smaller 10" wheels would benefit from a smaller inertia thus requiring less energy to accelerate it, however the 13" wheel were necessary to provide the space to fit the electric wheel hub motors. No specific wheel manufacture or model has been chosen as of yet, but Keizer Wheels offer 13" rims suitable for FSAE application with any offset desirable (Kiezer Wheels 2011).

6.1.9 Tyres

Goodyear D2696 tyres are specifically designed for FSAE application and feature a new compound that heats up more quickly than the older D2692 model (Goodyear 2011), a previous issue for UWAM according to communications made with the team. These tyres were selected because it was originally thought UWAM would be in possession of comprehensive data on these tyres which REV could also use. It later became apparent their data was purchased from Milliken Research Associate's Tire Test Consortium and they had entered into agreement not to on supply this information. Purchase of this information requires a $US500 contribution to the consortium. The tyre data was deemed not to provide enough value for the REV team considering its severely limited financial resources and relatively basic analytical set up of the spring damper units. The spring rate of the tyres was instead assumed to be 18367.97kg/m, equal to the Goodyear Eagle 20.0x6.5 13” from 2002 (Winzer 2002).

6.1.10 Anti-pitch geometry

No anti-dive or ant-lift geometry was incorporated into suspension design. This significantly reduces the complexity by allowing the kinematic design to occur in 2D and follows the ethos of keeping the design simple. The low centre of gravity of the vehicle means any effects of such geometry would not be greatly discernable. Furthermore adding any anti-pitch geometry means the suspension will be prone to act in a contrary manner when an opposite torque is applied. For example anti-dive geometry in the front suspension will make the vehicle prone to lift from the torque of the front motors. Not including anti-pitch geometry eliminates any such un-intended behaviour.

6.2 Dampers

As discussed prior in the literature review, the FOX Vanilla RC spring dampers originally designed for mountain bike application were found to have inappropriate damping characteristics for FSAE (Finalyson 2003). However the author does not wholly agree with Finalyson’s highly critical inference. It would have been in Finalyson’s interest to highly criticise the FOX dampers to justify the need for Kinetics system he was implementing. High initial compression damping characteristic should make a vehicle more responsive. Secondly it is appropriate that the compression damping is lower than the rebound damping because the dampers main role is to allow the suspension to quickly compress under a road surface bump and to than dissipate energy absorbed by the spring in rebound (Matschinsky 2000).

Even disregarding the author’s argument, the FOX Van RC 7.5” x 2.0” (Vanilla RC replacements) would have still been chosen because the advantages of:

- Low cost ($USD310 (eBay 2011)) relative to other dampers; for example Ohlins TTX25 FSAE are $US610 (Motorsport Spares 2011) and Penske FSAE Shock – Double Adjustable 50mm stroke are $USD675 (Kaz Technologies 2011). An extra cost of $USD1200 and $USD1460 respectively for a set of four.

- Reliability; there has be complaints regarding reliability of other dampers like those from Risse Racing (FSAE Forum 2008) whereas FOX have been commendable for their reliability.
- Contingency plan; in the worst case scenario of the team running out of money they may transplant the spring/damper units from the 2001 chassis and make a saving of over $USD1240.

The specified model has an eye to eye distance of 200mm and 50mm stroke.

6.3 Roll Dynamics

6.3.1 Anti-roll bars

No anti-roll bars were included in this design due primarily to extra layer of complexity this would add to the design and the extra strain it would place on the small team during construction. It was deemed more effective to spend time getting the other areas of the suspension correct.

Anti-roll bars are used by most experienced teams, although the 2001 and 2002 UWAM vehicles did not feature any and it is arguable that the effects of an anti-roll bar would be limited because of a FSAE vehicle's very low centre of gravity and correspondingly small roll angles. Secondly anti-roll bars do not reduce the amount of lateral weight transfer, they are merely a tuning tool used to adjust handling by varying the portion of lateral weight transfer on the front relative to the rear.

6.3.2 Roll centres

The roll centre is an imaginary point on the transverse plane of the front or rear tracks about which the sprung mass rotates. For a vehicle it's more accurately the roll axis (the line between the front and rear roll centres) that the vehicle rotates about (Aird 1997). In a force sense it's the point which any lateral load acting on it will not produce any roll of the vehicle's sprung mass.

The height of the roll centre affects the coupling distances (or moment arms) connecting it to the centre of gravity and the tyre contact patch. A high roll centre reduces the roll of the vehicle by reducing the moment arm of the centrifugal force acting on the centre of gravity during cornering. A high roll centre however increases the moment arm to the lateral force acting at the tyre contact patch creating undesirable jacking of the sprung mass, lateral tyre scrub (makes the vehicle jolt to the side when hitting a bump mid-corner) and inducing forces to pass through the wishbones rather than through the spring (Aird 1997). Modern racing vehicles usually have a roll centre between once inch (25.4mm) below the ground and two inches above ground (50.8mm) (Staniforth 1999). It is recommended that the higher roll centre be located at the heavier end of the vehicle (usually the rear) as this will result in a more stable vehicle by reducing the load transfer at this end (Staniforth 1999). In consideration of these points and comparison of 2001-2003 roll centre heights (Table 6-2) a roll centre of 25mm at the front and 50mm at the rear was selected (above ground).

Key Parameters	2001 front	2001 rear	2002 front	2002 rear	2003 front	2003 rear
Swing Arm Length	2000	2000	2300	1350	3000	1200
Roll Centre Height	25	50	28	56	28	46
Scrub Radius	50	50	30	50	36	72.46
Lower Arm Length	520	586	482	509	435	410
Upper Arm Length	400	393	355	403	309	297.25
upper/lower arm length	0.769	0.67	0.737	0.792	0.71	0.66
Upright Length	240	240	220	220	239.4	250
Vehicle Track	1350	1250	1300	1250	1200	1150
Caster	6	6	6	6	7	NA
Camber Properties						
Camber due to roll	0.95	0.98	0.85	1.38	0.6	1.37
Steering Camber	-1.25	NA	-1.25	NA	-1.5	NA
Static Camber	-2	-1	-2	-1.5	-0.4	-2
Total Camber	-2.3	-0.02	-2.4	-0.12	-1.3	-0.63

Table 6-2: Suspension parameters of 2001-2003 UWAM (Finalyson 2003)

6.3.3 Roll gradient

A roll gradient (θ) of 1.5 deg/g was selected as recommended by Milliken & Milliken (1995) for hard racing vehicle suspensions.

6.3.4 Roll stiffness

The roll stiffness (K_ϕ) is described as the roll moment per degree of chassis roll. The roll moment is generated by the lateral force acting at the centre of gravity about the kinematic roll centre. Roll stiffness is determined by Equation 6.1 and Equation 6.2 from Finalyson (2003). H is the vertical distance between the centre of gravity and the roll axis and m represents the mass of the vehicle.

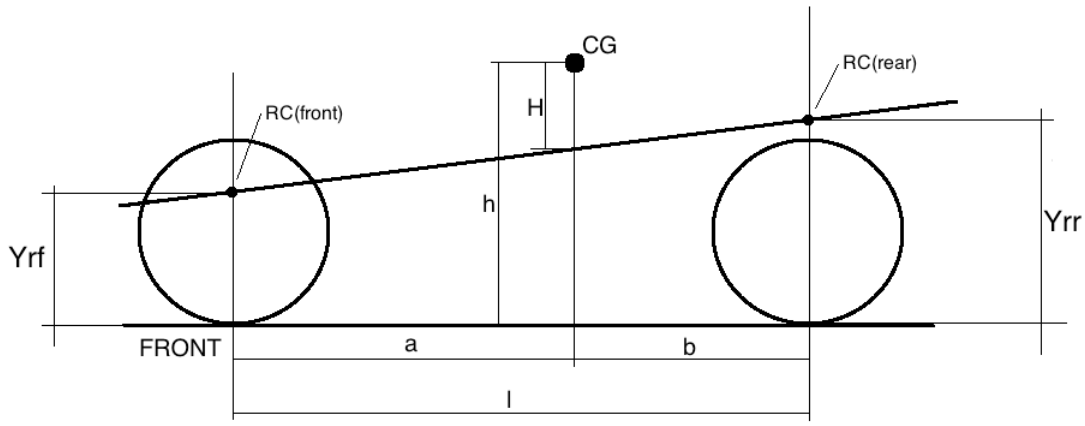

Figure 6-1: Roll axis

$$K_{\phi} = \frac{m \times H}{\theta} \qquad \text{Equation 6.1}$$

$$H = h - (Yrf + \frac{a}{l}(Yrr - Yrf)) \qquad \text{Equation 6.2}$$

$$H = 0.274 - (0.025 + \frac{(0.55 \times 1.6)}{1.6}(0.05 - 0.025)$$

$$H = 0.235m$$

$$K_{\phi} = \frac{310kg \times 0.235m}{(1.5 \times \pi \div 180)rad/g}$$

$$K_{\phi} = 2787.03kgm/rad$$

6.3.5 Roll stiffness distribution

The roll stiffness can be distributed disproportionately among the front and rear tracks to affect the relative front to rear lateral weight transfer. As the rear of the vehicle is heavier and the roll couple only marginally shorter, the lateral weight transfer at the rear would be larger given even roll distribution. Therefore in the pursuit of neutral handling, the roll distribution was marginally biased towards the front 52:48. This would help reduce the excessive loading on the outside rear tyre that lowers if coefficient of friction. The past UWAM vehicles have also favoured a roll distribution towards the front (Table 6-1). Even with the marginally front biased roll distribution, the lateral weight transfer is still larger at the rear. The roll distribution bias was not increased further as stiffening the lighter front end might make it too jittery and lose grip under bumpy conditions, secondly the chassis designers are endeavouring to balance the weight distribution and may improve on the estimated 45:55 split. The front and rear roll distributions are calculated as:

$$K_{\phi FS} = K_{\phi} \times 0.52 \qquad \text{Equation 6.3}$$

$$K_{\phi FS} = 2787.03 kgm/rad \times 0.52$$

$$K_{\phi FS} = 1449.24 kgm/rad$$

$$Kr = K_{\phi} \times 0.48 \qquad \text{Equation 6.4}$$

$$K_{\phi Fr} = 2787.03 kgm/rad \times 0.48$$

$$K_{\phi FS} = 1337.76 kgm/rad$$

6.3.6 Weight transfer

Reactive forces at the wheels attempt to counteract the lateral forces acting at the centre of gravity described before. As a result load is transfer from the inside to the outside wheels. This dynamic could be accurately modelled as three mass system consisting of the sprung weight and front and rear unsprung weights. However in Milliken & Milliken (1995) the system is simplified to a single mass representing the vehicle's centre of gravity thereby greatly reducing the complexity of the equation while only losing 2.7% accuracy (in their example). This shows separating the sprung and unsprung masses is not critical when the suspension is designed purely for roll, as a result the expected increase in unsprung weight from 14kg per corner to 17kg (on account of the motors and gearbox) can be ignored for these calculations.

The simplified lateral weight transfer (ΔW) formula given by Milliken & Milliken is displayed as Equation 6.5 and Equation 6.6. It still takes into account the different roll centres and roll stiffness at the front and rear. t represents the track width.

$$\Delta W_{YF} = A_x \times \frac{m}{t_F} \times \left[\frac{H \times K_{\phi F}}{K_{\phi}} + \frac{b}{l} \times Y_{rf} \right] \qquad \text{Equation 6.5}$$

$$\Delta W_{YF} = 1.2 \times \frac{310kg}{1.2m} \times \left[\frac{0.235m \times 1449.24 kgm/rad}{2787.03 kgm/rad} + \frac{1.6m \times 0.445}{1.6m} \times 0.025m \right]$$

$$\Delta W_{YF} = 41.33kg$$

$$\Delta W_{YR} = A_x \times \frac{m}{t_R} \times \left[\frac{H \times K_{\phi R}}{K_\phi} + \frac{a}{l} \times Y_{rr} \right] \qquad \text{Equation 6.6}$$

$$\Delta W_{YR} = 1.2 \times \frac{310kg}{1.2m} \times \left[\frac{0.235m \times 1337.76kgm/rad}{2787.03kgm/rad} + \frac{1.6m \times 0.555}{1.6m} \times 0.05m \right]$$

$$\Delta W_{YR} = 43.57kg$$

6.3.7 Ride rate

The ride rate (K_R) represents the force needed per unit of vertical displacement of the tyre contact patch (Milliken & Milliken 1995). Equation 6.7 and Equation 6.8 for ride rates are given in Milliken & Milliken (1995).

$$K_{RF} = \frac{2 \times K_{\phi FS}}{t_F^{\,2}} \qquad \text{Equation 6.7}$$

$$K_{RF} = \frac{2 \times 1449.24kgm/rad}{(1.2m)^2}$$

$$K_{RF} = 2012.83kg/m$$

$$K_{RR} = \frac{2 \times K_{\phi RS}}{t_R^{\,2}} \qquad \text{Equation 6.8}$$

$$K_{RR} = \frac{2 \times 1337.76kgm/rad}{(1.2m)^2}$$

$$K_{RR} = 1857.78kg/m$$

6.3.8 *Wheel rates*

The wheel rate (K_W) is the vertical force per unit of displacement of the wheel. For stiffly sprung racing suspension the tyres can provide up to half of the compliance (Milliken & Milliken 1995), therefore the compliance of the tyres must be taken out of the ride rate to calculate the necessary spring stiffness. The suspension can be modelled as a two mass system compromising the sprung and unsprung mass (Figure 6-2) thereby separating the tyre rate and wheel rate according to Equation 6.9 and Equation 6.10.

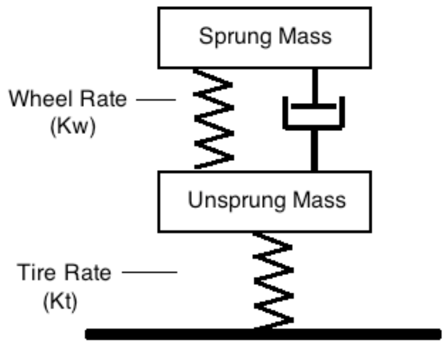

Figure 6-2: Unsprung and sprung mass in series

$$K_{WF} = \frac{K_T K_{RF}}{K_T - K_{RF}} \qquad \text{Equation 6.9}$$

$$K_{WF} = \frac{18367.97kg/m \times 2012.83kg/m}{18367.97kg/m - 2012.83kg/m}$$

$$K_{WF} = 2260.55kg/m$$

$$K_{WR} = \frac{K_T K_{RR}}{K_T - K_{RR}} \qquad \text{Equation 6.10}$$

$$K_{WR} = \frac{18367.97kg/m \times 1857.78kg/m}{18367.97kg/m - 1857.78kg/m}$$

$$K_{WR} = 2066.82kg/m$$

6.3.9 *Installation ratio*

The installation ratio (IR) relates the displacement of the spring/damper to the vertical displacement of the wheel. As the installation ratio reduces both the displacement and force at the wheel relative to the spring/damper, the ratio must be squared when relating the wheel and spring rates (Milliken & Milliken 1995) according to Equation

6.11 and Equation 6.12. Spring rates of 450lbs/in (8036.1kg/m) and 400lbs/in (7143.2kg/m) where used for the front and rear springs as this would allow for adjustment either way with regular spring sets for the Fox Van RC damper ranging from 300lbs/in to 600lbs/in in 50lbs/in intervals.

$$IR_F = \sqrt{\frac{K_{WF}}{K_S}} \qquad \text{Equation 6.11}$$

$$IR_F = \sqrt{\frac{2260.55kg/m}{8036.1kg/m}}$$

$$IR_F = 0.530$$

$$IR_R = \sqrt{\frac{K_{WR}}{K_S}} \qquad \text{Equation 6.12}$$

$$IR_R = \sqrt{\frac{2066.82kg/m}{7143.2kg/m}}$$

$$IR_R = 0.538$$

7 Kinematic Design

7.1 Wishbones

7.1.1 Camber angle

Goodyear recommends a static camber of -1 ° to -1.5 ° for the D2696 tyre (Goodyear 2011), -1 ° has been chosen for both the front and rear tyres and will be easily adjustable on the vehicle. However to simplify the kinematic design process the camber has been made zero with extra adjustment of camber being incorporated to compensate.

7.1.2 Swing arm length

As the swing arm length is the largest determinant of the camber curve (Hayward 2001), variation of this distance would be used to attain the desired camber gain. Based on the recommendations identified in Staniforth (1999) a medium swing arm length (1000mm – 1800mm) was chosen as the best compromise of minimising roll centre movement, getting the required camber gain and limiting scrub (track width variation, Figure 7-1). As the chassis rolls in a corner the roll centre migrates. This movement should ideally be minimised to improve the predictably of the handling (by limiting the variation of the moment arm length between the roll centre and centre of gravity). Such simulation was beyond the scope of this thesis. The UWAM vehicle dynamics specialist recommended a longer swing arm at the front (relative to the rear) to reduce camber gain as castor geometry on the steering would introduce extra negative camber on the outside wheel. Swing arm lengths of 1500mm on the front and 1200mm on the rear were finally selected for the kinematic design.

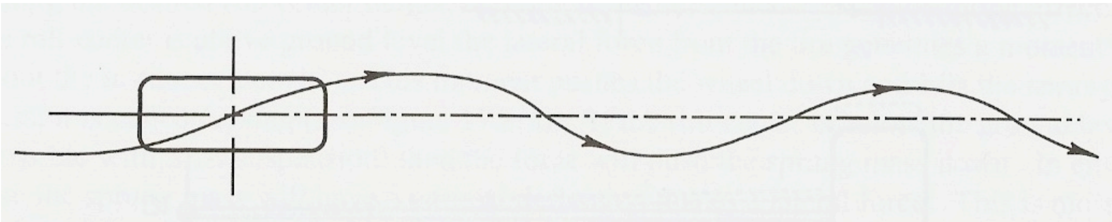

Figure 7-1: Tyre path on rough road with large scrub (Milliken & Milliken, p.616)

7.1.3 Scrub radius

The scrub radius at the front affects the forces felt through the steering wheel as a result of braking, acceleration and cornering forces. Any braking or accelerative forces will act as a moment about the scrub radius. On bumpy or inconsistent grip surfaces, it is beneficial for a FWD vehicle to have negative scrub radius as this will create a self steering effect during straight line travel. The opposite is true for a positive scrub radius but provides steering feedback of tyre slip angles. Large scrub radius can fatigue the driver’s arms (Pat’s Corner 2005). As the REV vehicle will only have a portion of its power transmitted through the front wheels and will transverse mainly smooth tracks, it was decided a small amount of positive scrub radius (40mm) would be the best compromise. It would also help avoid contact between the wishbones and wheel rim under max bump and jounce.

7.1.4 King pin inclination

Zero king pin angle is the ideal situation as positive kingpin angle causes the outside tyre to take on positive camber when the front wheels are steered which is highly undesirable. (Milliken & Milliken 1995) This undesirable effect can be countered by the negative camber gained during steering from castor angle. Zero king pin angle was selected for preliminary design, however kingpin angle is often introduced as a compromise to reduce the scrub radius and satisfying packaging requirements.

7.1.5 Outer pivots

The distance between the outer pivots should be maximised to reduce the forces placed on the wishbones from the brake and motor torques but is limited by the inside wheel diameter. On the 2001 vehicle the inside wheel diameter of the 13” rim was 300mm. 240mm between pivots was deemed the maximum vertical distance with sufficient clearance. A vertical distance of 115mm from the wheel centre would be used for the pivots that provide toe control (lower on the front, upper on the rear) and 125mm for the other pivots. Toe control pivots are spaced wider apart and therefore require to be placed closer to wheel centre for the same clearance from the wheel circumference (Figure 7-6).

7.1.6 Inner pivot distances / Arm lengths

The ratio of the upper arm length to the lower arm length affects the rate of camber gain in a suspension (shape of camber curve). Decreasing the upper arm lengths will increase the rate of camber gain in a suspension (Hayward 2001). Adams (1993) recommends upper arm length to be 50-80% of the lower arm. Adams (1993) also recommends making the lower arm as long as possible to reduce roll centre movement.

Lower inner pivot (suspension pivot on chassis) on the front was selected to be 200mm horizontal distance from on the vehicle centreline to clear the 350mm (175mm from centre) exclusion zone in the foot well incorporating an additional 25mm for the pivot mounting. This distance would also ease the design of the steering with the selected rack of 355mm between pivots (discussed in more detail in Chapter 8.4.3). To make the lower arms as long as possible, pivots at the centreline below the chassis were considered but eliminated on the need of bell crank steering to eliminate bump steer, which would require the writing of a simulation program to achieve Ackerman. On the rear a distance of 180mm from the centreline was selected to allow the 300mm jacking point (150mm from centreline) to fit in between the pivots and reduce the need for extra material on the chassis. 30mm was incorporated for the pivot mounts.

Upper inner pivot distances from centreline of 290mm and 300mm for the front and rear were selected to provide 65mm of distance from the chassis rails (for the fitment of the shims used to tune static camber). These values yield lower to upper arm ratios of 70% and 76% on the front and rear; within the ranges recommended.

7.1.7 Results

The finished kinematic design is exhibited in Appendix 13.4 with camber curves illustrated in the following figures. The graphs were obtained using the Motion Analysis feature in SolidWorks. The front design undergoes a camber gain of -1.05° and 0.92 ° for 25.4mm of jounce and rebound (Figure 7-2). The rear suspension undergoes more camber gain at -1.27° and 1.06° for 25.4mm of jounce and rebound (Figure 7-3). Both curves exhibit a very slight rising (negative) camber gain that is desirable as the vehicle rolls increases. A summary of the final kinematics design parameters is listed in Table 7-1.

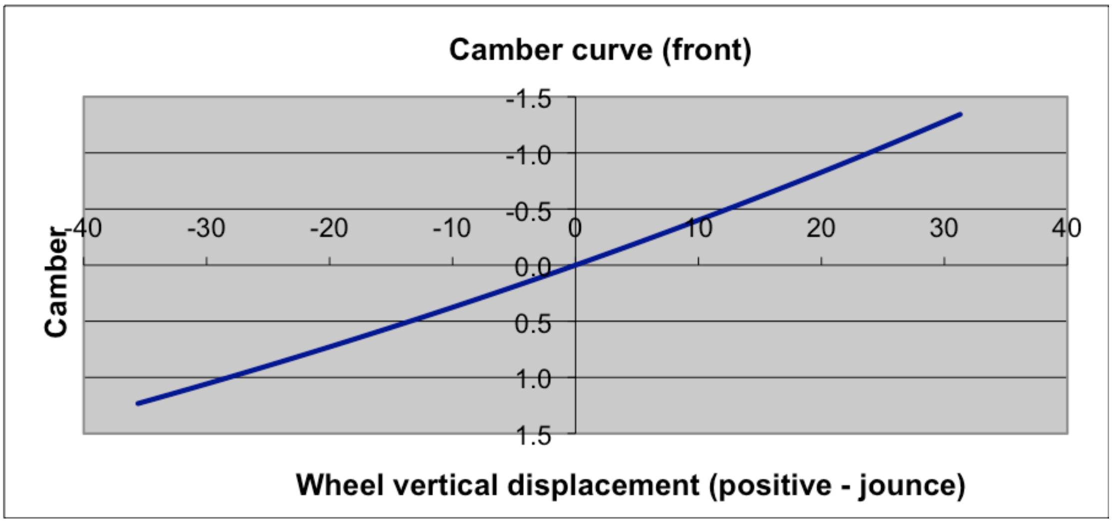

Figure 7-2: Camber curve (front)

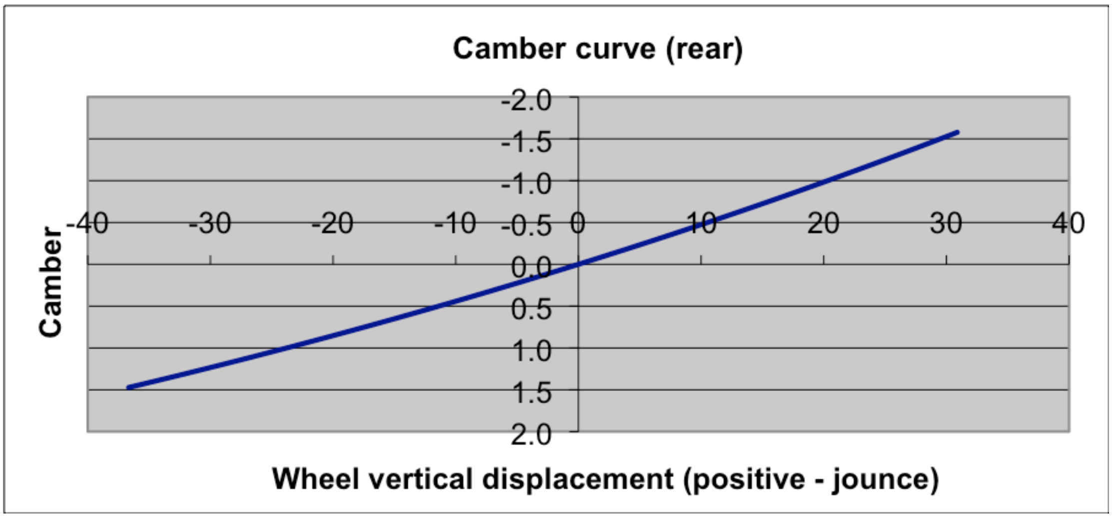

Figure 7-3: Camber curve (rear)

Static camber angle	-1	deg
Kingpin inclination (front)	0	deg
Scrub radius	40	mm
Caster angle	7	deg
Swing arm length (front)	1500	mm
Swing arm length (rear)	1200	mm
Roll centre (front)	25	mm
Roll centre (rear)	50	mm
Vertical distance between upright pivots	240	mm
Track	1200	mm
Ackerman steering	100	%

Table 7-1: Suspension geometry summary

7.2 Steering

7.2.1 Ackerman steering

Opinion on Ackerman is varying. For high speed corners reverse Ackerman; where the outside wheel steers more is sometimes employed based on the theory that the inside tyre is less loaded and therefore can only sustain lower slip angles than the heavy loaded outside tyre (Costin & Phipps 1971). Whereas for low speed corners, closer to 100% Ackerman is used to reduce scrub (Milliken & Milliken 1995) with recommendations for FSAE teams to even implement 50% more Ackerman than expected (Pat's Corner 2005). 100% Ackerman steering was deemed an appropriate preliminary design target.

7.2.2 Castor

Castor angle on the front wheels has the advantageous effect of making the outside wheel gain negative camber with increasing steering angle. Figure 7-4 displays the camber gain with 6° of castor. However it also has the downside of lifting the outside tyre and pushing down the inside tyre causing diagonal weight transfer from the outside tyre to the inside at the front and from the inside to the outside tyre on the rear (Pat's Corner 2008). A castor angle of 7° was selected, equal to that of the 2003 UWAM vehicle (Table 6-2). This was deemed the best compromise between the benefits of negative camber gain with steering and drawback of diagonal weight transfer.

