University of Southern Queensland
Faculty of Engineering and Surveying

Steering System and Suspension Design for 2005 Formula SAE-A Racer Car

A dissertation submitted by

Cristina Elena Popa

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towards the degree of

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Abstract

The dissertation documents the design project for the steering system and suspension of the 2005 Formula SAE-A racer car made at the University of Southern Queensland.

The dissertation includes a review of current automotive steering and suspension systems followed by the review of Formula SAE-A restrictions and design requirements.

A thorough analysis of 2004 USQ racer car has been included in order to establish the areas of design modifications followed by the actual design with all the technical specifications required.

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ENG4111/2 *Research Project*

Limitations of Use

Certification

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

CRISTINA ELENA POPA

Student number: 0050018250

Signature

Date

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CRISTINA POPA

University of Southern Queensland

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Introduction

1.1 Project objective

The aim of this project is to design and build the steering system and suspension for USQ SAE-A racer car. To help achieve this goal in the available time, various tasks were set. These include:

research of the Formula SAE-A rules to determine restrictions and design requirements;

research on currently used automotive steering systems and suspension and evaluation of feasible alternatives;

analysis of other team's design;

design an appropriate steering system and suspension for 2005 race car;

develop systems to evaluate the performance of the steering system and suspension.

1.2 Overview of the Formula SAE-A Competition

The main objective of Formula SAE-A competition is for students to conceive, design and build a small open wheel racer car according to the competition requirements that is to be compared with other competing designs in order to decide the best overall vehicle. The restrictions on the car are set up to challenge student's imagination and knowledge while giving them a meaningful project as well as good practice working in a team environment.

The design brief is for a prototype car made for evaluation as a production item for non-professional weekend autocross racers that should cost below $25,000. Therefore, the car must have high performance in terms of acceleration, braking and handling qualities as well as high reliability, low cost and easy maintenance.

1.3 Vehicle requirements

The vehicle is required to be an open-wheeled and open-cockpit design with four wheels not in a straight line. The minimum wheelbase must be 1525mm (60 inches), and minimum track width no less than 75% of the larger track.

The ground clearance requirement is that no part of the car other than tires touches the ground during truck events.

The chassis must have crush protection and roll hoops built into it at various points of safety.

The wheels should be a minimum of 203.2 mm (8.0 inches) in diameter and have a fully operational suspension package on all wheels with at least 50.8mm (2 inches) usable wheel travel.

The tire size and type is free.

The car must be equipped with breaking system that operates on all four wheels with two independent circuits operated by a single control.

The steering must affect at least two wheels. The steering system must have positive stops to prevent steering linkages from locking up and to prevent the tires from contacting the car at all times. The allowable free play is limited to 7 degrees total, and is measured at the steering wheel.

The engine used to power the car must be four-stroke piston engine with a maximum displacement of 610 cc per cycle. All engine airflow is to pass through a 20mm restrictor to limit its power capabilities.

Chapter 2

Background

2.1 Automobile suspension systems

In a practical suspension system, the wheel is connected to the body through various links that permit an approximately vertical motion of the wheel relative to the body, controlled by the springs and dampers.

The main components of a suspension system are:

1 springs,

2 shock absorbers,

3 struts

4 tires

When an additional load is placed on the springs or the vehicle meets a bump in the road, the springs will absorb the load by compressing as sown in Figure 1. The springs are very important components of the suspension system that provides ride comfort. Shocks and struts help control how fast the springs and suspension are allowed to move, which is important in keeping tires in firm contact with the road.

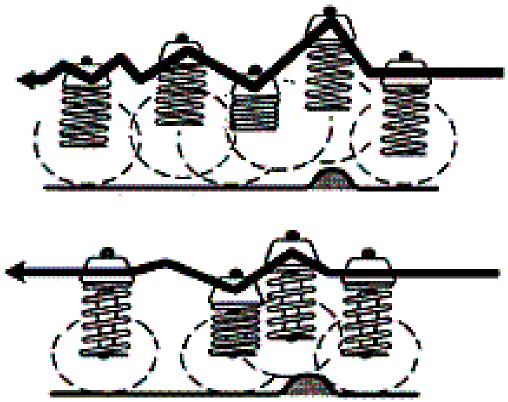


Figure 1 Spring movement

2.1.1 Springs

Springs are the flexible links that allow the frame and the body to ride relatively undisturbed while the tires and suspension follow the bumps in the road. The springs support the weight of the vehicle, maintain ride height, and absorb road shock.

The major spring designs in use today are:

1. coil springs
2. leaf springs
3. torsion bar
4. air springs

2.1.1.1 Terminology used in the study of springs

The term bounce refers to the vertical (up and down) movement of the suspension system.

The upward suspension travel that compresses the spring and shock absorber is called the jounce, or compression.

The downward travel of the tire and wheel that extends the spring and shock absorber is called rebound, or extension.

Spring rate is used to measure spring strength. It is the amount of weight that is required to compress the spring 1 inch.

Sprung weight is the weight supported by the springs. For example, the vehicle's body, transmission, frame, and motor would be sprung weight.

Unsprung weight is the weight that is not carried by springs, such as the tires, wheels, and brake assemblies.

2.1.1.2 Coil springs

Coil springs are the most commonly used springs. The diameter and length of the wire determine the strength of a spring. Increasing the wire diameter will produce a stronger spring, while increasing its length will make it more flexible.

Some coil springs are made with a variable rate as shown in Figure 2. This variable rate is accomplished by either constructing this spring from materials having different thickness or by winding the spring so the coil will progressively compress at a higher rate.

Variable rate springs provide a lower spring rate under unloaded conditions offering a smoother ride, and a higher spring rate under loaded conditions, resulting in more support and control.

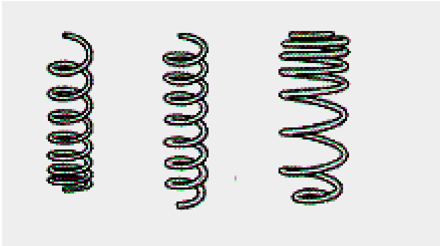


Figure 2 Coil spring types

2.1.1.3 Leaf springs

There are two major leaf spring design: multi-leaf and mono-leaf.
The multi-leaf spring as shown in Figure 3 is made of several steel plates of different lengths stacked together. During normal operation, the spring compresses to absorb road shock. The leaf springs bend and slide on each other allowing suspension movement.

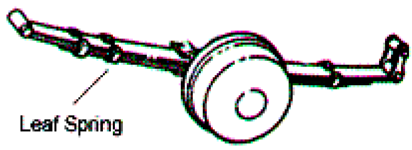


Figure 3 Leaf spring

In most cases leaf springs are used in pairs mounted longitudinally (front to back) but some of vehicle manufacturers are using a single transverse (side to side mounted leaf spring).

2.1.1.4 Torsion bar springs

The torsion bar is a straight or L shaped bar of spring steel. Most torsion bars are longitudinal, mounted solidly to the frame at one end and connected to a moving part of the suspension at the other. Torsion bars may also be transverse mounted. During suspension movement, the torsion bar will twist, providing spring action.

2.1.1.5 Air springs

The air spring is another type of spring that is becoming more popular on passenger cars, light trucks, and heavy trucks. The air spring as seen in Figure 4 is a rubber cylinder filled with compressed air. A piston attached to the lower control arm moves up and down with the lower control arm. This causes the compressed air to provide spring action. If the vehicle loads changes, a valve at the top of the airbag opens to add or release air from the air spring. An onboard compressor supplies air.

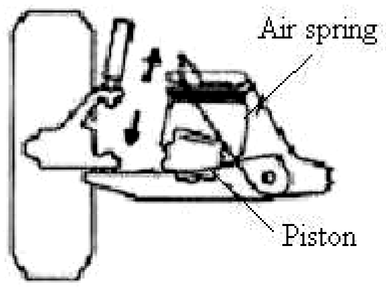


Figure 4 Air spring

2.1.2 Shock absorbers

If the suspension were equipped with just a spring, it would bounce up and down several times after each bump. When compressed by a bump, a suspension system needs a way to dissipate the energy that is stored in the spring. The shock absorber is the device that dissipates the energy and keeps the suspension from bouncing out of control.

Common types of shock absorbers illustrated in Figure 5 are:

1. oil filled
2. gas charged
3. reservoir

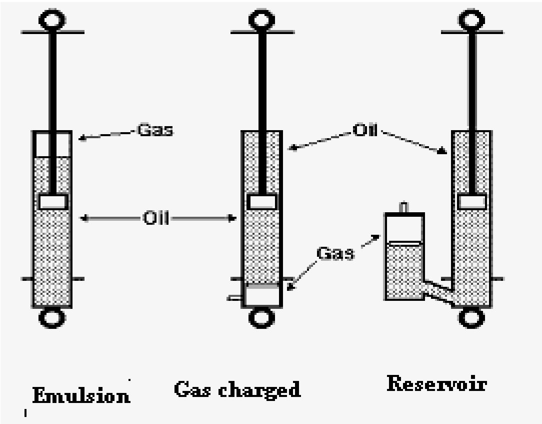


Figure 5 Types of shocks

2.1.2.1 Oil filled shock absorber

The most common type of shock absorber is oil filled. This type is used in car suspensions as well as bike suspensions. Many shock absorbers have adjustable spring rates and damping.

Some shocks have a rod extending into the centre of the shock that can be turned, allowing you to open or close more holes to adjust the damping. Some premium shocks have small one-way valves in the piston that open up holes when the piston is moving in one direction at different speeds, and close them when the piston is moving in the other direction.

Most shocks have a small amount of compression damping, and a much larger amount of rebound damping which means that the shock moves easily when it is compressing to absorb a bump, then slowly lengthens to release the energy stored in the spring.

2.1.2.2 Gas charged shock absorbers

There are shocks charged with gas, typically nitrogen. These shocks have an extra unattached piston in the bottom of the damper cylinder, with oil above the piston and high pressure gas below the piston. Typically, nitrogen at 30 to 300 psi is used because the oil would not combine (burn) with the nitrogen nearly as easily as it will with the oxygen in normal air.

2.1.2.3 Reservoir shock absorbers

The reservoir shocks have an extra oil reservoir attached with heat fins to help get rid of the heat.

2.1.3 Struts

Strut mounts are vehicle specific, and there are numerous designs in use today on both front and rear suspension systems.
The three most common designs are:

1. inner plate, illustrated in Figure 6;
2. centre sleeve;
3. spacer bushing.

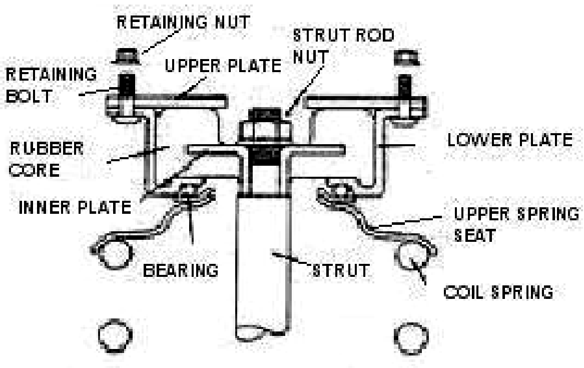


Figure 6 Strut design – inner plate

2.1.4 Tires as springs

Tires are air springs that support the total weight of the vehicle. The air spring action of the tire is very important to the ride quality and safe handling of the vehicle. Tire size, construction, compound and inflation are very important to the ride quality of the vehicle.

There are three basic types of tires shown in Figure 7, namely:

1. radial ply, bias ply and bias belted;
2. bias ply;
3. bias belted.

Figure 7: Tire Types

2.1.4.1 Radial Tires

Radial ply tires have ply cords, which run across the centreline of the tread and around the tire.
The two sets of belts are at right angles. The belts are made of steel wire, polyester or other substances. Today, radial tires come as original equipment on most passenger cars and light trucks.

2.1.4.2 Bias ply tires

Bias ply tires use cords that run at an angle across the centreline of the tire tread. The alternate ply cords cross at opposite angles.

2.1.4.3 Bias belted tyres

Bias belted tires are the same as bias ply, with the addition of layers of cords - or belts - circling the tire beneath the tread.

The air pressure determines the spring rate of the tire. An over inflated tire will have a higher spring rate and will produce excessive road shock. Over inflated tires will transmit road shock rather than reduce it. Over or under inflation also affects handling and tire wear.

2.1.5 Suspension types

Suspension systems can be broadly classified into two subgroups: dependent and independent. These terms refer to the ability of opposite wheels to move independently of each other.

2.1.5.1 Dependent suspension systems

A dependent suspension system normally has a simple beam axle that holds wheels parallel to each other and perpendicular to the axle. When the camber of one wheel changes, the camber of the opposite wheel changes in the same way.

Dependent systems may be differentiated by the system of linkages used to locate them, both longitudinally and transversely. Often both functions are combined in a set of linkages.

Examples of location linkages include:

Trailing arms
Panhard rod as seen in Figure 8
Watts linkage as seen in Figure 9
Inboard
WOB link
Mumford linkage
DeDion axle

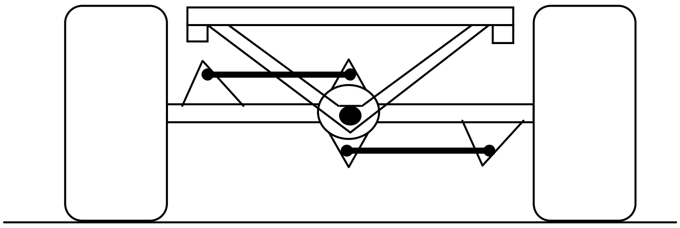


Figure 8 Panhard rod

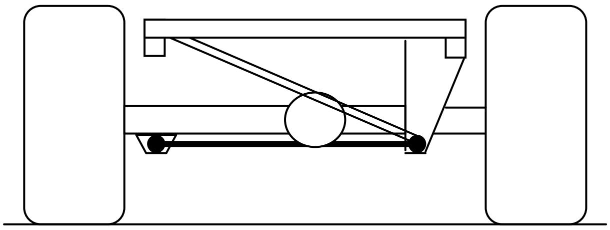


Figure 9 Watts link

2.1.5.2 Independent suspension systems

An independent suspension allows wheels to rise and fall on their own without affecting the opposite wheel. In this case, the wheels are either not connected at all or are connected through universal joints with a swing axle

Independent front suspensions entirely replaced the rigid axel type after World War II and numerous independent rear suspensions came into use, first on European cars.

The main advantage of an independent suspension system is that is permitting the wheels to move independent to each other so that when one wheel hits a bump in the road, only that wheel is affected.

The variety of independent systems is greater and includes:

Swing axle;
DeDion axle illustrated in Figure 10;

MacPherson strut
Wishbone;
Multi-link;
Torsion bar;
Semi-trailing arm.

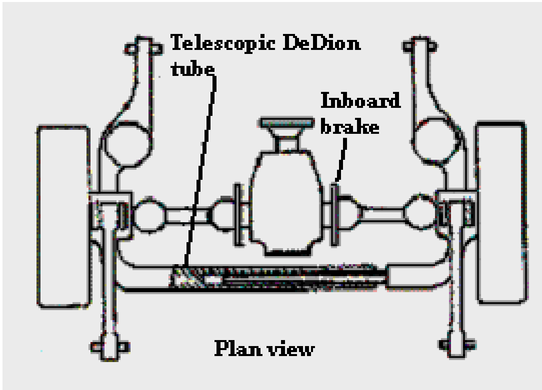


Figure 10 DeDion axle

However, a mixture of the two basic types is called a semi-dependent suspension. In this case, a swing axle is used, but the wheels are also connected with a solid tube, most often a deDion axle.

New suspension technologies in development include a system from the Bose Corporation that uses computer-controlled motors to automatically adjust the suspension to changing road surfaces, keeping a vehicle level and in contact with the road even at high speed over bumpy roads, or in hard cornering.

2.1.6 Front and rear suspension design

The weight of a vehicle is unequally distributed between front and rear therefore the separate front and rear suspension mechanisms have to meet different requirements. Although few automobiles have a non-independent rear suspension, every modern car has an independent front suspension.

The front suspension, which is connected to the steering system, has more critical role in controlling the vehicle's direction. The two systems must align the wheels so that the vehicle travels in the intended direction and the tires remain essentially perpendicular to the road and do not wear abnormally.

There are several different front and rear suspensions system designs, including double A-arms and McPherson strut types (both of which have coil springs), torsion-bar springs and leaf springs.

The double A-arm system has normally completely independent suspension in the front or rear and are commonly used in sports and racing cars. It is a more compact system that lowers the vehicle hood resulting in grater visibility and better aerodynamics.

The McPherson strut system is a coil spring wrapped around a shock absorber that acts as an upper control arm. It is less expensive than double A-arm system and is found in most modern front wheel-drive passenger vehicles.

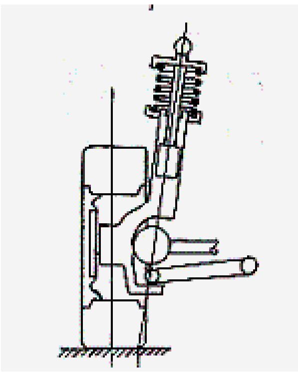


Figure 11 MacPherson strut

Leaf springs are typically used in non-independent rear suspension system and in sport utility vehicles.

2.2 Steering system

2.2.1 Manual steering systems

The manual steering system incorporates:

1. steering wheel and column;
2. a manual gear box and pitman arm or a rack and pinion assembly;
3. linkages, steering knuckles and ball joints;
4. wheel spindle assemblies.

There are several different manual steering gears in current use:

1. worm and sector type;
2. worm and tapered pin steering gear;
3. worm and roller steering gear
4. recirculating ball type where the balls acts as a rolling thread between the wormshaft and the ball nut;
5. rack and pinion type which is the choice of most vehicle manufacturers.

2.2.1.1 Worm and sector steering

The manual worm and sector assembly uses a steering shaft with a three-turn worm gear supported and straddled by ball bearing assemblies. In operation, a turn of the steering wheel causes the worm gear to rotate the sector and the pitman arm shaft and the movement is transmitted through the steering train to the wheel spindles.

2.2.1.2 Worm and tapered peg steering

The manual worm and tapered steering gear as illustrated in Figure 12 has a three-turn worm gear at the lower end of the steering shaft supported by ball bearings assemblies. The rocker shaft has a lever end with tapered peg that rides in the worm grooves. The movement of the steering wheel revolves the worm gear, and that causes the tapered peg to follow the worm gear grooves. Movement of the peg moves the lever on the rocker shaft that in turn moves the rocker arm and the steering linkage.

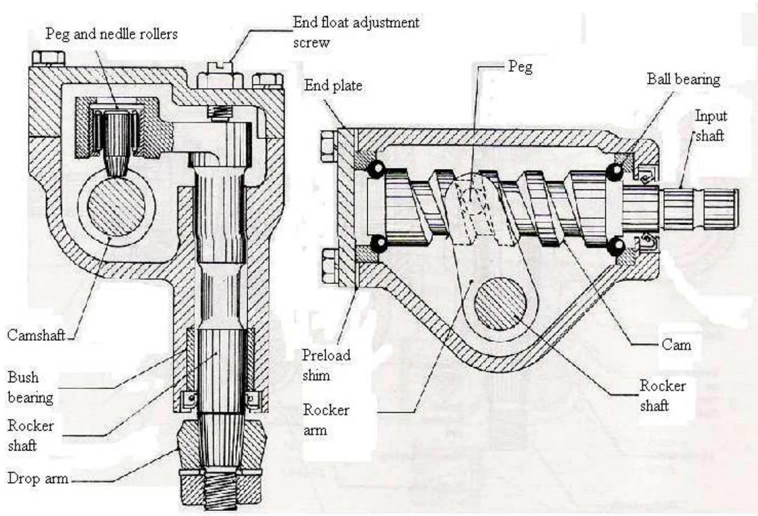


Figure 12 Worm and peg steering

2.2.1.3 Worm and roller steering gear

The manual worm and roller steering gear has a three-turn worm gear at the lower end of the steering shaft as seen in Figure 13. Instead of a sector or tapered peg on the pitman arm shaft, the gearbox has a roller assembly (usually with two roller teeth) that engages the worm gear. The assembly is mounted on anti-frictional bearings. When the roller teeth follow the worm, the rotary motion is transmitted to the pitman arm shaft, pitman arm and into the steering linkage.

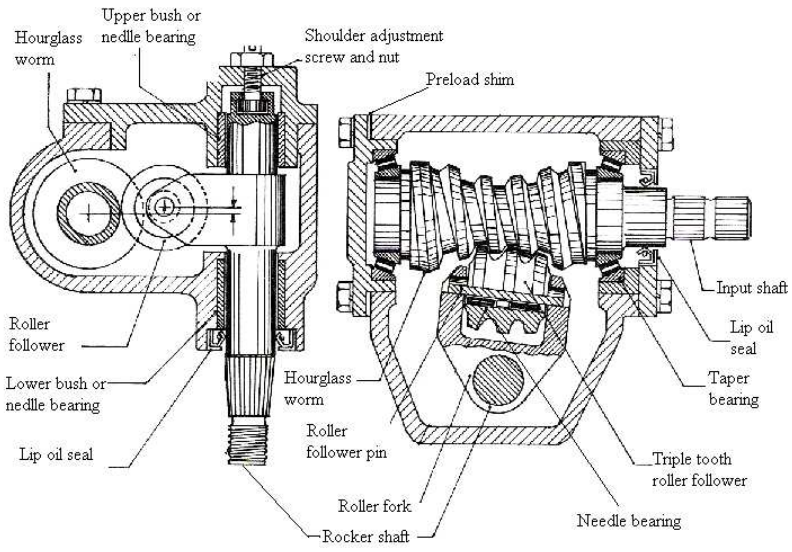


Figure 13 Worm and roller type steering gear box

2.2.1.4 Manual recirculating ball and sector steering

With the manual recirculating ball steering gear illustrated in Figure 14, turning forces are transmitted through ball bearings from a "worm gear" on the steering shaft to a sector gear on the pitman arm shaft. A ball nut assembly is filled with ball bearings, which "roll" along grooves between the worm teeth and grooves inside the ball nut. When the steering wheel is turned, the worm gear on the end of the steering shaft rotates, and movement of the recirculating balls causes the ball nut to move up and down along the worm. Movement of the ball nut is carried to the sector gear by teeth on the side of the ball nut. The sector gear then moves with the ball nut to rotate the pitman arm shaft and activate the steering linkage. The balls recirculate from one end of the ball nut to the other through ball return guides.

