University of Southern Queensland

Faculty of Engineering & Surveying

Steering Design for a Formula SAE-A Open Wheel Race Car

A dissertation submitted by

Leslie Rayner

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towards the degree of

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Abstract

This dissertation documents a design project for the steering system for the first Formula SAE-A car made at the University of Southern Queensland.

The documentation includes a description of the design processes adopted for the various parts of the steering system. This begins with a review of currently used steering designs in modern motor vehicles and the projects design constraints. The next few chapters deal with the actual design of the steering components and their integration together. This is followed by the technical specifications of the final design and finally the conclusion.

The steering system complies with all the relevant rules of the Formula SAE-A rule book.

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ENG4111/2 *Research Project*

Limitations of Use

Certification of Dissertation

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.

Leslie Rayner

Q12210982

Signature

Date

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LESLIE RAYNER

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Chapter 1

Introduction

1.1 Chapter Overview

An overview of the Formula SAE-A competition rules, the design methodology, the software packages used and the project objectives is presented in this chapter to give some background information on the project.

1.2 Overview of the Formula SAE-A Competition

The principle objective of Formula SAE-A is for students to conceive, design, build and compete with a small open wheel race car. The restrictions on the car are designed to maximize the use of the student's imagination and knowledge, and to give the students a meaningful project as well as good practice working in a team environment. The car is to be designed to maximize its acceleration, handling and braking. The maximum speed is kept to about 100kph by the track layout. The car needs to be easy to maintain, low in cost and reliable. The design brief is for a prototype of a car that is intended to be made as a production item for non-professional weekend autocross racers. As such the prototype should cost less than $25000. The challenge is to design and build a car that best meets these requirements that will then be compared with the other competing designs to determine the best overall vehicle.

1.3 Judging

The finished car is judged in a combination of static and dynamic events.

Static Events

Presentation	75
Engineering Design	150
Cost Analysis	100

Dynamic Events

Acceleration	75
Skid-Pad Event	50
Autocross Event	150
Fuel Economy Event	50
Endurance Event	350
Total Points	1000

1.4 Eligibility

To ensure fair competition, entrants must be undergraduates or recently graduate students only. The car must be conceived, designed and fabricated by the students without direct involvement of any professionals, but students are allowed to use information from said professionals as long as the information is used in a discussion of the pros and cons of all the alternatives. Students should perform all fabricating wherever possible to give them hands on experience with performing these tasks.

Driver. All drivers must use a 5 or 6 point safety harness at all times. A safety helmet, fire resistant suit, fire resistant gloves and shoes of an approved standard are also to be used at all times. All drivers must be at least 18 years of age and possess a valid government issued drivers licence.

1.5 Vehicle Requirements

1.5.1 Body

The vehicle is required to be an open style cockpit and open wheeled design. Openings into the drivers compartment must be kept to an absolute minimum. The car must have a minimum wheelbase of 1525mm (60 inches), and four wheels not in a straight line. The minimum track is to be no less than 75% of the maximum.

1.5.2 Chassis

Minimum ground clearance is to be such that no part other than the tyres touches the ground at any point during racing. The wheels should be a minimum of 203.2mm (8 inches) in diameter and have a fully operational suspension package on all wheels with at least 50.8 (2 inches) travel. The tyre size and type is free. The tyres are not allowed to contact the bodywork, suspension components or frame at any time during the event. The car must be equipped with a braking system that operates on all four wheels with two independent hydraulic circuits. Both circuits are to be operated by a single control. The chassis also must have crash protection and roll hoops built into it at various points for safety.

1.5.3 Engine and Drivetrain

The engine used to power the car must be four stroke with a maximum displacement of 610cc. Any transmission and driveline can be used. Forced induction is allowed but has restrictions on where it is placed. All engine airflow is to pass through a 20mm restrictor to limit the power capabilities of the engines.

1.5.4 Steering

The Formula SAE-A rules state:

- 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 tyres from contacting the car at all times; and
- the allowable free play is limited to 7 degrees total, and is measured at the steering wheel.

1.6 Design Methodology

The design process can be generally described as having the following steps (Ertas & Jones 1996):

- recognition of need;
- conceptualisation;
- feasibility;
- decision to proceed;
- preliminary design;
- cost analysis / redesign;
- detailed design;
- production planning and tooling design; and
- production.

This project should incorporate all phases except the tooling design phase as the steering system will be a one off part.

1.7 Solid Modeling Software

ProEngineer was chosen to create the solid models for the steering package. It was chosen for the following reasons:

- excellent visualisation capabilities (as models can be rotated, zoomed and moved around the screen at any time);
- the features library is very extensive allowing detailed and accurate models to be created;
- the drawing is created automatically from the solid model;
- availability at the University of Southern Queensland;
- previous experience and knowledge;
- package chosen by the team to create the full 'virtual car'; and
- allows easy capture of model into the Finite Element Analysis package, Ansys.