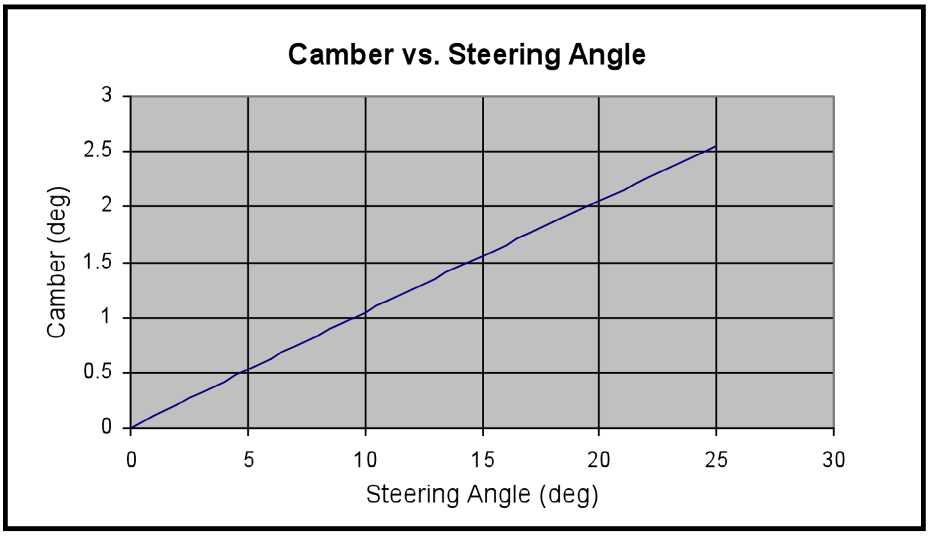

Figure 7-4: Camber gain with steering (6 ° of castor) (Hayward 2001)

7.2.3 Steering rack

Rack and pinion steering was selected due to several factors. It could be easily incorporated to produce close to 100% Ackerman using the method explained in 3.5.2. Secondly rack and pinion steering exhibits high reverse efficiency; the 'ability to pass road inputs back to the driver for feedback' (Milliken & Milliken 1995, p.719). Reverse efficiency is crucial in communicating the tyre's slip angle back to the driver that allows them to know when the front tyre's adhesion limit is being reached.

Bellcrank steering was rejected because of the need to write a simulation program to design for Ackerman.

A 14” (355mm pivot to pivot) dune buggy steering rack with 1.5 turns from lock to lock and 108mm of rack travel from Desertkarts.com was selected (Desert karts 2011). The steering rack diagram is available in Appendix 13.5. Costing only $USD98, this rack complies with REV’s limited resources and also the lower inner pivots could be designed to match the rack’s length (7.1.6). The steering rack could be placed in line with the lower inner pivots (front view) to help reduce bump steer by making the steering tie rods parallel. Pat’s Corner (2005) recommends bump steer should be zero to prevent unpredictable handling, however due to time constraints it was not possible to completely eliminate it. The bump steer characteristic of the final design can be seen in Figure 7-5. From full rebound to jounce toe changes a very marginal 1.5 ° (especially for a first year vehicle). Under heavy braking the front suspension will dive and toe in making it more stable than if it were to toe out. The low mounted rack (just behind the lower front wishbone) will now comply with the 350mm square exclusion zone and aid in keeping the centre of gravity low.

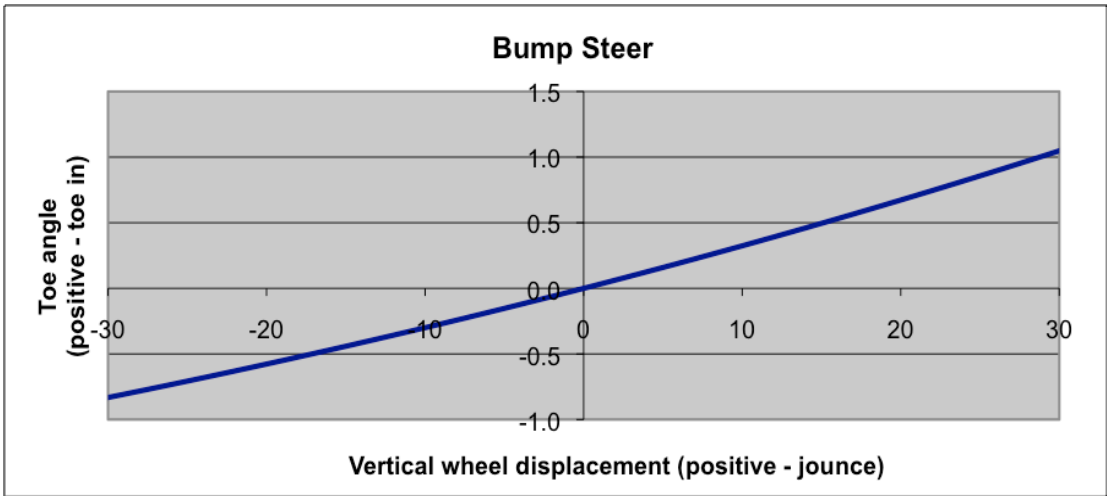

Figure 7-5: Bump steer of front suspension

The tie rod pivot on the hub should be horizontally spaced as far from the closest other pivot as physically possible to give more accurate control of wheel angle. Examination of various UWAM vehicles found a distance of 65mm to be appropriate. Next the steering arm angle was determined to be 20.55 ° with a lateral distance from the steering axis of 24mm (refer to Appendix 13.5). The proposed locations of the outer pivots and tie rod pivot can be seen in Figure 7-6.

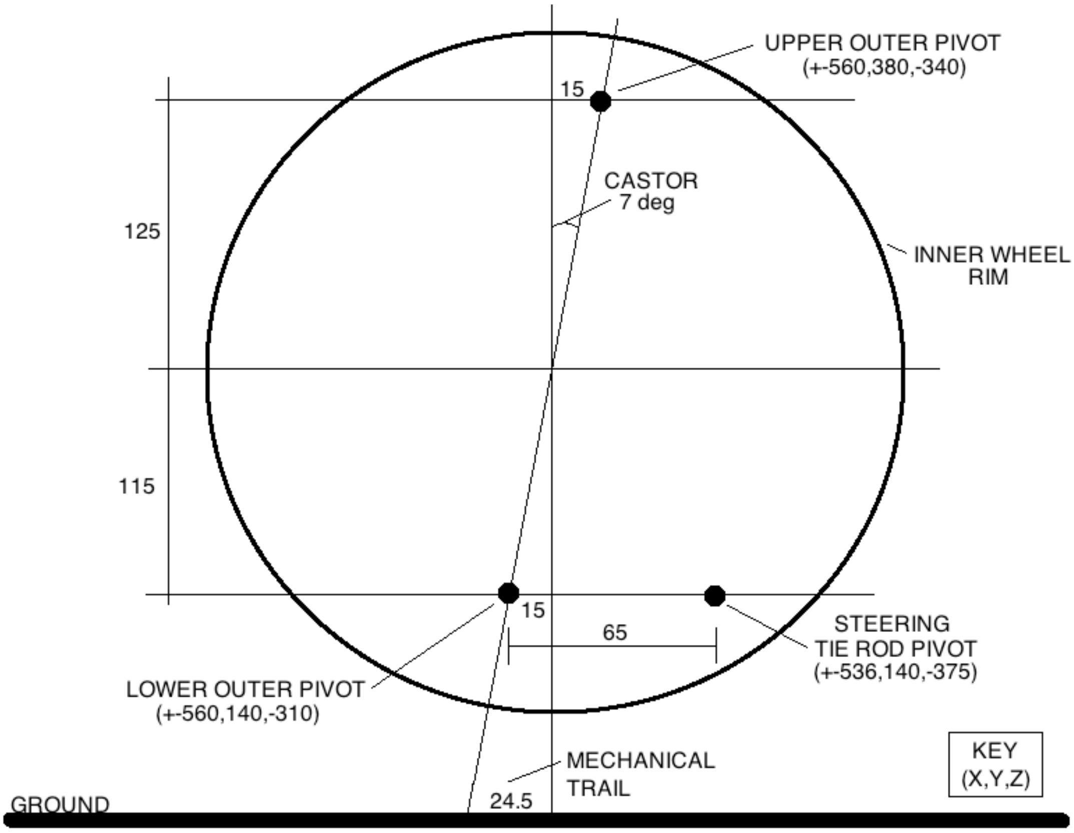

Figure 7-6: Outer pivot and tie rod pivot locations (side view)

7.2.4 Toe angle

Toe angles will be adjustable on the front and rear. Optimal angles will be determined from testing once the vehicle is built.

8 Component Detailed Design

8.1 Material selection

Carbon fibre is now commonly used for the wishbones by the more experienced teams as it offers a great improvement in strength to weight ratio and stiffness. Carbon fibre exhibits a yield strength approximately that of steel (dependent on the structure of the fibre) and density of approximately plastic (Dragon Plate 2011). However the main disadvantage and the reason it was excluded for the REV vehicle is its high cost, with a 22mmx3mm (O.D. x thickness) tube costing $91 per metre (CST Composites 2011).

Aluminium was considered due to its extensive use in lightweight, high-strength structures like airplanes. Even though the yield strength to weight ratio of some high strength steels approaches that of aluminium, its much lower specific gravity of 2.7 g/cm^3 compared to steel's 7.85 g/cm^3 (Black & Kosher 2008), allows for stronger structures to be built, especially in bending as the material will be further away from the neutral axis (higher second moment of inertia) (Engineers Edge 2002). However aluminium's disadvantages include a low modulus of elasticity (under identical loading an aluminium component will deflect three times as much), poor creep wear and poor resistance to fatigue (Black & Kosher 2008). Aluminium however was rejected for the two main reasons of difficulty in welding (especially for the inexperienced welder as there is a fine line between no penetration and blowing holes) (MIG Welding 2011) and because it would be 4-5 times expensive as steel for only a small weight saving (Black & Kosher 2008).

Finally Chrome Molybdenum (AISI 4130) high-strength steel was selected as the best compromise for REV's resource limited situation. The content composition is in the range of 0.3% Carbon, 0.5% Mn, 0.3% Si, 1% Cr and 0.2% Mo (eFunda (a) 2011). It has much higher yield strength of 650MPa as opposed to 300MPa for mild steel (AISI 1020) (eFunda (b) 2011) however welding can reduce the yield strength to 360MPa. Stiffness of AISI 4130 at 200GPa is similar to mild steel (eFunda 2011). Cost of appropriate size AISI 4130 tubing ranges form $15 to $30 per metre (Go Gear 2010). AISI 4130 is approximately double the price of mild steel but costs for a-arm tubing is estimated at $200 hence the extra cost is relatively small. 4130 can be TIG welded using ER80S-D2 filer material and it is not necessary to stress relieve tubing of 3mm thickness or less (Lincoln Electric, 2011). There are no major difficulties in welding 4130 as there are with aluminium. AISI 4130 was chosen as the material for all the components to be built in house including the wishbones, bearing housings, pull-rod, end pieces and rocker arms. The brackets and shims may be made from plain carbon steel.

8.2 Bearing selection

Three main types of bearings were considered for this suspension system, spherical bearings, spherical rod ends and bushings/bearings. Aurora PWB-5T 5/16" Spherical bearings where chosen for all the wishbone pivots. These are part of the performance racing series and are PTFE lined to reduce stiction. They allow up to 14° of misalignment (difference between the axis of the inner sphere and outer housing) and are limited to 41.8kN static radial and 7.3kN static axial loads (Aurora 2011). They were used successfully on the 2001-2003 UWAM vehicles. They are available from Go Gear racing for $34.39 each, with Aurora doing a special deal for FSAE teams offering buy one and get one free. With the suspension requiring 24 spherical bearing, the cost is reduced from $825 to $413. Bushings could have been used on the inner pivots of the wishbone like on the 2001 UWAM vehicle, but the cost saving would have only been $235 (assuming $2.50 a bushing) and could have caused stiction problems. Rod ends were not used as judges dislike them being placed in bending (Pat's Corner 2005). With difficulties of the pull-rods clearing the wheel hub motors, it was likely the wishbones could be placed in bending.

Figure 8-1: Aurora PWB series spherical bearing (Aurora 2011)

Maximum tyre loading was calculated for both 1.2g lateral acceleration and a combination of 1.5g braking and 0.5g lateral acceleration. Both conditions produced a maximum tyre loading of below 130kg (1274N). Of the spherical bearings, the upper wishbone's outer pivot would be the most critical due to the large vertical force it must resist in axial loading (spherical bearings are much weaker in resisting axial loading than radial). For this critical pivot the Aurora PWB-5T still displays a safety factor of 5.7 (please refer to Appendix 13.6 for detailed calculations). The rest of the spherical bearings will be resisting mostly radial loads (or smaller axial loads) of which are they are rated to much higher and will therefore not have any issues. The upper and lower wishbones are predicted to rotate ±5 ° and ±4 ° for 30mm of jounce and rebound thereby being well in the misalignment angle of 14 °.

Aurora AM-5T (right hand) and AB-5T (left hand) 5/16” threaded male rod ends were selected for the pull-rod ends. They are also PTFE lined and are part of the High Strength Alloy – Performance series. They are available from Go-Gear Racing for $23.45 (AM-5T) and $29.60 (AB-5T) with the same discount (total cost $159). They also allow 14 ° of misalignment and have a radial load limit of 34kN (Aurora 2011). They were used successfully for the pull-rods on the 2001 and 2002 UWAM vehicles.

Figure 8-2: Aurora AM series rod end (Aurora 2011)

Under the maximum tyre load of 130kg, the pull-rod will be subject to a tensile force of 2270N. The rod ends on the pull-rod will be resisting this load radially, thereby giving it a large safety factor of 15 (detailed calculations in Appendix 13.6). The safety factor might seem excessive but the rod end size was chosen to retain the same bolt size as other bearings to reduce necessary spare parts. In the future a smaller rod end may be chosen to reduce weight.

The same Aurora AM-5T and AB-5T rod ends will be used on the rear suspension toe arms. The steering arms will use 3/8” spherical bearings mounted on the steering rack and 3/8” rod ends supplied in the steering toe arm kit (Dan’s Performance 2011) (discussed further in 8.4.3).

SKF PCM081012 Teflon lined bush with a static load limit of 20kN was selected as the bearing surface for the rocker pivot. This bush has a 10mm O.D. and 8mm I.D. which will reduce the necessary spare parts by allowing the use of the same bolts as those on the wishbone pivots.

8.3 Adjustability

It is desirable for certain parameters on a race vehicle to be adjustable to cater for different tracks, track conditions and drivers. On this vehicle camber, toe, ride height, spring stiffness and damping will all be adjustable. Following Sands' (2003) recommendations the following adjustability methods for each parameter have been used:

- Camber: adjusted using 5mm removable shims placed in between the upper wishbone brackets and chassis. Design setup incorporates 25mm of shims for zero camber, thereby allowing up to -6 ° of camber by removing all shims. This method was selected based on easy manufacture of shims and desire not to use rod ends due to bending in the upper wishbone. Main downside is weight penalty of the shims. Shims were not used on the lower pivot to reduce centre of gravity because changes would have inadvertent affects on the steering geometry.
- Toe: the front steering toe arm will use a 3/8" rod end on the joint connecting to the upright. This will allow easy toe adjustment by turning the rod end to change the length of the steering toe arm. On the rear, two opposite threaded rod ends will allow adjustability of toe. This choice was made on the relatively simple design of the toe arm that could be used with rod ends.
- Ride Height: will be adjustable via two methods. The first and main method will be altering the length of the pull rod by winding or unwinding the opposite threaded rod ends. Secondly, the preload may be altered on the spring. By increasing preload, the vehicle will lift up. However there is a recommend limit placed on the preload for the dampers so this method should rather be used to prevent the spring from floating freely under full droop.
- Spring stiffness: the springs will be interchangeable with spring stiffness varying from 300lbs/in to 600lbs/in in 50lbs/in intervals. As the vehicle will not have any anti-roll bars this will be the main method of altering handling characteristics by changing the roll stiffness distribution.
- Damping: the Fox Van RC have both adjustability of rebound and compression damping. However as discussed in the literature review, low speed compression damping adjustability is very limited.

8.4 CAD design

8.4.1 Wishbones

Wishbone design would attempt to emulate as much as possible from the past 2001-2003 UWAM vehicle designs. These vehicles competed in the FSAE competition without serious failure of their suspension systems that leads to conclude they had been built sufficiently strong to endure such conditions. These vehicle's weight, weight distribution and roll stiffness were similar (±10%) to those proposed for REV's 2011 vehicle (Table 6-1). It was beyond the scope of this project to analyse in detail the safety factor of each suspension component, instead approximations were made to satisfy main concerns.

The inner wishbone pivots were spaced 400mm apart on both front and rear, similar to the UWAM 2001 vehicle. Increasing the distance between inner wishbone pivots reduces the forces acting in the wishbones under any longitudinal loads as shown in Figure 8-3. On the front the wishbones are asymmetrically sloping rewards as shown in (Figure 8-3 (C)) for several reasons:

- It was desirable for the wishbones to be attached as far forward to leave as much space possible for the battery pack to be mounted beside the driver for better weight distribution.
- Longitudinal braking would place the fore wishbone arm into tension thereby reducing the likelihood of buckling as braking was expected to be 36% larger than acceleration.
- Would allow easier fitment of rear steer that has the steering arm coming out of the wheel (less likely contact with wheel rim under full lock) as opposed to further inside as with front steer. With the rack being placed just behind the front wishbone the steering tie rod arms will be nearly parallel with the rack as desired for Ackerman steering in this design.

On the rear the wishbones are symmetrical when viewed from the top as there is no steering to concern and braking forces are much smaller due to longitudinal weight transfer.

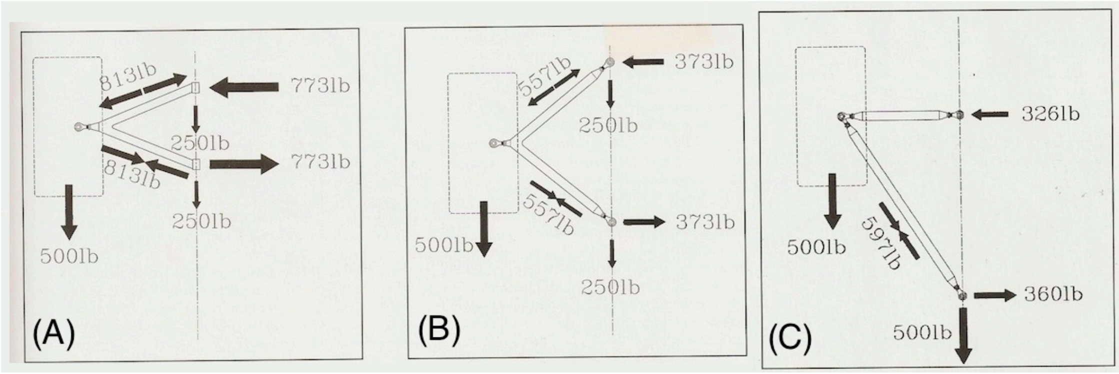

Figure 8-3: Wishbone forces, (A) narrow base, (B) wide base, (C) asymmetrical base (Aird, p.59)

The damper actuation was the largest design hurdle of the entire project. The problem lay with the electric wheel hub motors protruding out of the wheel centre 110mm (as shown in Appendix 13.1) that would come into contact with the commonly used pull-rod or push-rod used to actuate the dampers (Figure 8-4).

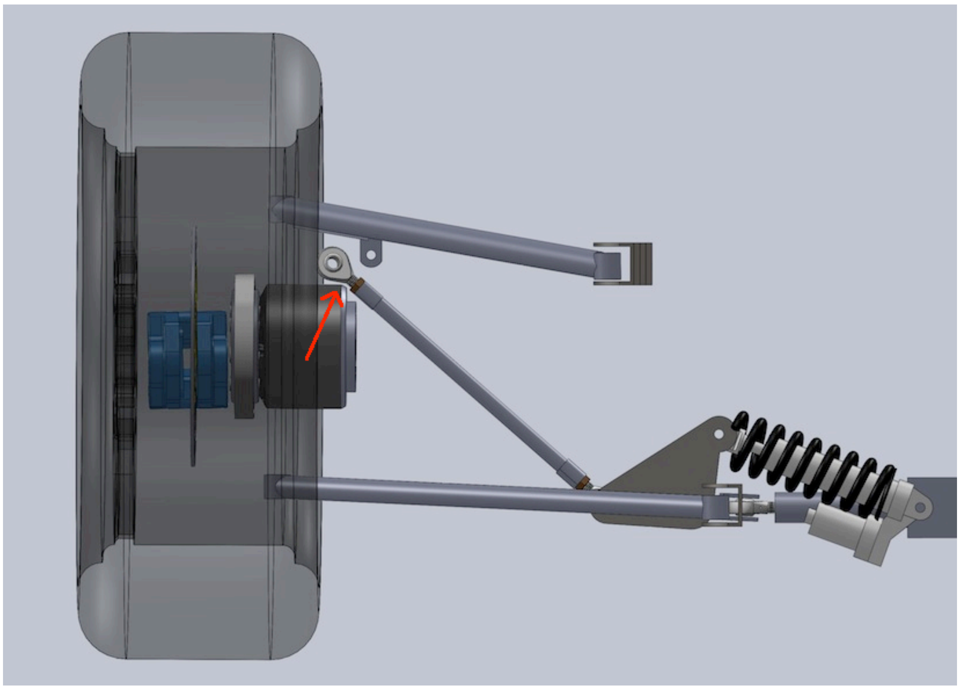

Figure 8-4: Pull-rod and motor contact

Usually the pull/push-rod is attached to the wishbone as close as possible to the outer pivot to reduce bending issue, this is especially the case when using rod ends. Many methods were attempted to overcome this issue but each presented a flaw:

- Top rocking arm: uses the top wishbone as a rocking arm to actuate the damper as shown in Figure 8-5. This was not possible on the front suspension as the damper would be inside the 350mm square footwell exclusion zone (FSAE rule, Appendix 13.2). Secondly the large bending forces placed on the wishbones would necessitate some sort of reinforcement above the wishbone that would protrude the driver's forward vision.

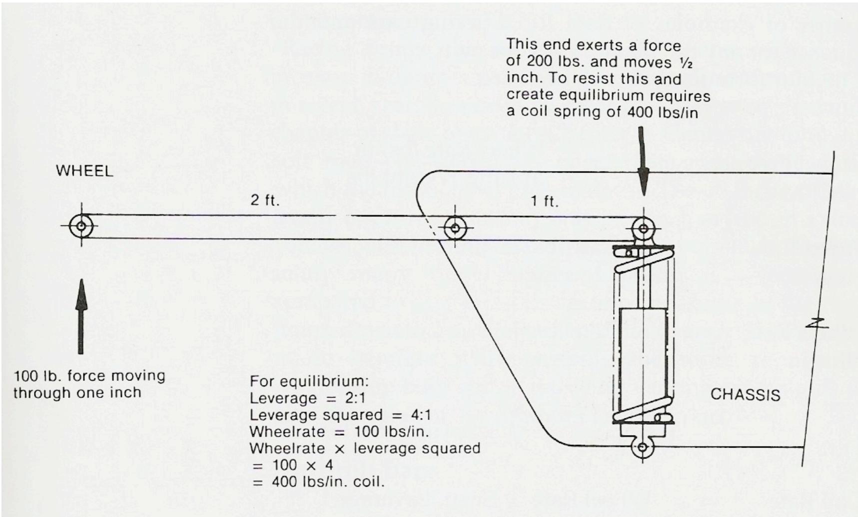

Figure 8-5: Top rocking arm actuation (Staniforth 1999, p.185)

- Push-rod mounted to upper wishbone: This would relocate the push rod from between the wishbones to above the top wishbone, however again the forward vision of the driver would be impaired and also the rocker mounting would be in an inconvenient location where additional chassis members would be necessary, increasing the weight of the vehicle and additionally not being a stiff location.
- Push/pull rod mounted to upright: This would alleviate bending stresses in the wishbones by loading forces straight into the upright. A long (~50mm) cantilever would be necessary to overcome the motor dimensions that would require a strong structure but still possible. However this method could not be implemented on the front, as steering angles would alter the length between the pull-rod upright mount and rocker. This solution is possible on the rear.
- Upper wishbone triangulation: This method removes the need of the pull/push-rod and rocker to connect the dampers as can be seen in Figure 8-6. This would ideally leave the entire space between the wishbones for the upright designer to place the motor wherever they please. However on the front this approach

would severely compromise the driver's visibility and even might contravene the rules and therefore again could not be implemented.

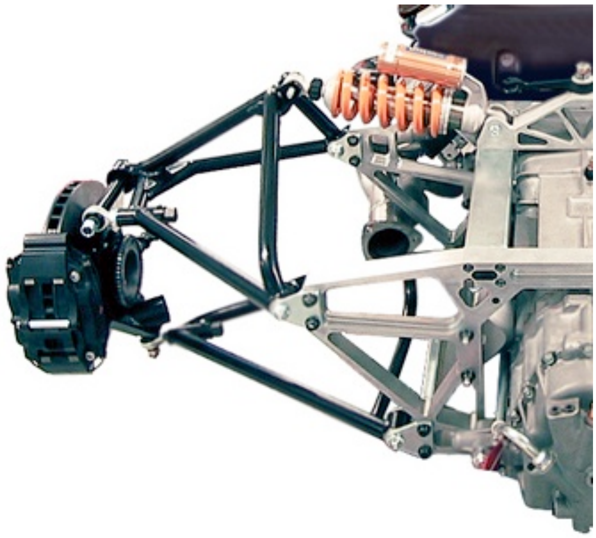

Figure 8-6: Koenigsegg upper wishbone triangulation (Solidworks Roadster 2011)

- Upper wishbone triangulation with pull rod: A unique approach implemented by the 1971 McLaren M19 GP car (Figure 8-7). The location of the front dampers would contravene the footwell exclusion zone and therefore could not be implemented on the front suspension.

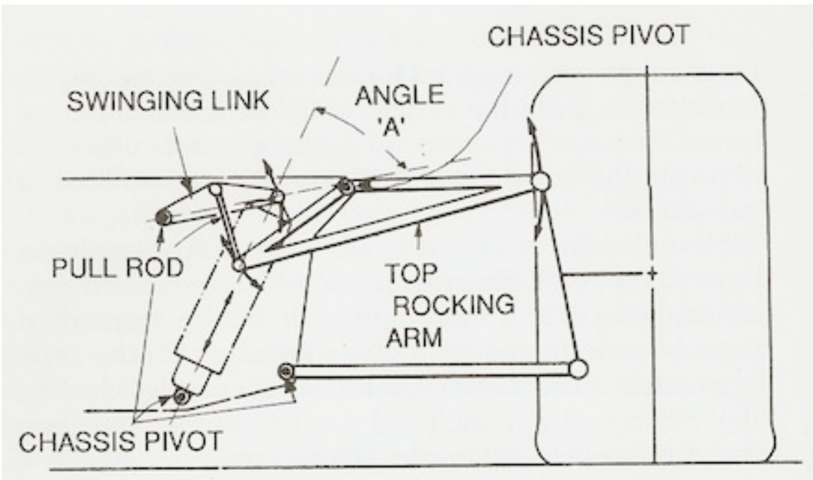

Figure 8-7: 1971 McLaren M19 suspension (Staniforth 1999, p.54)

- Pull-rod with increased distance from wheel pivot: this was the final solution employed. The main drawback of this solution is the increased bending experienced by the upper wishbone. The distance between pull-rod mount and wheel pivot was increased to 75mm to clear the electric motor. Because of limited time the force through the wishbones could not be calculated, however using this conventional approach, wishbone forces calculated for past UWAM vehicles could be used as approximations. This option was chosen for both the front and rear suspensions to limit variability and thereby reduce manufacturing time.