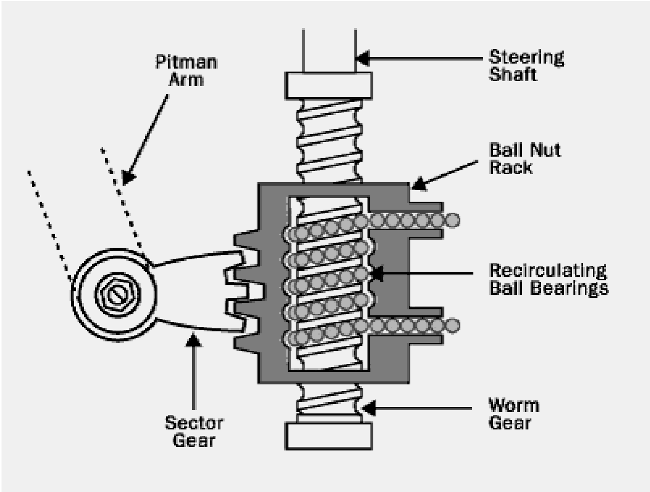


Figure 14 Manual recirculating ball and sector steering

2.2.1.5 Manual rack and pinion steering

A typical rack and pinion steering gear assembly consists of a pinion shaft and bearing assembly, rack gear, gear housing, two tie rod assemblies, an adjuster assembly, dust boots and boot clamps, and grommet mountings and bolts. When the steering wheel is turned, this manual movement is relayed to the steering shaft and shaft joint, and then to the pinion shaft. Since the pinion teeth mesh with the teeth on the rack gear, the rotary motion is changed to transverse movement of the rack gear. The tie rods and tie rod ends then transmit this movement to the steering knuckles and wheels.

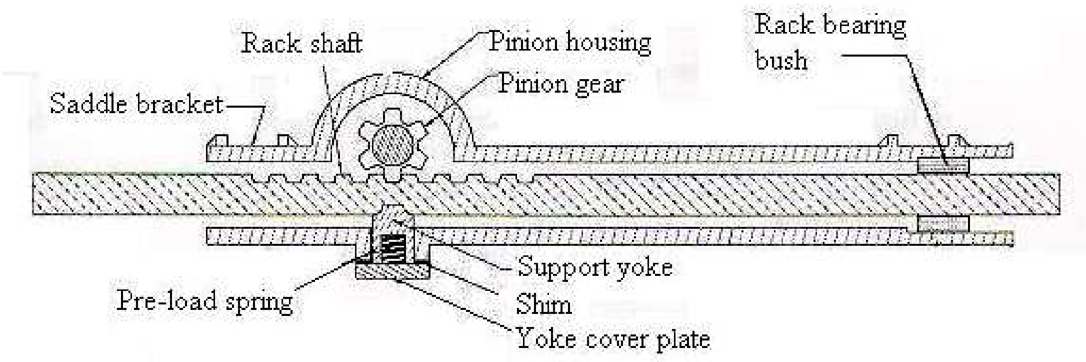


Figure 15 Rack and pinion steering

2.2.2 Power assisted steering system

Power steering makes a heavy car respond easily to the steering wheel, whether at highway speeds or inching into a narrow parking place, and it is normal equipment for large automobiles.

This system requires a power steering pump attached to the engine and driven by a belt, a pressure hose assembly, and a return line. Also, a control valve is incorporated somewhere in the hydraulic circuit. "Power steering" is really "power assisted steering." All systems are constructed so that the car can be steered manually when the engine is not running or if any failure occurs in the power source. Most power steering pumps contain a flow control valve, which limits fluid flow to the power cylinder, and a relief valve which limits pressure according to system demands.

Common types of power assisted steering systems are:

1. power rack and pinion;
2. power recirculating ball.

2.2.2.1 Power rack and pinion

Power rack and pinion steering assemblies are hydraulic/ mechanical unit with an integral piston and rack assembly. An diagram of a rack and pinion steering assembly is illustrated in Figure 16. An internal rotary valve directs power steering fluid flow and controls pressure to reduce steering effort. The rack and pinion is used to steer the car in the event of power steering failure, or if the engine (which drives the pump) stalls.

When the steering wheel is turned, resistance is created by the weight of the car and tire-to-road friction, causing a torsion bar in the rotary valve to deflect. This changes the position of the valve spool and sleeve, thereby directing fluid under pressure to the proper end of the power cylinder. The difference in pressure on either side of the piston (which is attached to the rack) helps move the rack to reduce turning effort. The fluid in the other end of the power cylinder is forced to the control valve and back to the pump reservoir. When the steering effort stops, the control valve is centred by the twisting force of the torsion bar, pressure is equalized on both sides of the piston, and the front wheels return to a straight ahead position.

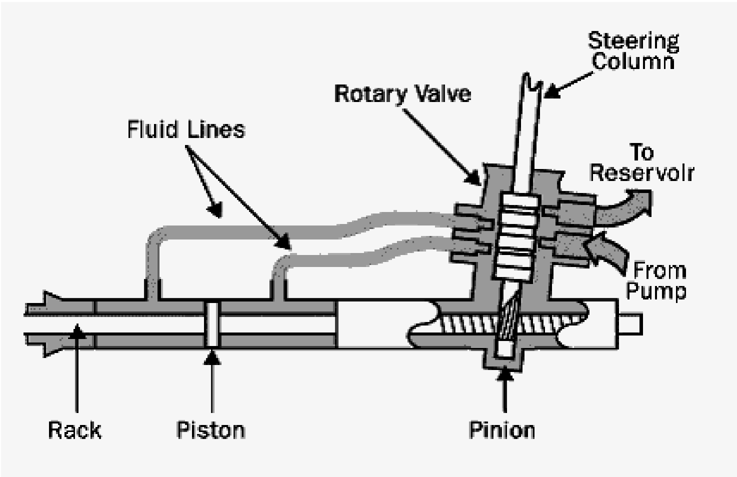


Figure 16 Power rack and pinion steering system

2.2.2.2 Power recirculating ball

This power steering gear uses a recirculating ball system in which steel balls act as rolling threads between the steering worm shaft and the rack piston. The key to its operation is a rotary valve that directs power steering fluid under pressure to either side of the rack piston. The rack piston converts hydraulic power to mechanical force. The rack piston moves up inside the gear when the worm shaft turns right. It moves down when the worm shaft turns left. During these actions, the steel balls recirculate within the rack piston, which is power assisted in movement by hydraulic pressure.

2.3 Geometry parameters involved in suspension and steering

2.3.1 Wheel camber

2.3.1.1 Camber angle

Camber angle is regarded as the inclination of the wheel plane to the vertical (SAE J670e). Negative camber inclines the top of the tyre toward the centreline of the vehicle as seen in Figure 17 and positive camber inclines the top of the tyre away from the centreline.

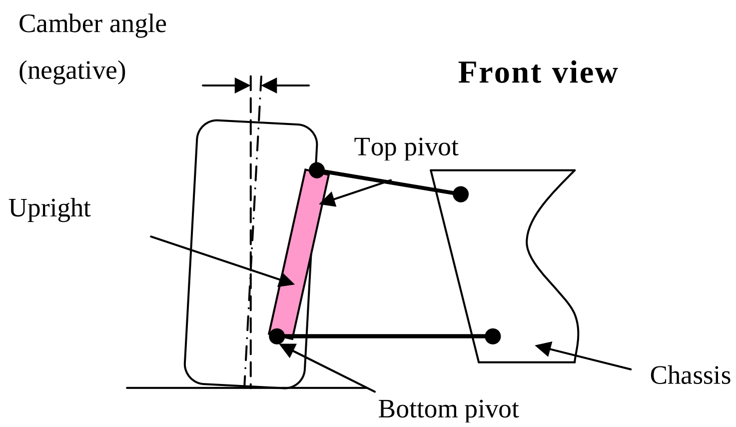


Figure 17 Camber angle

A small amount of negative camber of up to 1.5 degrees it is recommended in order to induce camber thrust (Smith 2004). However, changes in camber should be kept at minimum during chassis roll in order to reduce the loss of camber thrust and the change in wheel track load distribution during cornering.

2.3.1.2 Rate of camber change

The rate of camber change is the change of camber angle per unit vertical displacement of the wheel centre relative to the sprung mass (SAE J670e).

2.3.2 Wheel caster

2.3.2.1 Caster angle

Caster angle is the angle in side elevation between the steering axis and the vertical. It is considered positive when the steering axis is inclined rearward (in the upright direction) and negative when the steering axis is inclined forward (SAE J670e). Caster angle can be visualised on Figure 18.

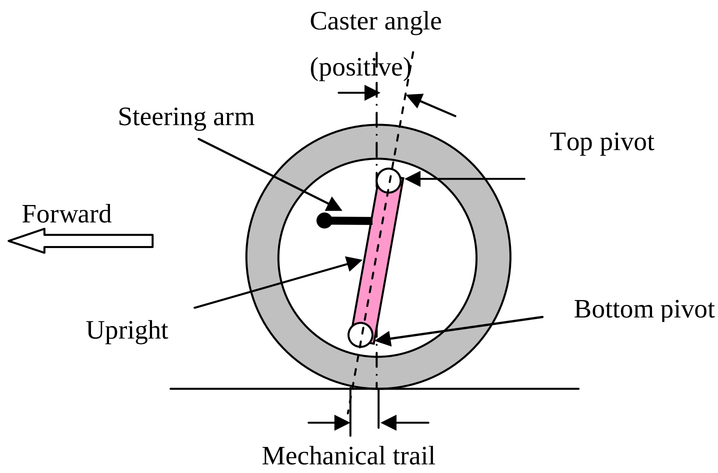


Figure 18 Caster angle

Positive caster induces a self correcting force that provides straight line stability, but increases steering effort. Caster ranges from approximately 2 degrees in racing vehicles up to 7 degrees in sedans (Smith 2004).

2.3.2.2 Rate of caster change

The rate of caster change is regarded as the change in caster angle per unit vertical displacement of the wheel centre relative to the sprung mass (SAE J670e).

2.3.3 Kingpin geometry

2.3.3.1 Kingpin inclination

The angle in front elevation between the steering axis and the vertical is regarded as kingpin inclination (SAE J670e). It is also known as steering axis inclination (SAI) and can be seen in Figure 19.

It is used to reduce the distance measured at the ground between steering axis and tyre's centre of pressure in order to reduce the torque about the steering axis during forward motion. A right kingpin inclination will reduce the steering effort and will provide the driver with a good 'road feel"

2.3.3.2 Kingpin offset

Kingpin offset measured at the ground is the horizontal distance in front elevation between the point where the steering axis intersects the ground and the centre of tyre contact (SAE J670e).

Kingpin offset it is also known as scrub radius. It is positive when the centre of tyre contact is outboard of the steering axis intersection point on the ground. Kingpin offset is usually measured at static conditions (zero degree camber).

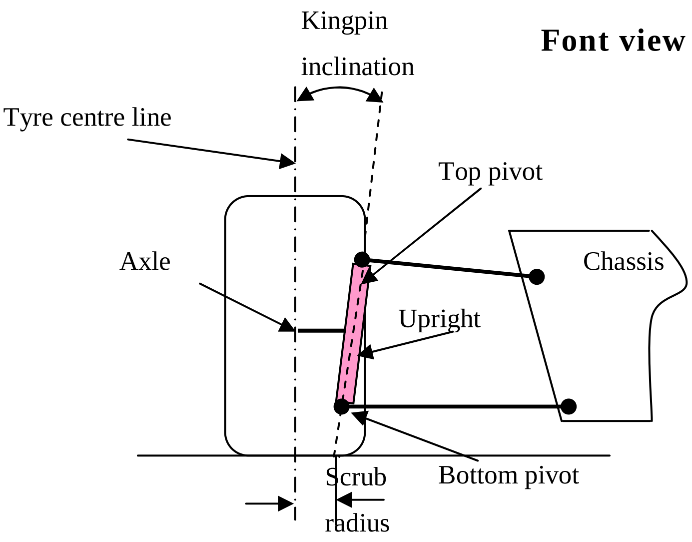


Figure 19 Kingpin inclination

The kingpin offset at the wheel centre is the horizontal distance in front elevation from the wheel centre to the steering axis (SAE J670e).

2.3.4 Wheel toe

2.3.4.1 Static toe angle

Static toe angle is measured in degrees and is the angle between a longitudinal axis of the vehicle and the line of intersection of the wheel plane and the road surface. The wheel is “toed-in” if the forward position of the wheel is turned toward a central longitudinal axis of the vehicle, and “towed-out” if turned away (SAE J670e).

2.3.4.2 Static toe

Static toe-in or toe-out of a pair of wheels is measured in millimetres and represents the difference in the transverse distance between the wheel planes taken at the extreme rear and front points of the tyre treads. When the distance at the rear is greater, the wheel is “toed-in” by this amount; and where smaller, the wheels are “toed-out” (SAE J670e) as illustrated in Figure 20.

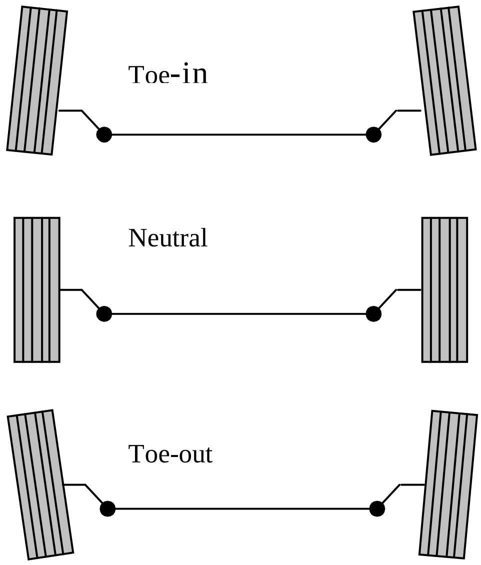


Figure 20 Toe in/out

It is necessary to set the static toe such way to prevent the tyres to become toe-out during maximum bump and roll in order to prevent the outboard tyre to steer the vehicle to the outside of the turn when cornering.

Toe-in produces a constant lateral force inward toward the vehicle centreline during forward motion that will enhance the straight line stability.

2.3.5 Ackermann steering

When a car goes round a corner, it turns around a point along the line of its rear axle, which means that the two front wheels will have to turn through slightly different angles so that they are also guiding the vehicle round this point, and not fighting the turn by scrubbing. This principle is illustrated in Figure 21.

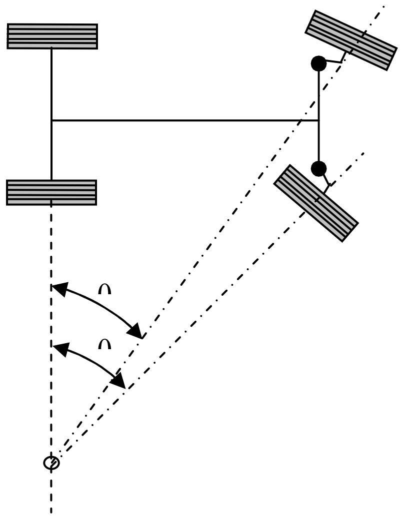


Figure 21 Akerman principle

Ackerman geometry results when the steering is done behind the front axle and the steering arms point toward the centre of the rear axle as seen on Figure 22

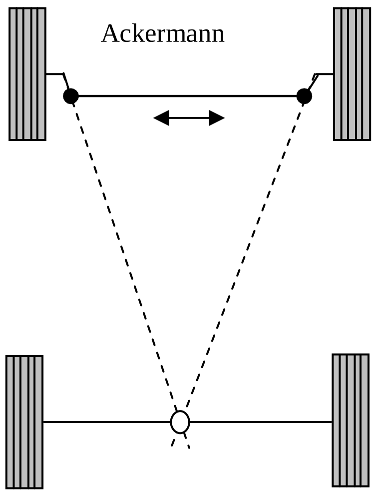


Figure 22 Ackermann construction

2.4. Suspension kinematic parameters

2.4.1. Instantaneous centre

The instantaneous centre of a suspension is the point through which an individual wheel rotates and is also referred to as the "swing centre" or virtual half-shaft" (Smith 2000). It is also the point through which the respective tyre force acts on the sprung mass.

The instantaneous centre of a four bar link independent suspension is located at the intersection of the lower and upper link extensions as seen in Figure 23. When analysing suspension kinematics, both left and right suspensions must be analysed together.

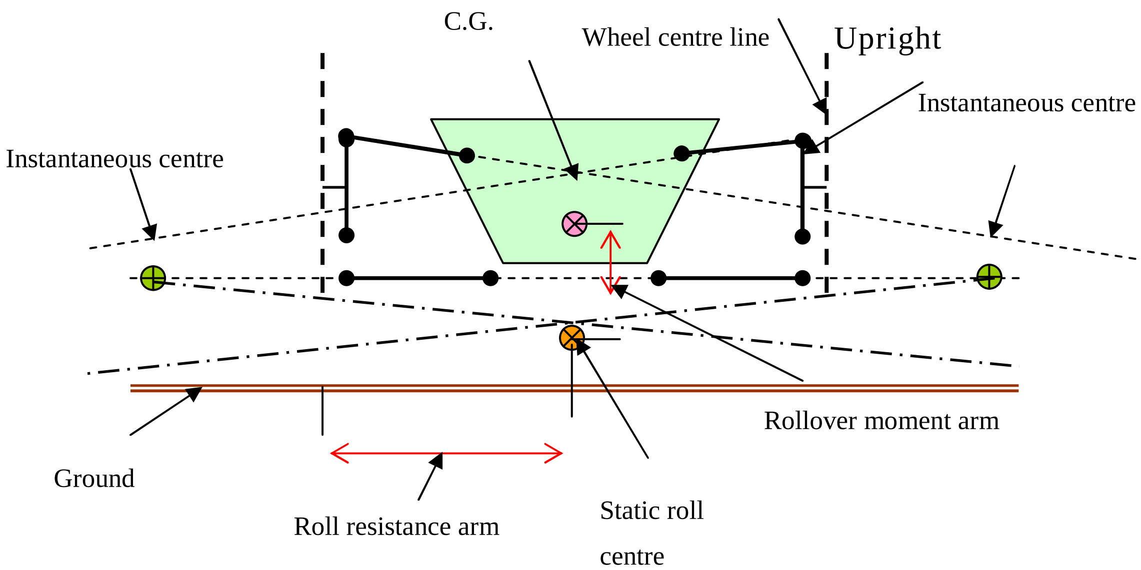


Figure 23 Instantaneous centre/ roll centre

2.4.2. Roll centre

The point in the transverse vertical plane through any pair of wheel centres at which lateral forces may be applied to the sprung mass without producing suspension roll (SAE J670e).

2.4.2.1. Roll centre location

The roll centre is located at the intersection of the lines formed by the tyre contact patches and their respective instantaneous centres. In Figure 23 the tyre contact patches are assumed at the intersection of wheel centre line with the ground, while in reality these two points may not coincide. The location of the roll centre is usually different for front and rear suspension.

The vertical location or height of the roll centre determines the resulting two moment arms formed between the roll centre and both the C.G. and the ground plane. These two moment arms determine the vehicle’s sensitivity to lateral acceleration by producing rollover moments and jacking forces (Smith 2000).

2.4.2.2. Roll resistance arm

The roll resistance arm is the lever arm formed between the thread’s centre of pressure and the vehicle centre line. This moment arm creates a roll resisting torque when acted on by the reaction forces generated at the tyre contact patch by the spring and anti-roll bars (Smith 2000).

2.4.2.3. Rollover arm

Rollover moment arm is the summation of three components that results from three force and roll moment arm pairs. These are lateral acceleration of the sprung mass acting on the arm formed by the CG and roll centre, vertical accelerated sprung mass acting on the arm formed by the lateral displacement of CG and vehicle’s centre line and finally the jacking forces acting on the arm formed by lateral displacement of the roll centre from the centre line during roll.

2.4.2.4 Jacking

The tyre reaction forces generated when the vehicle is accelerated during cornering are transmitted to the vehicle through the suspension links. In suspension that place the roll centre above the ground, the upward tyre reaction force generated by the outside tyre is greater than the downward tyre reaction force generated by the inside tyre. Summing these forces the resultant will be positive upward acting through its roll centre. This upward jacking force lifts or “Jacks” the sprung mass upward when cornering.

Chapter 3

Analysis of the 2004 race car for design changes

The USQ 2004 racer car has been tested in the competition last year as well as on couple occasions this year. After minor modifications on the car to overcome the understeer effect and the uneven distribution of weight on the front and rear pair of wheels the test result this year were good in terms of handling and manoeuvrability. However, the overall performance of the car needs to be improved in order to make it competitive.