ProEngineer is parametric based, which means that when a parameter that defines a feature in the model is modified, the model is automatically updated. Relations can also be specified between different parts so that when one part is changed the others that are related change automatically. This is a very useful feature when creating a large, complicated model.

ProEngineer uses predefined features such as protrusions, cuts, rounds and chamfers that are defined in space using a cartesian coordinate system. Using these features it is possible to build on the original base feature in any number of combinations, to form the final assembly. This is usually done in a similar order to which you would assemble the exact part. These parts can then be assembled into a complete model by mating different features on the various parts together to form the final complete assembly.

Items like the uprights and axle are most easily designed in solid modeling packages such as Solidworks and ProEngineer. Traditional two-dimensional packages are not as powerful for this type of project. A parametric package allows the updating of any dimension at any time and then can be regenerated instantly. It also allows addition of new features at any time. Creating a solid model was very useful to optimise the parts and to make sure that they fit together properly.

1.7.1 Approach Used for Solid Modeling

The solid model of the steering system was built by creating the separate parts of steering system and then assembling them all together to create a full assembly. The modeling approach that was used to create each part is as follows:

- create default part datum planes;
- protrude base feature of part; and
- create features to add to original base feature to systematically build up the entire part.

The actual steps used to create the axle were as follows:

- create base drawing and revolve around 360 degrees;
- extrude cut one of the four cutouts in the front of the axle and circular pattern this cut;
- extrude cut one of the six holes for mounting the brake disk rotor and circular pattern this cut;
- extrude cut one of the six cutouts from between the brake rotor mounting points and circular pattern this cut;
- extrude cut the brake disk rotor locating lugs; and
- extrude cut one of the four road wheel stud holes and then circular pattern this hole.

The assembly was created using a similar approach:

- create default assembly datum planes;
- import base part; and
- import additional parts and mate them together by defining their positions with respect to the original base feature.

The features of each part were then checked and modified (if required) as part of the optimisation process. This procedure generated the parts illustrated in Figure 5.9, Figure 5.10, Figure 5.11 and Figure 5.12.

1.8 Finite Element Analysis Software

Finite element analysis was used to optimise the strength to weight ratio and to check if the part was satisfactory.

Ansys was chosen to perform the finite element analysis for the steering package. It was chosen for the following reasons:

- allows easy capture of model from the solid modeling package, ProEngineer;
- availability at the University of Southern Queensland;
- previous experience and knowledge; and
- package chosen by the team run all the finite element analysis.

Other packages were available for purchase but this was considered unnecessary as Ansys works well with the files output from ProEngineer and it is a very powerful tool if used correctly.

1.8.1 Finite Element Analysis Background

The basic premise of the finite element method is that a part can be analytically modeled or approximated by assembling it with an assemblage of discrete elements.

Because these elements can be put together in many ways, they can be used to represent extremely complex shapes. An advantage of the finite element method over other methods is the relatively ease with which the boundary conditions of the problem are handled. (Ertas & Jones 1996) These elements are predefined, and each has different characteristics and properties.

The elements are grouped into three main categories:

- one dimensional elements (see fig 1.1(a));
- two dimensional elements (see fig 1.1(b)); and
- three dimensional elements (see fig 1.1(c)).

The most common finite elements are the one dimensional elements with two and three nodes. See Figure 1.1(a). They have a single degree of freedom at each grid point and are used for one-dimensional problems involving two force (truss) members. In the two-dimensional domain the two general elemental families are the triangular and quadrilateral elements. See Figure 1.1(b). The lower order elements only have nodes at the corners and the displacement and strain is constant along the boundaries, hence they are called linear elements. The higher order elements have a node in the centre, which means they can have straight or curved edges and this allows quadratic displacement along each boundary, hence they are called quadratic elements. Two-dimensional elements are used to model membranes, plates and shells. Quadrilateral elements are preferred because of their greater accuracy. Three-dimensional elements are used to model thick plates, thick shells and three-dimensional solid media. See Figure 1.1(c). (Ertas & Jones 1996)