The 2003 UWAM vehicle employed spherical bearings for the wishbones and a push-rod set up with the push-rod's mounted ~50mm away from the wheel pivot. AISI 4130 steel tubes of 19.05mm x 1.25mm (O.D. x thickness) were used for the wishbones. Safety factors ranged from 2.1 to 4.3 for the wishbones under loads of 146kg on the front and 160kg on the rear respectively and including bending (Sand 2003). With the pull-rod's mounted 50% further on the REV vehicle, bending loading was assumed to also increase by 50%. To account for the expected larger bending moments in the upper wishbone, increasing the outside tube diameter or increasing wall thickness over the 2003 choice was considered. Bending stresses were simply assumed to equal:

$$\sigma = \frac{Moment_{(constant)} \times radius}{I} \qquad \text{Equation 8.1}$$

Based on the results in Table 8-1 increasing the tube diameter to 22.22mm while retaining a wall thickness of 1.25mm was selected due to the larger reduction in bending stress, smaller weight increase and reduction in cost compared to a thicker tube. Going to an even larger diameter was not pursued as it would cause contact issues with the wheel rim under full rebound. Lower wishbone tube specifications remain at 19.05mm x 1.25mm as they are not subject to bending.

Tube	**OD (mm)**	**Thickness (mm)**	**Bending stress reduction**	**Weight increase**	**Cost per meter**
Original (2003)	19.05	1.25	0%	0%	$13.72
Thicker	19.05	1.65	20%	30%	$15.77
Larger	22.22	1.25	39%	18%	$12.83
** costs from Appendix 13.7*					

Table 8-1: Various tube diameters performance

Bearings housings will have an O.D. of 22.22mm and I.D. of 17.46mm (equal to O.D. of Aurora PWB-5T). Housings will be cut from 22.22mm x 3.05mm tubes and turned on a lathe to the correct I.D. Housings are to be 20mm high for the lower wishbones and 22mm high for upper wishbones to account for the larger tube. Housings will than be welded on to the tubes using 4mm fillet weld (Winzer 2002) as recommended under 8.1 Material selection.

Originally on the rear suspension the toe control arms where to be located parallel to the upper wishbone to provide easier access to adjustment. The 2001 UWAM vehicle also had this configuration as the damper was located on an angle, the pull-rod mount would need to be displaced longitudinally from the outer wishbone pivot to have forces from the pull-rod directed at the bearing. On the REV vehicle the damper was planned to be inline with the rear track, therefore the pull-rod mount would also be located inline with the rear track. Therefore the toe control arm was moved to be parallel with the lower wishbone (Figure 8-8). This reversed the vertical distances of the outer pivots from the wheel centre to allow for the now wider spaced lower outer pivots to maintain clearance from the wheel rim under jounce. The lower outer pivots are now 115mm from the wheel centre and the upper outer pivot is 125mm.

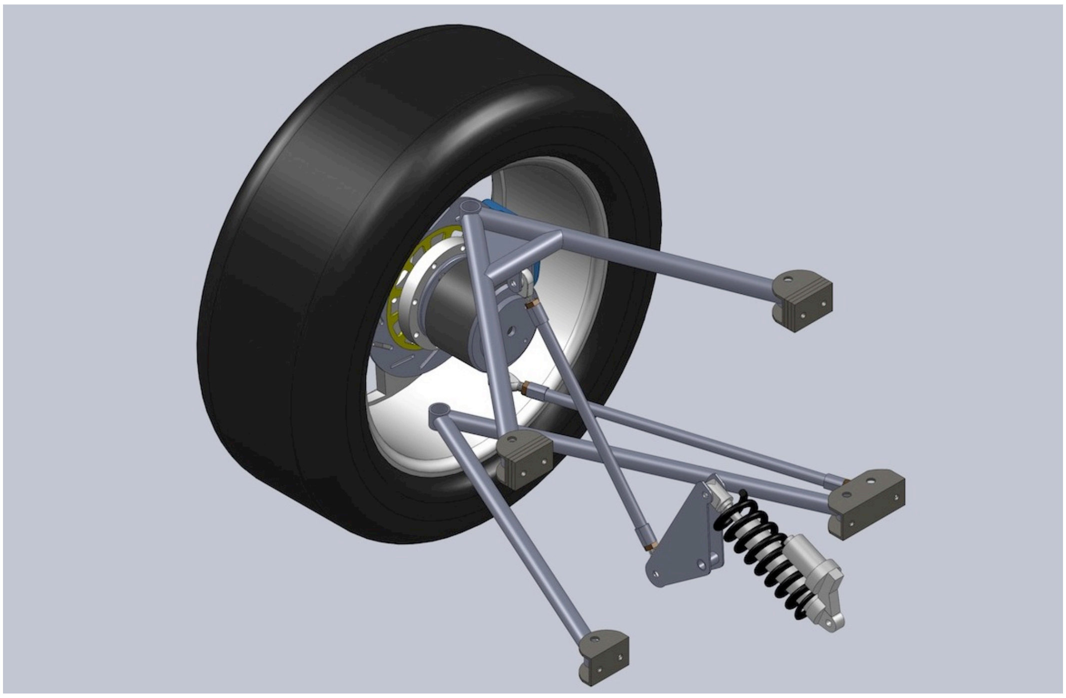

Figure 8-8: Rear suspension showing toe control

The wishbone pivots are attached to the chassis using brackets (Figure 8-8). The brackets will be made from 3mm AISI 4130, laser cut to correct form and bent into a C shaped clevis. The bend radius is yet to be determined but will be based on feedback from the workshop technicians. This design requires minimum work while capturing the spherical bearings in double shear and also explains the reasoning for the orientation of bearings. 5/16” (7.94mm) bolts with nylon nuts will be used to attach the brackets to the wishbones and chassis. The UWAM 2003 vehicle uses Unbrako’s bolts of property class 12.9 (Sands 2003), it is recommended the same be used on the Rev vehicle. The wishbone will be attached to upright using the same 5/16” size (7.94mm) bolts but will require a cone washer (Figure 8-9) to comply with the rule requirement for spherical bearings to be captured by a washer of larger O.D. if not in double shear.

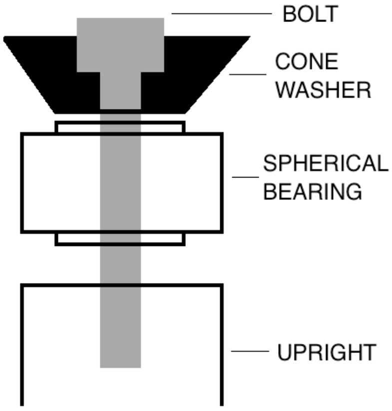

Figure 8-9: Cone washer on outer spherical bearing

8.4.2 Damper Actuation

Placement of the damper was another major design hurdle. Original plans were to place the damper vertically in between the wishbones, attaching it to the chassis rail connecting the upper wishbone brackets and the rocker mounted on the lower rail. This orientation can be seen on the 2001 UWAM vehicle (Figure 8-10).

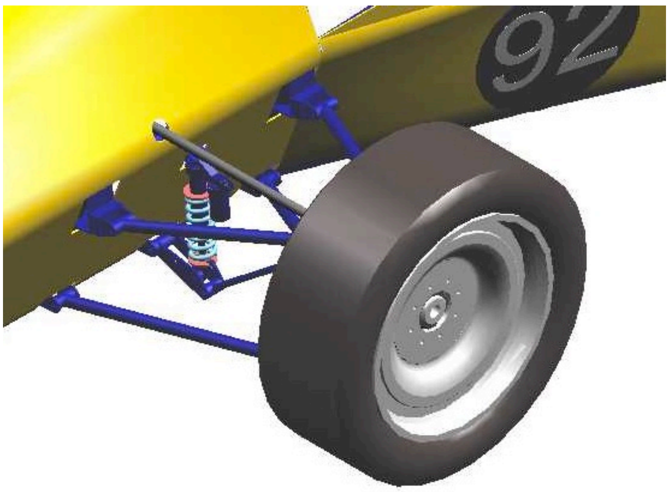

Figure 8-10: 2001 UWAM damper location (Hayward 2001)

When this was attempted, it was impossible to achieve the required installation ratio as the wishbones are shorter on the REV vehicle and the pull-rod mount has been moved further away (amplifying the effect at the wheel). An innovative solution was found to attach the dampers to the bottom of the chassis in the centre as can be seen in Figure 8-11. This has benefits of lowering the centre of gravity and having the forces of the left and right dampers oppose each other.

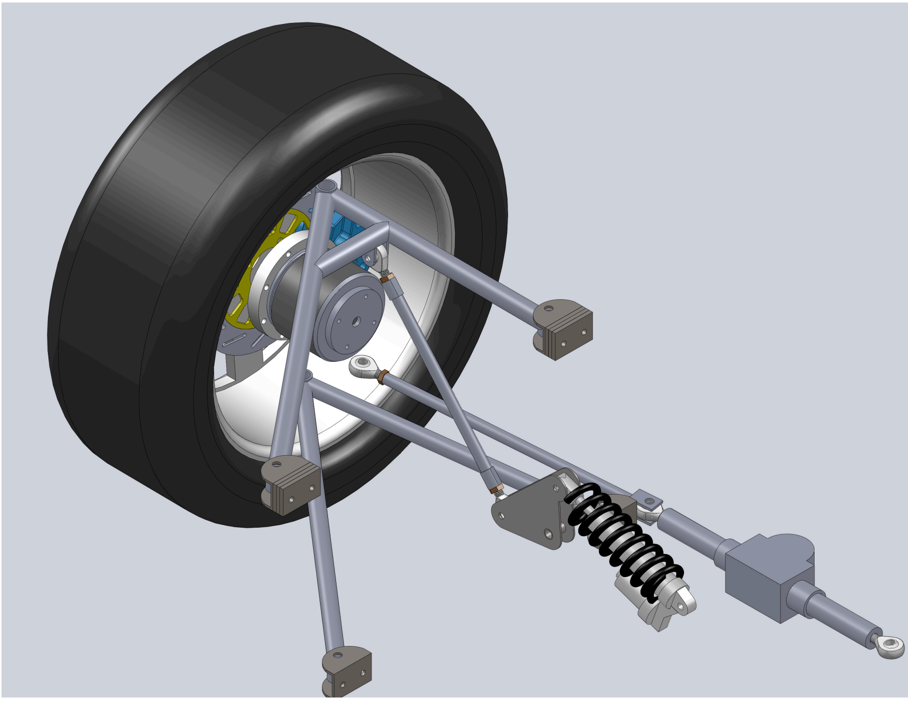

Figure 8-11: REV damper location

Unfortunately it has drawbacks on the chassis design by having to raise the footwell exclusion zone higher. At the time of writing, the chassis design had not be finalised but drafts showed the upper areas of the chassis may not need to be raised as they will still provide 350mm from the top of the dampers. It will be possible to orientate the reservoir of the damper downwards thereby reducing the exclusion zone height but this could leave the dampers vulnerable to being hit by something on the track. Orientation of the damper reservoir can be changed without consequence to the suspension so will be determined on feedback from the chassis design.

The rockers are to be made of 3mm AISI 4130 sheets laser cut to correct form. They are than welded to a tube of 12.7mm x 1.35mm that houses the bush (tube is made from 12.7mmx1.65mm tube and machined to correct I.D.) (Figure 8-12). A 5/16” bolt and nylon nut will hold the rocker attached to the chassis. FE analysis was carried out on the rockers and showed a safety factor of 4 and 3 for the front and rear rockers respectively (Appendix 13.8), with the largest forces expected at the sharp cut out below the damper connection. This cut out was necessary for the rocker not to come into contact with the spring. Originally the rockers were designed with 2mm AISI 4130 sheets but this displayed an uncomfortably low safety factor of 2 for the rear rocker (Appendix 13.8). Both front and rear rockers are to be made from 3mm sheets to allow all flat components including the brackets to be cut from the same sheet.

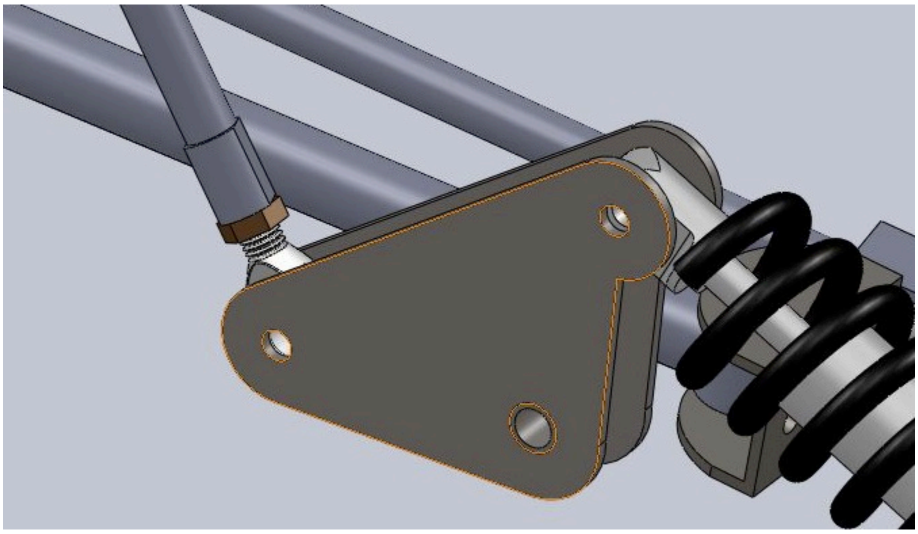

Figure 8-12: Front rocker

The rocker dimensions selected produced an installation ratio of 0.525 on the front, marginally below the 0.530 calculated in the preliminary design. The design of a rocker displays a desirable rising installation ratio (Smith 1978), however the slope may be a little too steep with the I.R. increasing to 0.57 under 25mm of jounce (Figure 8-13). This was mainly a consequence of trying to keep the damper as low as possible to reduce the height of the foot well exclusion zone. On the rear the I.R. is 0.535, marginally smaller than the 0.538 calculated during preliminary design. The rear also features an increasing I.R. although with a smaller slope (Figure 8-14).

Smith (1978) recommends for those without large funds and that want to focus on racing rather than over endowing engineering to use a gentle slope (less than 20%) rising rate for the front and a very gentle (less than 5%) set up at the rear. Smith's main reasoning for this is that a both rising rates at the front and rear will produce an unpredictable handling car on road circuits, although it is unclear how applicable this is to AWD vehicles. With the design complying with these recommendations they were not further changed.

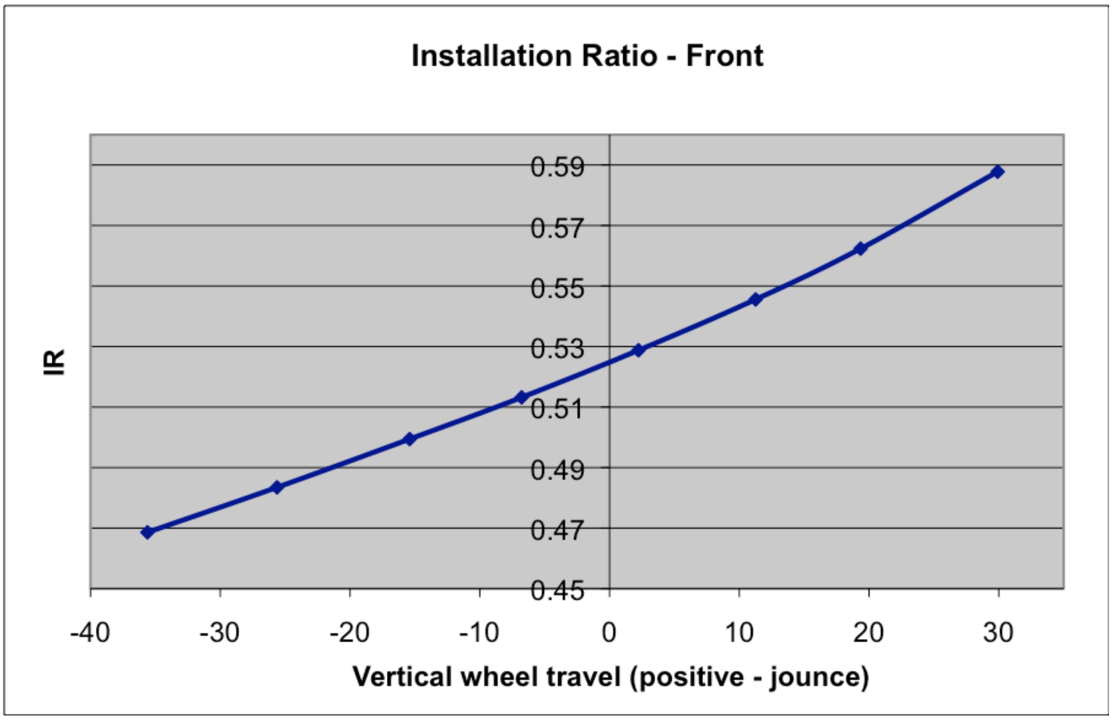

Figure 8-13: Installation ratio – Front (Equation 3.1 - Equation 3.4)

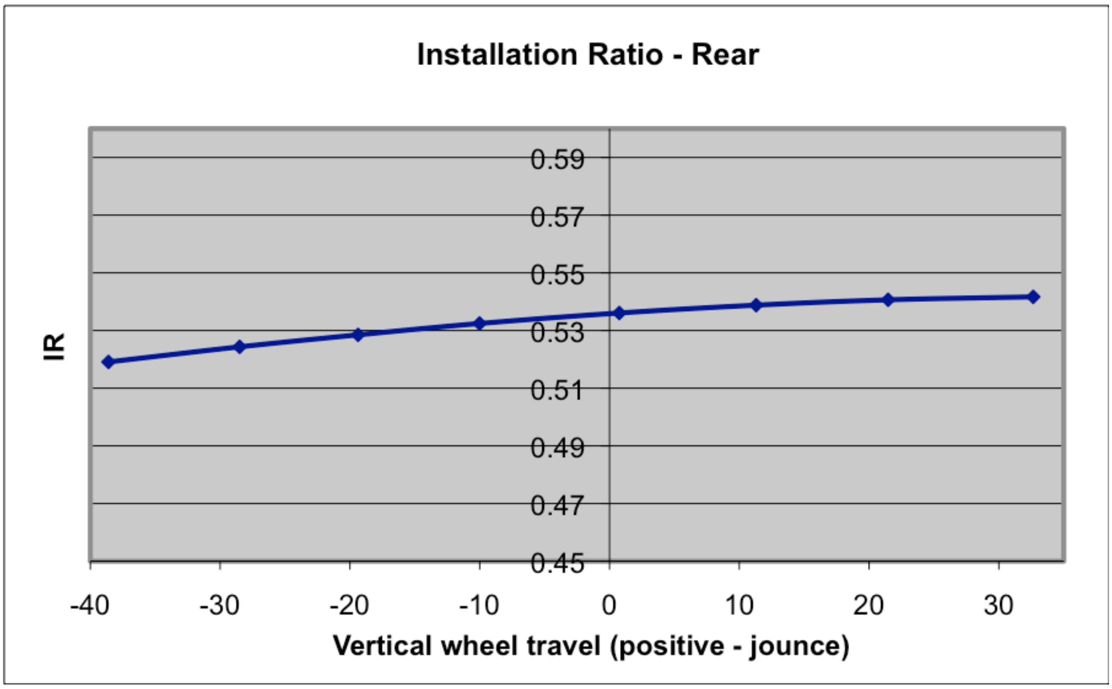

Figure 8-14: Installation ratio – Rear (Equation 3.1 - Equation 3.4)

The pull-rod compromises of several components. The middle section is made from 12.7mm x 0.9mm AISI 4130 tube. Under the expected loading, the safety of factor on this tube is 5.3. The tube size could have been reduced to get the safety factor down to four, but it was decided to maintain this tube size as it had been used successfully on the 2002 UWAM vehicle.

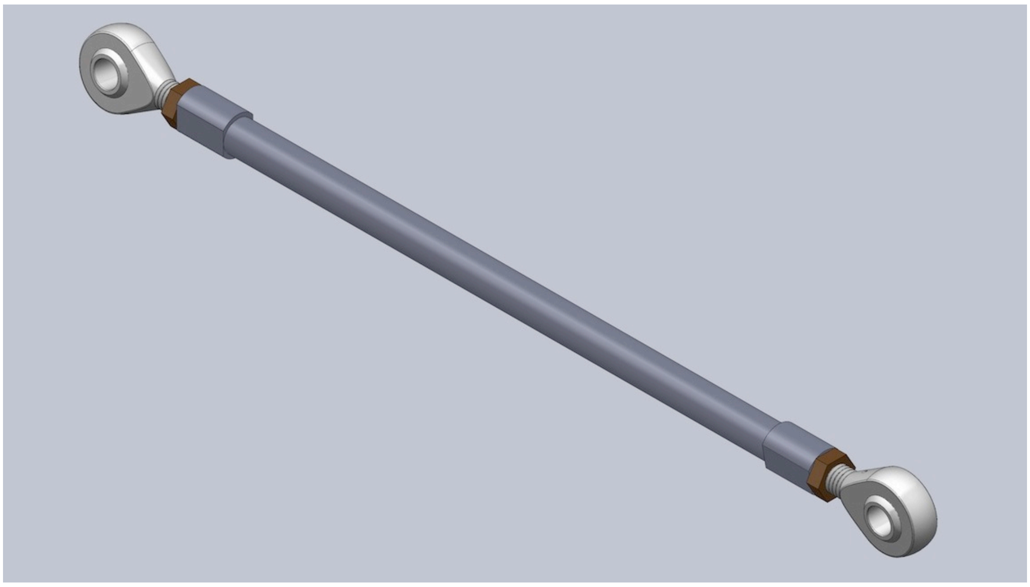

Figure 8-15: Pull-rod

As the rod-ends 7.94mm thread is smaller than the inside diameter of the pull-rod's main section, smaller inside diameter end pieces need to be fabricated and welded on (Figure 8-16). These will be made from 12.7mm x 3.05mm tubes with the inside diameter threaded out using tapping. Having the end pieces it will make it possible to use a taper tap as it will be a through hole, this will reduce the torque needed allowing the use of a hand turned tap (Black & Kosher 2008). Flat edges will be cut into the end pieces to allow the tubes to be held with a wrench without damage. Finally jam nuts will be used to prevent the rod ends from unwinding.

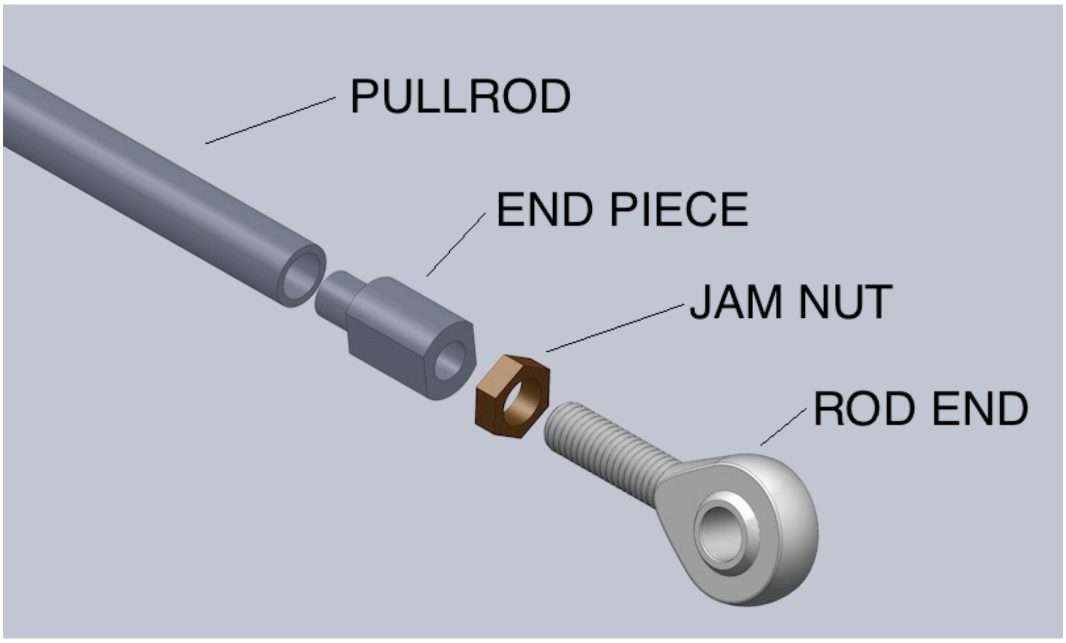

Figure 8-16: Pull-rod components exploded view

Positive bump stops on the suspension will prevent the wishbones from exceeding their maximum travel. Jounce will be limited to 30mm using silasto bump rubbers on the dampers. These will provide an increasing resistance just before reaching maximum and should reduce the effects of unpredictable handling when the suspension bottoms out (effectively making the ride rate infinite)(Smith 1978). However during competition the suspension should not reach the bump stops, if this does happen stiffer springs should be fitted. In rebound, the rockers will be restrained also to 30mm of travel by detachable steel wires attached to the chassis. These wires will only need to support the weight of the unsprung mass and a minor load from the spring.

8.4.3 Steering

The steering system was designed only up to the rack leaving the steering wheel and column placement to the chassis design team. The specifications of the steering rack are discussed in 7.2.3. Tie rod kits specifically for the 14" steering rack can be bought over the internet for $US47.95 (Dan's Performance 2011) and include 3/8" rod ends, variable length tie rods, clevises, nuts and bolts. This represents exceptional value for money and will greatly reduce the amount of time spent compared to manufacturing them in house. The steering system design has incorporated the use of this kit with confidence in its strength having been intended for heavy duty, off-road buggy use.

Positive bump stops will be incorporated into the rack (ie. stops welded on) but cannot be designed until the rack is purchased and examined.

When the rack was placed behind the front wishbones, the tie rods were at a large angle (relative to the rack) producing more than the intended 100% Ackerman. To resolve this issue the front track was shifted back 25mm, shortening the wheelbase to 1575mm. With the steering rack placed 425mm from the front bulkhead the angle was now acceptable (Figure 8-17), although still producing more than 100% Ackerman. This should not be an issue as recent UWAM vehicles have been running more than 100% Ackerman to scrub the front inside tyre helping it reach operating temperature and get better turn in response. Some teams have adjustable mounting of the steering rack (forwards and backwards) to allow tuning of Ackerman for specific tracks (Pat's Corner 2005). This will depend however on the chassis design and space available around the steering rack. This change displays the importance of incorporating a small margin during preliminary design, as the 1600mm chosen for the wheelbase was longer than the rules stipulated but allowed a packing issue to be overcome with minimal impact on the suspension geometry.