The overall car design was very robust with many components over designed and many afterthoughts chassis trusses and suspension brackets. Although very reliable it proved to add some unnecessary weight therefore reducing the chances to meet the weight criteria.

3.1. Analysis of 2004 car steering system

3.1.1. Steering system geometry

The steering system geometry used was 100% Ackermann geometry with a steering angle of about 18 degrees and the steering box placed at a distance of 50 mm behind the front axle and just above the upper chassis rail. This position was to low for a driver taller than 1.80 meters who had his/her shins right against the steering mechanism. In a more serious impact this would cause leg injuries.

3.1.2. Steering system mechanism

The steering mechanism used is shown in Figure 24 and consisted of a modified rack and pinion gearing in a custom made steering box. This mechanism was a good choice since the car is light and does not require power assistance, it is cheap, takes up a very small amount of space and is very responsive. However, the assembly had an amount of free play at the steering wheel above the design limit set by the competition rules.

Figure 24 2004 Steering box

3.2. Analysis of 2004 car suspension system

3.2.1. Suspension system type

The suspension system for 2004 racer car was four wheel independent suspension with double wishbone, activated by push rod. The suspension layout was parallel arms with unequal lengths.

3.2.2. Suspension system components

3.2.2.1. Hub axle

The front hub/axle was designed to accommodate brake components and it was machined from one piece of mild steel as seen in Figure 25. Analysing this component with Finite Element software it proved to be over designed.

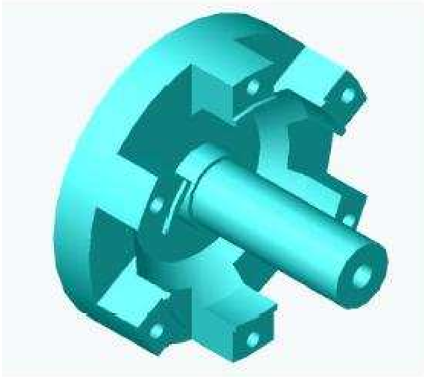


Figure 25 2004 Front hub axle

3.2.2.2 Uprights

The uprights were fabricated from mild steel. The front upright has an incorporated 4.7 degrees of kingpin inclination. The design shown in Figure 26 allowed single shear top and bottom suspension mounting points, and that induced bending in the lower wishbone spherical rod end that is under the greatest load.

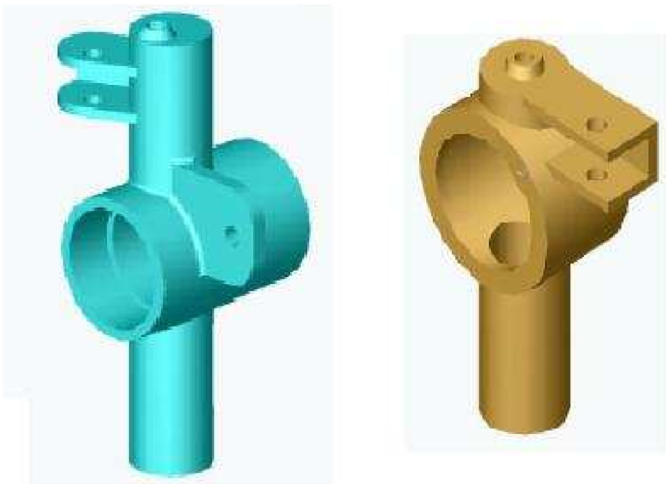


Figure 26 2004 Front and rear uprights

3.2.2.3 Control arms and suspension arm

All control arms or so called wishbones, and suspension arm as well as steering tie rods were manufactured from circular tubing steel ERW, 19 mm diameter and 2mm wall thickness. Adjustable spherical rod ends of different material and different dimensions were used. This variety of rod ends did not provided consistency in the design that makes it difficult when tuning the car.

3.2.2.4 Springs and dampers

The motorbike spring/shock assembly used consists of coil spring mounted over oil filled adjustable shock absorber. The mounting position of the assembly was almost horizontally due to packaging constrains but was not according to the shocks specifications that require vertical mounting therefore reducing damper performance and causing oil liking. Therefore the position of the spring/shock should be carefully considered to comply with the shocks specifications and avoid such incidents.

3.2.2.5 Mounting points

For mounting points location some extra rails has been welded on to the chassis that added extra weight to the design. The brackets were fabricated from flat mild steel with 5 mm thickness therefore the design proved to be robust but heavy.

Chapter 4

Design constrains

4.1 Design constrains set by the Formula SAE rules

4.1.1 Wheelbase and vehicle configuration

The car must have a wheelbase of at least 1525 mm (60 inches).
The vehicle must have four wheels that are not in a straight line.

4.1.2 Vehicle track

The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track.

4.1.3 Wheels

The wheels of the car must be 203.8 (8.0 inches) or more in diameter.
Any wheel mounting system that uses a single retaining nut must incorporate a device to retain the nut and the wheel in the event that the nut loosens.

4.1.4 Suspension

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 (1 inch) redound, with driver seated (2005 Formula SAE rules).

4.1.5 Steering

The steering system must affect at least two wheels and must have positive steering stops placed either on the uprights or on the rack. This is to prevent the steering linkages from locking up and the tyres from contacting suspension, body or frame members during track events.

Allowable steering free play is limited to 7 degrees total measured at the steering wheel. Rear wheel steering is permitted only if mechanical stops limit the turn angle of the rear wheels to ± 3 degrees from straight ahead position. The steering wheel must be mechanically connected to the front wheels.

4.1.6 Steering wheel

The steering wheel must have a continuous perimeter that is near circular or near oval. The steering wheel must be attached to the column with a quick disconnect that can be operated by the driver in the normal driving position with gloves on (2005 Formula SAE rules)
The steering wheel must be attached to the column with a quick disconnect that can be operated by the driver in the normal driving position with gloves on (2005 Formula SAE rules)

4.1.7 Fasteners

All threaded fasteners utilised in the steering, breaking, safety harness and suspension must meet or exceed, SAE Grade5, Metric Grade M 8.8 and/or AN/MS specifications (2005 Formula SAE rules).

All critical bolt, nuts and other fasteners on the steering, breaking, safety harness and suspension must be secured from loosening by use of positive locking mechanisms (2005 Formula SAE rules).

All spherical rod ends 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. Adjustable tie-rod ends must be constrained with a jam nut to prevent loosening (2005 Formula SAE rules).

4.2 Other design constrains

Apart from the design constrains set by the Formula SAE rules the steering system and suspension design must consider the implied constrains for materials and parts availability, weight, cost as well as packaging constrains. Although all are equally important for evaluating the design in the competition, many times compromises have to be made. In this case the judging criteria should be addressed and the solution that will bring the most advantage to the overall score of the competition to be selected.

Chapter 5

Steering system design

5.1. Steering system requirements

A steering system must offer sufficient precision for the driver to actually sense what is happening at the front tyres contact patch as well as enough “feel” to sense the approach to cornering limit of the front tyres. It must be structurally stiff to avoid components deflections.

The steering must be fast enough so that the vehicle’s response to steering and to steering correction to happen almost instantaneous and it must also have some self returning action.

The feel, feedback and self returning action are function of the kingpin inclination, scrub radius, castor angle and self aligning torque characteristics of the front tyre.

5.2. Design of the steering system geometry

Although modern cars do not use 100% Ackerman since it ignores important dynamic and compliant effects, the principle is sound for low speed manoeuvres. The competition track set up allows only for low cornering speed. In this case the tyres are at small slip angles therefore, 100% Ackerman is the best option.

In consultation with the team, in our primary phase of the design we decided the wheelbase and the track width. However, at the beginning of the second semester a major decision was made to use for this year competition the previous year chassis.

Since the geometry used last year proved to work well, the decision was made to use for this year project same 100% Ackermann geometry.

5.2.1 Ackermann condition

For the Ackermann analysis the Ackermann condition is used to determine the relationship between inner and outer wheel in a turn and the radius of turn.

General equation:

$$\frac{1}{\tan \theta o} - \frac{1}{\tan \theta i} = \frac{B}{L}$$

Where: θo= turn angle of the wheel on the outside of the turn

θi= turn angle of the wheel on the inside of the turn

B= track width

L= wheel base

b= distance from rear axle to centre of mass

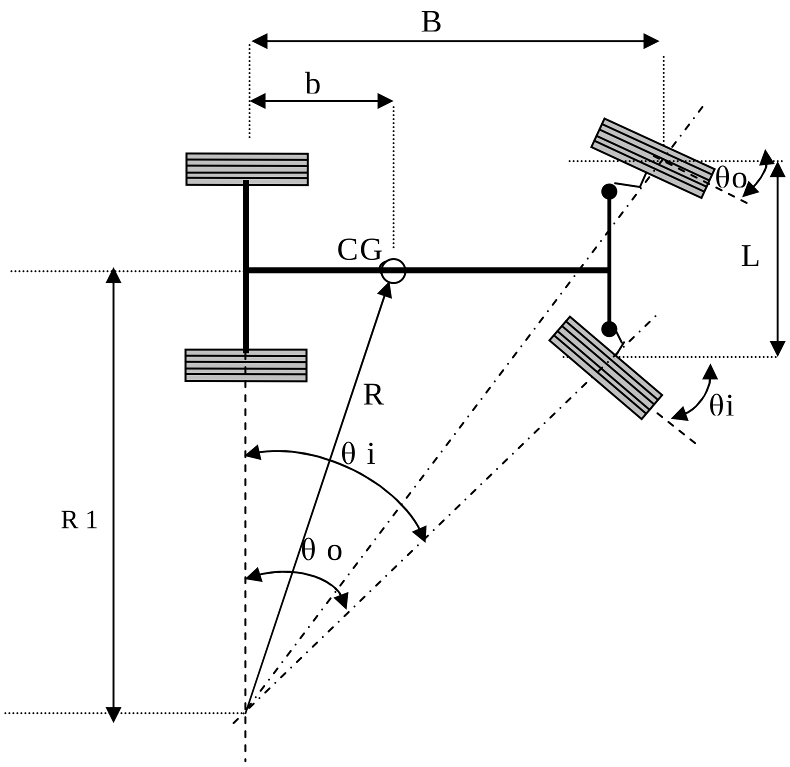


Figure 27 Ackermann condition

From the general equation we can calculate the turn angle of the wheel on the outside of the turn for a given inside wheel angle as follows:

B=1800 mm

L=1320 mm

θi=30˚

$$\frac{1}{\tan\theta o}=\frac{1}{\tan\theta i}+\frac{L}{B}=\frac{1}{\tan 30}+\frac{1320}{1800}=2.465$$

$$\theta o = 22.12°$$

The minimum radius of turn R can be determined from the geometry:

$$R1 = \frac{B}{\tan \theta i} + \frac{L}{2} = \frac{1800}{\tan 30} + \frac{1320}{2} = 3778mm = 3.7m$$

$$R = \sqrt{R1^2 + b^2} = \sqrt{3778^2 + 800^2} = 3862mm = 3.86m$$

Therefore the minimum radius of turn of the vehicle around its centre of gravity for a maximum inside wheel turn of 30 degrees is about 4 meters.

5.2.2 Selection of the steering parameters

The initial decision of zero degree kingpin inclination had to be reconsidered since the 56 mm of scrub radius resulted is large and will give an excessive feedback to the driver. Therefore 4 degree kingpin inclination is to be build in the front upright design that will result in an amount of scrub radius of 30mm calculated for last year wheel offset. Since this amount is still grater than 10% of the thread width (Heisler 1989), new wheels with less offset have been found therefore the resulting scrub radius is about 20 mm that is the amount we aimed for.

The amount of castor angle was set to 3.5 degree and is also build in the front uprights. However, castor angle can be adjusted by adjustment of the upper wishbone. This requires that one arm of the wishbone to be shortened while lengthening the other arm by screwing in or out the adjustable spherical rod ends. Another possible adjustment is to assemble the upright in an inclined position on the hub axle but this is not a handy method of adjustment.

5.3. Selection of the steering mechanism

From all manual steering systems the more suitable is rack and pinion steering for the following reasons:

-has a simple construction;

- is cheap and readily available;
- has a high mechanical efficiency;
- has a reduced space requirement.

Since last year rack and pinion steering mechanism had an undesirable amount of free play the decision was made to modify one of the two steering mechanisms sourced by the team members as donations for the project.
The rack and pinion steering box selected is from a Honda Civic 1983 and has a 5 teeth pinion gear and a pitch on the rack of 4.5mm.

The steering box assembly have been modified by Bruce Llewellyn, one of the team members. The rack has been shortened and the assembly was kept in the original steering box. The input shaft is not in a central position therefore the steering column will be connected to the input shaft through a universal joint.

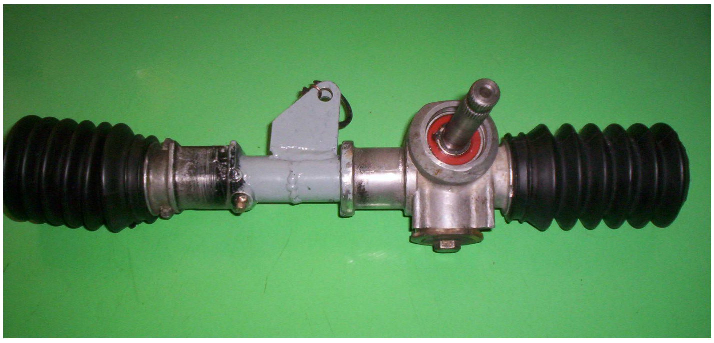


Figure 28 Modified steering box assembly

5.4. Position of the steering mechanism

5.4.1 Position relative to the front axle

The steering box position in a horizontal plane is to remain at 50 mm behind the front axle as seen in Figure 29.

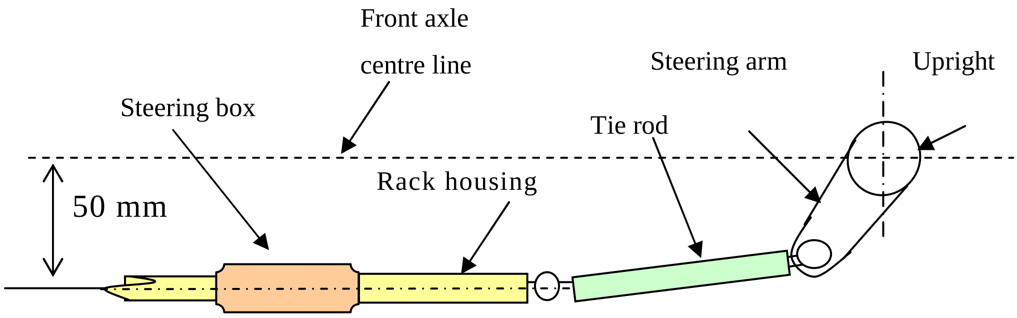


Figure 29 Top view diagram of steering system geometry

5.4.2 Position in a vertical plane

In a vertical plane, two placement possibilities were investigated. To place the steering box at the bottom of the chassis just above the bottom chassis rails or to place it above the actual position such as the steering column will be horizontal. The first possibility would cause trouble in getting into the car when the driver's feet must pass first over the steering mechanism. Even worse getting out of the car when the hell of the foot will be caught by the mechanism. Therefore, this option is inconvenient leaving the only possibility to place the steering box above the actual position.

For this location the steering assembly is at a higher position than the steering arm even though the steering arm is attached to the upright above the centre of the wheel. Therefore, the steering linkages had to be redesigned. The solution was to use a rocker arm but this will result in a reversed movement of the tie-rod relative to the rack as seen in Figure 30.

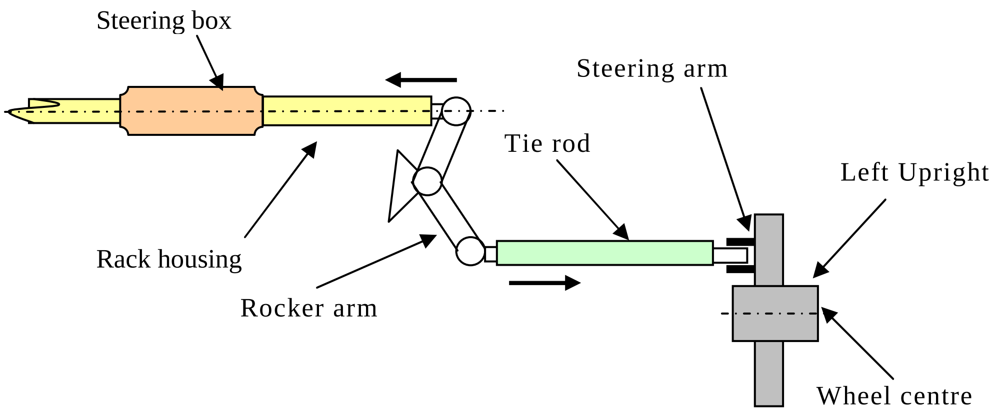


Figure 30 Front view diagram of steering system geometry

The problem can be overcome by reversing the position of the rack relative to the pinion as seen in Figure 31.

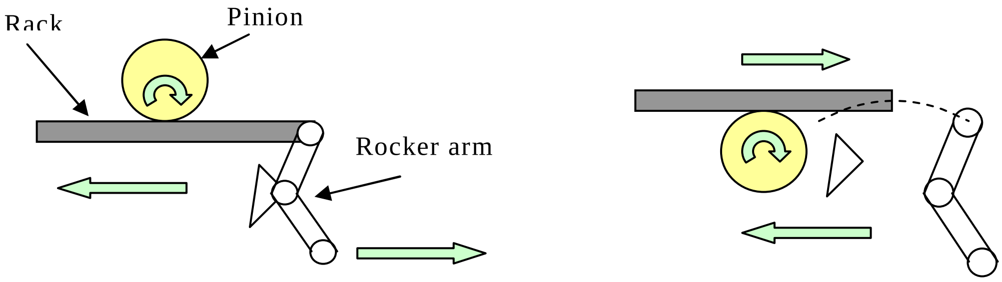


Figure 31 Position of the pinion relative to the rack

Another problem encountered was that the pivot points of the rocker arm will follow a circular path producing a displacement of the rack and pinion assembly therefore the assembly has to have a small amount of rotational freedom. The solution for this problem is to have a rather floating steering box attached to the chassis upper rail seen in Figure 32 or in Appendix E.

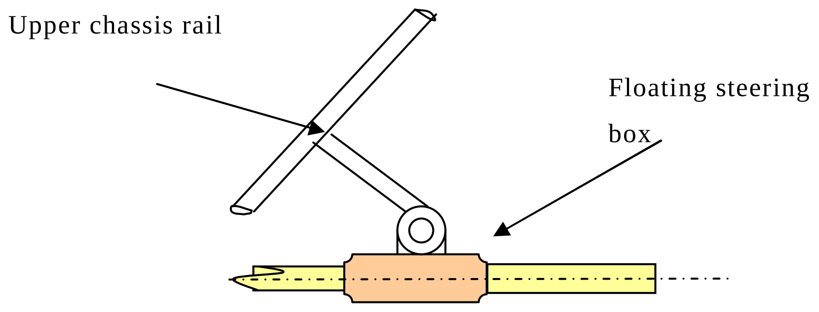


Figure 32 Mounting of the steering box

After the suspension components are assembled on the chassis and the steering mechanism in place, the distance between lower rocker arm pivot point and steering arm mounting point can be measured, hence the length of the tie rod can be determined. The tie rod will have left and right threaded spherical rod ends for adjustment of the link length as a method of adjustment of the bump steer.

5.5 Bump steer geometry

In the design of the steering geometry, the bump steer and roll steer effect should be taken into account. Bump steer results from the combination of wheel toe in or toe out with wheel vertical travel while the roll steer is produced by the combination of toe in/out and body roll. The suspension and steering links should be placed such way to minimise the distortion of the steering geometry with suspension movement. For this reason, to minimise the bump steer, the steering tie rod should be parallel with the upper wishbone and to minimise roll steer the upper wishbone inboard pivot points should be in the same vertical plane with tie rod inboard pivot point.

5.6 Steering movement ratio

The rack and pinion mechanism is designed to transfer the circular input motion of the pinion into linear output movement of the rack.

It was measured that for a full travel of the rack of 137 mm the pinion has to be rotated $3^1/_8$ turns
Therefore for one turn, the rack travel will be:

$$x_o = \frac{137}{3.125} = 43.84mm$$

Considering the pinion to make one revolution then the input steering movement is:

$$x_i = 2\pi R$$

Where, R = 155 mm is the radius of the steering wheel.

And the output rack movement is:

$$x_o = 2\pi r = 43.84 \Rightarrow r = \frac{43.84}{2\pi} = 6.97 = 7$$

Then, the movement ratio can be calculated as input movement over output:

$$MR = \frac{x_i}{x_o} = \frac{2\pi R}{2\pi r} = \frac{155}{7} = 22$$

Therefore the movement ratio is 22:1

We needed to know the movement ratio in order to determine the output load transmitted to the tie rods for a given input load.

For an effort of 20 N applied by each hand on the steering wheel and considering no friction, the output load will be:

$$F_o = F_i xMR = 2x20x22 = 880N$$

Therefore the load transmitted to the tie rods is 880 N.

5.7 Quick release mechanism

As a requirement of the design for the competition the steering wheel must be present with a quick release mechanism. Following discussion with the team it was decided to make use of the last year quick release mechanism. The mechanism is simple and consists of a hexagonal socket with a spring loaded plunger release.