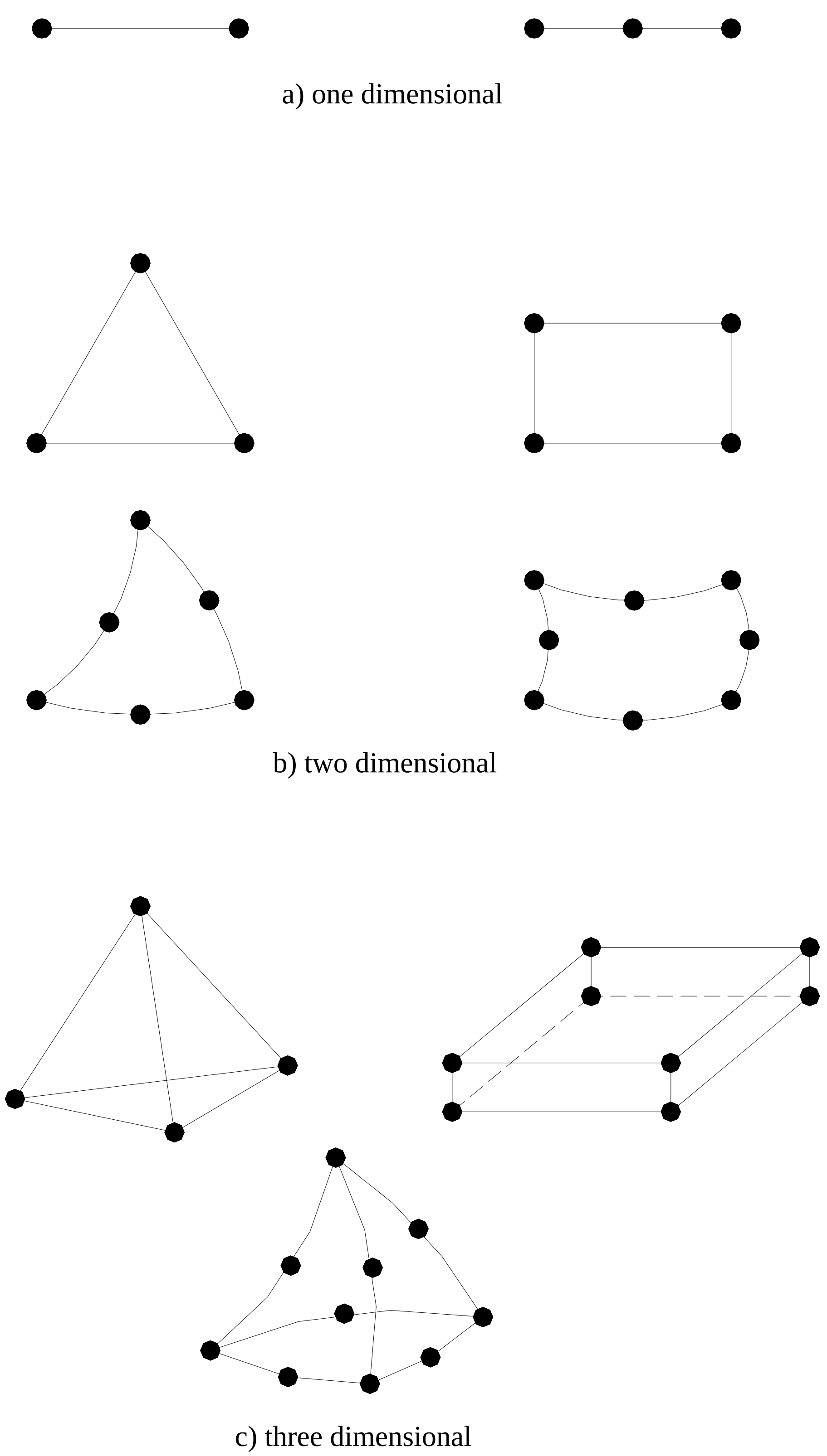


Figure 1.1: Common finite elements.

All objects are constrained in space, and in finite element analysis this is called a boundary condition. Boundary conditions are applied at nodes and can be an applied force or displacement. If the displacement is set at zero then the node is fixed from translation or rotation.

The relationship between the forces and displacement for every element in the part is termed the stiffness matrix and is denoted [K]. The forces that act on every node in the part is described by the force vector which is a column matrix and is denoted {F}. The displacement of every node in the object is described by the matrix {d}. Thus the relationship between the applied force and the resultant displacement can be described by Equation 1.1.

$$\{F\} = [K] * \{d\} \tag{1.1}$$

The most common approach used in finite element analysis solves for the displacements at each node and these displacements are then used to determine the strains and stresses in the part. (Snook 2003)

The main steps in the finite element approach are as follows (Snook 2003):

- model definition (nodal geometry and element type);
- material properties definition;
- load and displacement definition;
- formulate stiffness matrix;
- solve for nodal displacement, stress and strain;
- results output;
- results interpretation; and
- model updated and optimised.

The engineer completes the first three steps in what is known as the pre-processing stage. The software completes the next two steps in the solver stage and the final three stages are completed in the post-processing stage. This is where the engineer draws conclusions from the results to determine how the model can be modified to give the desired results. The stub axle components were designed using this procedure. Optimisation of the stub axle components consisted of removing weight in areas where possible without compromising the structural integrity of the components.

FEA is a very powerful tool for use in the design of components. It enables us to calculate the stresses on parts that would be extremely difficult to calculate using any other method.