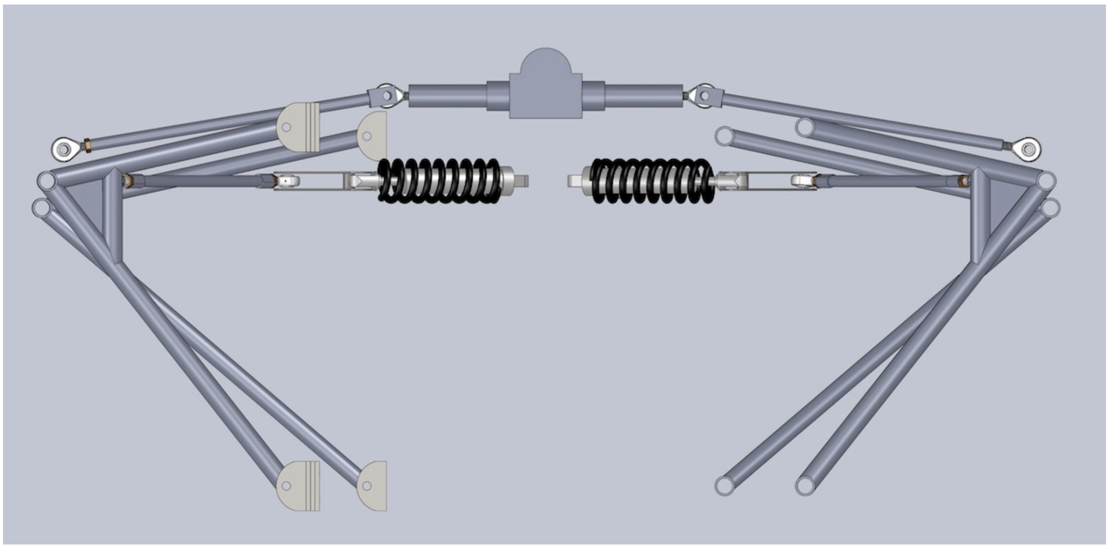

Figure 8-17: Steering rack location (top view of front suspension)

When the suspension was tested for contact under full steering lock it was discovered the asymmetrical design of the wishbone meant it would come into contact with the outside wheel very early (10 °) under steering lock. Serious redesign of the suspension would be needed to counter this problem. Changing the asymmetrical design of the wishbone was not a viable option because it would increase the weight of the chassis with additional members and add even more Ackerman to the steering. Three concurrent adjustments would be made to minimise the impact of changes on the suspension geometry. Firstly the front track was increased by 60mm to 1260mm. This meant the outer suspension pivots would be 30mm further away from the wheel centre line, greatly reducing contact issues especially under full jounce and rebound conditions. Increasing the track was chosen because it had minimal impact on the suspension geometry, only increasing the scrub radius by 30mm to 70mm. Milliken & Milliken (1995) states race cars that run on smooth tracks can get away with a relatively large scrub radius without much repercussion although fatigue of the drivers arms may be a concern during endurance. In case 70mm scrub radius was too large, 1 ° of kingpin angle was incorporated that reduced the scrub radius to 63.5mm. Furthermore the upper outer suspension pivot was lowered 10mm (to 115mm from the wheel centre) as the upper wishbone was still making contact under full steering lock. This would have the effect of raising the roll centre and increasing swing arm length thereby changing the camber curve. However both effects would have been marginal and were not analysed further due to insufficient time. After these changes the suspensions' maximum steering angles are sufficient to make an 8m diameter turn and are listed in Table 8-2 below.

	Outside wheel (deg)	**Inside wheel (deg)**
Required for 8m diameter turn	19.2	25.2
25mm Jounce	24	37
25 mm rebound	23	36

Table 8-2: Steering angles before contact

9 Safety

9.1 Lab safety

The REV team conducts it work in Lab G50 of the Electrical Engineering building. This lab is vulnerable to many potential hazards and it is therefore a requirement that all students undergo a safety induction prior to obtaining access. Safety induction covers:

- emergency contact numbers (on campus: 2222, off campus: 6488 2222)
- location of fire extinguishers, first aid (G62EE), emergency exits and assembly muster points in case of fire
- PPE (Personal Protective Equipment) requirements of always wearing closed shoes, leather gloves and safety glasses when carrying out mechanical work and rubber gloves when carrying out electrical work
- requirement to tag any faulty equipment
- potentially hazardous work must be conducted in pairs and can not be after hours
- emergency stop switch must be pushed in when working on a electric vehicle.
- use car stands when working under a vehicle
- visitors must be supervised by a person that has conducted safety induction
- if someone is electrocuted: push person away with a non-conducting object, seek medical advice regardless of injury, report the incident (legal requirement)
- no eating, drinking or smoking in the lab

9.2 Motorsport: Track Racing

Motorsport is a dangerous activity and will always involve some risk no matter how well thought out are attempts at controlling such risks. Major health risks to the driver of the vehicle are from the result of a crash with another vehicle or into an object. In limiting said risks posed by competitors to themselves and others are the safety requirements of the FSAE rules. These include the impact attenuator on the front of the vehicle designed to absorb energy from a frontal crash incident with minimum requirements of decelerating a vehicle from 7.0 m/s with less than 40 g's peak and 20 g's average (FSAE Rules 2011). There are also minimum requirements for the steel tube size and layout of the driver's cockpit including the main roll hoop, front roll hoop and intrusion bars to protect the driver. There is a multitude of PPE that must be worn by drivers to further protect their safety including helmets and fire resistant clothing.

Normal road vehicles are usually set up to understeer once they exceed their tyres maximum grip. This produces a more predictable handling vehicle for the regular drivers natural tendency to add more steering angle or step on the brakes. Motorsport vehicles are set up to have neutral steer to fully utilise the grip of all four tyres which makes the vehicle more prone to oversteer under certain load transferring conditions like lift-off oversteer, power on or applying the brakes mid corner. Oversteer results in the vehicles rear-end sliding towards the outside of the turn and possibly spinning the vehicle. Team members will have driver training to teach and practice them how to handle the vehicle in such conditions reducing the likelihood of crashing. The expected rear weight bias of the REV vehicle should provide it with more grip at the rear therefore making it more likely to display safer understeering dynamics.

9.3 Suspension failure

Failure of a suspension component during competition could result in the driver losing control of the vehicle. Therefore the suspension system has been designed with a safety factor of four described in more detail in chapter 3.6.4 Strength analysis and chapter 8 Component design. A major concern was the possibility of the suspension arms intruding into the cabin and injuring the driver in the event of large impact at the wheel. Anti-intrusion bars were added to the wishbones to prevent such an occurrence. These are not specifically required under FSAE rules but highly recommend by Pat’s Corner (2005). The anti-intrusion bars were added late in the design process and therefore do not appear in other figures or the SolidWorks drawings. On the lower wishbone the anti-intrusion bar is further outboard than usual because of the rocker position (Figure 9-1). In case of the wishbone being disjoined, the anti-intrusion bar would catch on the rocker and not protrude dangerously into the cabin area.

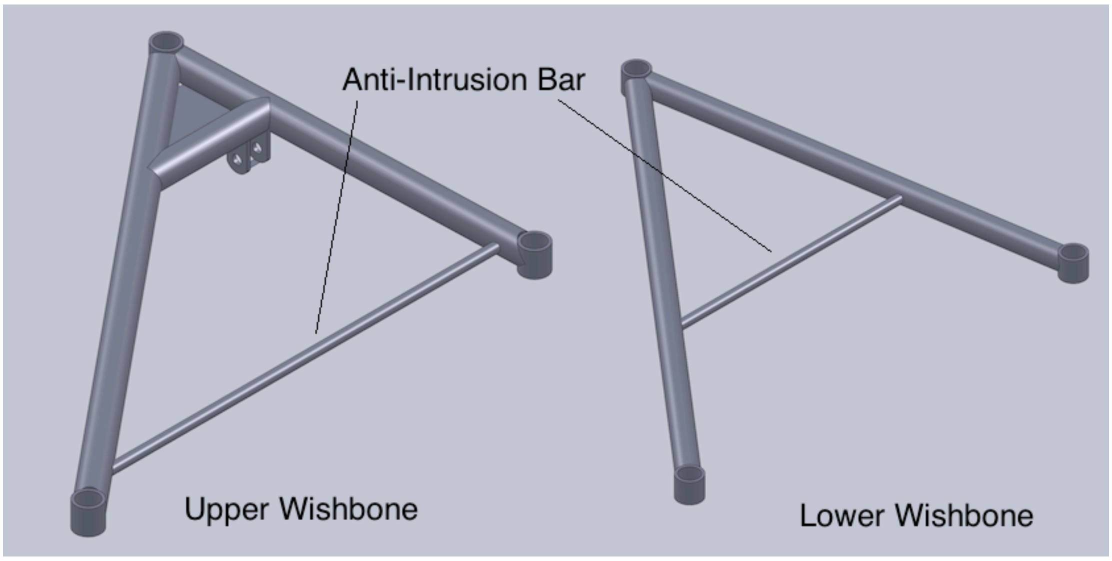

Figure 9-1: Wishbone anti-intrusion bars

10 Recommendations

With the limited team members involved (especially mechanical engineering students) the design focused on simplification in order to get a vehicle constructed and ready for competition in time. Therefore there are many aspects of the suspension that will allow for optimisation in the pursuit of greater performance. For 2011, there was only one person (the author) involved in designing the suspension system. Given the complexity of interaction of all the suspension parameters it is advisable more than one person be engaged in the design. For example, one may focus on the kinematic and component design while another optimises the spring and damper performance.

The technical improvements that can be made to this suspension design will be greatly dependent on the performance its displays after construction and physical testing. If the vehicle demonstrates excessive roll during cornering retro-fitting of anti-rolls bars may be advantageous. Anti-roll bars would also be useful if the torque-vectoring is unable to provide sufficient adjustability to achieve neutral handing. Utilisation of simulation software like ADAMS or Optimum K to model roll centre movement during cornering may be very useful in addressing unpredictable/skittish handling.

To aid fine tuning of the vehicles suspension, verification of the vehicles weight distribution and centre of gravity should be conducted prior to competition following the process described in '18.2: Centre of Gravity Location' (Milliken & Milliken

1995). The spring stiffness can than be adjusted accordingly. Current design stipulates spring stiffness of 450lbs/in at the front and 400lbs/in at the rear which might result in turn-in understeer, therefore it highly recommended the team also try with the stiffer springs at the rear and compare performance.

On future vehicles, further review of damper actuation methods would be useful as the current pull-rod design is knowingly not the most favourable solution because of the resulting bending forces. In pursuit of weight reduction, force analysis on the wishbones can be conducted using the max accelerations recorded at competition and allow the reduction of the wishbone tube sizes. If performance of the kinematic design proves to be competitive, more focus could be placed on investigating cheaper manufacturing techniques of exotic materials, such as producing carbon fibre tubes in house.

Another compromised area of this design was the selection of the FOX Van RC dampers. As the team's financial constraints were the determining factor in selection of these dampers, the financial position of the team next year may permit more expensive dampers specifically designed for FSAE. For the next vehicle, the budget for the dampers should be decided in the early stages to determine if the FOX dampers will be used again and to allow the team to re-valve them as recommended by Finalyson (2003).

As the kinematic design took place early in the project, many of the parameters were based on guidelines and past UWAM vehicles. With greater knowledge of the suspension as interacting systems, the author is questioning some of his design decisions. The following two items are the author's opinion and are not supported by specific literature nevertheless present interesting points to think about in the future. Firstly castor angle was not increased beyond 7 ° because of inadvertent effects of diagonal weight transfer. In a rear wheel drive vehicle this is undesirable because it unloads the inside rear tyre and making the vehicle easy to spin when coming out of a corner with large throttle. However in an AWD vehicle the effects of diagonal weight transfer may not be as detrimental as a smaller portion of the drive is at the rear wheels and the drive at the front wheels will aid it from spinning. Also the weight transfer to the inside front tyre from the mechanical lift will be beneficially balancing the weight transfer to the outside wheel from the lateral acceleration. Increasing castor to 10 °

could be further investigated in AWD application however without reducing to the large scrub radius currently on the vehicle, the mechanical lifting of the tyre may already be at the acceptable limit.
Secondly vehicles fitted without anti-roll bars might benefit from a higher roll centre to reduce the amount of roll during cornering. However testing this theory would probably be less time effective and more risky than designing and building anti-roll bars.

Being the first FSAE vehicle built by a new inexperienced team, there was always going to be a long list of improvements that could be made to the vehicle, particularly in regard to the complex suspension system. However all the recommendations listed may prove futile if the REV and UWAM team merger plans for next year are successful. In this case the experience gained from running an AWD vehicle with four independent motors will be combined with the expert mechanical knowledge of the UWAM team. The suspension system will likely take an extreme leap in sophistication and make most if not all the technical recommendations mentioned for future vehicles unfortunately redundant.

11 Conclusion

The design of REV's 2011 FSAE vehicle was largely influenced by the team's financial and human resources. Consequently the thesis objectives were to design a low cost, easy to manufacture suspension system that would exhibit predictable handling and thereby provide a good development base for the innovative AWD drive train.

The preliminary design stage determined many of the suspension parameters based upon the objectives of the team and restrained by the FSAE rules. The vehicle was primarily set-up for roll conditions as the tight twisty track of the FSAE competition has the vehicle cornering the majority of the time. Kinematic design followed the processes described in Milliken & Milliken (1995) using established guidelines allowing the design to be conducted on paper without sophisticated simulation software. The component design was done in CAD software SolidWorks to test for contact issues, verify kinematic design and perform rudimentary stress analysis on some of the components. AISI 4130 steel was determined to be the most appropriate material for the wishbones, pull-rods and rockers.

The design attempted to emulate the 2001-2003 UWAM suspension systems as much as possible as the author was responsible for the REV vehicle's entire suspension system that involved selecting a compromises between complexly interrelating factors. This difficulty manifested in the constant changes needed during the design process. Many of the solutions decided upon are not ideal (like the pull-rod actuation that results in extra bending stress on the wishbones) and therefore there are many opportunities for improvement to future vehicles.

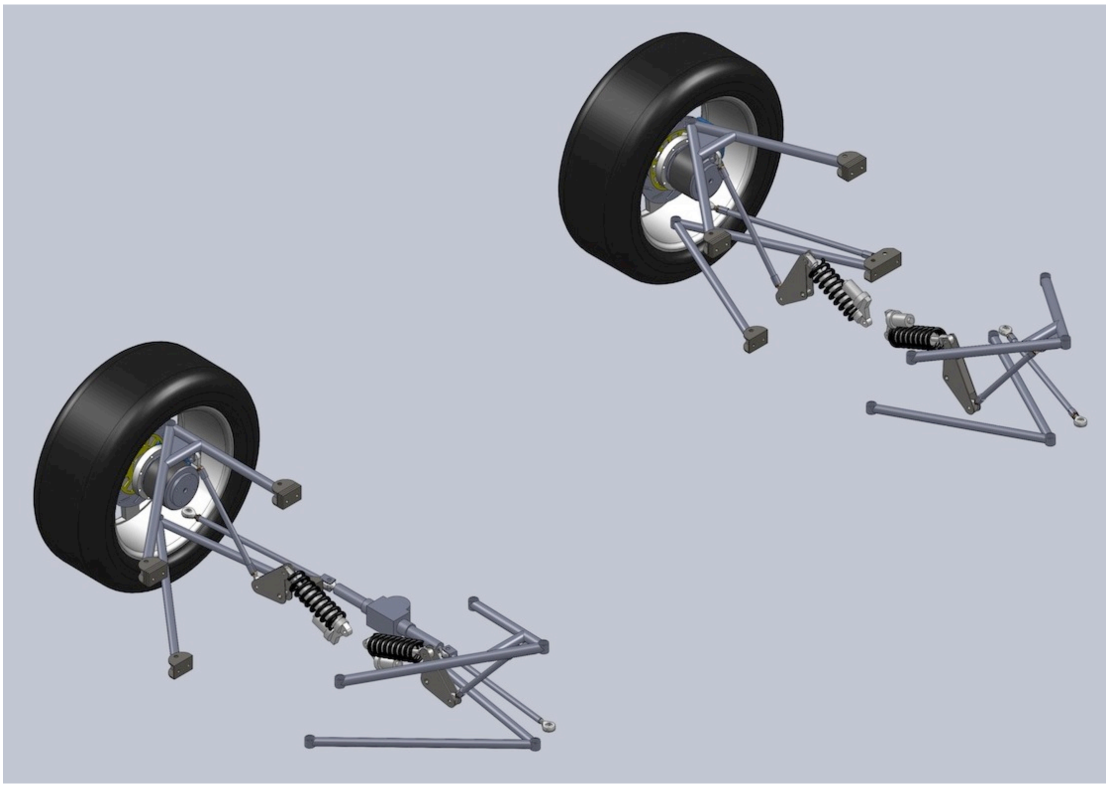

Figure 11-1: Complete suspension design

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13 Appendix

13.1 Wheel-hub / Upright dimensions

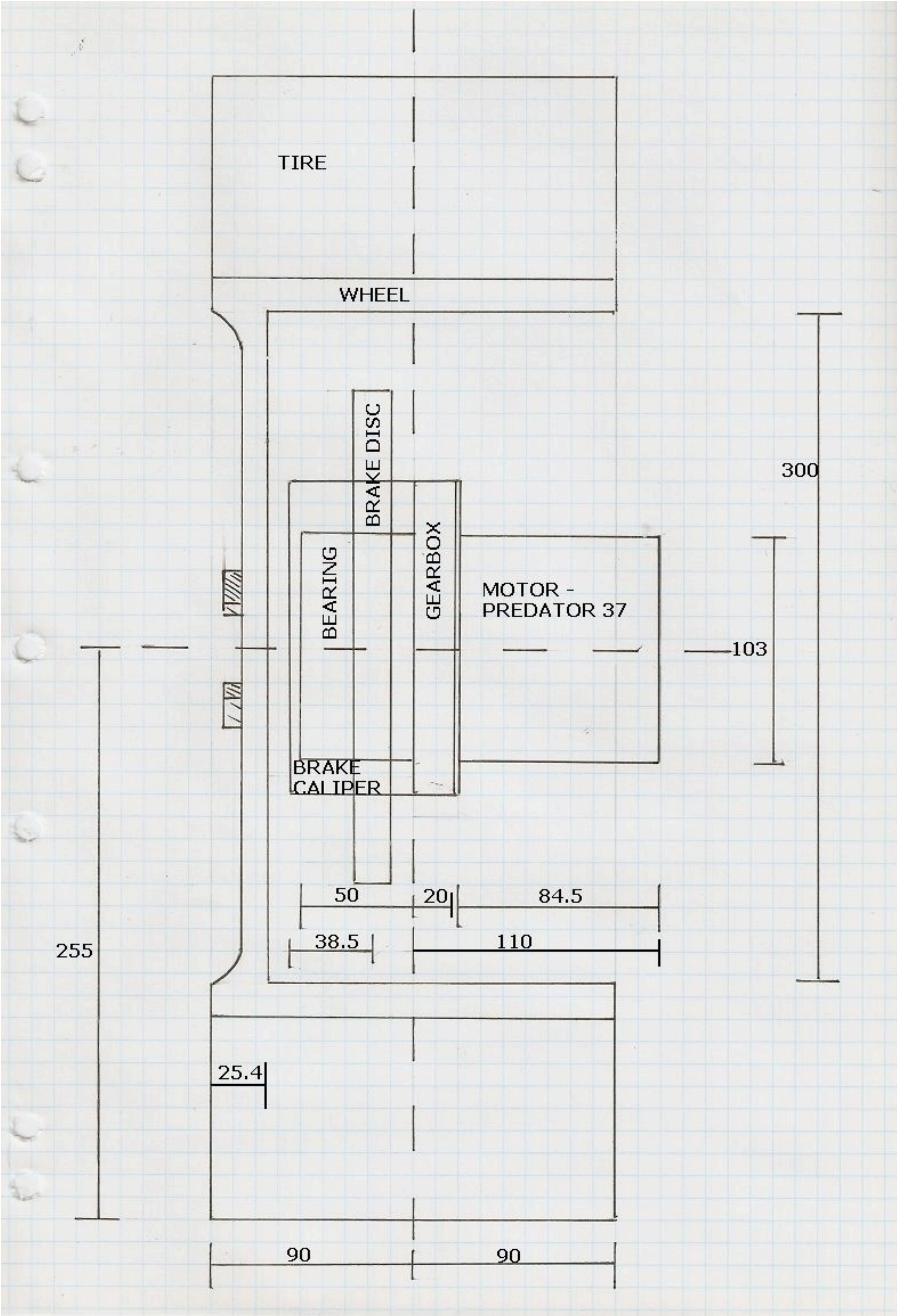


Figure 13-1: Upright and motor dimensions

13.2 FSAE Rules

Rule #	Description	Condition met
A	**Events**	
A6.1	**Student Developed Vehicle** Vehicles entered into Formula SAE competitions must be conceived, designed, fabricated and maintained by the student team members without direct involvement from professional engineers, automotive engineers, racers, machinists or related professionals.	√
A6.4	**Student Fabrication** It is the intent of the SAE Collegiate Design Series competitions to provide direct hands-on experience to the students. Therefore, students should perform all fabrication tasks whenever possible.	√
B	**Technical Requirements**	
B2.3	**Wheelbase** The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the center of ground contact of the front and rear tires with the wheels pointed straight ahead.	√
B2.4	**Vehicle Track** The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track.	√
B3.4.2	Titanium tubing on which welding has been utilized cannot be used for any tubing in the Primary Structure. This includes the attachment of brackets to the tubing or the attachment of the tubing to other components.	√
B3.22.1	*Except as allowed by B3.22.2, a*ll non-crushable objects (e.g. batteries, master cylinders, hydraulic reservoirs) must be rearward of the bulkhead. No non-crushable objects are allowed in the impact attenuator zone.	√
B6.1	**Suspension**	
B6.1.1	The car must be equipped with a fully operational suspension	

	system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.	√
B6.1.2	All suspension mounting points must be visible at Technical Inspection, either by direct view or by removing any covers.	√
B6.2	**Ground Clearance** There is no minimum ground clearance requirement. However, teams are reminded that under Rule D1.1.2 any vehicle condition which could, among other things, "... compromise the track surface" is a valid reason for exclusion from an event. Any vehicle contact that creates a hazardous condition or which could damage either the track surface or the timing system is cause for declaring a vehicle DQ.	√
B6.3	**Wheels** The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter.	√
B6.4.1	Vehicles may have two types of tires as follows: • Dry Tires – The tires on the vehicle when it is presented for technical inspection are defined as its "Dry Tires". The dry tires may be any size or type. They may be slicks or treaded. • Rain Tires – Rain tires may be any size or type of treaded or grooved tire provided: 1. The tread pattern or grooves were molded in by the tire manufacturer, or were cut by the tire manufacturer or his appointed agent. Any grooves that have been cut must have documentary proof that it was done in accordance with these rules. 2. There is a minimum tread depth of 2.4 mms (3/32 inch). **Note:** Hand cutting, grooving or modification of the tires by	√

	the teams is specifically prohibited.	
B6.5	**Steering**	
B6.5.1	The steering system must affect at least two (2) wheels.	√
B6.5.2	The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four-bar linkage at one of the pivots). The stops may be placed on the uprights or on the rack and must prevent the tires from contacting suspension, body, or frame members during the track events.	√
B6.5.3	Allowable steering system free play is limited to seven degrees (7°) total measured at the steering wheel.	√
B6.6	**Jacking point**	
B6.6.1	A jacking point, which is capable of supporting the car's weight and of engaging the organizers' "quick jacks", must be provided at the very rear of the car.	√
B6.6.2	The jacking point is required to be: • Visible to a person standing 1 metre (3 feet) behind the car. • Painted orange. • Oriented horizontally and perpendicular to the centerline of the car • Made from round, 25 – 29 mm (1 – 1 1/8 inch) O.D. aluminium or steel tube • A minimum of 300 mm (12 inches) long • Exposed around the lower 180 degrees (180°) of its circumference over a minimum length of 280 mm (11 in) • The height of the tube is required to be such that: - There is a minimum of 75 mm (3 in) clearance from the bottom of the tube to the ground measured at tech inspection. - With the bottom of the tube 200 mm (7.9 in) above ground, the wheels do not touch the ground when they are in full rebound.	√
B6.7	**Rollover Stability**	

B6.7.1	The track and center of gravity of the car must combine to provide adequate rollover stability.	√
B6.7.2	Rollover stability will be evaluated on a tilt table using a pass/fail test. The vehicle must not roll when tilted at an angle of sixty degrees (60°) to the horizontal in either direction, corresponding to 1.7 G's. The tilt test will be conducted with the tallest driver in the normal driving position.	√
B14.1	**Fastener Grade Requirements**	
B14.1.1	All threaded fasteners utilized in *the driver's cell structure,* and the steering, braking, driver's harness and suspension systems must meet or exceed, SAE Grade 5, Metric Grade 8.8 and/or AN/MS specifications.	√
B14.2.1	All critical bolt, nuts, and other fasteners on the steering, braking, driver's harness, and suspension must be secured from unintentional loosening by the use of positive locking mechanisms. Positive locking mechanisms include: • Correctly installed safety wiring • Cotter pins • Nylon lock nuts • Prevailing torque lock nuts **Note:** Lock washers and thread locking compounds, e.g. Loctite®, DO NOT meet the positive locking requirement.	√
B14.2.2	There must be a minimum of two (2) full threads projecting from any lock nut.	√
B14.2.3	All spherical rod ends and spherical bearings on the steering or suspension must be in double shear or captured by having a screw/bolt head or washer with an O.D. that is larger than spherical bearing housing I.D.	√
B14.2.4	Adjustable tie-rod ends must be constrained with a jam nut to prevent loosening.	√

Table 13-1: Relevant FSAE Rules Checklist (FSAE 2011)

Static Events	**Points**
Cost and Manufacturing	100
Presentation	75
Design	150
Sub-Total	*325*
Dynamic Events	
Acceleration	75
Skid Pad	50
Autocross	150
Fuel Economy	100
Endurance	300
Sub-Total	*675*
Total	*1000*

Table 13-2: FSAE Point Scoring (FSAE 2011)

13.3 Preliminary Calculations

Weight Distribution and Centre of Gravity of Vehicle

CENTRE OF GRAVITY

**Z Direction measured in 'mm' from front bulkhead.*
Y dierection measured in 'mm' from bottom of chassis.

Ref.	Item	Mass		Z	Y	Mz	My
D	driver		70	1250	350	87500	24500
B	battery		65	1200	100	78000	6500
C	chassis		40	1220	280	48800	11200
S	suspension arm		3.5	250	200	875	700
			3.5	250	200	875	700
			3.5	1900	200	6650	700
			3.5	1900	200	6650	700
W	wheel	4					
	tyre	3					
	brakes	1					
	upright	4					
	gearbox	2					
	motor	3					
	TOTAL		17	300	195	5100	3315
			17	300	195	5100	3315
			17	1900	195	32300	3315
			17	1900	195	32300	3315
H	shocks		1	300	250	300	250
			1	300	250	300	250
			1	1900	250	1900	250
			1	1900	250	1900	250
P	pedal box		3	200	100	600	300
T	steering rack		2	25	300	50	600
O	steering column		2	700	360	1400	720
E	seat+harness		7	1250	120	8750	840
R	electronics		10	1900	200	19000	2000
Y	body		20	1220	280	24400	5600
I	wiring		5	1100	50	5500	250
	TOTAL		310			368250	69570
					CG	**1187.9**	**224.4**