Chapter 6

Suspension system design

6.1 Design approach

Choosing suspension geometries and components involves a wide range of choices and compromises. An analysis of the tyre, chassis and road interaction is required to decide the trade-offs that will result in an optimum configuration for the type of vehicle and the nature of the race track for which the vehicle has to perform.

The basic steps in designing a vehicle’s suspension are:

- selection of the suspension type to be employed;
- selection of the wheels;
- establish the vehicle’s dimensions – wheel base and track width(s);
- set up suspension parameters;
- model the suspension geometry;
- design components.

6.2 Suspension type and geometry selection

The starting point in designing the geometry of suspension system was to first select the type and the geometry of the suspension. Double wishbone independent suspension is the common option for racer cars because it is light and can be designed with a large freedom of adjustment.

Two options were investigated for the geometry of the suspension: parallel arms with unequal lengths layout used for last year design and unparallel arms with unequal lengths. Although previous year design was a good choice, for this year project the second option was selected. The unparallel arms will result in a lower roll centre that will produce a bigger roll moment arm, hence more roll for the sprung mass and less jacking force.

6.3 Wheel selection

Then a decision had to be made about the wheels to be used. One of the team members investigated the availability and cost of wheels and the team decision was that a 13 inches wheel with at least 20 mm offset is to be used.

6.4 Wheelbase, track width and pivot points location

The wheelbase is the distance between the front and rear axles and has to be determined prior to chassis design. The advantages of a relatively long wheelbase are increased straight line stability, reduced longitudinal load transfer and more room to pack components in while the advantages of a relatively short wheelbase is reduced overall weight and increased manoeuvrability (Smith 1978).

The track width is the distance between a pair of wheels centre lines and is determined by the lengths of the links. The advantages of wide track width are reduced lateral load transfer for a given amount of centrifugal acceleration and less camber change. The increase in the frontal area will represent a disadvantage only for high speeds where the aerodynamics plays an important role.

Therefore in making a selection one shall keep in mind that a race car with long wheelbase and narrow track width will be very stable in a straight line at the expenses of cornering power and manoeuvrability while a race car with short wheelbase and wide tracks will be lass stable, will develop more cornering power and be more manoeuvrable. In addition, a wider track width at the front than at the rear will provide more stability in turning the car into corners decreasing the tendency of the car to trip over itself on corner entry and more resistance to diagonal load transfer.

Giving all the above considerations, the wheelbase agreed on was 1650 mm, which is slightly shorter than last year's design and a track width of 1370mm at the front and 1280mm at the rear.

The pivot points location was decided initially on the drawings of the chassis provided by Tony O'Neal who is designing the chassis. The chassis has been manufactured by the beginning of second semester but it was thought that due to time and cost constrains a decision have to be made in order to ensure the completion of the project. Therefore, the team decided to use for this year application the previous year chassis because had already components that can be used like fuel tank, fire wall, brake pedals, etc which do not require time to manufacture and assemble.

This had been a critical decision in terms of suspension design that had to be entirely reconsidered. The wheelbase is now 1800mm while but the track width is not affected and so the link lengths can be calculated.

6.5 Geometry parameters selection

Camber is one of the suspension parameters that should be considered. Negative camber will produce a better contact patch shape, producing additional lateral force without a large increase in slip angle and tyre heating. A small amount of negative camber is desired such that on corners the outside wheel does not go into an excessive negative camber angle.

Since camber is easily adjustable by the lengths of the control arms, an initial three degrees static negative camber angle is to be trial on the test track. The goal is to achieve the minimum change in camber over the range of suspension travel.

The change in camber with 10 mm bump/drop and 1 and 3 degrees of roll on the final suspension geometry has been modelled in AutoCAD as can be seen in Appendix F.

Another important suspension parameter is the roll centre. Both roll centre and camber angle have been defined in 2.4.2. Roll centre. A position of the roll centre has been also modelled with AutoCAD software and the result as seen in Appendix F is 70 mm below the ground.

	Camber change	Track width change
10 mm bump	0.48 deg.	1.35 mm
1 deg. roll	0.42 deg	2.02 mm
3 deg. roll	2.25 deg	15.25 mm

6.6 Suspension components design

6.6.1 Hub axle

Since this year identical break components have been sourced, the decision was to use the same hub axle because it has been designed to accommodate those particular brake components. However, the hub was too heavy and a Finite Element Analysis showed that the component was over designed.

After removing some of the material in not so critical areas and reducing the shaft diameter from 30 mm to 20 mm an estimated reduction of 1.2 Kg per component weight can be achieved. However, the reduction in shaft diameter had to be checked for the stress concentration in the shoulder area of the shaft. Consequently, a second analysis was performed and the report is attached in

Appendix B. The detail drawing of the component showing the dimensions to be modified is attached in Appendix D.

6.6.2 Uprights

The 2004 uprights design have been analysed and thorough consideration for the competition rules that allows for spherical rod ends in single shear captured by an appropriate screw/bolt had been given. However, the judges did criticise this practice and strongly encouraged for the double shear design of the spherical rod ends pivot points.

Considering this and the fact that the modified hub axle shaft diameter is smaller the decision was to entirely redesign the uprights.

6.6.2.1 Method of fabrication

In order to decide the method of fabrication of the components, three options were investigated.

Option 1 – Machine the component from one piece of material.

This option can result in a compact design with intricate shape if necessary and allows the use of a wide range of materials like aluminium alloys which will result in a much lighter component.

However, later modifications of the design will not be possible therefore the design must be from the beginning very precise. Apart from that it requires skilled operator and together with the cost of the material will result in a relatively high total cost of the component.

Option 2- Fabricate the component by casting

This method of fabrication has same advantages as option one. In addition, the method requires some machining but the amount of metal to be removed is minim. Also the method requires the design of special tools like dies as well as special facilities for casting process. Although the initial cost of the dies is high, it is divided by the number of components produced so, for a thousand units production the cost per unit will be relatively low. Once again, the design dimensions must be accurate.

Option 3 – Fabricate the component by welding parts together

The advantage of this method is that the materials are inexpensive and available in the workshop shelves and does not require skilled labour to fabricate the component. It is also possible to modify the design at a later stage if necessary. This option will result in a relatively heavier component compared with option one but with a lower total cost.

The decision was to compromise on weight of the component to gain in cost and in the advantage of easily modify and adjust the upright design in the case of other components design changes. Hence, the selected method for fabrication of the uprights was by welding.

6.6.2.2 Material selection

The criteria used in the material selection for manufacture of the uprights was:

1. the material have to be easy to be machined, cut and welded;
2. have to be relatively light for its strength;
3. have to be relatively inexpensive and available.

After discussion with the USQ workshop it was found that only mild steel fitted all the criteria and that a wide range of dimensions and cross sections are available.

6.6.2.3 Design concepts

Design concept 1

The first general design concept of the uprights consisted of a circular section with machined inside diameters to suit the wheel bearings. Two parts of "C" channel with appropriate manufactured end configuration for the wishbones assembly are to be welded on top and bottom of the circular part. The steering arm, and brake calliper mounting bracket is then welded on the upright. This concept is illustrated in Figure 33. The front upright provides a zero kingpin inclination and an adjustable zero or three degrees caster.

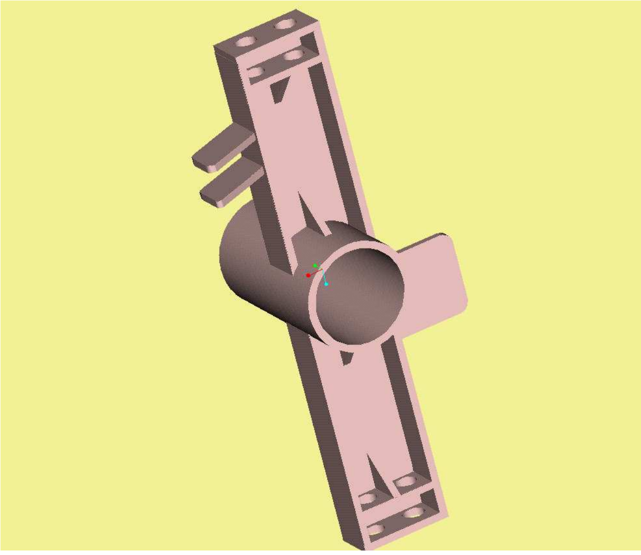


Figure 33 Upright design concept 1

Although, the front and rear upright are different the same method of fabrication and materials are to be used for both components.

Design concept 2

The second design concept uses the same circular section for the bearings housing but has a RHS square section welded on the top and bottom. A different end configuration for the wishbone pivot points is to be welded on the other ends of this section. This configuration allows for a kingpin inclination and caster to be build in. This design is illustrated in Figure 34.

Final design

Both designs have a manufacture medium to low degree of difficulty since they uses simple parts. The cost of fabrication is expected to be near the same. However using an open section like the “C” channel used in the first design has the disadvantage of less torsional stiffness than a closed section. The addition of ribs to reinforce the structure will add more parts into the component hence increasing the weight and complexity without achieving same degree of stiffness expected from a closed section. Moreover the first design does not allow for later modification of the wishbones pivot points locations, while for the second design the brackets can be easily removed and welded back in a different position. This provision for easy adjustment of the pivot points it eliminates the necessity to remanufacture the whole component in the eventuality of later changes in geometry of the suspension.

Therefore, the second design concept illustrated in Figure 34 has been selected. The detailed design drawings of both front and rear uprights are attached in Appendix D.

Figure 34 2005 Front and rear final design

6.6.3 Control arms (wishbones) and suspension arm

6.6.3.1 Material selection

In selecting the material for the wishbones same criteria had to be considered as for the uprights material selection.

ERW circular steel tube appeared to be a good choice and after discussion with the USQ workshop staff it was found that is low-priced and the common sizes can be delivered in just a week time therefore, the material matched up the selection criteria.

Hence, the material selected was ERW circular steel tube with an OD of 19 mm and thickness of 1.6 mm that will allow an overall links weight reduction of about 1.5 Kg comparing with last year design.

The suspension has been designed in metric units and the sizes chosen for the spherical rod ends were M10 except for the lower wishbones outboard ends for which M12 spherical bearings were to be used. That is because the lower pivot points on the uprights are under greatest load.

However, for the rod ends we have been offered sponsorship from Linear Bearings Pty Ltd for industrial grade and imperial units equivalent with the metric ones initially selected. Although some minor modifications had to be made to the uprights pivot points due to small differences in dimensions, the rod ends offered were suitable for our application in terms of strength and cost. Therefore the offer has been gratefully accepted.

The spherical rod ends and spherical bearings specifications are as follows:

Type	Bore and thread UNF	Head diameter (inch)	Thread length (inch)
INDUSTRIAL ROD ENDS			
AM6GP (steel on steel)	3/8 x 3/8	0.406	1.250
AML6GP (steel on steel)	3/8 x 3/8	0.406	1.250
STAINLESS STEEL SPHERICAL PLAIN BEARINGS			
Type	Bore diameter	Outside diameter	Ball width
ABWT8	1/2	1.000	.625

6.6.3.2 Design concept

Once the track width was known, the lengths between pivot points could be calculated. Thus, taking in consideration distances between pivot points on the chassis and the dimensions of the rod ends as well as the geometric configuration of each component, the exact lengths of the wishbones was establish.

Front suspension

For the front suspension the upper wishbone design consists of two members pin jointed for easy adjustment as seen in Figure 35. The links have a boss with a threaded hole to match the spherical rod ends welded at each end.

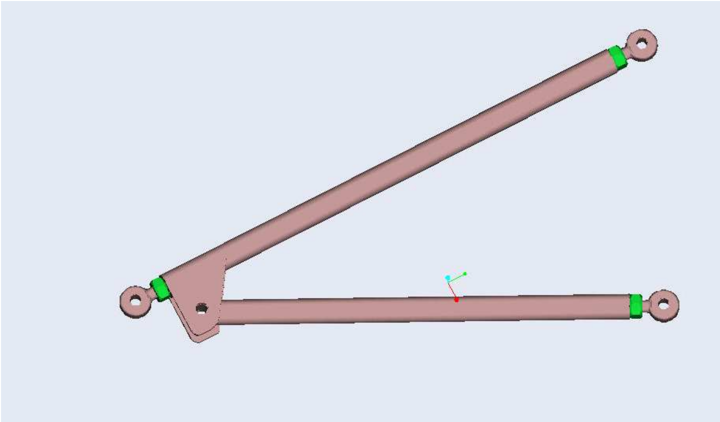


Figure 35 Upper wishbone - front suspension

The lower wishbone consists also of two links and as I mention before has spherical bearings at the outboard ends. For this spherical bearing a casing had to be designed. The links are welded on to this casing therefore the component is rigid with no possibility of adjustment. On the casing are also welded brackets for the suspension arm pivot point. The design is illustrated in Figure 36.

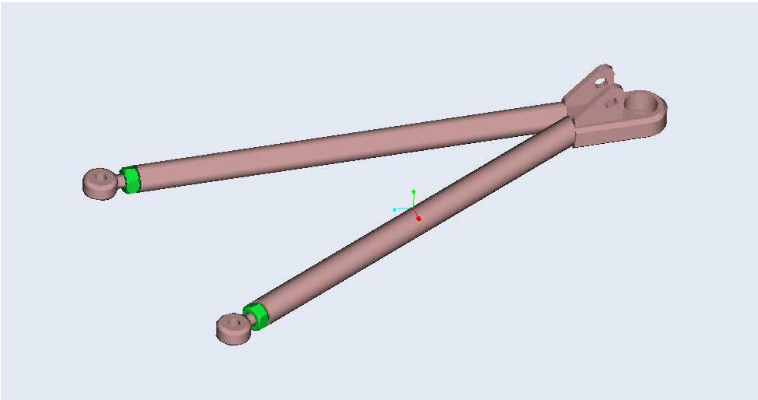


Figure 36 Lower wishbone - front suspension

Rear suspension

The rear suspension, have to accommodate a toe control rod therefore provision for the outboard pivot point was included in the rear upright design. For this reason the design of the upper rear wishbone is different than for the upper front as can be seen in Figure 37.

However, the lower rear wishbone is similar to the lower front one as seen in Figure 36.

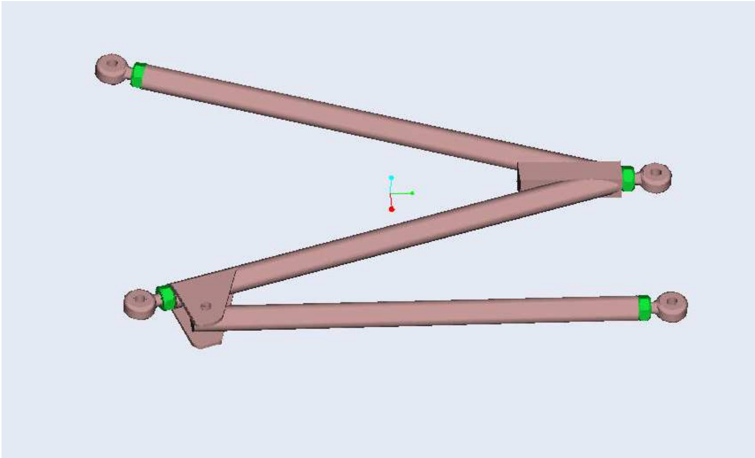


Figure 37 Upper wishbone - rear suspension

Suspension arm

The suspension arm is a single rod with two spherical rod ends fitted in a welded boss.
At this stage, the position of the shock/spring assembly and rocker arm is known with approximation, hence the exact dimension of this component can't be yet calculated. Once the spring/shock assembly is mounted on the chassis and all other suspension components are in place, an exact measure between the rocker arm pivot point and the outboard pivot point will establish the length of this component.

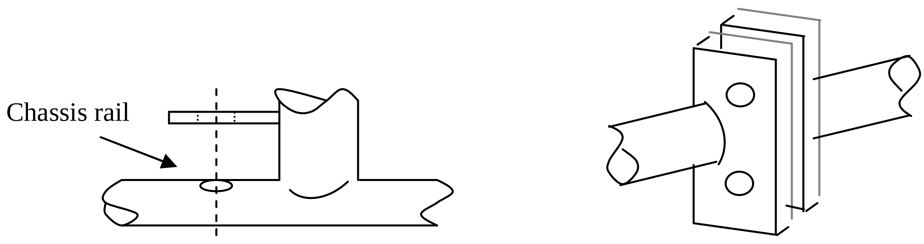
6.6.4 Mounting points

A decision had to be made if the brackets on the last year chassis are to be kept or removed.

Because last year design had compromise on weight to guarantee the reliability, all the brackets were robust and heavy. Following discussions with the team, it was decided that is worth the trouble to remove and replace them with lighter ones and better design.

It was suggested that for the lower mounting points to make use of the chassis bottom rail as seen in Figure 38. This idea has been considered in the design since it has the benefit of reducing the amount of brackets.

For the upper mounting points the brackets are to be welded in a vertical plane to allow for more movement freedom of the control arms. The design will also allow for two mounting positions as seen in Figure 38 for adjustment of the inclination of upper wishbone relative to the lower one.



Lower mounting points Upper mounting points

Figure 38 Mounting points

6.6.5 Springs, shock absorbers and anti-roll bars

6.6.5.1 Position of the spring and shock absorbers

It is a common practice to have the springs mounted over the shocks because it minimises space requirements. For the purpose of this project, this will be regarded as spring/shock assembly.

The chassis roll is restricted by the compression of the springs hence the physical placement of the suspension springs determines how much roll resistance they will offer.

Options for the location of spring/shock assembly

For selection of the spring/shock assembly location there are multiple options depending on the application and the packaging constrains. However there are two main possibilities: to mount the spring/shock assembly inboard (within the chassis) or outboard (outside the chassis).

For the inboard mounting, the assembly is at some distance for the wheel centre line and that requires a link and a rocker arm. The link, called suspension arm, can be either a pull rod if will extend the spring while the wheel goes over a bump or the opposite, a push rod if the spring will be compressed.

Another possibility is outboard mounting when the upper spring/shock assembly pivot is attached to the main chassis structure and the lower pivot is attached to either the upright or the lower wishbone.

Position of the spring/shock assembly

Outboard mounting for front spring/shock assembly is not a viable option since between the upper and lower wishbones is going to be the steering tie rod. Hence, the only choice was inboard mounting of the assembly that will be actuated by a push rod.

After a thorough examination of the last year chassis and all the components assembled on it, and discussion with the team, the conclusion drown was that at the front between the upper and lower chassis rails is the most appropriate location for the spring/shock assembly as seen in Figure 39. While the location will not permit for a vertical position of the shocks, it will allow for more room inside the front chassis to better position of the steering and breaking components and make more space for the driver legs.

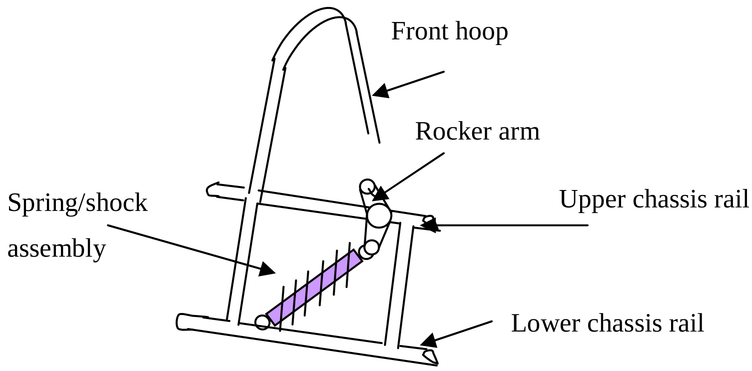


Figure 39 Spring/shock position

At the rear again an outboard mounting position of the spring/shock assembly is not an option since the space it is been taken by the rear axle CV joints. Hence the rear spring/shock assembly will be also attached between the upper and lower chassis rails.

6.6.5.2 Spring rates and wheel rates

The relationship between wheel rate and spring rate is a function of motion ratio between wheel travel and spring axis travel and the formula is (Smith1978):

$$WR = \frac{SR}{MR^2}$$

where: WR= wheel rate
SR= spring rate
MR= motion ratio

For the given wheel travel of 50.8mm (2 inches) we can calculate the spring travel for a given motion ratio.

$$MR = \frac{WT}{ST} \Rightarrow ST = \frac{WT}{MR}$$

where: WT= wheel travel
ST= spring travel
MR= motion ratio

MR	WT(mm)	ST(mm)	WT(mm)	ST(mm)
1.3	25.4	19.53846	50.8	39.07692
1.4	25.4	18.14286	50.8	36.28571
1.5	25.4	16.93333	50.8	33.86667
1.6	25.4	15.875	50.8	31.75
1.7	25.4	14.94118	50.8	29.88235
1.8	25.4	14.11111	50.8	28.22222
1.9	25.4	13.36842	50.8	26.73684
2	25.4	12.7	50.8	25.4

Table 1 Motion ratio

Function of the free length of the spring and how much it is permitted to travel we can choose the motion ratio. As seen in the Table 1 as the motion ratio increases we have less spring travel.

The springs to be used have the following characteristics:

	Front suspension	Rear suspension
Free length (mm)	165	125
Coil diameter (mm)		
Spring rate (lb/inch)	400	450
Spring rate (N/mm)		80.4

Table 2 Springs characteristics

The springs can accommodate a large amount of trave therefore a small motion ratio can be selected.

Then, the wheel rates can be calculated as follows:

REAR					FRONT			
SR	80.4				SR	71.4		
MR	1.3	1.4	1.5		MR	1.3	1.4	1.5
WR=SR/MR^2	47.57	41.02	35.73		WR=SR/MR^2	42.24852	36.42857	31.73333

Table 3 Wheel rates

Optimum wheel rates vary with gross vehicle weight, power to weight ratio, tire width, aerodynamic down force generation and track characteristics (Smith 1978).