WEIGHT DISTRIBUTION		
	FRONT	**REAR**
	44.5%	55.5%

Table 13-3: Weight and centre of gravity estimation

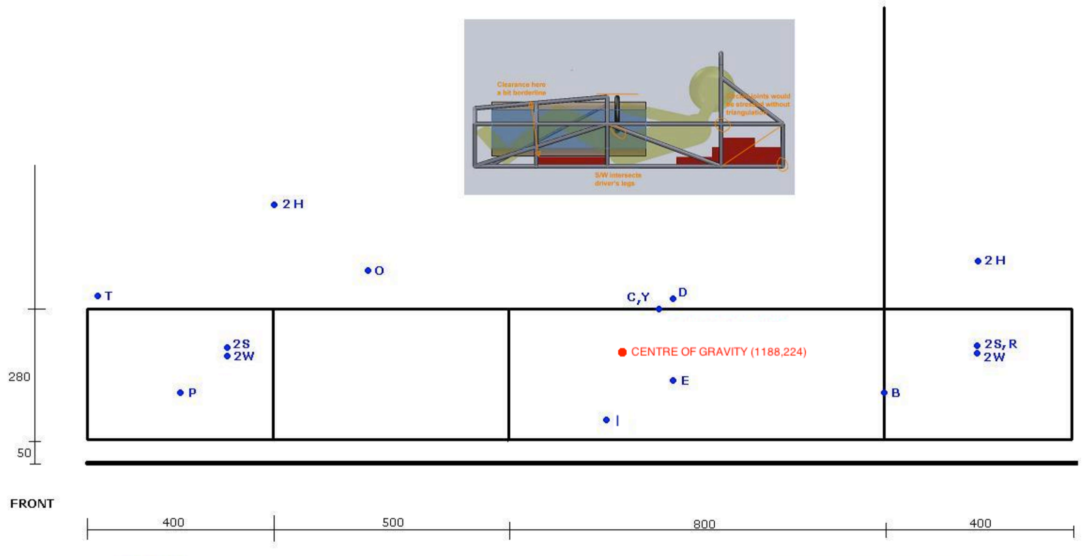


Figure 13-2: Weight and centre of gravity estimation

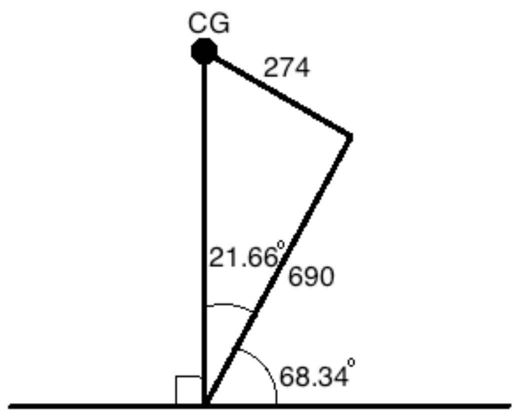


Figure 13-3: Tilt test angle

Roll Stiffness

$$K_{\phi} = \frac{m \times H}{\theta}$$

$$K_{\phi} = \frac{310kg \times 0.235m}{(1.5 \times \Pi \div 180) rad / g}$$

$$K_{\phi} = 2787.03 kgm / rad$$

Roll Stiffness Distribution

$$K_{\phi FS} = K_{\phi} \times 0.52$$

$$K_{\phi FS} = 2787.03 kgm / rad \times 0.52$$

$$K_{\phi FS} = 1449.24 kgm / rad$$

$$Kr = K_{\phi} \times 0.48$$

$$K_{\phi Fr} = 2787.03 kgm/rad \times 0.48$$

$$K_{\phi FS} = 1337.76 kgm/rad$$

Weight Transfer

$$\Delta W_{YF} = A_y \times \frac{m}{t_F} \times \left[\frac{H \times K_{\phi F}}{K_{\phi}} + \frac{b}{l} \times z_{RF} \right]$$

$$\Delta W_{YF} = 1.2 \times \frac{310kg}{1.2m} \times \left[\frac{0.235m \times 1449.24 kgm/rad}{2787.03 kgm/rad} + \frac{0.712m}{1.6m} \times 0.025m \right]$$

$$\Delta W_{YF} = 41.33kg$$

$$\Delta W_{YR} = A_y \times \frac{m}{t_R} \times \left[\frac{H \times K_{\phi R}}{K_{\phi}} + \frac{a}{l} \times z_{RR} \right]$$

$$\Delta W_{YR} = 1.2 \times \frac{310kg}{1.2m} \times \left[\frac{0.235m \times 1337.76 kgm/rad}{2787.03 kgm/rad} + \frac{0.888m}{1.6m} \times 0.05m \right]$$

$$\Delta W_{YR} = 43.57kg$$

Ride Rate

$$K_{RF} = \frac{2 \times K_{\phi FS}}{t_F^{\;2}}$$

$$K_{RF} = \frac{2 \times 1449.24 kgm/rad}{(1.2m)^2}$$

$$K_{RF} = 2012.83 kg/m$$

$$K_{RR} = \frac{2 \times K_{\phi RS}}{t_R^{\;2}}$$

$$K_{RR} = \frac{2 \times 1337.76 kgm/rad}{(1.2m)^2}$$

$$K_{RR} = 1857.78 kg/m$$

Wheel Rates

$$K_{WF} = \frac{K_T K_{RF}}{K_T - K_{RF}}$$

$$K_{WF} = \frac{18367.97 kg/m \times 2012.83 kg/m}{18367.97 kg/m - 2012.83 kg/m}$$

$$K_{WF} = 2260.55 kg/m$$

$$K_{WR} = \frac{K_T K_{RR}}{K_T - K_{RR}}$$

$$K_{WR} = \frac{18367.97 kg/m \times 1857.78 kg/m}{18367.97 kg/m - 1857.78 kg/m}$$

$$K_{WR} = 2066.82 kg/m$$

Installation Ratio

$$IR_F = \sqrt{\frac{K_{WF}}{K_S}}$$

$$IR_F = \sqrt{\frac{2260.55 kg/m}{8036.1 kg/m}}$$

$$IR_F = 0.530$$

$$IR_R = \sqrt{\frac{K_{WR}}{K_S}}$$

$$IR_R = \sqrt{\frac{2066.82 kg/m}{7143.2 kg/m}}$$

$$IR_R = 0.538$$

13.4 Kinematic Design

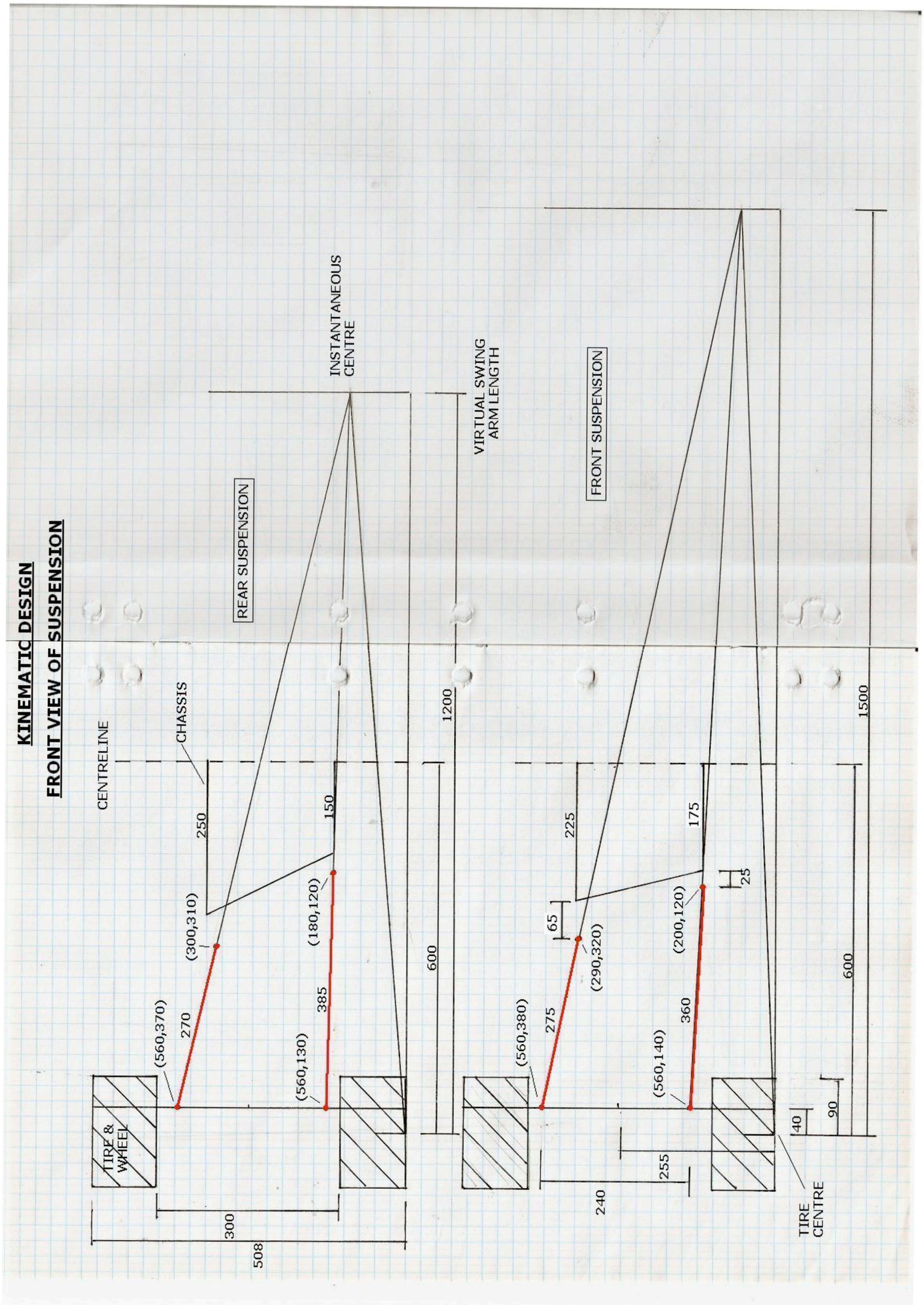


Figure 13-4: Kinematic Design

13.5 Kinematic Design Calculation

- Steering

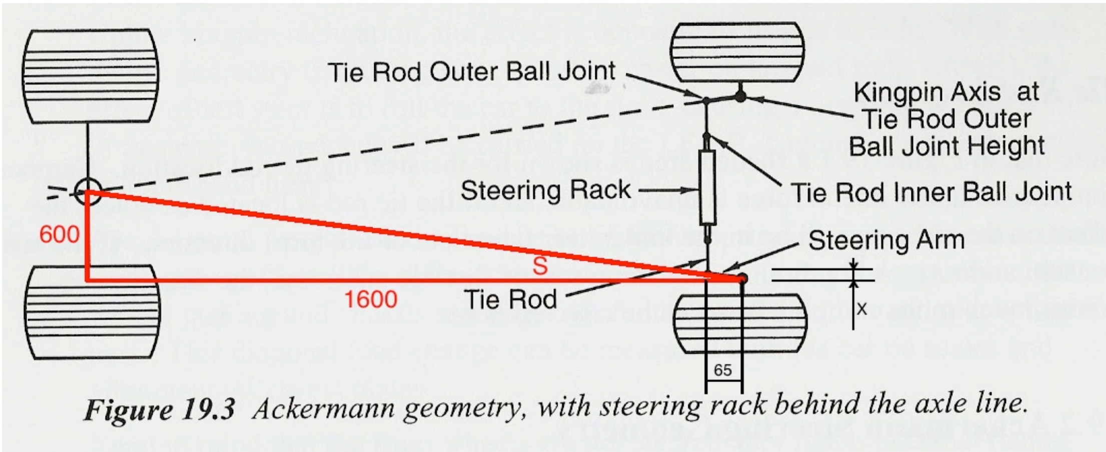


Figure 13-5: Steering angle calculation (Milliken & Milliken 1995, p.714)

Steering arm angle:

$$\tan(S) = 600/1600$$

$$S = 20.55\ °$$

Horizontal distance between tyre rod outer pivot and steering axis:

$$\tan(S) = X / 65$$

$$X = 24.375\text{mm}$$

- 14” Dune Buggy Steering Rack

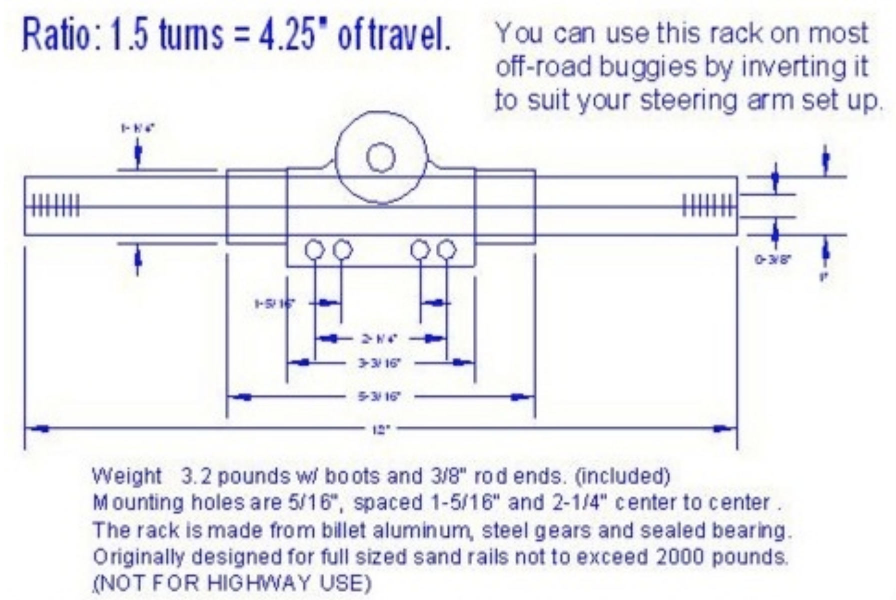


Figure 13-6: 14" dune buggy steering rack (Desert Karts 2011)

13.6 Component Design Calculations

Figure 13-7: Tyre loading under 1.2g lateral acceleration and (1.5g braking combined with 0.5g lateral acceleration)

Maximum expected load on wheel = 130kg = 1274N

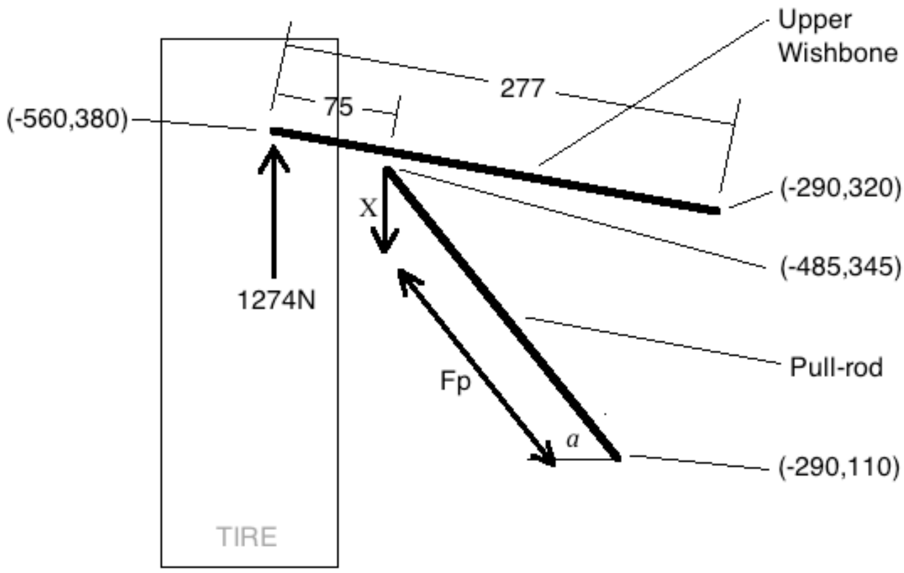


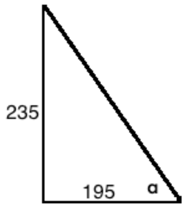
Figure 13-8: 130kg load on front suspension

Moment considered about inner pivot:

$X(277-75)=1274N(277)$

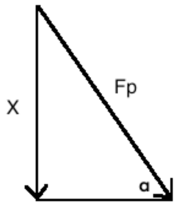
$X=1747N$

Angle of pull-rod:



$$\tan\alpha = 235/195$$
$$\alpha = 50.31°$$

Tension force in pull-rod necessary to create X vertical force:



$$\sin\alpha = 1747N/Fp$$
$$Fp = 2270N$$

Spherical Bearing (Aurora PWB-5T) safety factor:

$$SF = 7.3kN/1274N = 5.7$$

Rod end (Aurora AM-5T) safety factor:

$$SF = 34kN/2270N = 15.0$$

Pull-rod safety factor:

$$\sigma = \frac{F}{A} = \frac{F}{\pi(O.D.^2 - I.D.^2)/4}$$
$$\sigma = \frac{2270N}{\pi(0.0127m^2 - 0.010922m^2)/4}$$
$$\sigma = 68815703Pa = 68.8MPa$$

SF = 68.8MPa/360MPa = 5.23

13.7 Go-gear 4130 tube pricing for December 2010

	4130 CHROME MOLY TUBE & SHEET			20/12/2010			
				Price per m	full length	Price per m	cut length
	Size O.D.	Size I.D.	Material	full length	p/m inc GST	cut length	p/m inc GST
	1/4" .049	0.152	4130 Round tube	21.81	23.99	23.99	26.39
	5/16" .035	0.243	4130 Round tube	19.12	21.03	21.03	23.13
	5/16" .049	0.215	4130 Round Tube	20.42	22.47	22.47	24.71
	3/8" .035	0.305	4130 Round Tube	17.97	19.77	19.77	21.75
	3/8" .049	0.259	4130 Round Tube	15.85	17.43	17.43	19.18
	3/8" .058	0.277	4130 Round Tube	17.65	19.41	19.41	21.35
	3/8" .065	0.245	4130 Round Tube	18.30	20.13	20.13	22.14
	7/16" .065	0.308	4130 Round Tube	21.08	23.18	23.18	25.50
Pullrod	1/2" .035	0.43	4130 Round Tube	17.97	19.77	19.77	21.75
	1/2" .049	0.402	4130 Round Tube	16.83	18.51	18.51	20.36
	1/2" .058	0.384	4130 Round Tube	17.07	18.78	18.78	20.66
Rocker	1/2" .065	0.37	4130 Round Tube	15.52	17.07	17.07	18.78
	1/2" .083	0.334	4130 Round Tube	22.47	24.71	24.71	27.18
End piece	1/2" .120	0.26	4130 Round Tube	31.21	34.33	34.33	37.76
	9/16" .065	0.433	4130 Round Tube	34.07	37.47	37.47	41.22
	5/8" .035	0.533	4130 Round Tube	11.68	12.85	12.85	14.14
	5/8" .049	0.527	4130 Round Tube	15.19	16.71	16.71	18.39
	5/8" .058	0.509	4130 Round Tube	13.07	14.38	14.38	15.82
	5/8" .065	0.495	4130 Round Tube	18.46	20.31	20.31	22.34
	5/8" .083	0.459	4130 Round Tube	17.07	18.78	18.78	20.66
	5/8" .120	0.385	4130 Round Tube	24.34	26.78	26.78	29.46
	3/4" .035	0.68	4130 Round Tube	13.48	14.83	14.83	16.31
Lower Wishbone	3/4" .049	0.652	4130 Round Tube	13.72	15.10	15.10	16.61
	3/4" .058	0.634	4130 Round Tube	16.75	18.42	18.42	20.26
	3/4" .065	0.62	4130 Round Tube	15.77	17.34	17.34	19.08
	3/4" .083	0.584	4130 Round Tube	19.36	21.30	21.30	23.43
	3/4" .095	0.56	4130 Round Tube	19.44	21.39	21.39	23.53
	3/4" .120	0.51	4130 Round Tube	26.14	28.76	28.76	31.63
	3/4" .156	0.438	4130 Round Tube	28.59	31.45	31.45	34.60
	7/8" .035	0.805	4130 Round Tube	14.05	15.46	15.46	17.00
Upper Wishbone	7/8" .049	0.777	4130 Round Tube	12.83	14.11	14.11	15.52
	7/8" .058	0.759	4130 Round Tube	16.34	17.97	17.97	19.77
	7/8" .065	0.745	4130 Round Tube	21.08	23.18	23.18	25.50
	7/8" .083	0.709	4130 Round Tube	23.12	25.43	25.43	27.97
	7/8" .095	685	4130 Round Tube	18.95	20.85	20.85	22.93
Bearing Housing	7/8" .120	0.635	4130 Round Tube	31.53	34.69	34.69	38.16
	1" .035	0.93	4130 Round Tube	17.40	19.14	19.14	21.05
	1" .049	0.902	4130 Round Tube	17.97	19.77	19.77	21.75
	1" .058	0.884	4130 Round Tube	17.32	19.05	19.05	20.96
	1" .065	0.87	4130 Round Tube	17.07	18.78	18.78	20.66
	1" .083	0.834	4130 Round Tube	21.81	23.99	23.99	26.39
	1" .095	0.81	4130 Round Tube	20.34	22.38	22.38	24.61
	1" .120	0.72	4130 Round Tube	30.96	34.06	34.06	37.46
	1" .250		4131 Round Tube				
	1 1/8" .049	1.027	4130 Round Tube	20.01	22.02	22.02	24.22
	1 1/8" .058	1.009	4130 Round Tube	17.81	19.59	19.59	21.55
	1 1/8" .065	0.995	4130 Round Tube	17.16	18.87	18.87	20.76
	1 1/8" .083	0.959	4130 Round Tube	17.16	18.87	18.87	20.76
	1 1/8" .095	0.935	4130 Round Tube	19.44	21.39	21.39	23.53
	1 1/4" .049	1.152	4130 Round Tube	21.24	23.36	23.36	25.70
	1 1/4" .058	1.134	4130 Round Tube	16.67	18.33	18.33	20.16
	1 1/4" .065	1.12	4130 Round Tube	23.28	25.61	25.61	28.17
	1 1/4" .083	1.084	4130 Round Tube	21.24	23.36	23.36	25.70
	1 1/4" .095	1.06	4130 Round Tube	22.55	24.80	24.80	27.28
	1 1/4" .120	1.01	4130 Round Tube	24.92	27.41	27.41	30.15
	1 3/8" .049	1.277	4130 Round Tube	16.34	17.97	17.97	19.77
	1 3/8" .058	1.259	4130 Round Tube	20.01	22.02	22.02	24.22
	1 3/8" .065	1.245	4130 Round Tube	18.79	20.67	20.67	22.73
	1 3/8" .083	1.209	4130 Round Tube	29.98	32.98	32.98	36.28
	1 3/8" .095	1.185	4130 Round Tube	28.76	31.63	31.63	34.79

Figure 13-9: Go-Gear 4130 tube pricing for December 2010 (Go Gear 2010)

13.8 Rocker FE Analysis

Force Calculation:

Force from pull-rod per rocker face:

2270N (Appendix 13.6) / 2 = 1135N

Force from damper/spring:

$$IR = \sqrt{\frac{K_W}{K_S}} \Rightarrow IR^2 = \frac{K_W}{K_S} \Rightarrow K_S = \frac{K_W}{IR^2}$$

$$F_{SF} = \frac{F_{WF}}{IR_F^{\ 2}} = \frac{1274N}{0.53^2} = 4535N$$

$$F_{SR} = \frac{F_{WR}}{IR_R^{\ 2}} = \frac{1274N}{0.538^2} = 4402N$$

Force from damper/spring per rocker face:

$$F_{SF}/2 = 2268N$$
$$F_{SR}/2 = 2201N$$

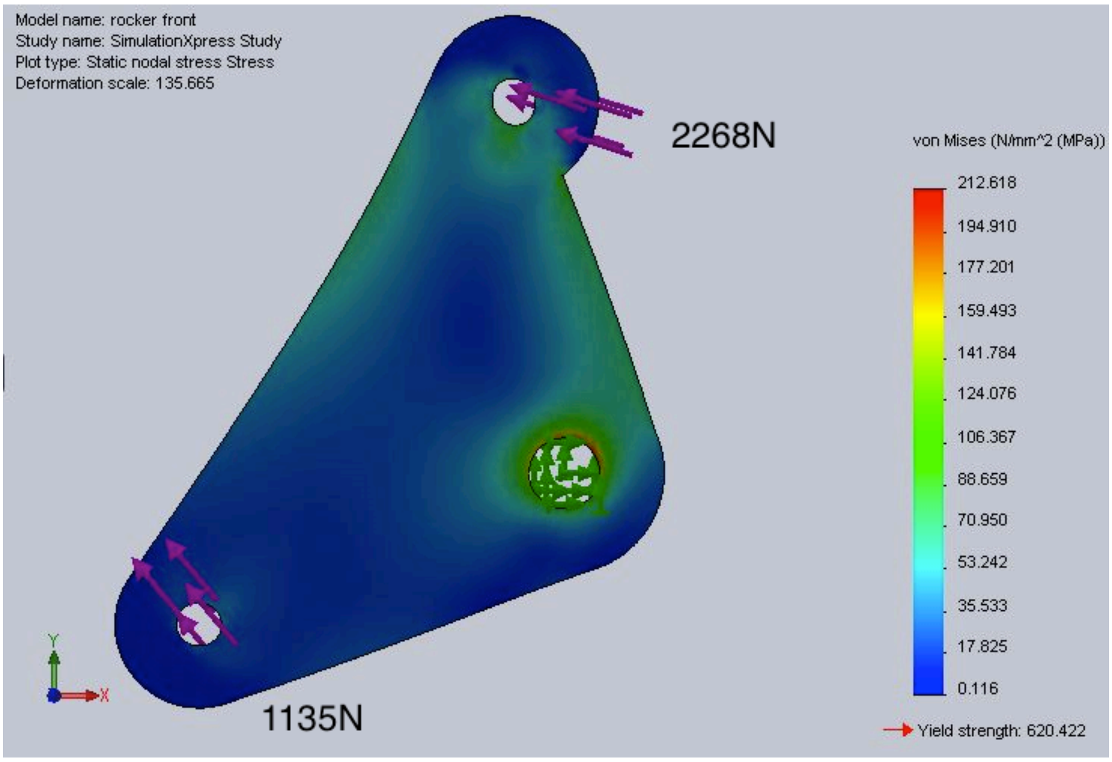


Figure 13-10: Front rocker - Von Mises stresses

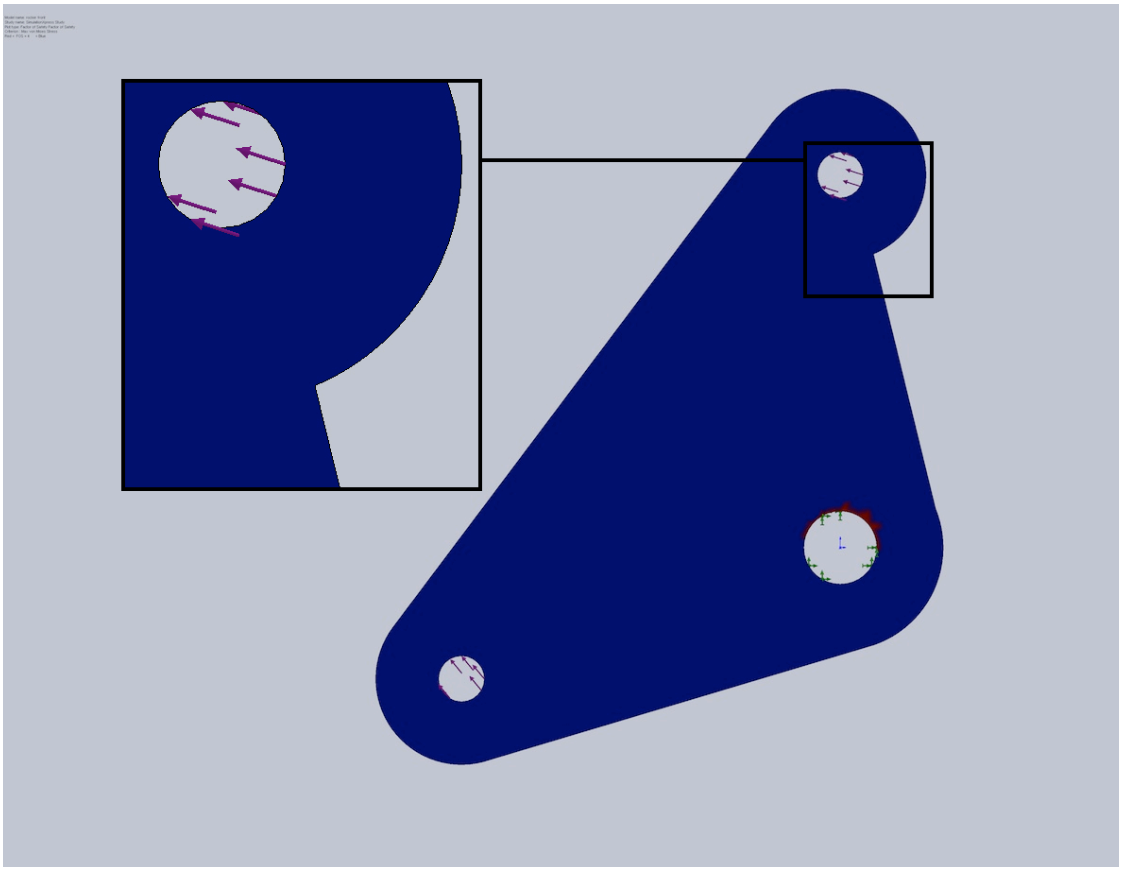


Figure 13-11: Front rocker – Factor of safety of 4

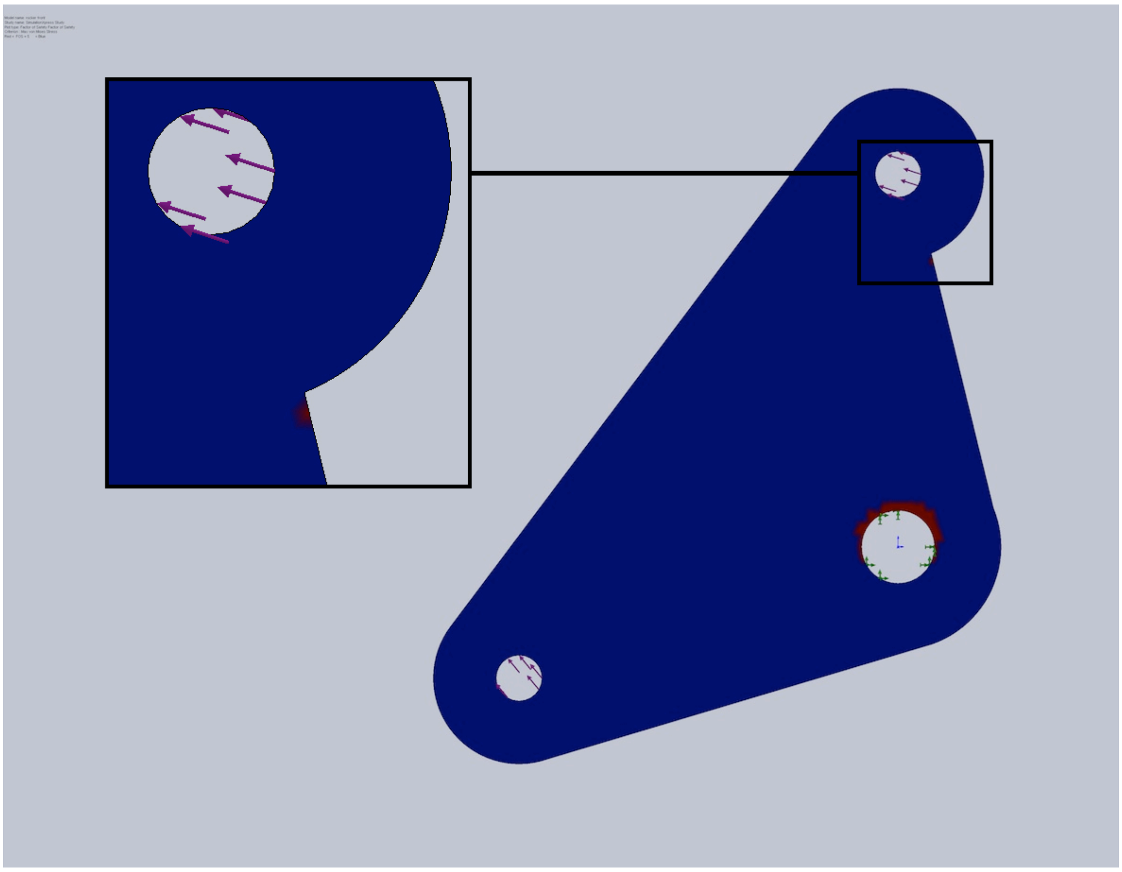


Figure 13-12: Front rocker – Factor of safety of 5

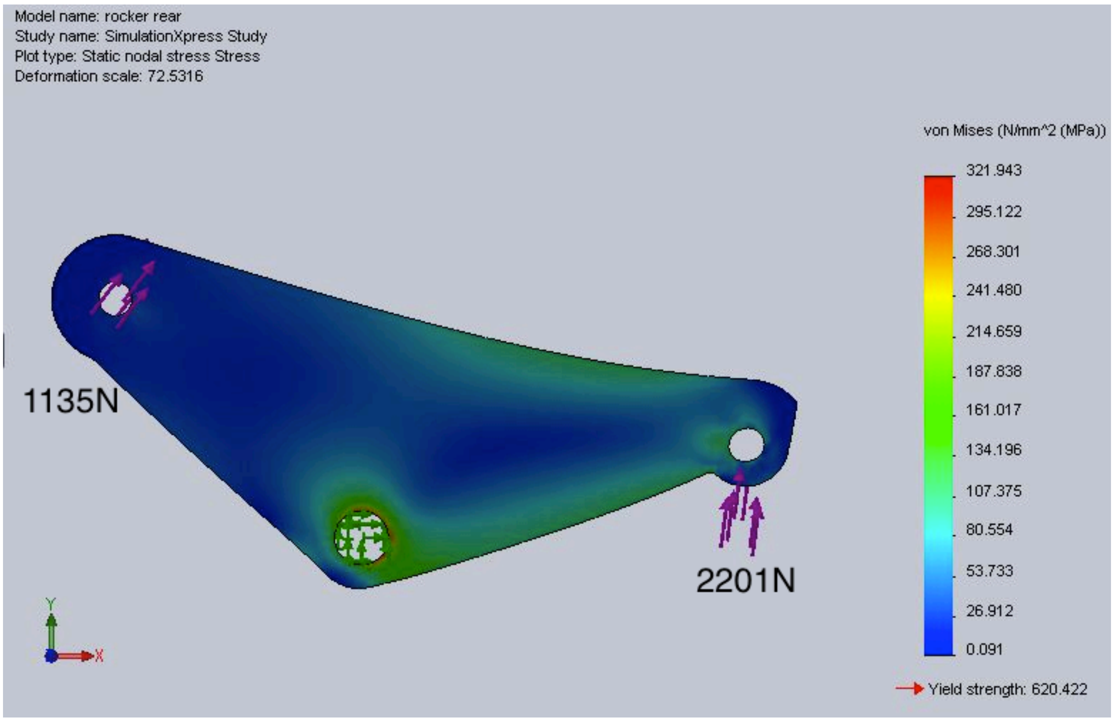


Figure 13-13: Rear rocker - Von Mises stresses

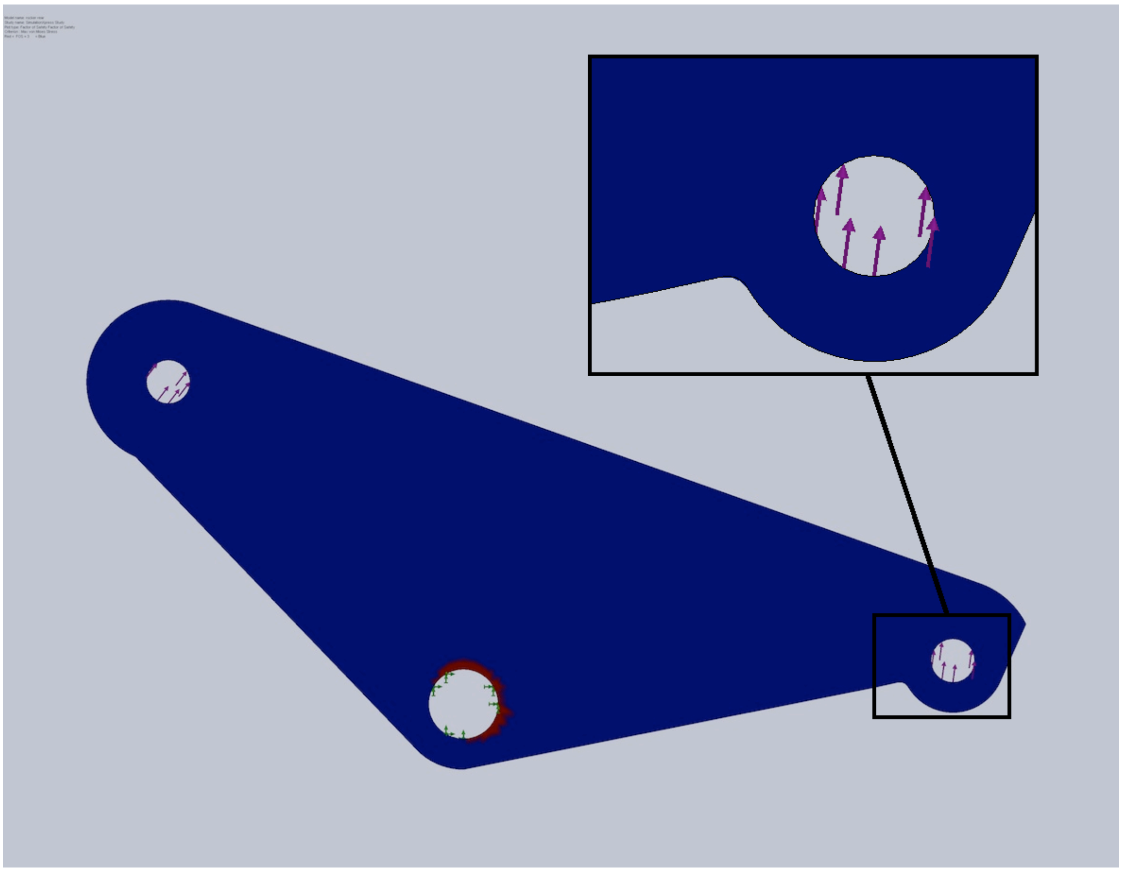


Figure 13-14: Rear rocker – Factor of safety 3

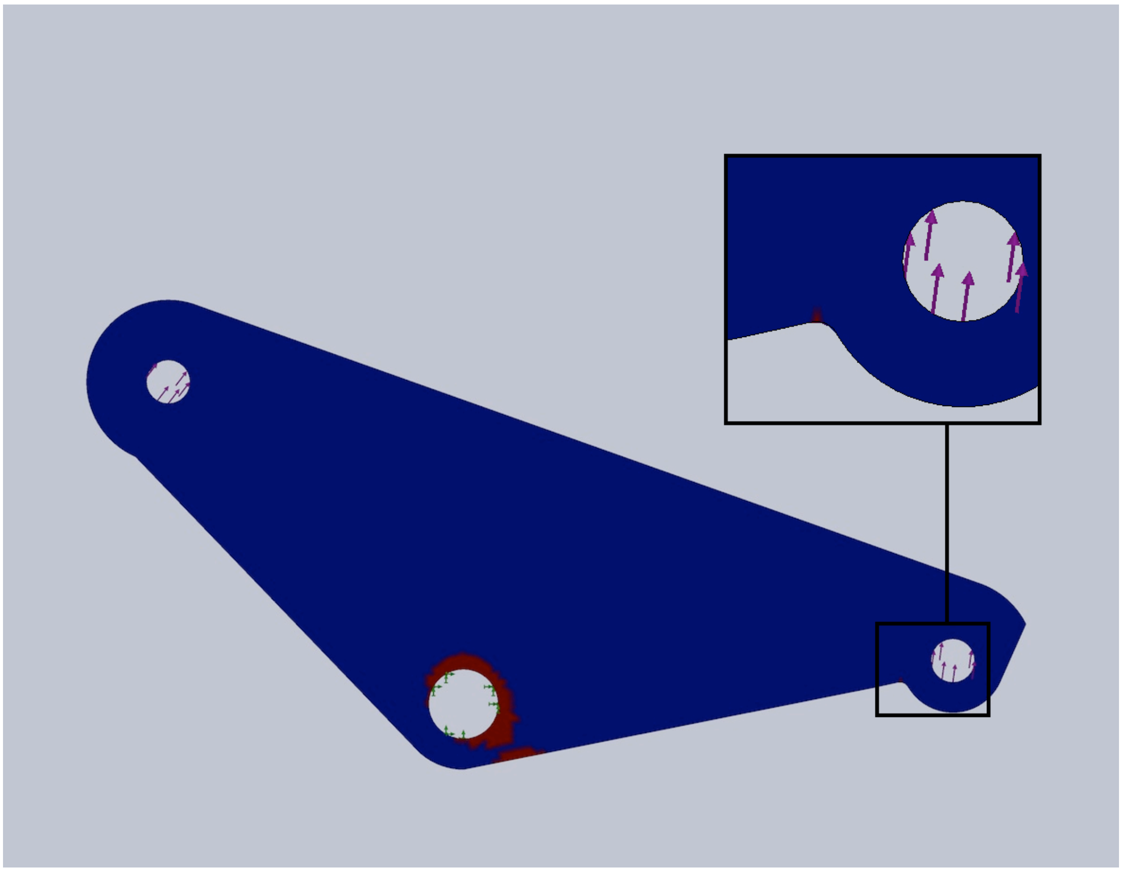


Figure 13-15: Rear rocker – Factor of safety 4

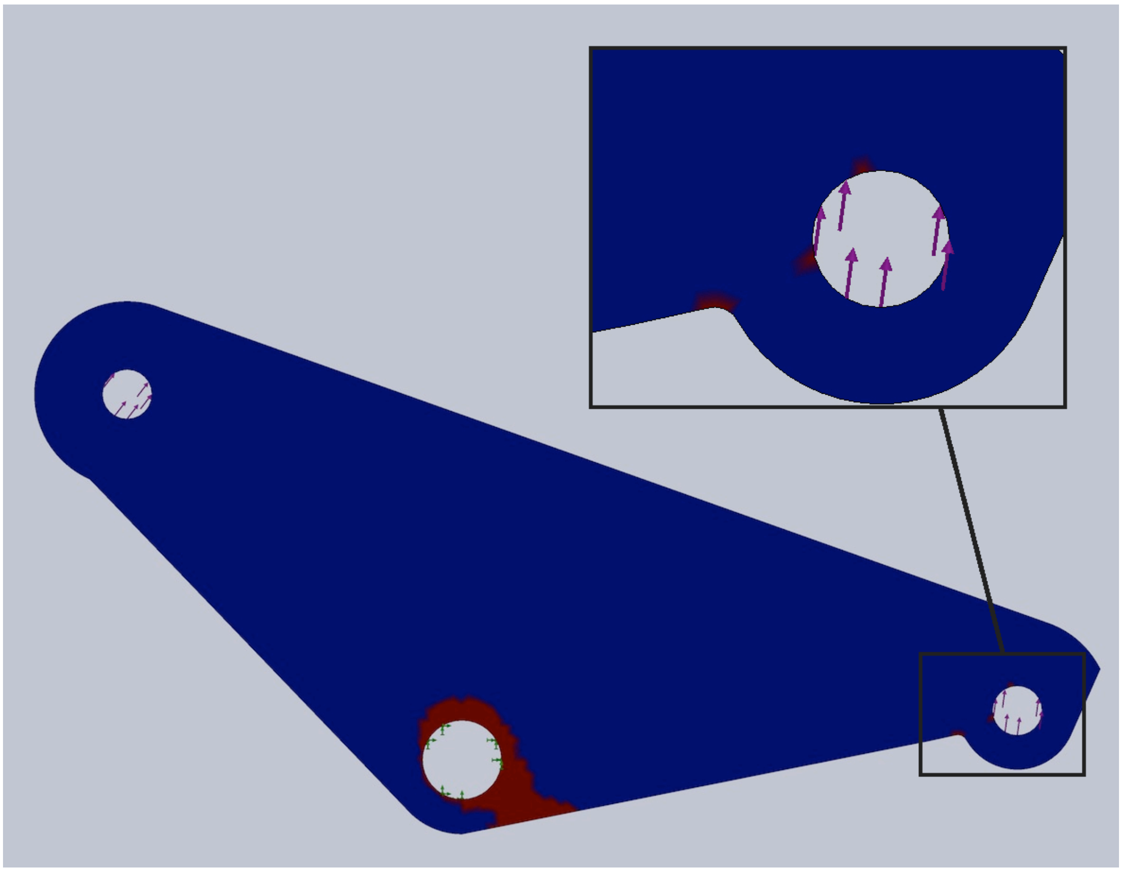


Figure 13-16: Rear Rocker 2mm sheet: Safety factor of 3 shown

13.9 Suspension pivot locations

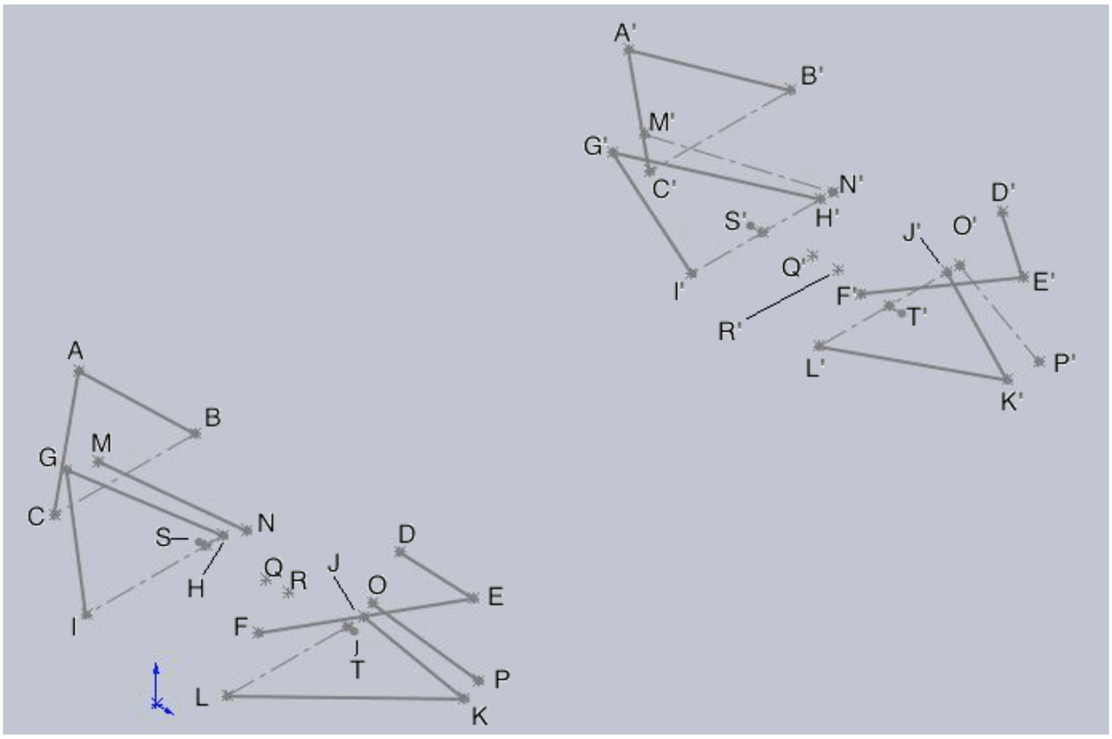
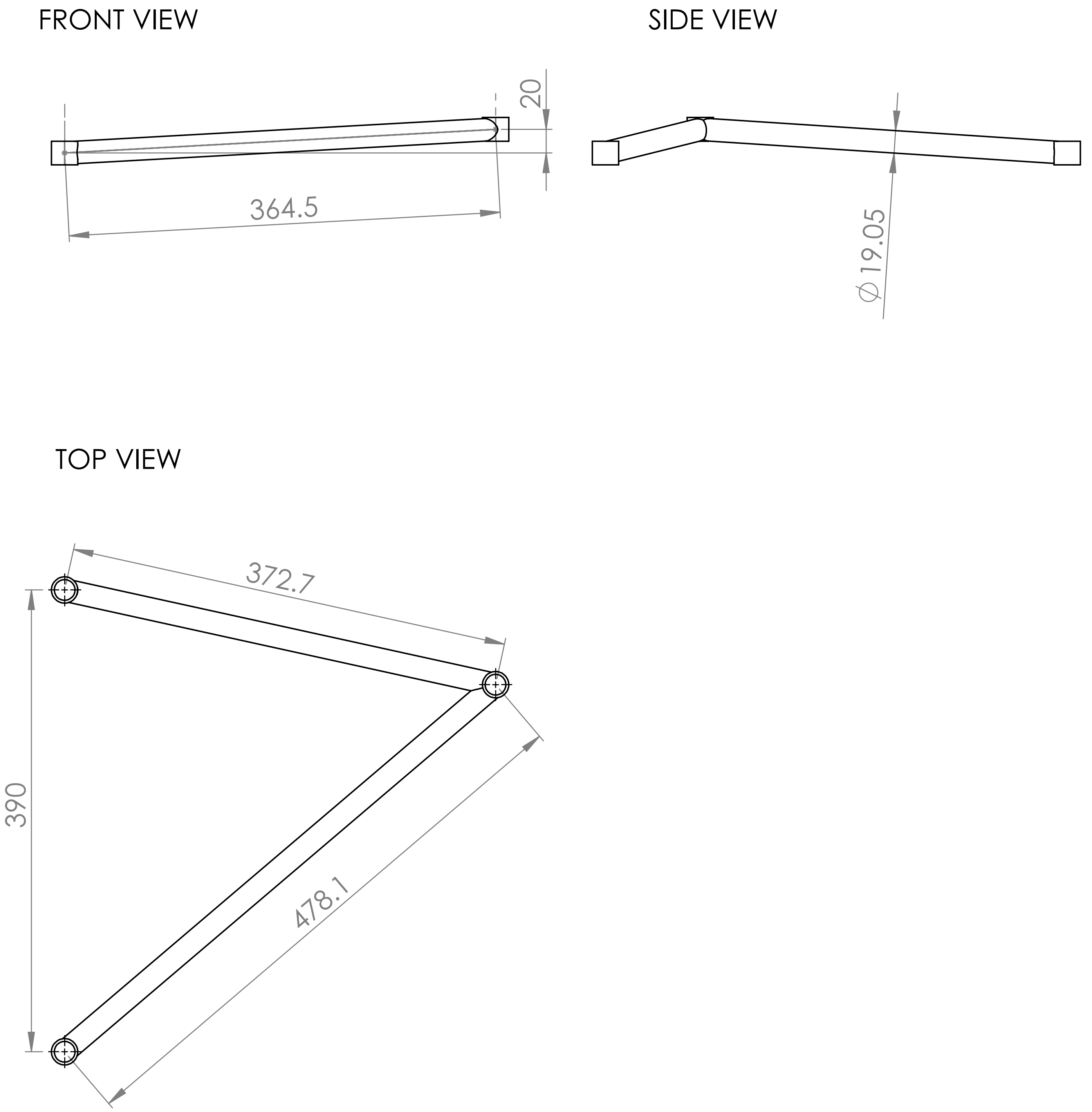


Figure 13-17: Suspension pivot locations

Label	Description	X	Y	Z		Label	Description	X	Y	Z
A	Wishbone	-560	370	-340		A’	Wishbone	-560	380	-1900
B	Wishbone	-290	320	-400		B’	Wishbone	-300	310	-2100
C	Wishbone	-290	320	0		C’	Wishbone	-300	310	-1700
D	Wishbone	290	320	-400		D’	Wishbone	300	310	-2100
E	Wishbone	560	370	-340		E’	Wishbone	560	380	-1900
F	Wishbone	290	320	0		F’	Wishbone	300	310	-1700
G	Wishbone	-564	140	-310		G’	Wishbone	-560	150	-1855
H	Wishbone	-200	120	-400		H’	Wishbone	-180	120	-2065
I	Wishbone	-200	120	0		I’	Wishbone	-180	120	-1700
J	Wishbone	200	120	-400		J’	Wishbone	180	120	-2065
K	Wishbone	564	140	-310		K’	Wishbone	560	150	-1855
L	Wishbone	200	120	0		L’	Wishbone	180	120	-1700
M	Steering	-540	140	-375		M’	Toe	-560	150	-1945
N	Steering	-178	120	-435		N’	Toe	-180	120	-2100
O	Steering	178	120	-435		O’	Toe	180	120	-2100
P	Steering	540	140	-375		P’	Toe	560	150	-1945
Q	Damper	-30	120	-340		Q’	Damper	-35	135	-1900
R	Damper	30	120	-340		R’	Damper	35	135	-1900
S	Rocker	-220	120	-340		S’	Rocker	-215	120	-1900
T	Rocker	220	120	-340		T’	Rocker	215	120	-1900

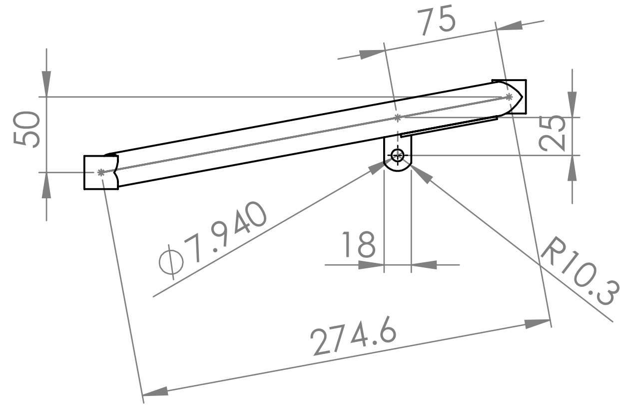
Table 13-4: Suspension pivot locations

13.10 CAD (SolidWorks) Drawings

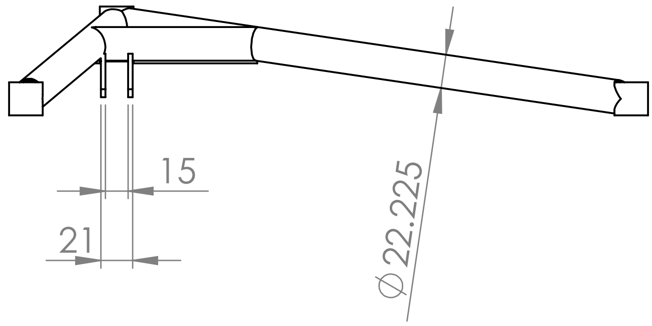


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE		REV FSAE 2011 - Suspension Design	
DRAWN					TITLE: Wishbone - Front Lower Left	
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL: AISI 4130	DWG NO.	A4
				WEIGHT:	SCALE:1:5	SHEET 1 OF 8

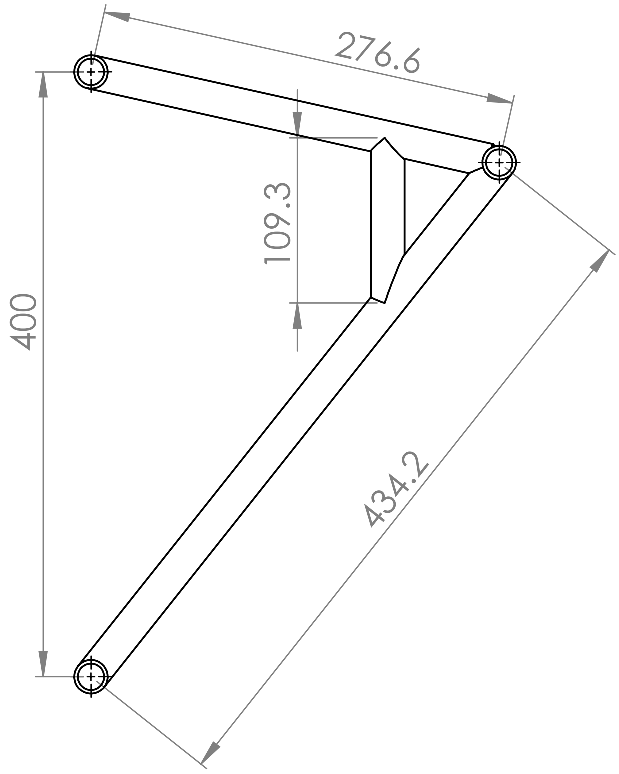
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Wishbone- Front Upper Left

MATERIAL: AISI 4130

WEIGHT:

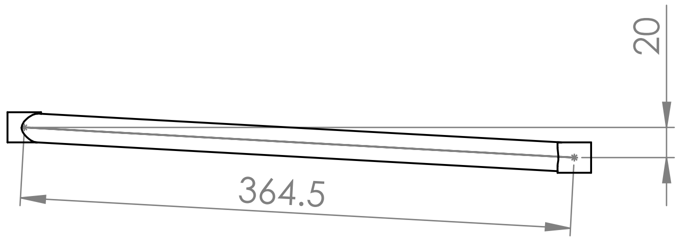
DWG NO.