6.6.5.3 Shock absorbers

In selecting shock absorbers the length and mounting of the shocks must have provision for adequate suspension travel. Hence, the shock shaft travel must be more than 50.8 mm that is the maximum wheel travel. It is also better to choose shocks with adjustable bump and rebound for a good damping control.

In consultation with the team the decision was made to buy a new pair of shock absorbers for the rear suspension and for the front suspension to use a pair of shocks from last year.

The characteristics of the shocks are as shown in Table 4.

	Front	Rear
Type	Single piston double acting-oil filled	Oil filled – gas reservoir
Length eye to eye(mm)	265 mm	210 mm
Adjustability	Rebound only	Both rebound/bump

Table 4 Shock absorbers characteristics

6.6.5.4 Anti roll bars

While the suspension springs determine the vehicle’s ride rate and roll resistance, the anti roll bar will add torsional resistance to the roll resistance of the springs to determine the total roll resistance of the vehicle. Hence, the anti roll bars are used to limit body roll of a vehicle.

A thorough investigation of anti roll bars is necessary in order to understand and quantify the benefits of it to the suspension system. However, due to time constrains, the anti roll bars are not considered for this project.

Chapter 7

Steering and suspension alignment

7.1 Equipment

The equipment needed to align suspension is simple and consists of:

machinist’s rule;
measuring tape;
trammel pins;
spirit level with protractor;
carpenter’s level;
floor shims of 500 mm square and 3 mm thickness;
two accurate scales;
a piece of straight edge rigid tubing long enough to extend across the tops of both front and rear wheels;
a ball of strong string;
custom made simple devices to measure the caster and camber.

7.2 Procedures for measuring and adjusting suspension

7.2.1 Static alignment

The firs step is static alignment and it is performed to ensure that the track, wheelbase, camber, castor and kingpin inclination are equal in both sides. This can be done by taking all the suspension links out of the car and assemble them to the designed dimensions making sure that the lengths of the components on the left side are exactly the same with the ones on the right side of the car. The exact lengths of the links can be easily determined using the trammel pins and a measuring tape. Also make sure that the chassis pivot point locations are at the same distance to the vehicle centreline and same longitudinal distance to the bulkhead.

7.2.2 Establishing a level surface

To establish a level surface the tyres must be placed on shims after checking that they are the same diameter left and right. Then a rigid straight edge must be placed across the tops of each pair of tyres normal to the vehicle's centre line. With a carpenter's level placed on the straight edge the shims can now be added or removed until the level is zero.

7.2.3 Adjusting the springs

To achieve equal ride height on the right and left side of the car, the distance between upper and lower spring attachments must be equal in both sides. Therefore, measure the distance from the centre of the lower mounting pivot point to the lock nut and adjust the lock nut on the shock absorbers such that this distance becomes equal on both sides. This adjustment works providing that the left and right springs are heaving the same spring loaded height.

7.2.4 Measuring and adjusting the ride height

The adjustment of the ride height must be done with half full fuel tank and the driver in.
The measurements must be taken to a string stretched across the shims, that is the established level area and not to the ground. Adjust again the left and right lock nuts on the shocks in equal increments until reach the desired ride height.

7.2.5 Measuring and adjusting castor

For castor measurement a simple device as in Figure 40 can be made that is to be placed against the upright pivot points or any machined surface that will give you a surface normal to the pivot point axes. A spirit level laid across this surface will give a direct measurement of the caster.

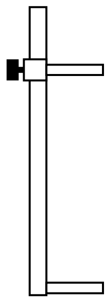


Figure 40 Castor measuring device

The device can be made out of RHS 30x30 with one pin welded at the bottom and another pin the same length welded on a sleeve that is fixed in position by a screw.

It is recommended to adjust the castor only to get it even in both sides of the car. Further adjustment will affect the camber and toe in of the wheel.
The camber can be adjusted on the upper wishbone by lengthening one arm and shortening the other.

7.2.6 Measuring and adjusting camber

Same device made to measure the caster and same procedure can be used to measure caster.
The device can be placed against the wheel making sure that it is not against a bent portion of the wheel rim. The spirit level with protractor placed on the device will give the camber angle.

Adjustment of camber is it done by varying the lengths of the upper and lower wishbones relative to each other.

7.2.7 Measuring and adjusting toe

To measure the toe of the wheel it is necessary to construct a rectangle with sides parallel and equidistant to the centre line. Therefore the reference centre line must be first established.

To find the centre line, carefully measure between front lower wishbones pivot points on the chassis and mark the centre on the lower cross member. Repeat the procedure at the rear. The centre line is then obtained by extending a tight string beneath the car exactly under the two marked points.

To set up the lines parallel to the reference centre line a pair of square tube (like arms) must be clamped on the front and rear chassis. The tube will have an adjustable sleeve on which a pin is welded as seen in Figure 41. Simply tie the string on the pin and stretch it over to the other arm’s pin.

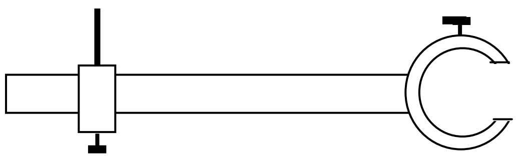


Figure 41 Arm used to measure toe

The lateral distance from the centre line should be great enough to allow for insertion of a camber gauge without disturbing the set up.

To determine the toe, use a machinist’s rule to measure from the reference line to a machined surface on the wheel rim as close to the outside diameter of the rim as possible.
The actual measurement of toe is illustrated in Figure 42.

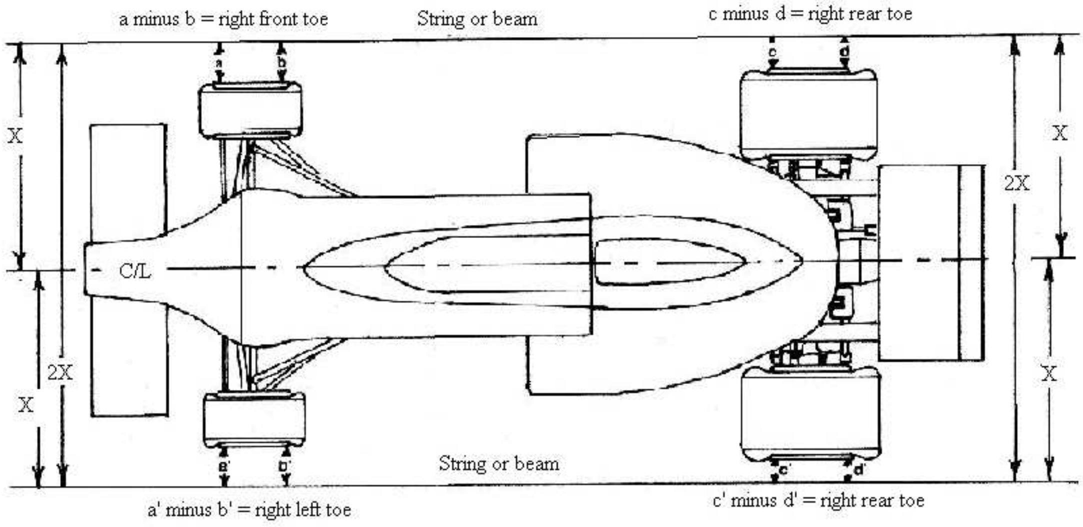


Figure 42 Toe adjustment

To adjust toe modify the lengths of the tie rods making sure that same number of rod end turn is applied in both sides.

7.2.8 Bump steer

An easy way to measure the bump steer is to use the reference line constructed before, swing the wheels and measure and record the change in toe against the reference strings.

After the change in toe for a given vertical wheel travel is determined it is necessary to reduce that change to preferably zero. This can be again achieved by changing the length of the tie rod but the static settings will be affected. Hence the toe change is better adjusted by changing the relative height of the inboard and outboard tie rod ends.

7.2.9 Recording data

After adjusting bump steer it is good to check the static alignment again and record all the distances, link lengths and set up parameters in the car's log book.

Convert a turn of rod end in a linear measure so that it is known by how much length a link is shortened or lengthened for each turn of the rod end. While making adjustments, record the number of turns and effect of it on camber, caster toe-in, toe out so that one can come easily back to initial settings. Also record the installed height of the spring perch and the effect of one turn of spring perch adjustment for each side of the car.

Chapter 8

Conclusions

8.1 Achievement of objectives

The project have been both challenging and rewarding. It required the knowledge gathered in many of the subjects pertaining to engineering studies and offered valuable practice in team working environment as well as good professional practice.

Although the topic covered two large areas of racing car design, all the tasks set up for this project have been completed in the available time.

At present, all the components have been manufactured, and the implementation of the two systems in the overall design will be soon finalised. The design complies with the SAE-A rules.

A complete test of all the components is planned when the vehicle's construction is completed.

Recommendations

An estimated overall weight reduction of 6 kg for both systems has been achieved. However, as cost constrains permits, it is recommended that lighter materials and methods of fabrication to be used in order to further minimise the weight of the component.

It is recommended to investigate and quantify the benefits of anti roll bars in suspension design and how these will improve the handling and manoeuvrability characteristics of the vehicle.

Although both systems have adjustments included in design, a even higher degree of adjustability can be achieved e.g. by use of spacers (shims) for the suspension pivot points to easily change geometry parameters like roll centre, camber, toe.

In modelling the suspension, the use of a computer software will be a great benefit since will allow a more rapid and accurate study of suspension design and settings.

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2005 Formula SAE Rules

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Appendix A

Project Specification

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR: **Cristina Elena POPA**
TOPIC: Steering System and Suspension Design
SUPERVISOR: Chris Snook

PROJECT AIM: This project aims to design and build the steering system and suspension for USQ SAE-A racer car

PROGRAMME: **Issue A, 09 March 2005-03-08**

1. Research Formula SAE-A rules to determine restrictions and design requirements.
2. Research information on currently used automotive steering systems and suspension designs; evaluate feasible alternatives.
3. Critically analyse designs used by other teams
4. Determine an appropriate steering arrangement for the car
5. Determine an appropriate suspension system for the car
6. Detailed design of components and assemblies and FEA for critical components
7. Liase with team members, Sponsors and Faculty Workshop Staff in manufacture and assembly
8. Develop systems to evaluate the performance of steering system and suspension
9. Document design and cost analysis of steering system and suspension
10. Write dissertation

AGREED: E Popa (Student) ________ (Supervisor)

(dated) 16/3/05

Appendix B

Finite Element Report For Hub Axle

Job name: HUB AXLE ANALYSIS

Job number: 1-2005 Steering System and Suspension Project

Analyst: Cristina Popa (w0018250)

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1. Problem description

Hub axle is one of the components of the front stub/axle assembly of the USQ 2005 SAE racer car.
The hub axle component is fixed on the wheel with four nuts and bolts. Therefore, the loads on the tires are transferred to the hub through the bolts. The shaft part carries two bearings that are mounted in the upright bearing housing.

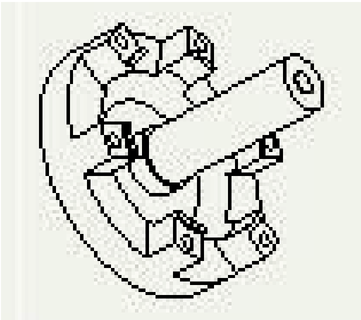


Figure 43 Front Hub axle

2. General approach

2.1 Analysis type: 3D Structural Analysis

2.2 Package used: ANSYS 7 on Solaris

2.3 Element type used

The element type used was Solid – Tet 10 node 187

2.4 Material properties

The hub was fabricated from mild steel with following material properties:

Modulus of elasticity	200 GPa
Tensile strength	455 MPa
Yield strength	250 MPa
Poisson’s ratio	0.3

2.5 Basic model

A solid model of the component was created in ProEnginer and imported in ANSYS for analysis.
Units used for analysis are N, mm, MPa.

2.6 Meshing the model

For meshing the model I have used the free mesh with a tetrahedral shape as can be seen in Figure 44

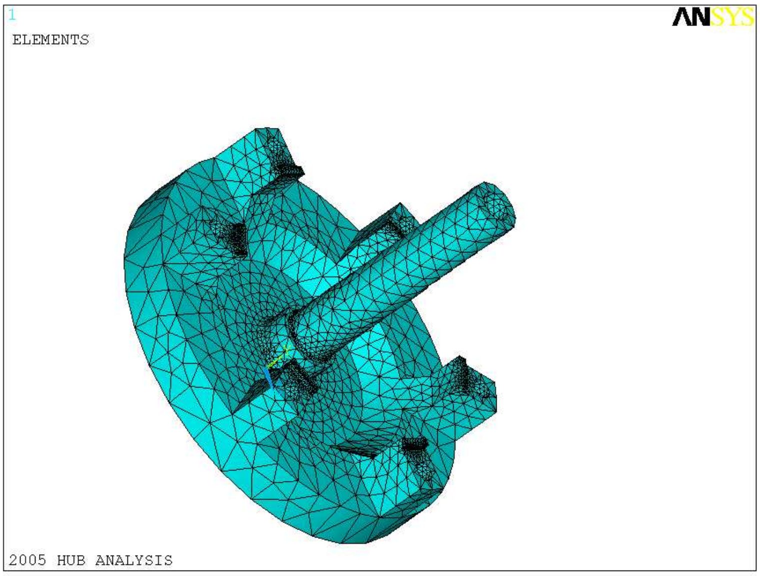


Figure 44 Hub mesh

2.7 Constrains and loads

The model was constrained with all DOF’s on 66 nodes on the surface of the shaft were the location of the middle section of the bearings was assumed to be.

The maximum accelerations are (Fenton 1980):

- ± 3g vertical (hitting a bump);
- ± 1g lateral (cornering);
- ± 1g longitudinal (breaking/accelerating);

Multiplying this accelerations with a factor of safety of 1.5 the resulting design loads for the wheel are:

-4500 N in vertical direction applied at the centre of the wheel;

-1500 N in lateral direction applied at the centre of tyre contact patch;

-1500 N in longitudinal direction applied also at the centre of the tyre contact patch.

The loads are transmitted to the hub through the bolts that holds the hub on the wheel. Therefore the loads were applied on the surface of the hub holes that match with the bolts. The number of selected nodes attached to these surfaces were 1184 hence the applied loads were divided by this number to give the appropriate force per node.

The applied constrains and loads can be seen in Figure 45.

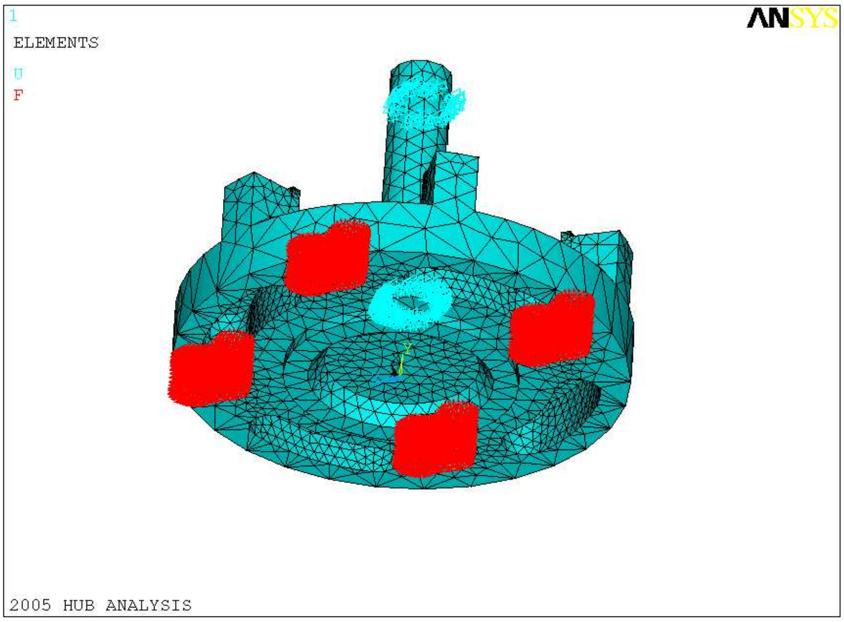


Figure 45 Hub loading

3. Solution

3.1 ANSYS analysis solution

The element solution was obtained by plotting stresses using von Misses theory of failure.

The plot and the result can be seen in Figure 46.

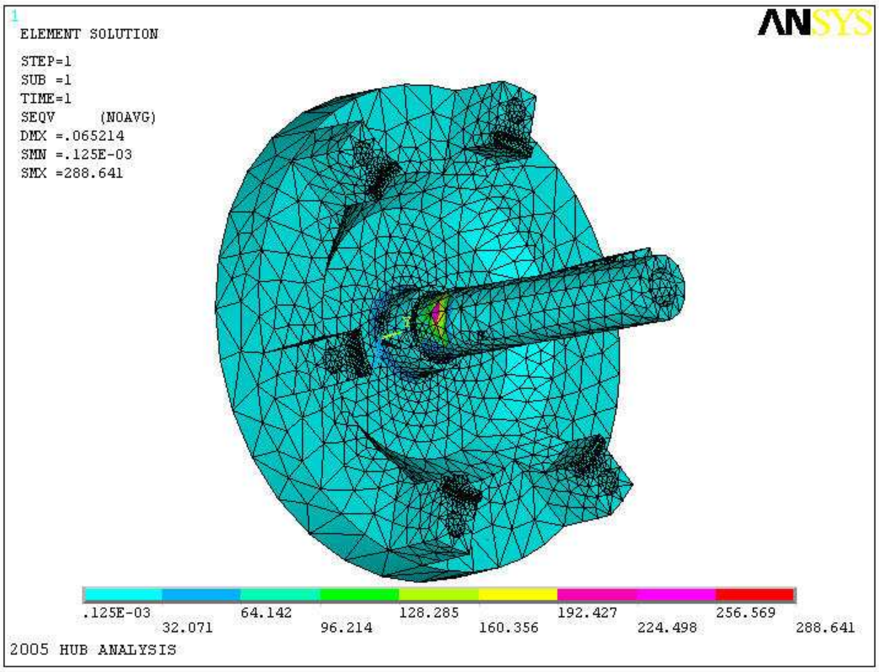
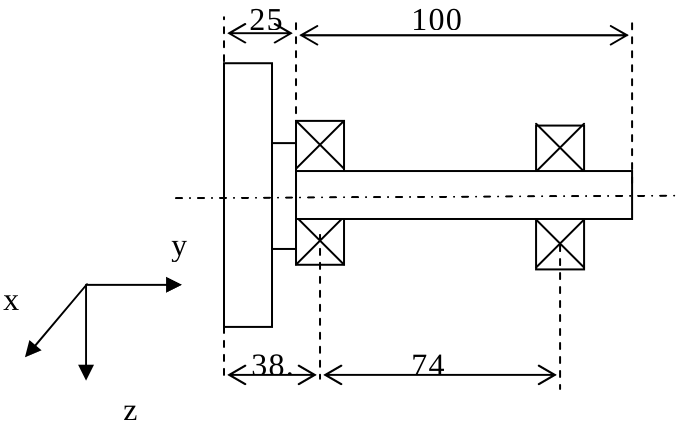


Figure 46 Hub analysis result

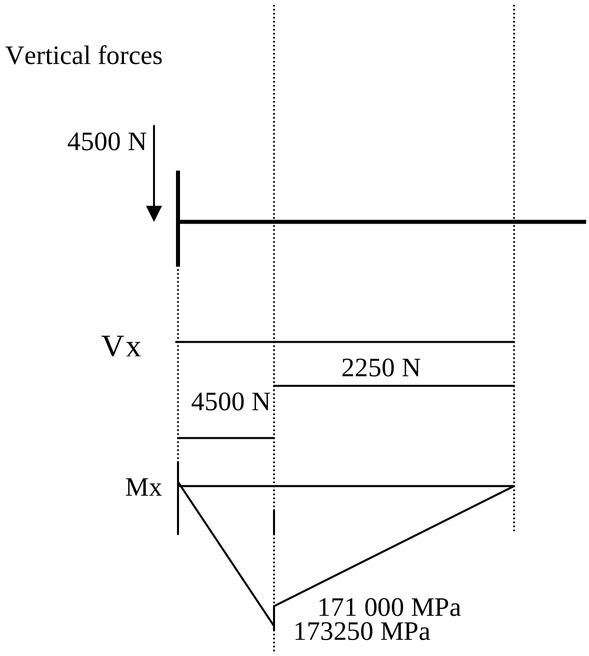
3.2 Analytical solution

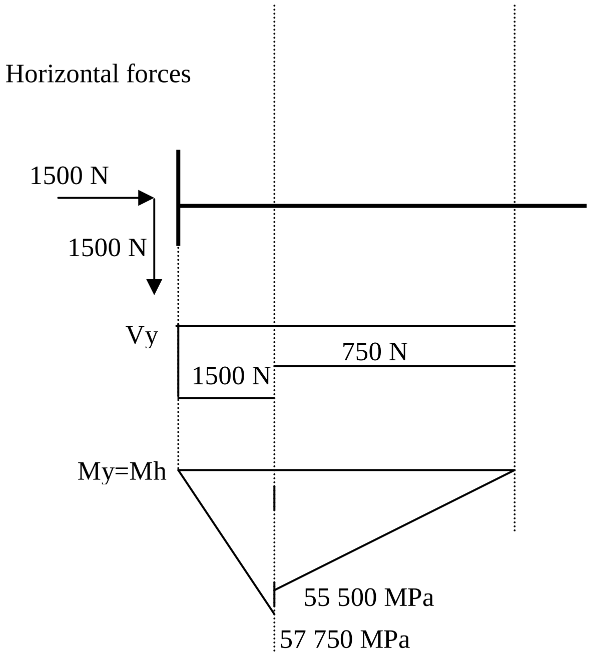


Fx=4500 N

Fy=1500 N

Fz=1500 N





$D = 20mm; c = 10mm$

$$A = \frac{\pi D^2}{4} = \frac{\pi x 20^2}{4} = 314.16mm^2$$

$$I = \frac{\pi D^4}{64} = \frac{\pi x 20^4}{64} = 7853.98mm^4$$

$$\sigma_x = \pm\frac{P}{A} \pm \frac{Myc}{I} \pm \frac{Mzc}{I} = 0 + \frac{57750x10}{7853.98} + 0 = 73.53MPa$$

$$\sigma_z = \pm\frac{P}{A} \pm \frac{Mxc}{I} \pm \frac{Myc}{I} = \frac{1500}{314.16} + \frac{173250x10}{7853.98} + 0 = 225.3634MPa$$

$$\sigma_{\max} = \frac{\sigma_x + \sigma_z}{2} + \sqrt{\frac{(\sigma_x - \sigma_z)^2}{2} + \tau_{xz}{}^2} = \frac{73.53 + 225.3634}{2} + \sqrt{\frac{(73.53 - 225.3634)^2}{2} + 0} =$$

$$= 149.4467 + 107.3624 = 256.8091MPa = \sigma_a$$

$$\sigma_{\min} = \frac{\sigma_x + \sigma_z}{2} - \sqrt{\frac{(\sigma_x - \sigma_z)^2}{2} + \tau_{xz}{}^2} = 149.4467 - 107.3624 = 42.0843MPa = \sigma_b$$

Maximum distortion energy criterion also known as von Misses criterion states that the component is safe as long as:

$$\sigma_a^2 - \sigma_a\sigma_b + \sigma_b^2 < \sigma_y^2$$

$\sigma_a, \sigma_b =$ principal stresses.