A4

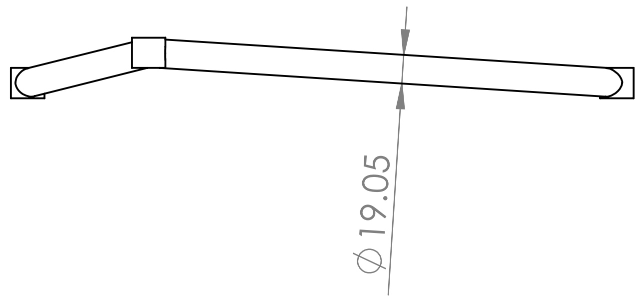
SCALE:1:5

SHEET 2 OF 8

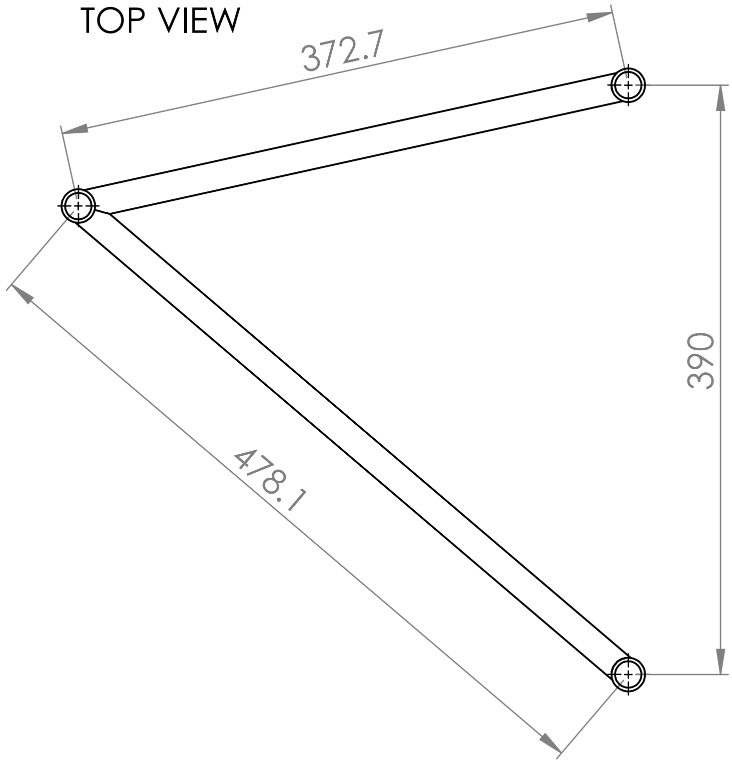
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Wishbone - Front Lower Right

MATERIAL: AISI 4130

WEIGHT:

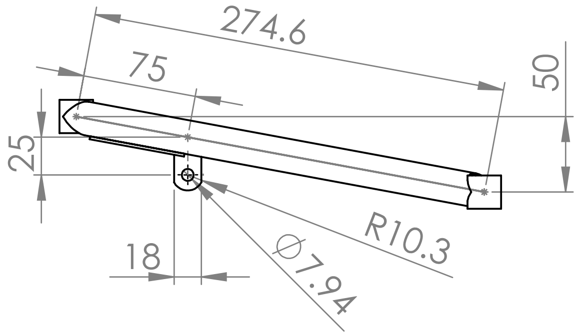
DWG NO.

A4

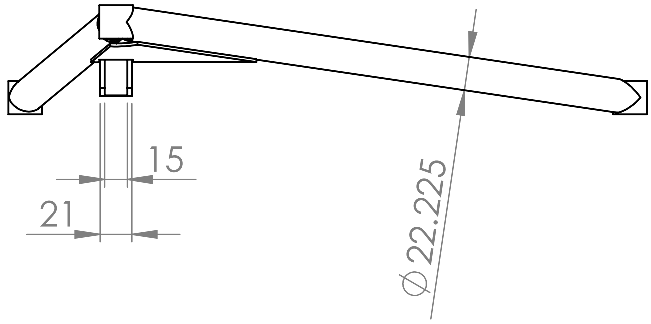
SCALE:1:5

SHEET 3 OF 8

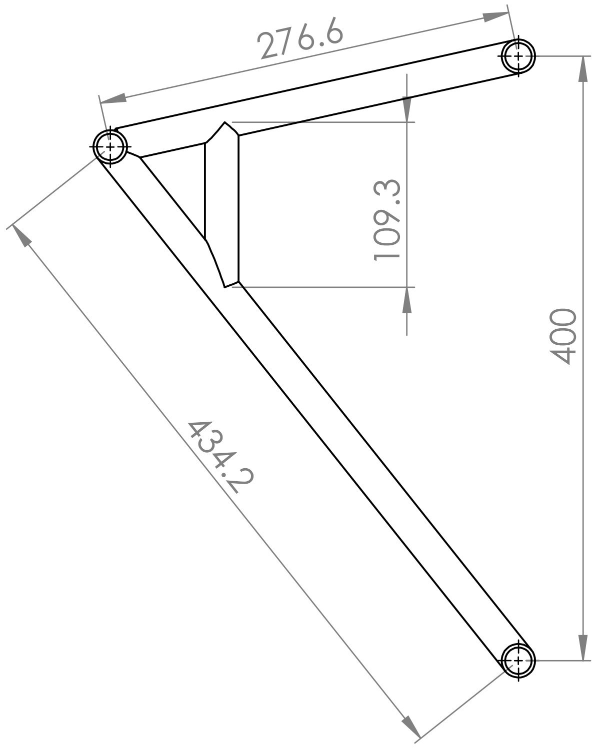
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Wishbone - Front Upper Right

MATERIAL: AISI 4130

WEIGHT:

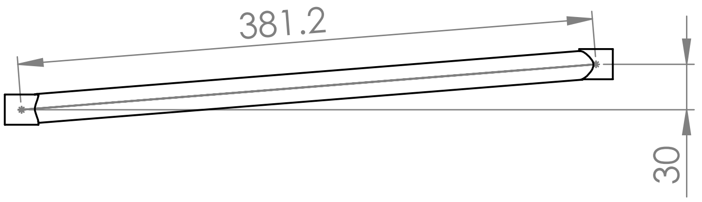
DWG NO.

A4

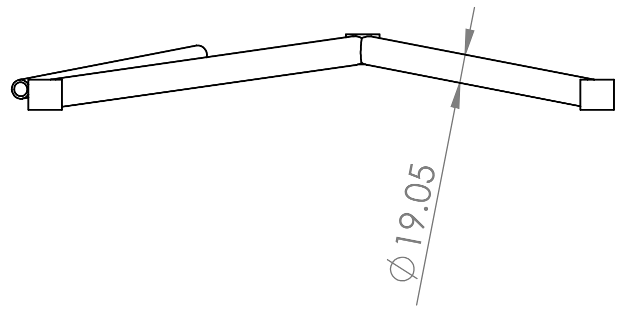
SCALE:1:5

SHEET 4 OF 8

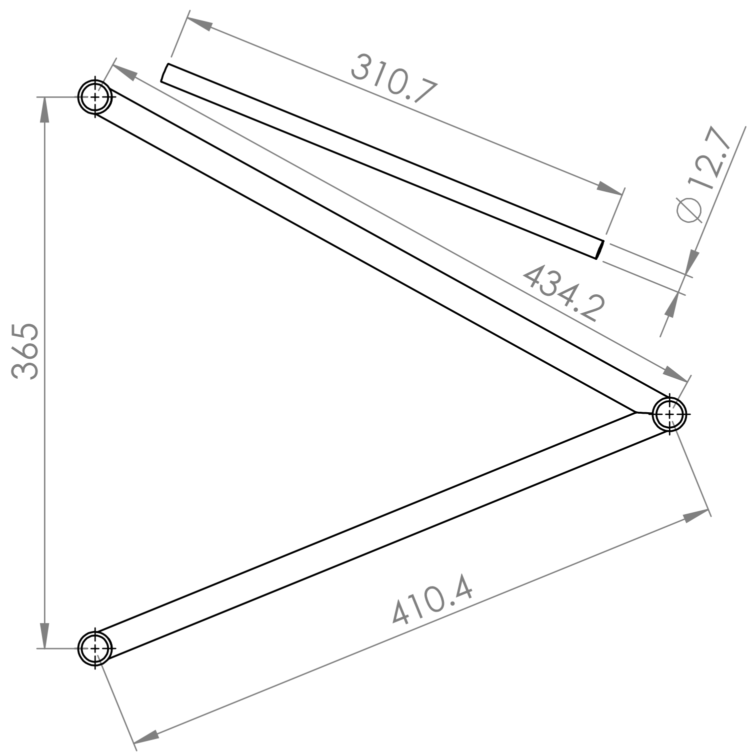
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
			REV FSAE 2011 - Suspension Design	

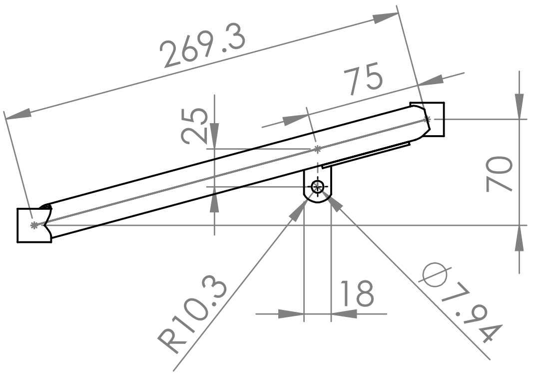
	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL: AISI 4130	
				WEIGHT:	

TITLE:

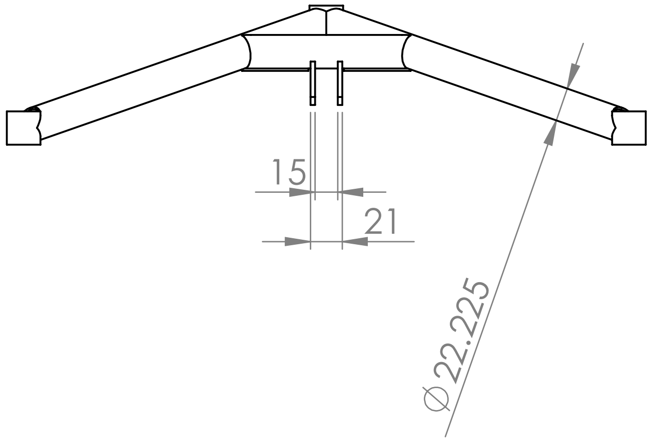
Wishbone - Rear Lower Left

DWG NO.	A4
SCALE:1:5	SHEET 5 OF 8

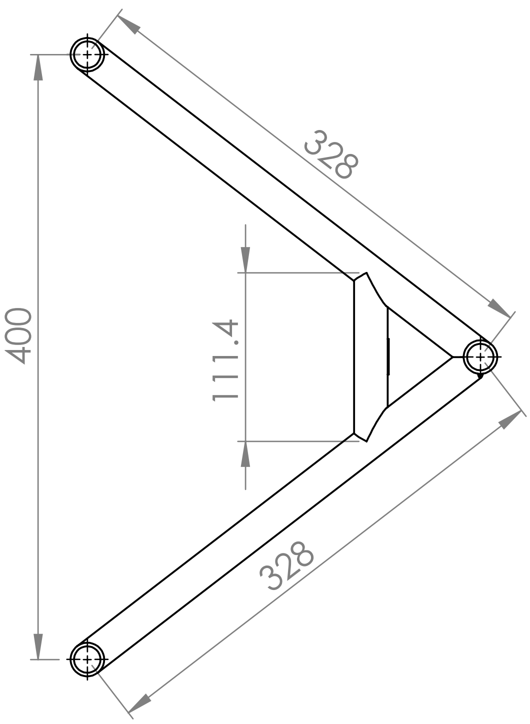
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE			
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A						

TITLE:

Wishbone - Rear Upper Left

MATERIAL: AISI 4130

WEIGHT:

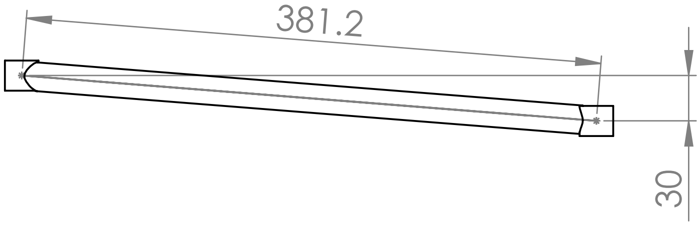
DWG NO.

A4

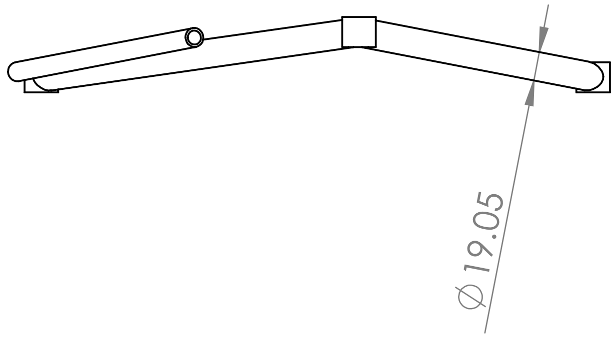
SCALE:1:5

SHEET 6 OF 8

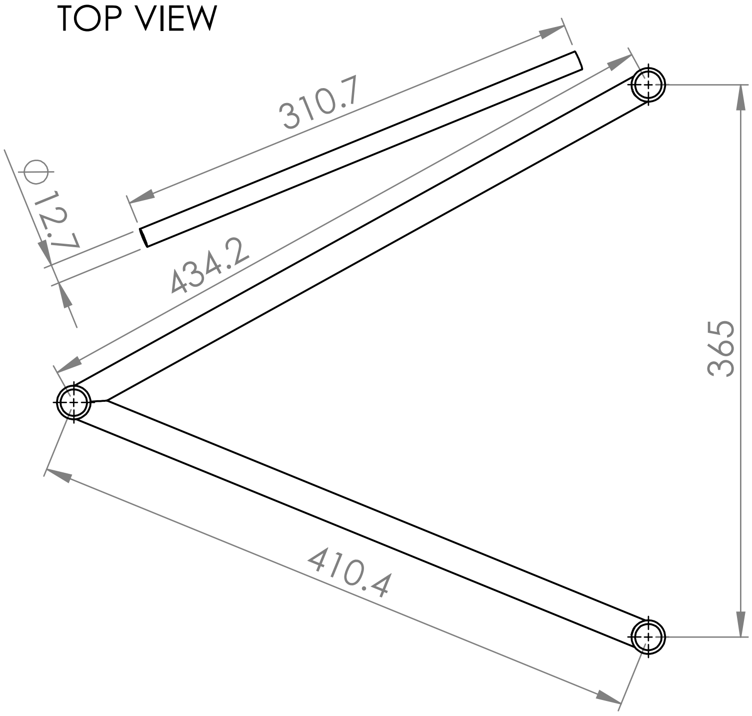
FRONT VIEW



SIDE VIEW

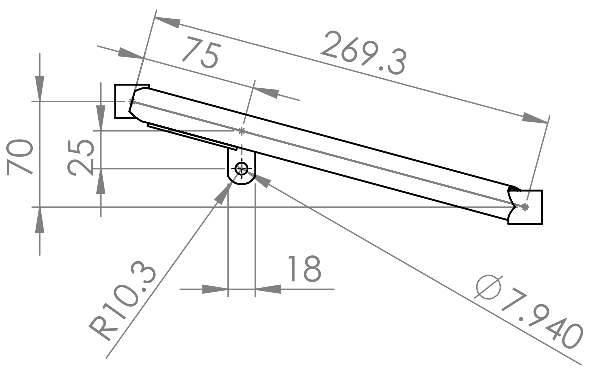


TOP VIEW

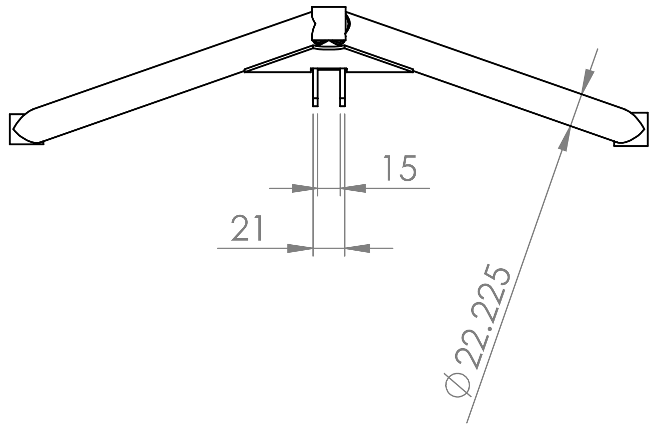


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
			REV FSAE 2011 - Suspension Design		
	NAME	SIGNATURE	DATE	TITLE:	Wishbone - Rear Lower Right
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL: AISI 4130	DWG NO.	A4
				WEIGHT:	SCALE:1:5	SHEET 7 OF 8

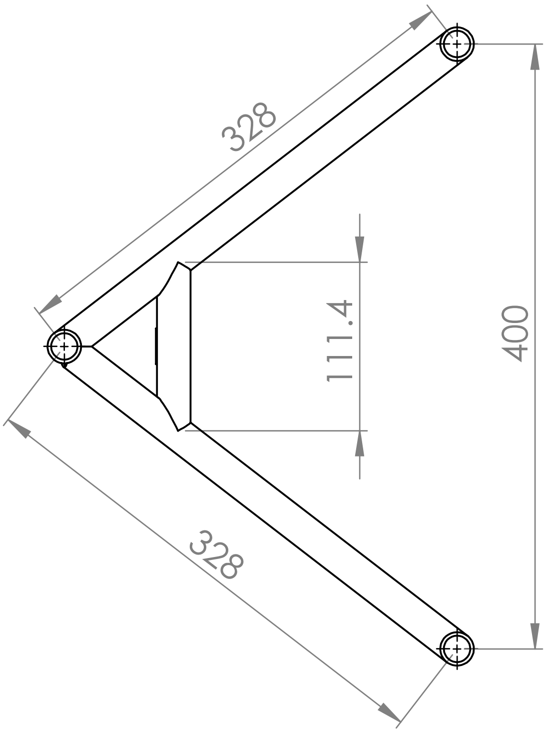
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING / REVISION
			REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE			
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A						

TITLE:

Wishbone - Rear Upper Right

MATERIAL: AISI 4130

WEIGHT:

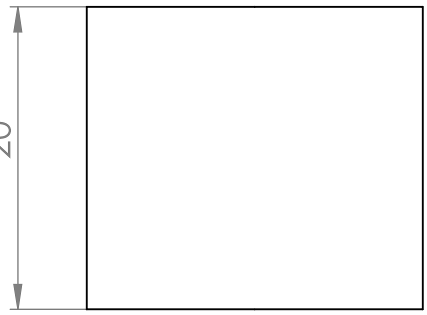
DWG NO.

A4

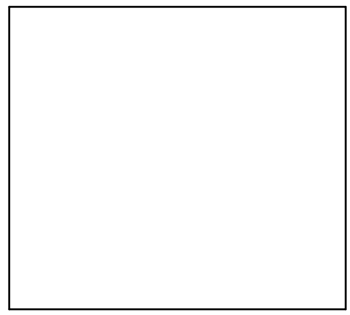
SCALE:1:5

SHEET 8 OF 8

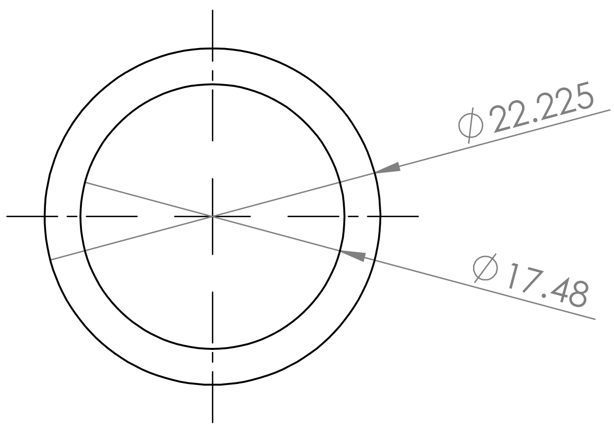
FRONT VIEW

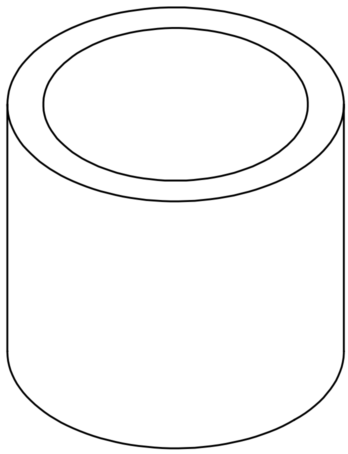


SIDE VIEW



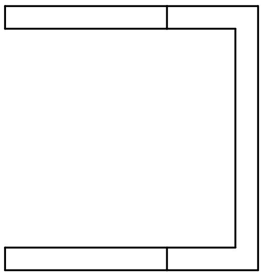
TOP VIEW



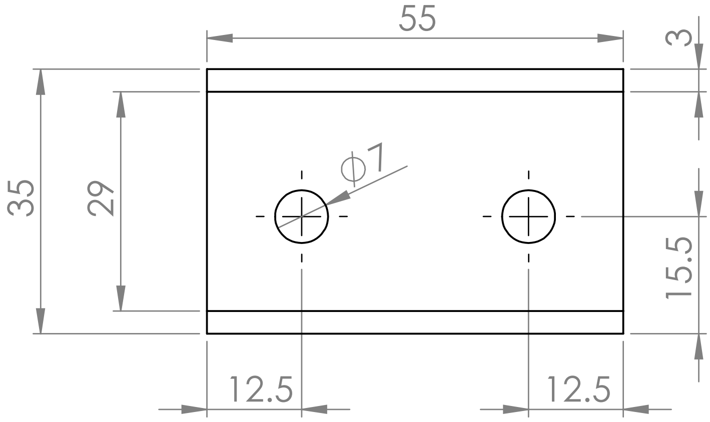


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:		DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
				REV FSAE 2011 - Suspension Design		
	NAME	SIGNATURE	DATE	TITLE: Spherical Bearing Housing		
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL: AISI 4130	DWG NO.	A4
				WEIGHT:	SCALE:2:1	SHEET 1 OF 1

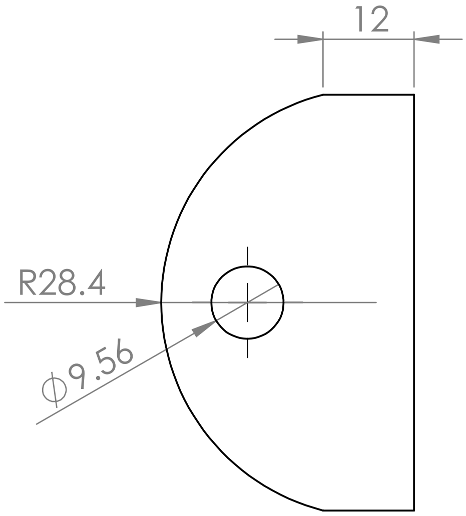
SIDE VIEW

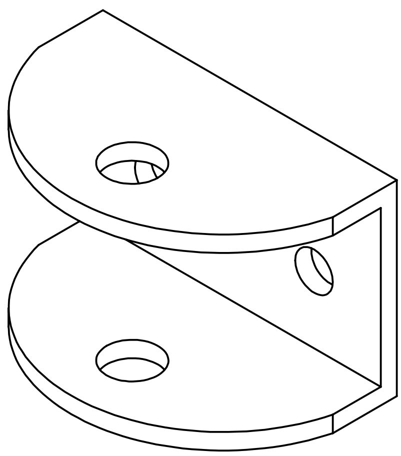


FRONT VIEW



TOP VIEW





UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Bracket

MATERIAL:

AISI 4130

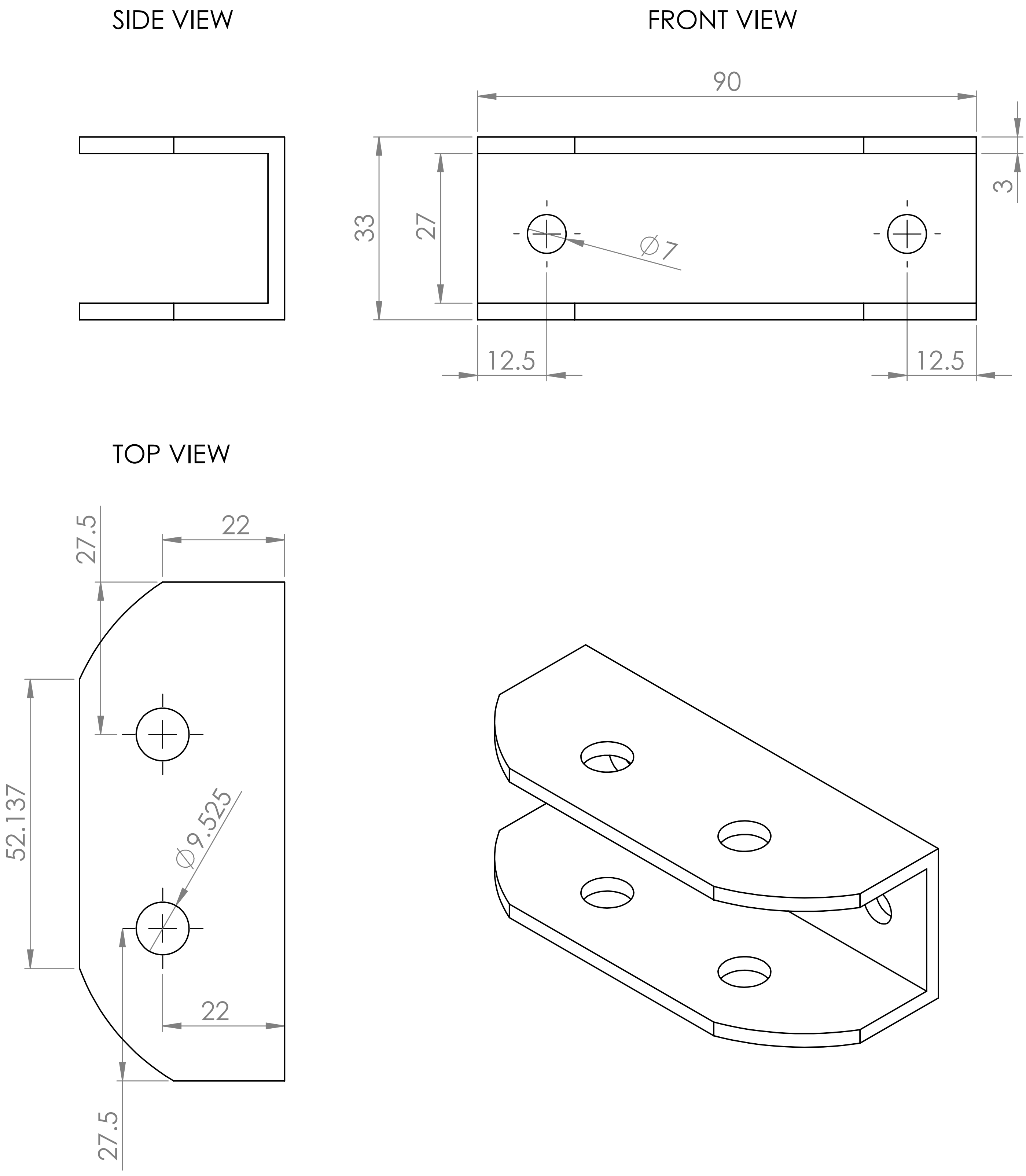
WEIGHT:

DWG NO.

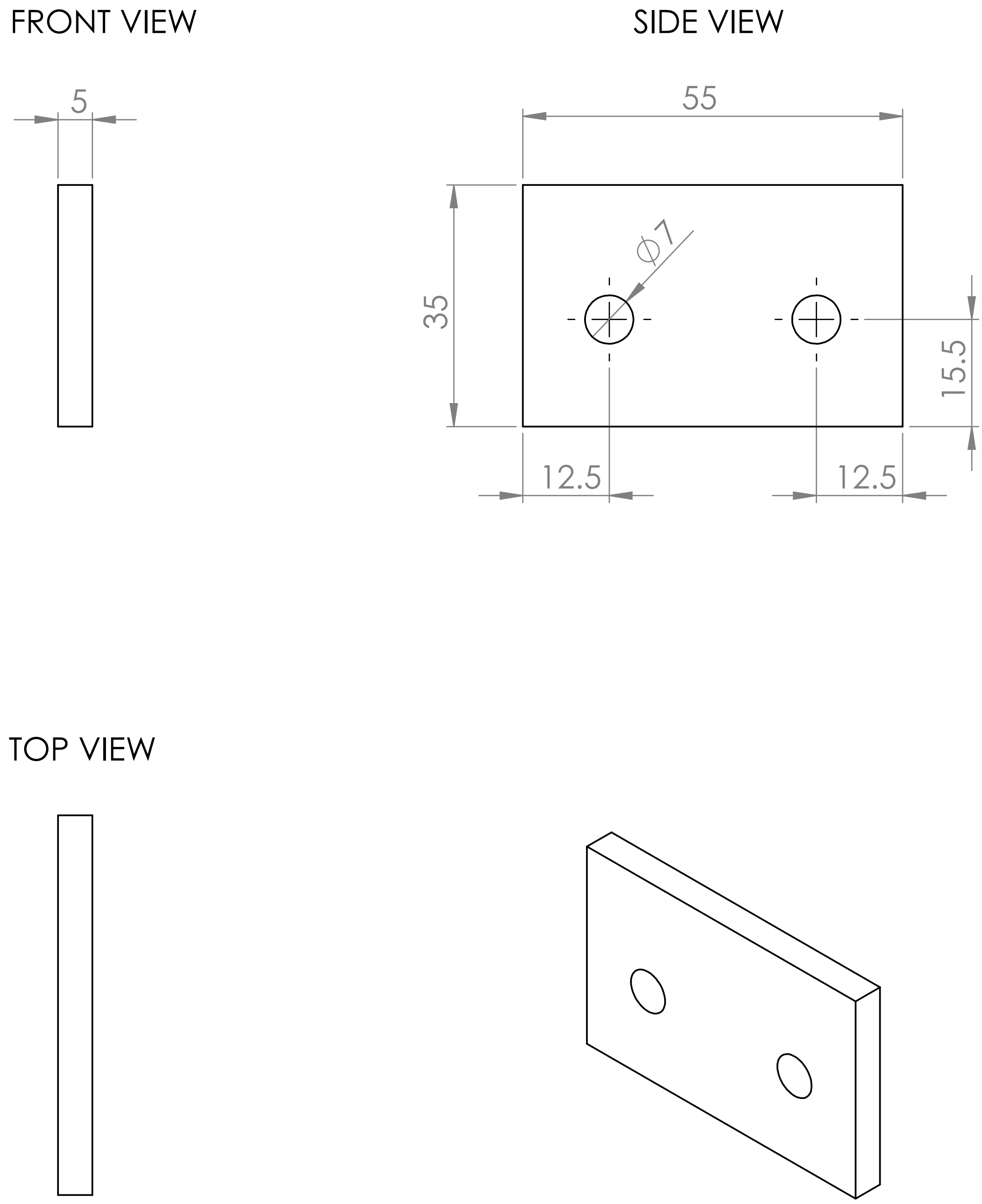
A4

SCALE:1:1

SHEET 1 OF 2

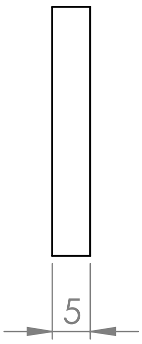


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:		DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING		REVISION
				REV FSAE 2011- Suspension Design		
	NAME	SIGNATURE	DATE	TITLE:	Bracket - Rear Toe Control	
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A			MATERIAL: AISI 4130	DWG NO.		A4
			WEIGHT:	SCALE:1:1	SHEET 2 OF 2	

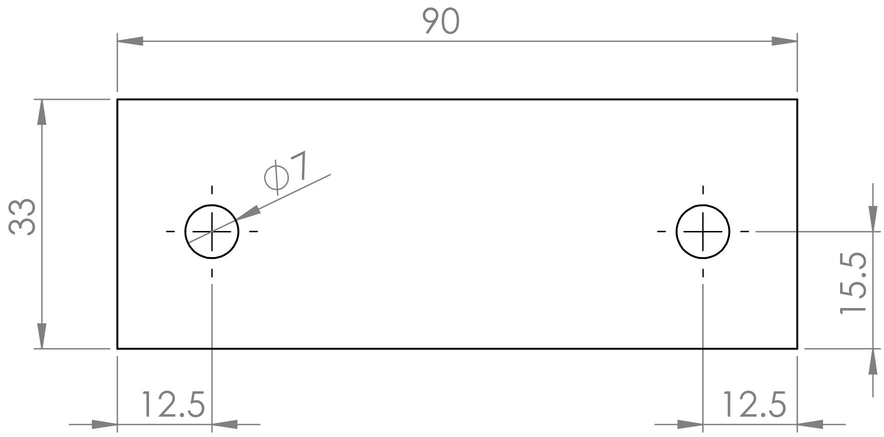


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:		FINISH:			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING | REVISION
						REV FSAE 2011 - Suspension Design
	NAME	SIGNATURE	DATE			TITLE:
DRAWN						Shim
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL: AISI 4130		DWG NO. | A4
				WEIGHT:		SCALE:1:1 | SHEET 1 OF 2

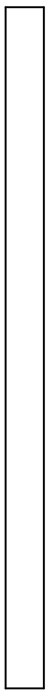
FRONT VIEW

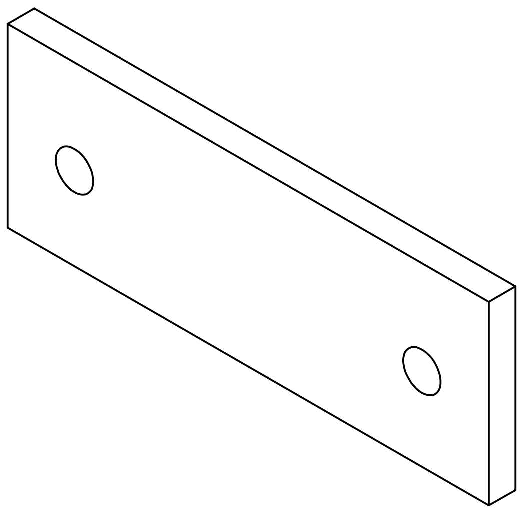


SIDE VIEW



TOP VIEW





UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011 - Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Shim - Rear Toe Control

MATERIAL:
AISI 4130

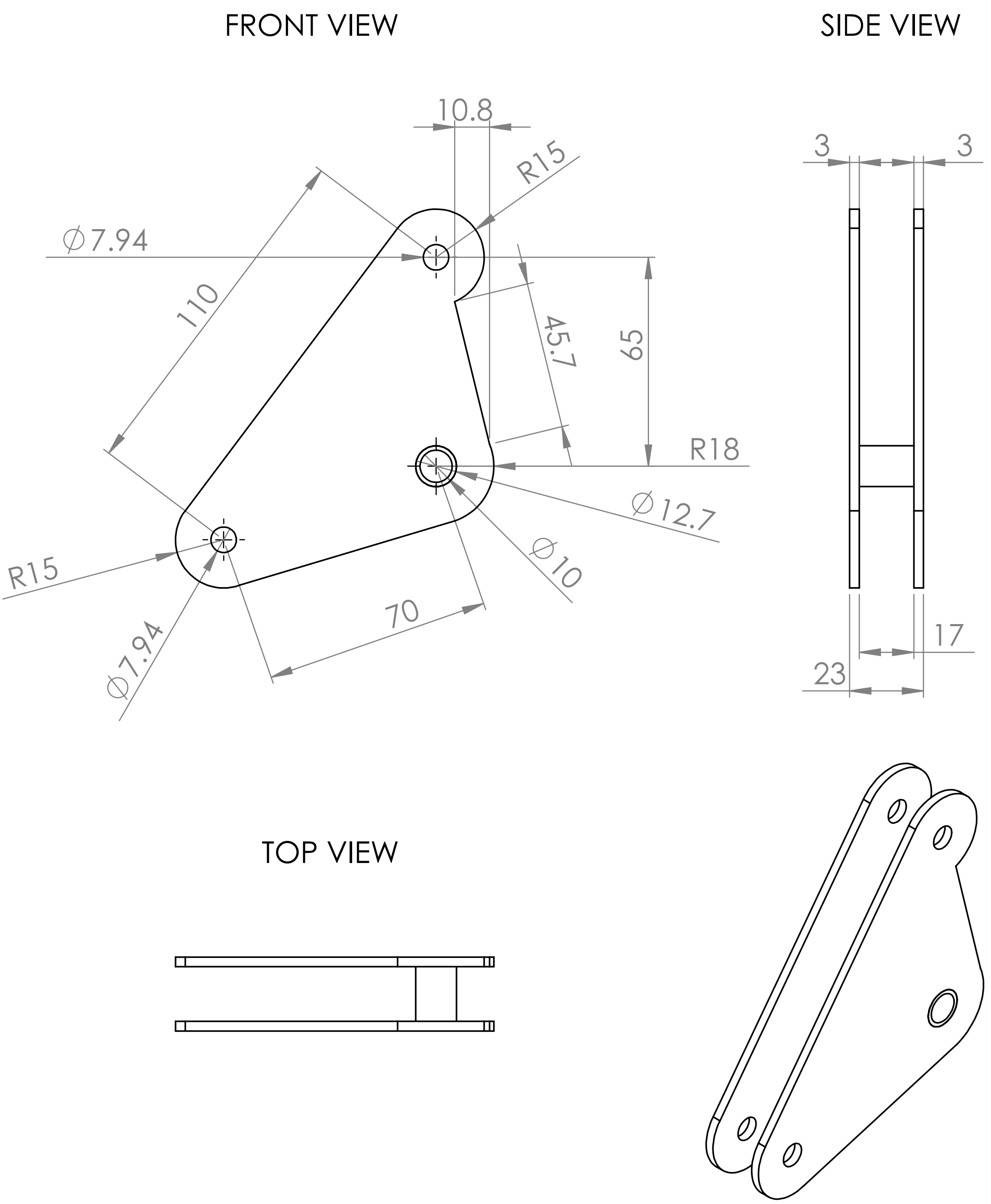
WEIGHT:

DWG NO.

A4

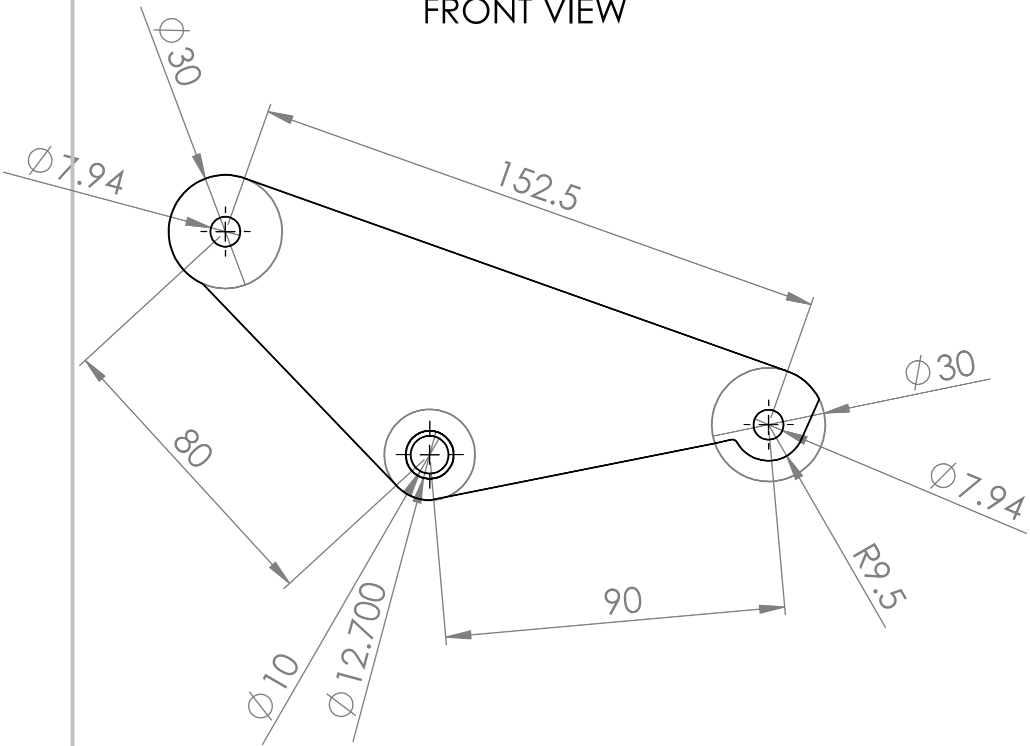
SCALE:1:1

SHEET 2 OF 2

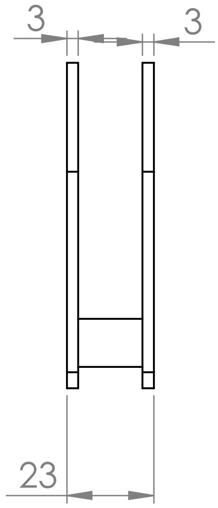


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:				DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
						REV FSAE 2011 - Suspension Design	
	NAME	SIGNATURE	DATE			TITLE:	
DRAWN						Rocker - Front	
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL: AISI 4130		DWG NO.	A4
				WEIGHT:		SCALE:1:2	SHEET 1 OF 2

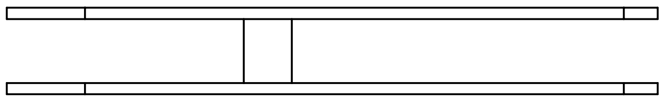
FRONT VIEW

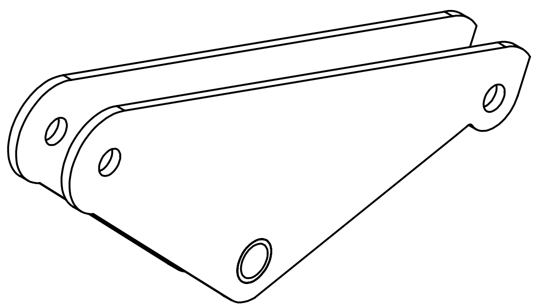


SIDE VIEW



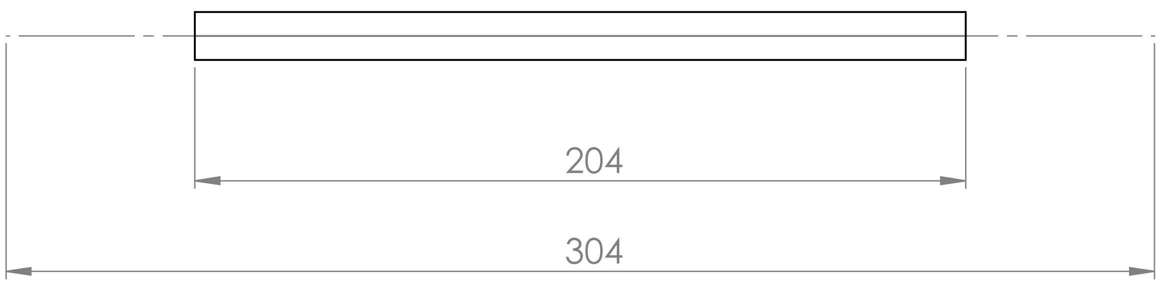
TOP VIEW



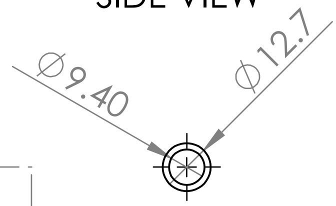


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
					REV FSAE 2011 - Suspension Design	
	NAME	SIGNATURE	DATE		TITLE:	
DRAWN					Rocker - Rear	
CHK'D						
APPV'D						
MFG						
Q.A			MATERIAL: AISI 4130		DWG NO.	A4
			WEIGHT:		SCALE:1:2	SHEET 2 OF 2

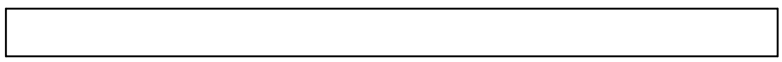
FRONT VIEW



SIDE VIEW



TOP VIEW



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011- Suspension Design

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Pullrod - Front

MATERIAL:
AISI 4130

WEIGHT:

DWG NO.

A4

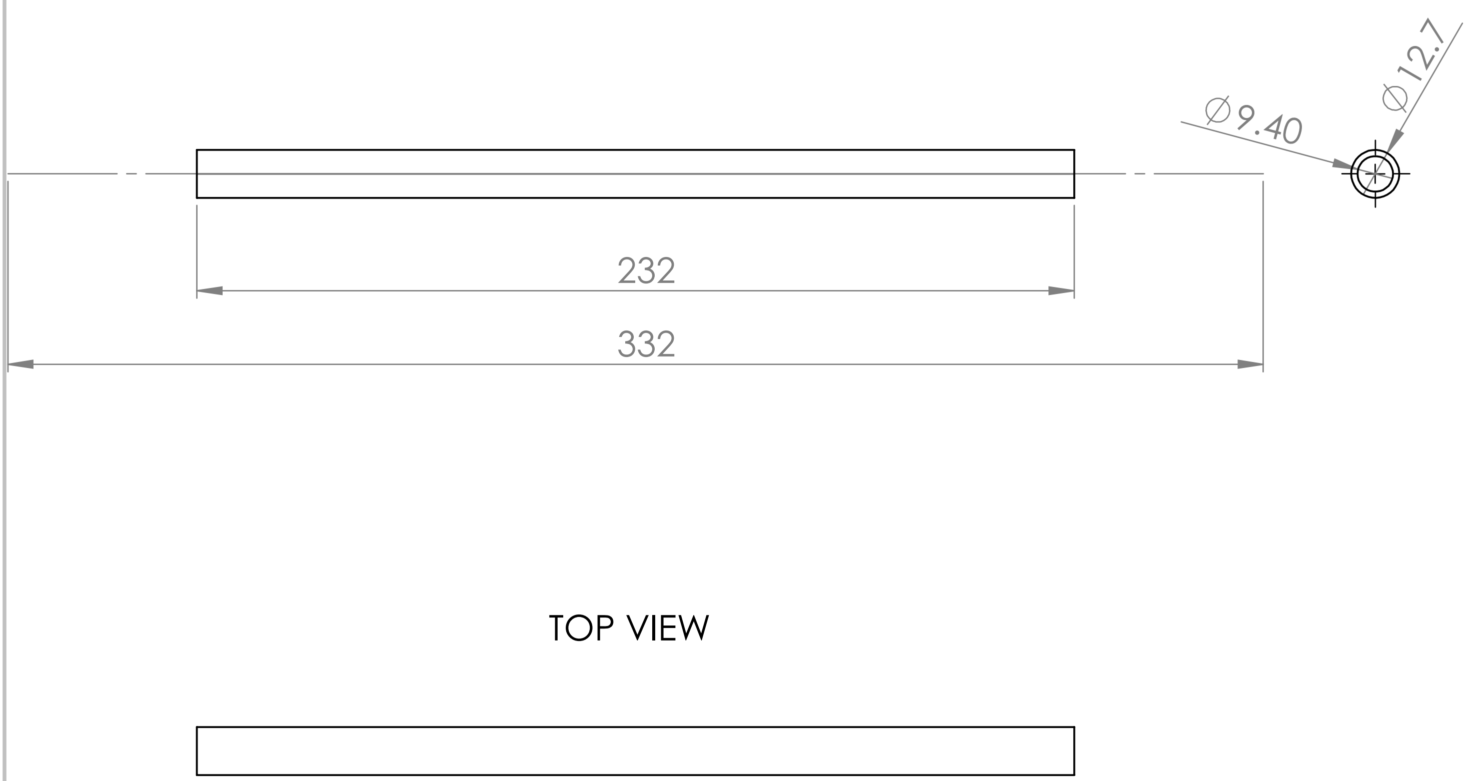
SCALE:1:2

SHEET 1 OF 3

FRONT VIEW

SIDE VIEW

TOP VIEW



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

REV FSAE 2011- Suspension Design

NAME | SIGNATURE | DATE

DRAWN

CHK'D

APPV'D

MFG

Q.A

TITLE:

Pullrod - Rear

MATERIAL:

AISI 4130

DWG NO.

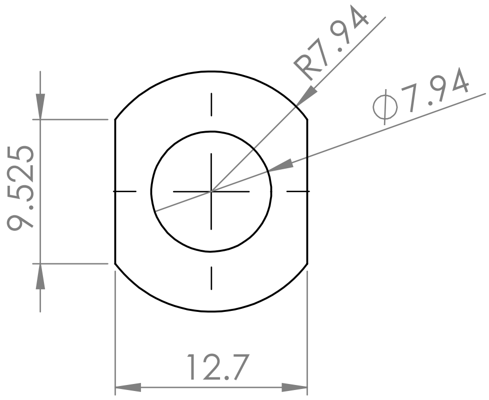
A4

WEIGHT:

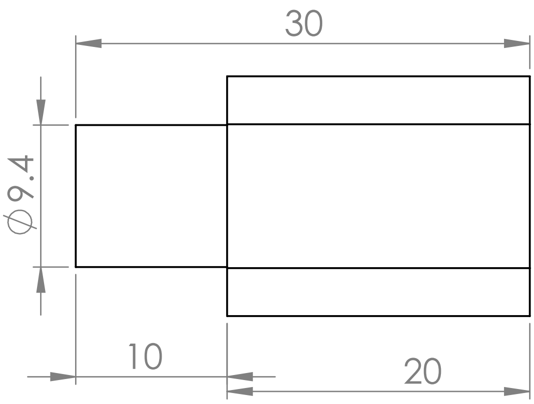
SCALE:1:2

SHEET 2 OF 3

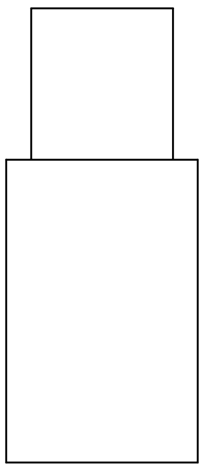
FRONT VIEW

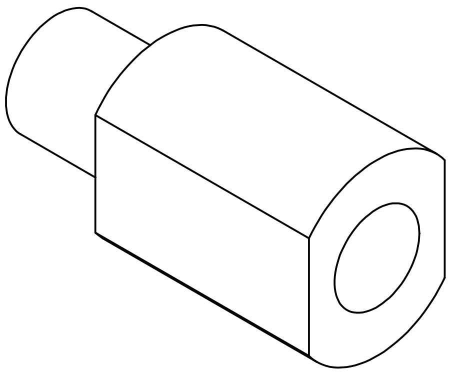


SIDE VIEW



TOP VIEW





UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION		
			REV FSAE 2011- Suspension Design			
	NAME	SIGNATURE	DATE			TITLE: Pullrod/Rear Toe End Piece
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL: AISI 4130		DWG NO.	A4
				WEIGHT:		SCALE:2:1	SHEET 3 OF 3