Hence,

256.8091^2-256.8091x42.0843+42.0843^2=52914.37094<65500

238,567<250 MPa = σ_y for mild steel

4. Conclusions

The maximum stress occurred around the area where the shaft is protruded out the hub as expected.
The value obtained with ANSYS analysis for maximum stress is 288.641 MPa that is above the yield strength of the material with about 40 MPa.

The conclusion based on von Misses criterion is that the component is not safe. However the loading case considered all maximum loads occurring at the same time while in reality this is not likely to happen. On the other hand, the loading case was simplified by considering the forces acting directly through the bolts of the wheel while this is not the case.

The theoretical value of 239 MPa obtained is below the yield strength of the material but again the loading case was simplified.

5. Recommendations

Another analysis should be carried with a radius included on the shoulder where the maximum stresses occurred, refining the mesh in the area to get more accurate results in the stress concentration zone. The geometry of the component should be also simplified to reduce the solution time and disk space required for the solution.

The loading case should be reconsidered hence, analysis and appropriate calculation should be performed and once again the results compared with the yield strength of the material. I believe that values up to 15 MPa above the yield strength should be acceptable since, like I said before, it is unlikely that all maximum loads to happen on the same time.

If the results will not confirm acceptable maximum stresses values, then the component should be designed with a bigger diameter closed to the shoulder to ensure that failure does not occur.

Appendix C

Finite Element Analysis for Front Upright

Job name: UPRIGHT ANALYSIS

Job number: 2- 2005 Steering System and Suspension Project

Analyst: Cristina Popa (w0018250)

1. Problem description

The upright is a component of front stub/axle assembly of the USQ 2005 SAE racer car. It consists of a bearing housing and two attached protrusions as seen in Figure 47 on which various brackets that holds the suspension components are welded.

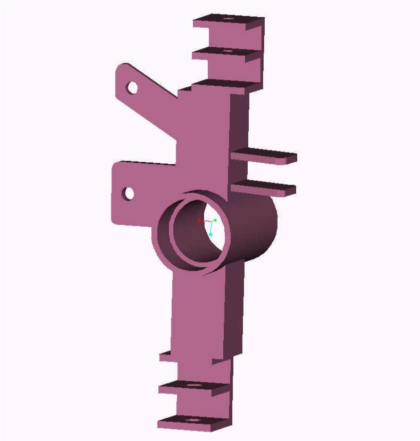


Figure 47 Front upright

2. General approach

2.1 Analysis type

The analysis performed on the component was a 3D Structural Analysis

2.2 Package used

The software use was ANSYS 7 on Solaris

2.3 Element type used

The element type used was Solid 10 node (187) tetrahedral

2.4 Material properties

The upright was fabricated from mild steel having the following material properties:

Modulus of elasticity	200 GPa
Tensile strength	455 MPa
Yield strength	250 MPa
Poisson's ratio	0.3

2.5 Basic model

The upright is designed to support suspension components, steering arm and brake components.

The forces acting on the tyre contact patch as well as the vertical force acting at the centre of the wheel are transferred to the upright pivot points and to the brake mounting points. The force acting on the steering arm is the output steering rack load transmitted through the tie rods.

The approach used was to simplify the geometry to minimise the number of elements used, hence the solution time and disk space required, and perform an analysis for each of the loading cases.

Units used for the analysis are: N, mm, MPa.

3. Analysis of suspension pivot points

3.1 Meshing the model

A simplified model containing only the basic structure and the brackets for suspension pivot points was created in ProEngineer and imported in ANSYS. The model was free meshed using tetrahedral elements as seen in Figure 48

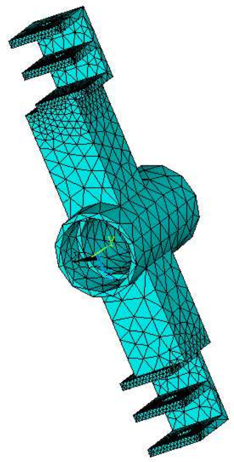


Figure 48 Upright mesh

3.2 Appling constrains and forces

The model was constrained on the surfaces that mate with the bearings. Only 58 nodes were selected on a centre line of each surface. The nodes were constrained for all degrees of freedom.

In order to calculate the forces to be applied on the upper and lower pivot points, first the forces that acts on wheel must be determined and then the moments about the wheel centre.

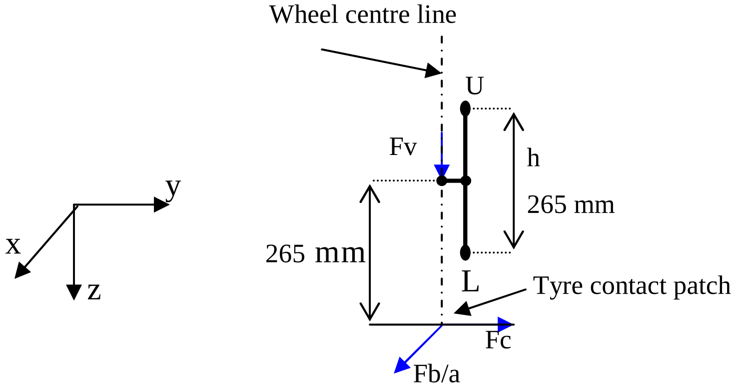


Figure 49 Loads on the upright

The maximum accelerations are (Fenton 1980):

- ± 3g vertical (hitting a bump);
- ± 1g lateral (cornering);
- ± 1g longitudinal (breaking/accelerating);

Multiplying the accelerations with a factor of safety of 1.5 the resulting design loads for the wheel are:

4500 N= F_v in vertical direction applied at the centre of the wheel;
1500 N= F_c in lateral direction applied at the centre of tyre contact patch;
1500 N= $F_{b/a}$ in longitudinal direction applied also at the centre of the tyre contact patch.

Moments about the centre of the wheel are:

$M_{O1}=M_{O2}= F_c \times 265 = F_{b/a} \times 265 = 1500 \times 265 = 397500$ Nmm = 397.5 Nm

Hence, forces at "U" are: $F_{xu} = F_{bu} = M_{O1}/h/2 = 397500/128 = 3105.5$ N

$F_{yu} = F_{cu} = M_{O2}/h/2 = 3105.5$ N

$F_{zu} = F_{vu} = 0$ N

And forces at "L" are: $F_{xl} = F_{bl} = -3105.5$ N

$F_{yl} = F_{cl} = -3105.5$ N

$F_{zl} = F_{vl} = 4500$ N

The area that defines the holes on the upper and lower bracket was selected and the forces were applied on the nodes attached to these areas.

3.3 Solution

The element solution using von Misses criterion is illustrated in Figure 50. and Figure 51

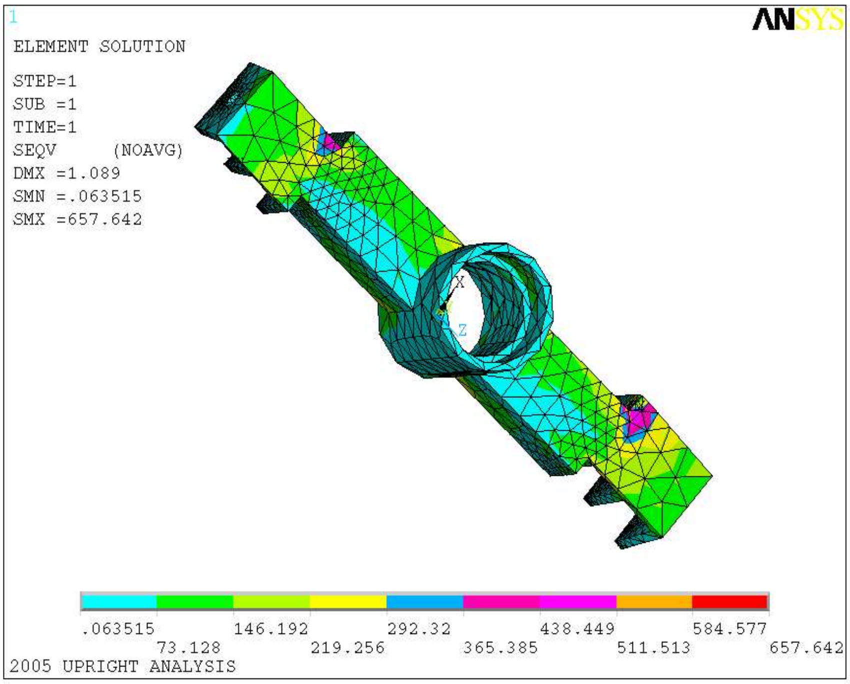


Figure 50 Element solution for pivot points loading case-rear view

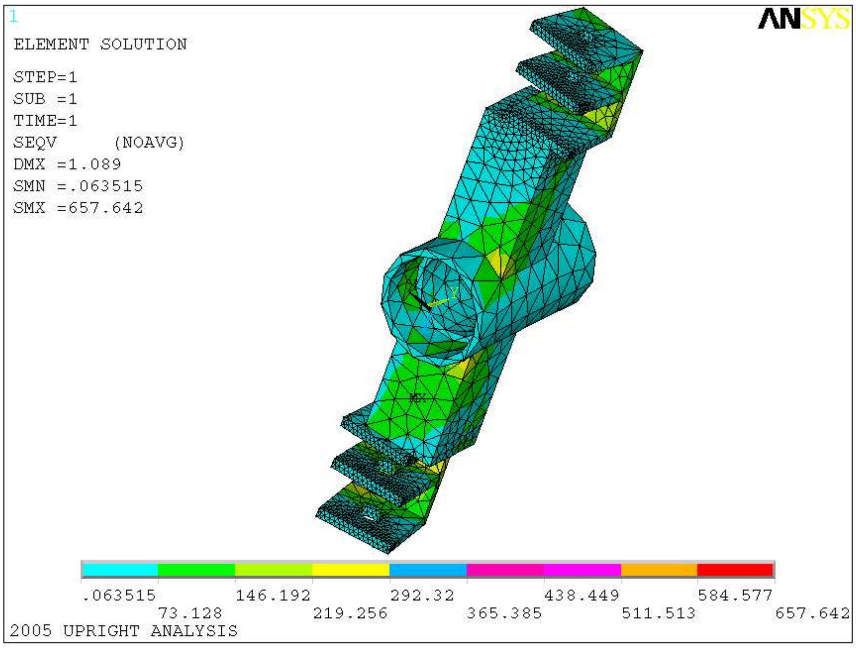


Figure 51 Element solution for pivot points loading case - front view

3.4 Conclusions

The element solution obtained for maximum stresses it is 657.642 MPa. As seen in the above pictures, the stresses on the component are in general below the yield limit of the material. However, on the corners, where the brackets are attached to the main structure, high stress concentrations occurs which means that this is a critical area.

One reason for such high stresses is that the model does not take into account the fact that the welds will add material such that there are not straight edges and sharp corners. Moreover, the welds will add strength to the structure.

Another reason has to do with the loads applied that are the maximum of all loads at the same time. In reality this is unlikely to happen.

3.5 Recommendations

The model can be improved by modelling the welds. For the welds another element type should be assigned with different material properties. Also refining the mesh in critical areas should be considered.

If the result is not within safe limits it means that the welds alone will not fix the problem hence, the component must be reinforced with some extra material. A flat plate with a 3 mm thickness should be welded at the back of the bracket with half of the length on the bracket and half on the upright main structure.

4. Analysis of brake calliper mounting points

4.1 Meshing the model

A simplified model containing only the basic structure and the brackets for the brake calliper mounting points was created in ProEngineer and imported in ANSYS.
The model was free meshed using tetrahedral elements.

4.2 Applying constrains and forces

The model was constrained as for the previous model for all degrees of freedom on the nodes that represent the centre line of the bearing mating surface.
Since there's two mounting points for the brake calliper, an average distance between this points and the centre of the wheel was considered in order to calculate the braking force.

Hence,

$$F = M_{O2}/45 = 397500/45 = 8833.3 \text{ N}$$

The force was applied on the nodes attached to the brackets holes surface. The mesh, constrains and applied force can be seen in Figure 52.

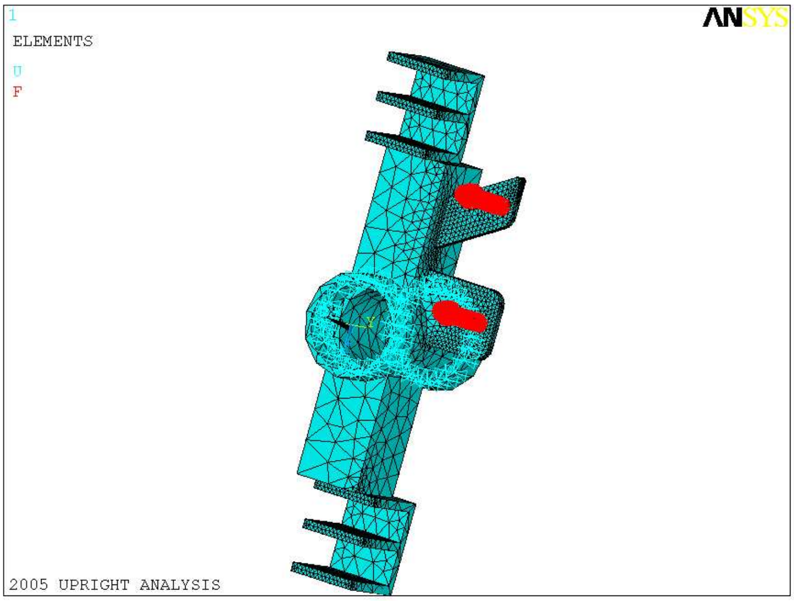


Figure 52 Analysis of brake calliper mounting points- mesh, constrains and loads

4.3 Solution

The result of the element solution plotted for von Misses stresses is 1944 MPa. This high stress occurs on few elements that are at the mating face of the bracket with the upright body, in the closest point to the applied loads seen in Figure 53.

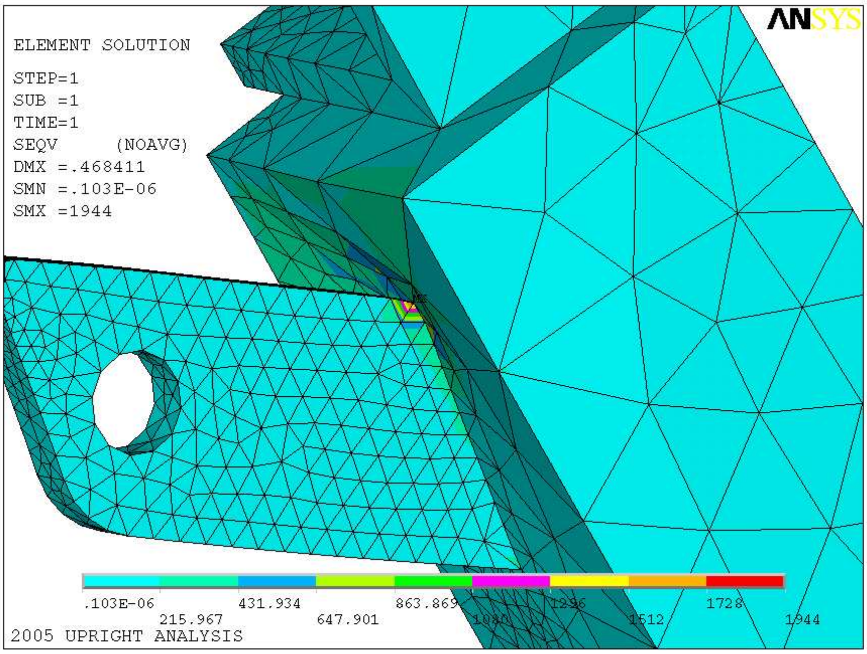


Figure 53 Solution for brake calliper mounting points

4.4 Conclusions

Except for few elements that are around the upper part of the bracket, the body of the upright does not show stresses above 216 MPa. However, a high stress concentration it is sown at the corner where the bracket is attached to the upright. Although it was expected and it appears in just a few elements, the value is quite high.

As for the previous case, the model did not considered the welds of the bracket to the upright. Usually the welds help increase the strength in sharp corners joints by adding material which, many time will have better properties than the material of the components to be welded.

4.5 Recommendations

Another analysis should be performed on a model that includes the welds.

It is also recommended to refine the mesh in critical area as well as on the area on the upright around the bracket.

If the result does not achieve an acceptable decrease in the value of stress then, the component should be redesigned in terms of the geometry of the bracket, thickness of it as well as the thickness of the material of the upper part of the upright.

5. Analysis of steering arm

5.1 Meshing the model

The solid model created in ProEngineer features the main structure and the steering arm brackets.

As before, the model was imported in ANSYS and meshed with a free coarse mesh using tetrahedral elements. The mesh was then refined on the areas of the brackets.

5.2 Applying constrains and lads

The same constrains on the model has been applied as in previous analysis.
The force from the steering shaft is transmitted through the tie rod to steering arm that is attached to the upright. Therefore the load was applied on the nodes attached to the areas of the holes on the steering arm brackets.

The force applied was 880 N as calculated in 5.6 Steering movement ratio, on “y” direction.

5.3 Solution

The plot and the result of the element solution for von Misses stresses is shown in Figure 54.

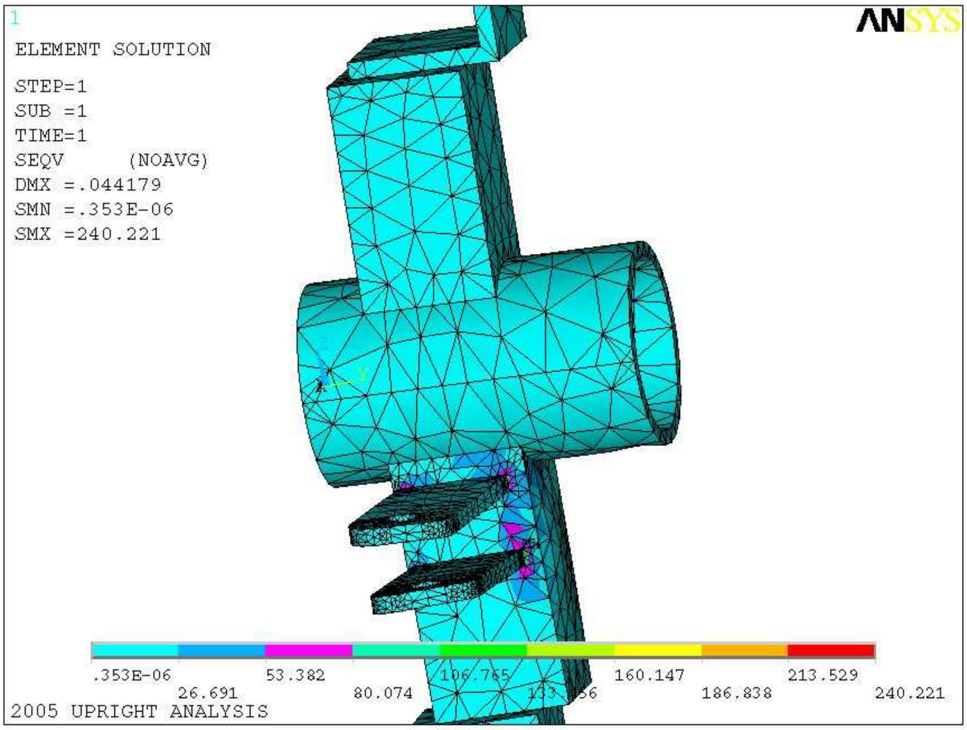


Figure 54 Steering arm analysis solution

5. 4 Conclusions

The maximum stress obtain is 240.221 MPa that is below 250 MPa, the yield value for the material. Hence the brackets on the component are safe.

6. Final Conclusions and recommendations

The simplified model used in all analyses did not included welds that would have reduced the level of stress concentrated in straight and sharp joins.

The results are as follows:

	Analysis	Max. stress (MPa)
1.	Suspension pivot points	657.642
2.	Brake calliper mounting brackets	1944
3.	Steering arm	240.221

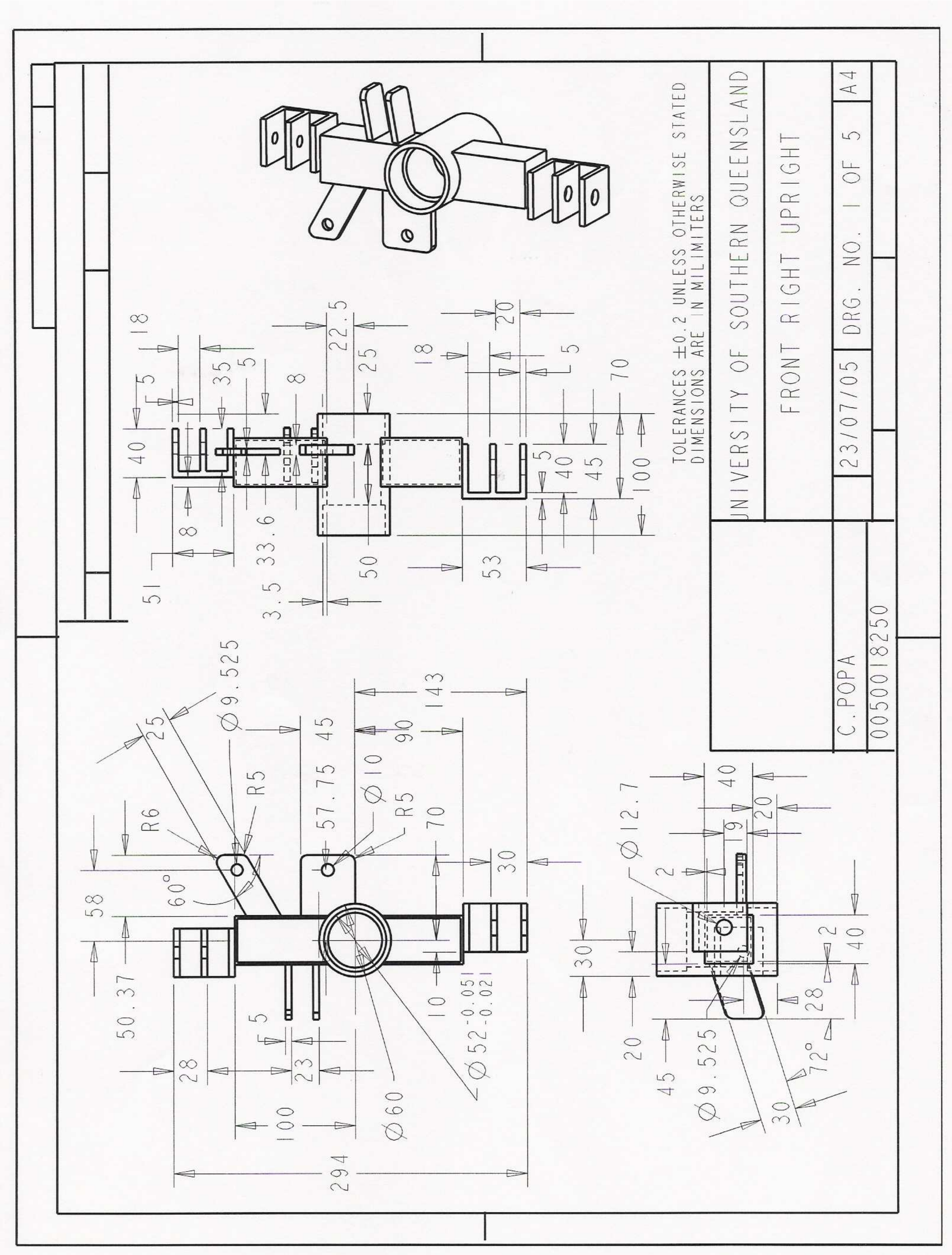
The analysis of the suspension brackets showed stress concentration above the limit due to the geometry of the component that presents sharp corners. Therefore the geometry should be improved by modelling the welds, and another analysis carried out. However, if maximum stresses will exceed the yield strength of the material then more material should be added by welding 3mm flat to reinforce the brackets on the upright.

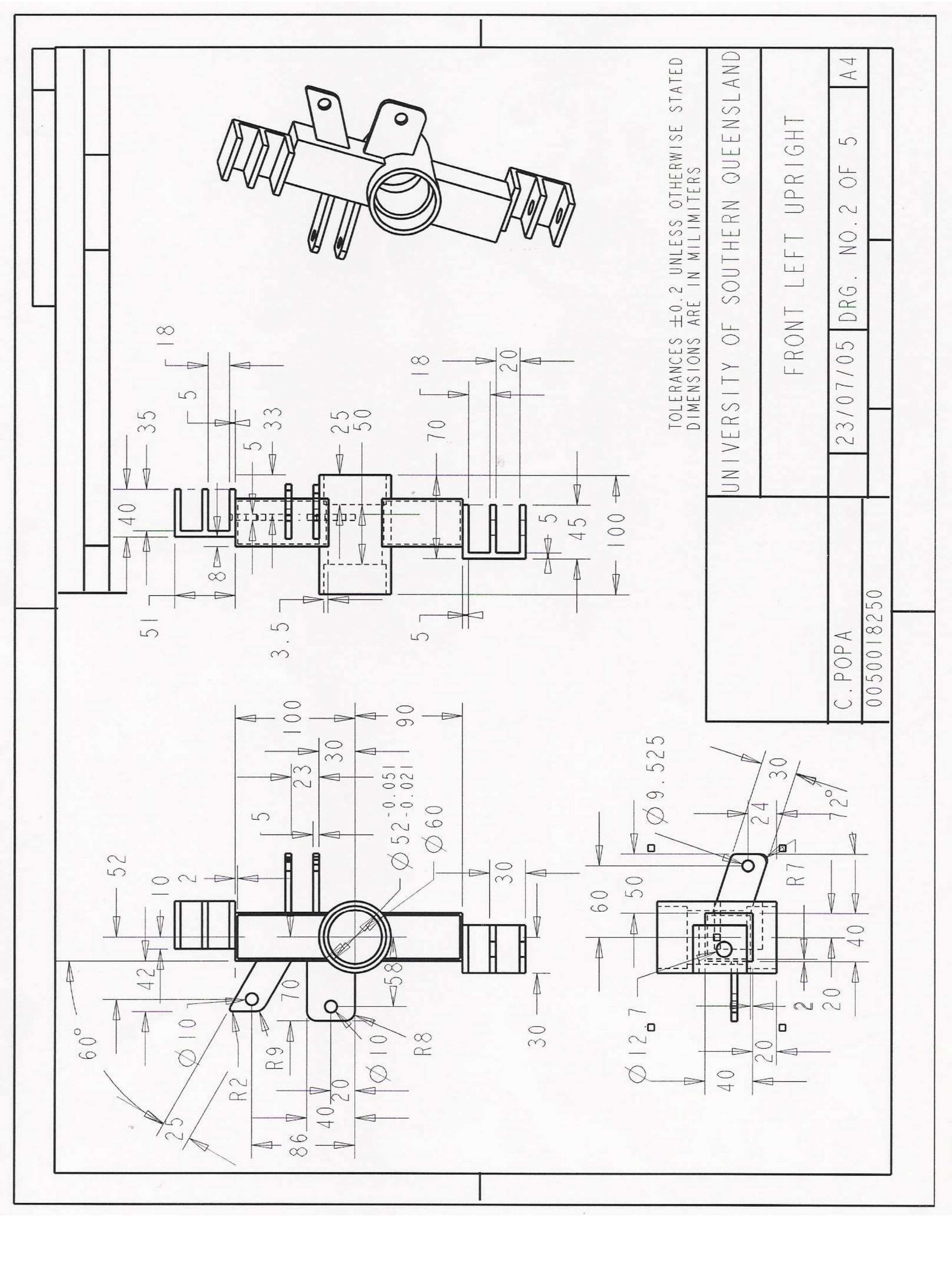
The analysis of the brackets for the brake calliper showed an excessive stress concentration on few elements of the upper bracket. Therefore, the point where these stress concentrations occurred should be reinforced. Consideration should be given to the upper bracket geometry and location that can also be improved. The lower bracket does show a low level of stress therefore the size and thickness of the bracket can be reduced.

The analysis results for the steering arm showed a lower value of stress than the yield strength of the material hence this part of the component is safe under the applied load.

Appendix D

Production Drawings


TOLERANCES ±0.2 UNLESS OTHERWISE STATED
DIMENSIONS ARE IN MILIMITERS
UNIVERSITY OF SOUTHERN QUEENSLAND
FRONT RIGHT UPRIGHT
23/07/05
DRG. NO. 1 OF 5
A4
C.POPA
0050018250


TOLERANCES ±0.2 UNLESS OTHERWISE STATED
DIMENSIONS ARE IN MILIMITERS
UNIVERSITY OF SOUTHERN QUEENSLAND
FRONT LEFT UPRIGHT
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DRG. NO. 2 OF 5
A4
C. POPA
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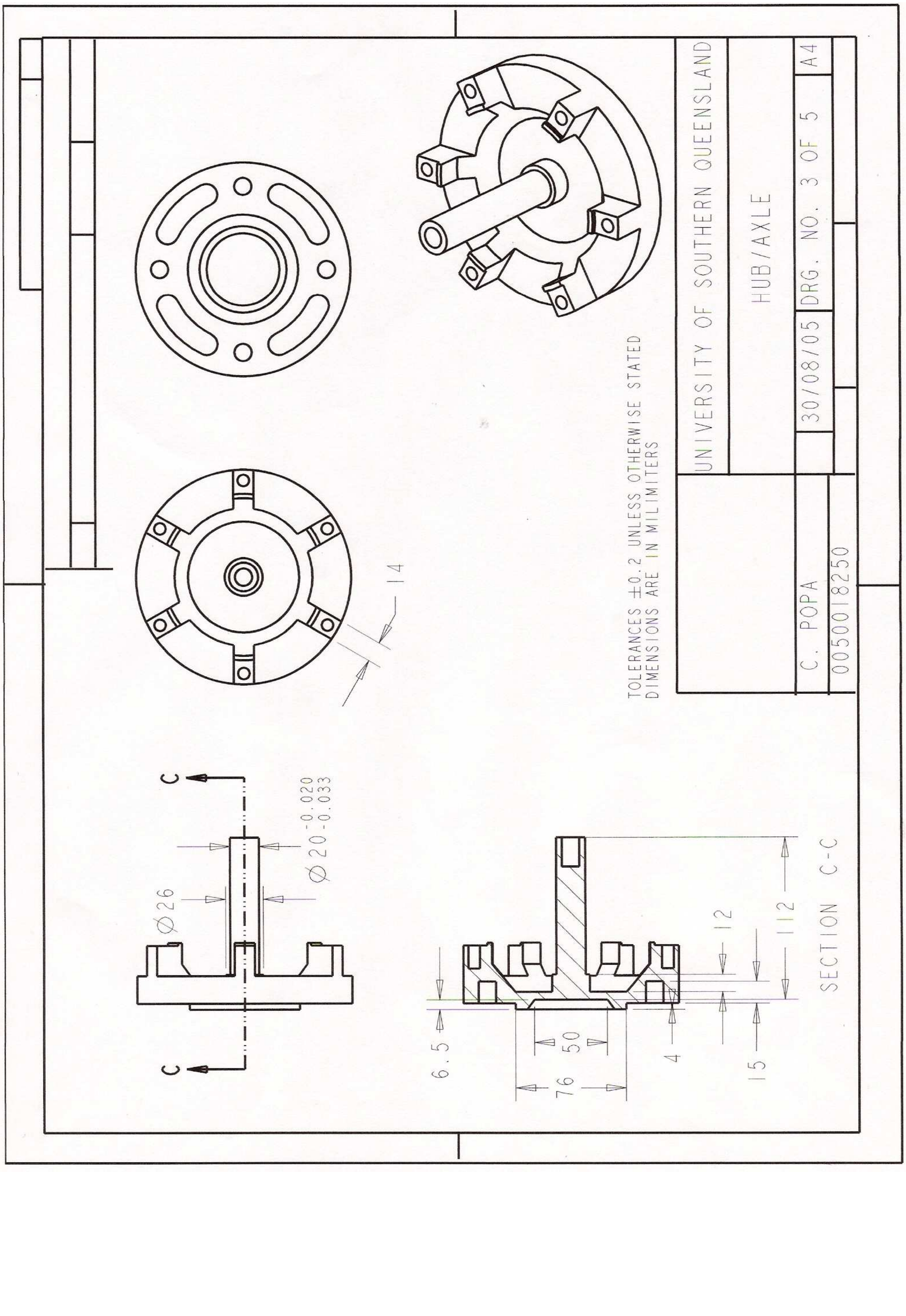
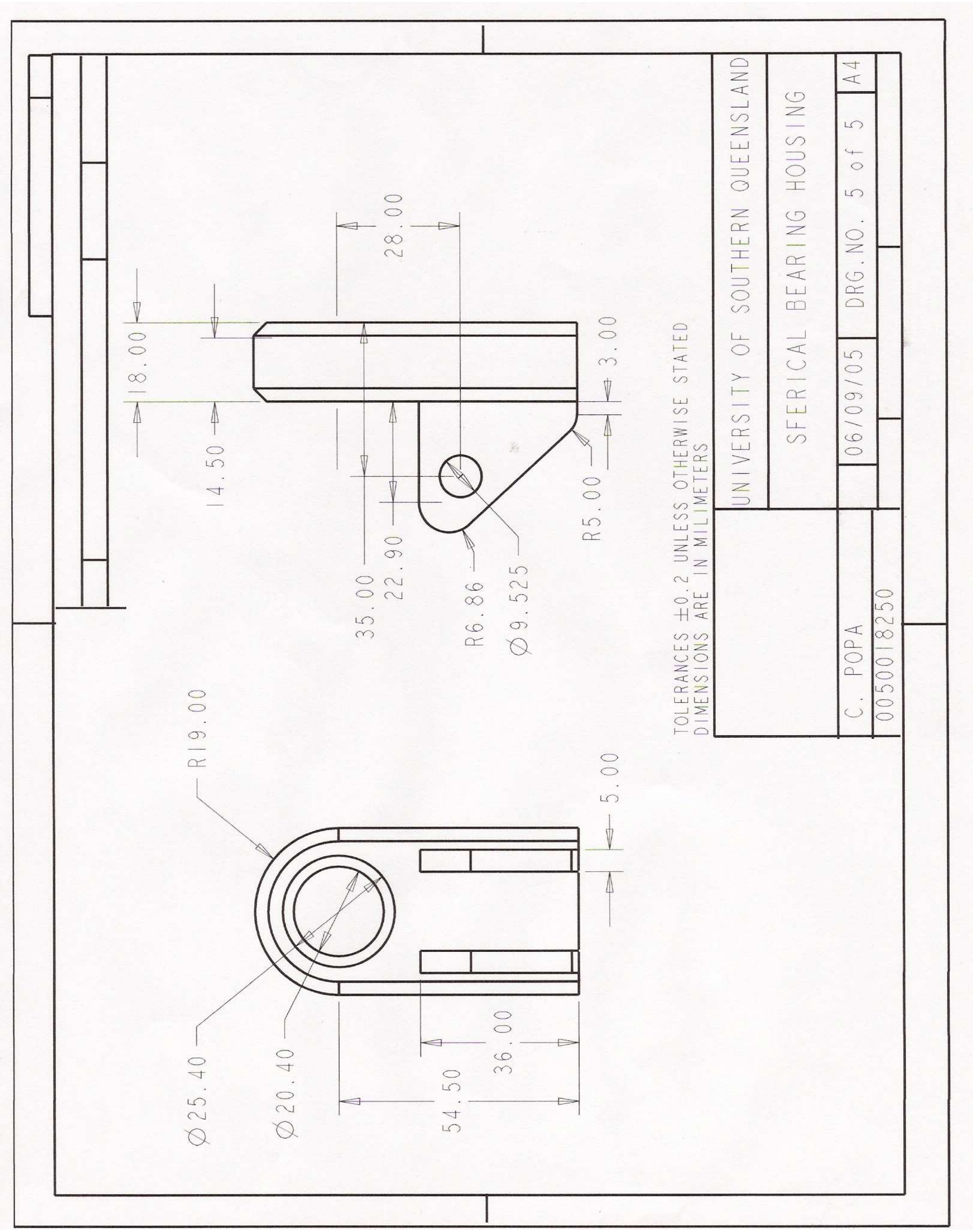

UNIVERSITY OF SOUTHERN QUEENSLAND
HUB/AXLE
30/08/05
DRG. NO. 3 OF 5
A4
C. POPA
0050018250
TOLERANCES ±0.2 UNLESS OTHERWISE STATED
DIMENSIONS ARE IN MILIMITERS
14
C
C
Ø26
Ø20 -0.020 -0.033
SECTION C-C
112
12
15
4
50
76
6.5


UNIVERSITY OF SOUTHERN QUEENSLAND
SFERICAL BEARING HOUSING
06/09/05
DRG.NO. 5 of 5
A4
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TOLERANCES ±0.2 UNLESS OTHERWISE STATED
DIMENSIONS ARE IN MILIMETERS
28.00
18.00
14.50
3.00
35.00
22.90
R6.86
Ø9.525
R5.00
R19.00
5.00
Ø25.40
Ø20.40
54.50
36.00

Appendix E

Photographs

Figure 1E - Front Right Upright

Figure 2E Front Right Suspension Assembly

Figure 3E Rear Suspension Assembly

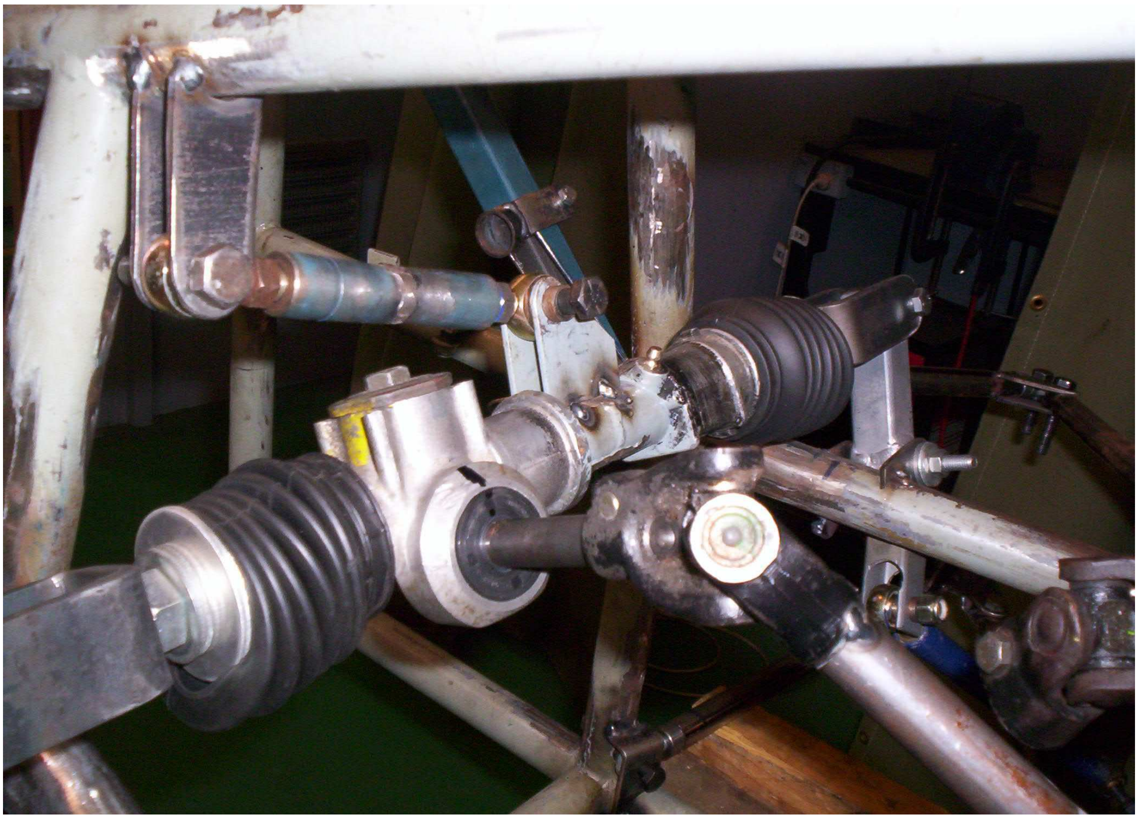

Figure 4E Steering Box Assembly on the Chassis

Figure 5E Mounting Position of the Rear Shock Absorber on the Chassis

Appendix F

Roll Centre and Camber Change Models

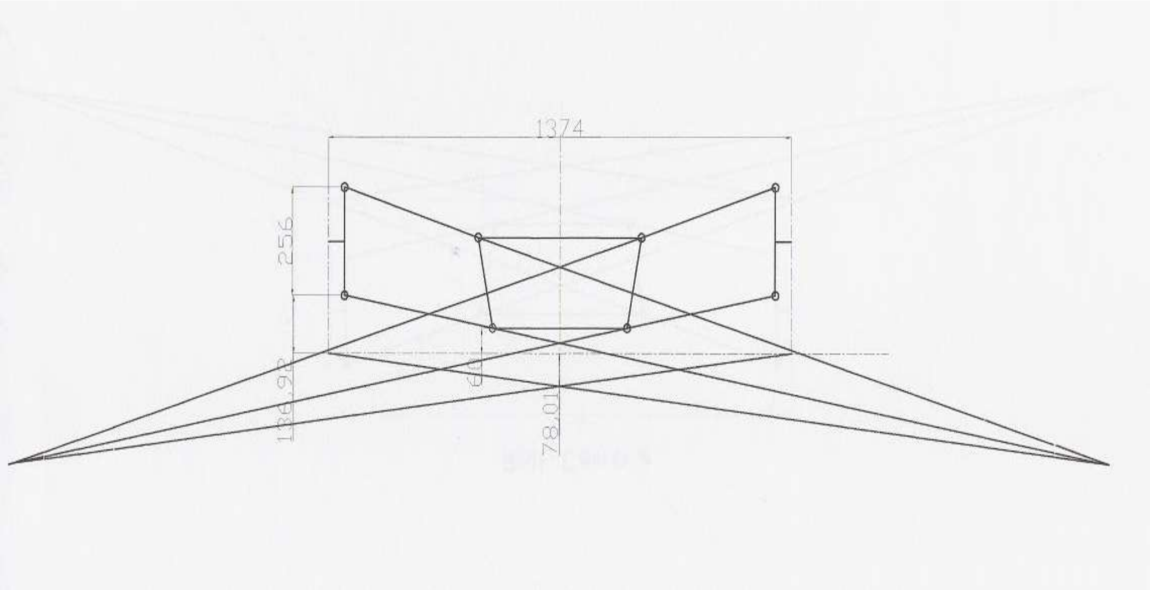

Figure F 1Position of the roll centre

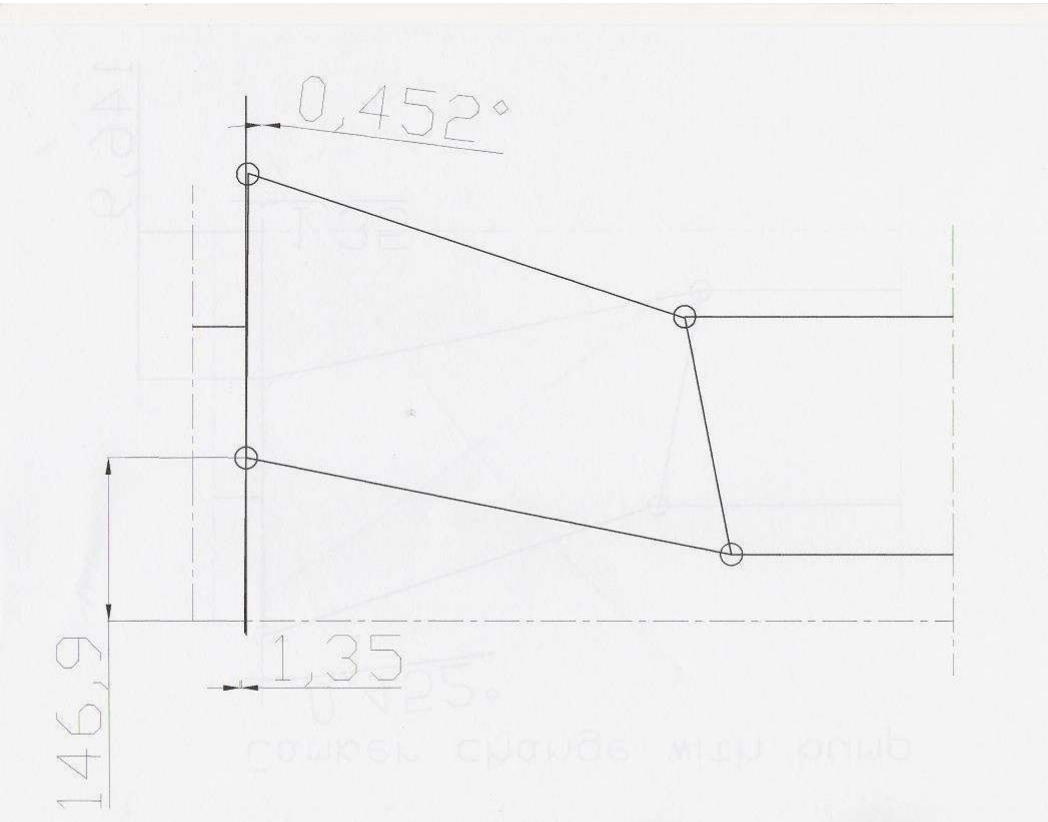

Figure F 2 Camber change with 10 mm bump

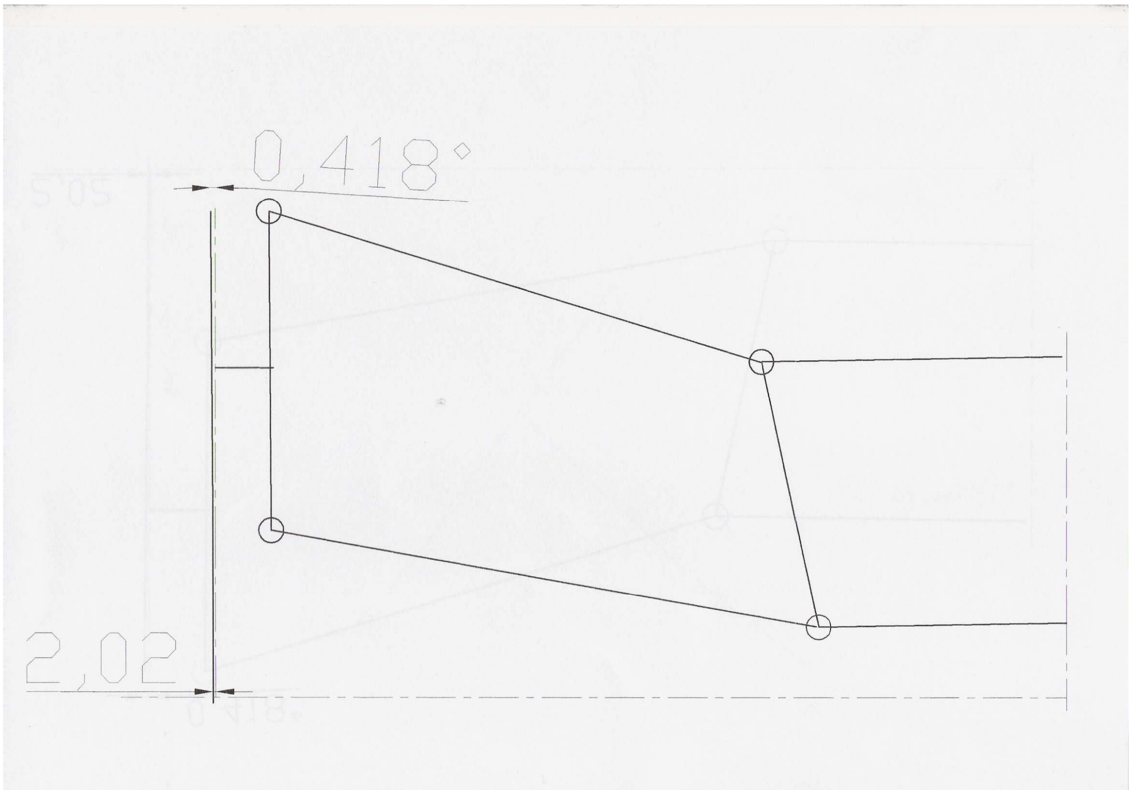

Figure F 3 Camber change with 1 degree of roll

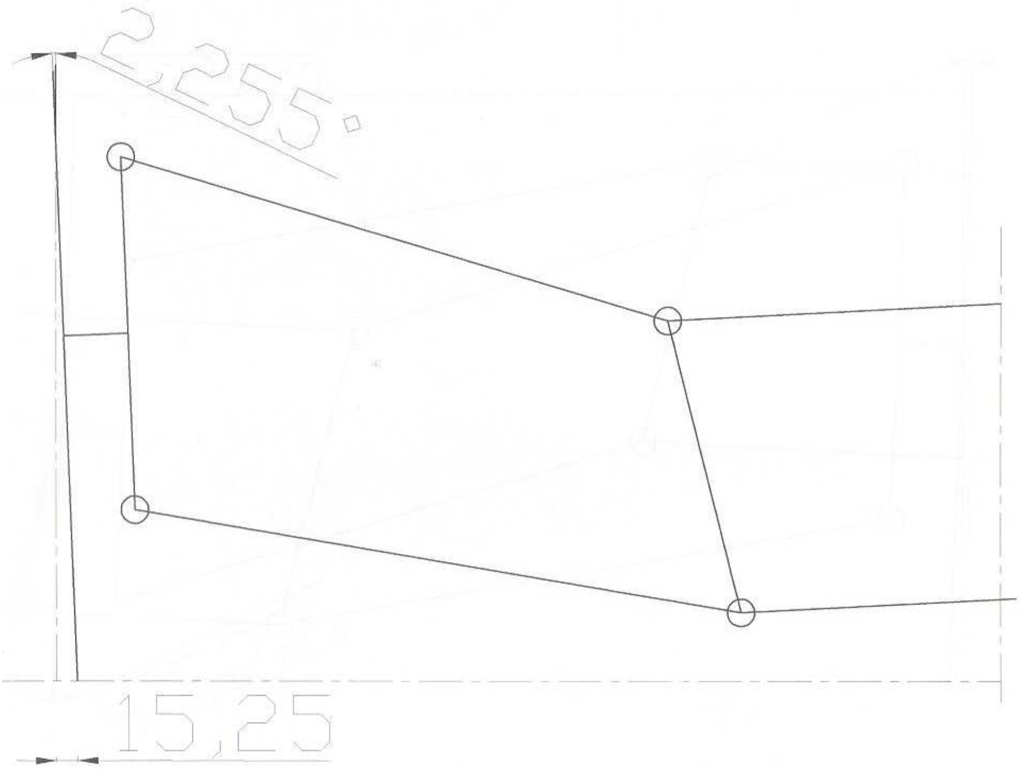

Figure F 4 Camber change with 3 degrees of roll

Appendix E

Cost Report

COST REPORT ESTIMATE

1. Cost calculations for the manufactured components

Cost calculations have been prepared using operations cost table from 2005 Formula SAE Rules book.

LOWER FRONT CONTROL ARMS

Materials	**Qty.**	**Description**	**Unit**	**$/Unit**	**Cost**
ERW circular steel tube	2x0.4 m	19 mm dia..x 1.6	m	$2.15	$ 1.72
RMS	3x0.04 m	20 mm dia.	m	$4.35	$ 0.52
BMS square	1x0.06 m	45x45		$42.90	$ 2.57
				Subtotal	**$ 4.80**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut tube	2	2	cm	$0.16	$ 0.64
Prepare tube for welding	2	2	end	$ 0.75	$ 3.00
Cut circ. sec.	2	2	cm	$ 0.16	$ 0.64
Drill hole	2	1	hole	$ 0.35	$ 0.40
Tap hole	2	1	hole	$ 0.35	$ 0.40
Weld boss to tube	2	6	cm	$ 0.14	$ 1.68
Machine housing	1	15	min.	$ 1.16	$17.50
Prepare housing for welding	1	5	Min.	$ 0.58	$ 2.90
Weld tubes to housing	1	12	cm	$ 0.14	$ 1.68
				Subtotal	**$28.84**
				TOTAL	**$33.65**

UPPER REAR CONTROL ARMS

Materials	Qty.	Description	Unit	$/Unit	Cost
ERW circular steel tube	3x0.4m	19 mm dia.x 1.6	m	$2.15	$ 2.58
RMS	3x0.04 m	20 mm dia.	m	$4.35	$ 0.52
FMS 35x3 flat	2x0.04m	35x3	m	$1.81	$ 0.15
				Subtotal	**$ 3.25**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut tube	2	2	cm	$0.16	$ 0.64
Prepare tube for welding	2	2	end	$ 0.75	$ 3.00
Cut circ. Sec.	4	2	cm	$ 0.16	$ 1.28
Drill hole	4	1	hole	$ 0.35	$ 1.40
Tap hole	4	1	hole	$ 0.35	$ 1.40
Weld boss to tube	4	6	cm	$ 0.14	$ 3.36
Cut flat 35x3	2	7	cm	$ 0.16	$ 1.12
Prepare for welding	2	0.08	h	$35.00	$ 2.80
Weld lugs	2	8	cm	$ 0.14	$ 1.12
				Subtotal	**$16.12**
				TOTAL	**$19.40**

FRONT UPRIGHT

Materials	Qty.	Description	Unit	$/Unit	Cost
RHS	1x0.3m	40x40x2	m	$ 9.00	$ 2.70
RMS circular section	1x0.205m	60 mm dia.	m	$47.9	$ 9.82
FMS 35x5 flat	1x0.22m	35x5	m	$ 2.86	$ 0.63
FMS 50x5 flat	1x0.3m	50x5	m	$ 4.47	$ 1.34
FMS 35x3 flat	1x0.08m	35x3	m	$ 1.81	$ 0.15
BMS square	1x0.11m	45x45	m	$ 42.9	$ 4.72
				Subtotal	**$19.35**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut circular section BMS	1x0.205m	0.17	h	$35.00	$ 5.95
Cut RHS to length	2	8	cm	$ 0.16	$ 1.28
Prepare for welding	3	0.25	h	$ 35.00	$ 8.75

Weld RHS to BMS	2	32	cm	$ 0.14	$ 4.48
Machine bearing housing	1	0.58	h	$70.00	$40.60
Cut BMS to length	2	0.17	h	$35.00	$ 5.95
Machine suspension brackets	2	0.75	h	$70.00	$52.50
Prepare for welding	2	0.17	h	$35.00	$ 5.95
Weld brackets	2	32	cm	$ 0.14	$ 4.48
Cut flat 35x5	2	7	cm	$ 0.16	$ 1.12
Cut flat 50x5	1	5	cm	$ 0.16	$ 0.80
Cut flat 35x3	1	5	cm	$ 0.16	$ 0.80
Prepare cuts for welding	4	0.25	h	$35.00	$ 8.75
Weld brackets on upright	4	17	cm	$ 0.14	$ 2.38
				Subtotal	**$143.79**
				TOTAL	**$163.15**

REAR UPRIGHT

Materials	**Qty.**	**Description**	**Unit**	**$/Unit**	**Cost**
RHS	1x0.2m	40x40x2	m	$ 9.00	$ 1.8
BMS circular section	1x0.200m	70mm dia.	m	$52.30	$10.46
FMS 40x5 flat	1x0.20m	40x5	m	$ 3.15	$ 0.63
FMS 160x5	1x0.11m	160x5	m	$ 5.90	$ 0.65
BMS square	1x0.05m	45x45	m	$ 42.9	$ 2.15
				Subtotal	**$15.70**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut circular section BMS	1x0.200m	0.17	h	$35.00	$ 5.95
Cut RHS to length	2	8	cm	$ 0.16	$ 1.28
Prepare for welding	3	0.25	h	$ 35.00	$ 8.75
Weld RHS to BMS	2	32	cm	$ 0.14	$ 4.48
Machine bearing housing	1	0.58	h	$70.00	$40.60
Cut BMS to length	2	0.17	h	$35.00	$ 5.95
Machine suspension brackets	2	0.75	h	$70.00	$52.50
Prepare for welding	2	0.17	h	$35.00	$ 5.95

Weld brackets	2	32	cm	$ 0.14	$ 4.48
Cut flat 40x5 to length	2	8	cm	$ 0.16	$ 1.28
Cut flat 160x5	1	16	cm	$ 0.16	$ 2.56
Prepare cuts for welding	4	0.25	h	$35.00	$ 8.75
Weld brackets on upright		68	cm	$ 0.14	$ 9.52
				Subtotal	**$152.05**
				TOTAL	**$167.80**

HUB AXLE

Materials	**Qty.**	**Description**	**Unit**	**$/Unit**	**Cost**
S 1020 Ø 160	0.1 m		m	$480.00	$48.00
				Subtotal	**$48.00**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Turning		5	h	$/35.00	$175.00
CNC set up time		2/1000	h/piece	$70.00	$0.14
CNC Milling		5	h	$70.00	$350.00
				Subtotal	**$525.14**
				TOTAL	**$573.17**

STEERING TIE ROD

Materials	**Qty.**	**Description**	**Unit**	**$/Unit**	**Cost**
ERW	1x0.3		m	$2.15	$0.65
RMS	1x0.04		m	$4.35	$0.17
				Subtotal	**$0.80**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut tube	1	2	cm	$0.16	$ 0.32
Prepare tube for welding	1	2	end	$ 0.75	$1.50
Cut circ. Sec.	2	2	cm	$ 0.16	$ 0.64
Drill hole	2	1	hole	$ 0.35	$ 0.70
Tap hole	2	1	hole	$ 0.35	$0.70
Weld boss to tube	2	6	cm	$ 0.14	$1.68
				Subtotal	**$5.54**
				TOTAL	**$6.35**

SUSPENSION ROCKER ARMS

Materials	Qty.	Description	Unit	$/Unit	Cost
FMS	4x0.08	50x3	m	$3.65	$1.20
RMS tube	1x0.02	20 diax1.6	m	$2.15	$0.05
				Subtotal	**$1.25**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut flat	4	0.25	h	$35.00	$8.75
Cut bush to length	1	2	cm	$0.16	$0.32
Prepare flat and bush for welding	5	0.17	h	$ 35.00	$5.83
Weld flats and bush	2	6	cm	$ 0.14	$1.68
				Subtotal	**$16.58**
				TOTAL	**$17.85**

STEERING ROCKER ARMS

Materials	Qty.	Description	Unit	$/Unit	Cost
RHS	4x0.04	20x20	m	$2.85	$0.45
RMS tube	3x0.02	20 diax1.6	m	$2.15	$0.13
				Subtotal	**$0.58**
Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Cut RHS	4	0.25	h	$35.00	$8.75
Cut bush to length	3	2	cm	$0.16	$0.96
Prepare flat and bush for welding	5	0.17	h	$ 35.00	$5.83
Weld flats and bushes	3	6	cm	$ 0.14	$2.52
				Subtotal	**$18.06**
				TOTAL	**$18.65**

2. Cost calculations for modified components

STEERING BOX

Materials	Qty.	Description	Unit	$/Unit	Cost
Steering box assembly	1	Honda	piece	$200.00	$200.00
				Subtotal	**$200.00**

Process Labour	**Qty.**	**Amount**	**Unit**	**$/Unit**	**Cost**
Dismantle and clean assembly	1	0.33	h	$35.00	$11.66
Cut rack	1	0.08	h	$35.00	$ 2.80
Face, drill, tap		0.5	h	$35.00	$17.50
Cut housing	1	0.166	h	$35.00	$ 5.81
Weld housing		0.166	h	$35.00	$ 5.81
Assembly		0.25	h	$35.00	$ 8.75
				Subtotal	**$46.48**
				TOTAL	**$246.48**

3. Cost calculations for assemblies

FRONT SUSPENSION

Assembly component	**Qty**	**Manufactured**	**Modified(M) Purchased (P)**	**Cost per component**	**Total cost**
Front upright	2	USQ workshop		$163.15	$326.30
Upper wishbone	2	USQ workshop		$ 18.50	$ 37.00
Lower wishbone	2	USQ workshop		$ 33.65	$ 67.30
Hub axle	2	USQ workshop		$573.17	$1146.34
Suspension rocker arm	2	USQ workshop		$ 17.85	$ 35.70
Suspension rod	2	USQ workshop		$ 6.35	$ 12.70
Spherical rod ends AM6GP	5		(P) Linear Bearings Pty Ltd	$ 10.30	$ 51.50
Spherical bearing	2		(P) Linear Bearings Pty Ltd	$ 63.00	$126.00
Angular contact bearings	4		(P) Toowoomba Bearings and Seals	$ 25.00	$100.00
Spring/shock	2				
				TOTAL	**$1941.90**

REAR SUSPENSION

Assembly component	**Qty**	**Manufactured**	**Modified(M) Purchased (P)**	**Cost per component**	**Total cost**
Rear upright	2	USQ workshop		$167.8	$335.60
Upper wishbone	2	USQ workshop		$ 19.40	$ 38.80

Lower wishbone	2	USQ workshop		$ 33.65	$ 67.30
Hub axle	2		(P)		
Suspension rocker arm	2	USQ workshop		$ 17.85	$ 35.70
Suspension tie rod	2	USQ workshop		$ 6.35	$ 12.70
Spherical rod ends AM6GP	7		(P) Linear Bearings Pty Ltd	$ 10.30	$ 72.10
Spherical bearing	2		(P) Linear Bearings Pty Ltd	$ 63.00	$126.00
Angular contact bearings	4		(P)Toowoomba Bearings and Seals	$ 25.00	$100.00
Spring/shock	2		(P)	$150.00	$300.00
				TOTAL	**$1088.20**

STEERING SYSTEM

Assembly component	**Qty**	**Manufactured**	**Modified(M) Purchased (P)**	**Cost per component**	**Total cost**
Steering box	1		(M)	$246.48	$246.48
Steering rocker arm	2	USQ workshop		$ 18.65	$ 37.30
Steering tie rod	2	USQ workshop		$ 6.35	$ 12.70
Rocker arm bearings	6		(P)	$ 2.50	$ 15.00
				TOTAL	**$310.50**