1.9 Design of the Upright and Axle

Designing and optimisation of the upright and axle components required a knowledge of the following:

- suitable factors of safety;
- material properties;
- boundary conditions;
- suitable loading; and
- analysis types.

Both components were de-featured to remove unnecessary complications and to save a huge amount of time. Initially the axle was tested with a fully featured model and it took approximately twenty minutes to solve, whereas after de-featuring this dropped to approximately five minutes with little to no loss in accuracy.

1.9.1 Factor of Safety

In order that the components don't fail prematurely, they should be designed using a suitable factor of safety. Because the parts are for a race car, the factor of safety should not be too large or the weight will be excessive. On the other hand if the factor of safety is too small, premature failure may result with possible catastrophic results. A factor of safety of approximately two was chosen for this project. This was incorporated into the calculations and FEA by using a force that was double what was expected.

1.9.2 Material Properties

The material properties used for the FEA analysis are shown in Table 5.1

1.9.3 Boundary Conditions and Loading

In both the upright and axle the bearing surfaces were fixed from translation and rotation in all directions by specifying a displacement of zero. In the case of the upright the forces were then placed on the top and bottom suspension mounting points, and in the case of the axle, on the outside face to which the road wheel is bolted. See Figure 5.6 and Figure 5.7. In reality this is not the case but it is a good approximation.

1.9.4 Analysis Type

The analysis type was a three dimensional, structural, solid, linear, elastic, isotropic analysis made up of tetrahedral 10 node (92) type elements.

1.10 Dissertation Generation

The author had the choice of a few software tools to generate this dissertation. The two most readily available were Microsoft Word and LaTeX. The author had no previous knowledge of LaTeX but after using Word for other large documents and having had problems, the decision was made to learn and use LaTeX.

1.11 Project Objectives

The final aim of this project was the design and construction of a steering system for a Formula SAE-A open wheel race car. In order to achieve this goal in the available time various tasks were set. These include:

- the research of information on currently used systems for steering motor vehicles;
- the research of the specifications and rules for Formula SAE-A;
- evaluating all the current steering systems with a view to finding the best design for a formula SAE-A race car;
- choosing a design and developing a preliminary proposal to integrate with the chassis and suspension designs;
- testing the design using finite element analysis and optimising the design;
- having the design manufactured and then testing the design in the vehicle; and
- the redesign and reanalysis of the steering system if needed.

Chapter 2

Background

2.1 Chapter Overview

An overview of the current situation in automobile steering systems is presented, so that an informed decision on the type of steering system to be used in the Formula SAE-A car can be made.

2.2 Background of Automobile Steering Systems

2.2.1 Turntable Steering

In the late 1800's and early 1900's most of a car's weight was concentrated over the rear axle. Steering was initially accomplished by turning a tiller that simply pivoted the entire front axle assembly as on a typical horse cart (Heisler 1997). See Figure 2.1. This is called turntable steering after the large turntable that is needed for the axle to rotate. Turntable steering achieves true rolling in all positions, that is, without any lateral slip, but creates a lot of other problems. Steering away from centre causes a large twisting moment about the central pivot, which requires a large diameter turntable to give support. A huge amount of clearance is needed when the axle is rotated to give adequate room for the axle-beam and inner road wheel to clear the side of the vehicle.

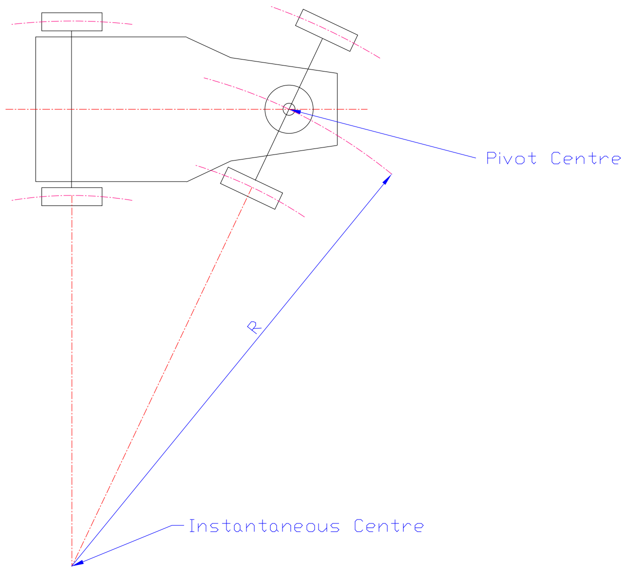


Figure 2.1: Turntable steering.

2.2.2 Lateral Slip

As can be seen in Figure 2.2 motion along the XX axis is purely rolling, motion along the YY axis is purely slip, and motion along any other plane is a combination of rolling and slip.

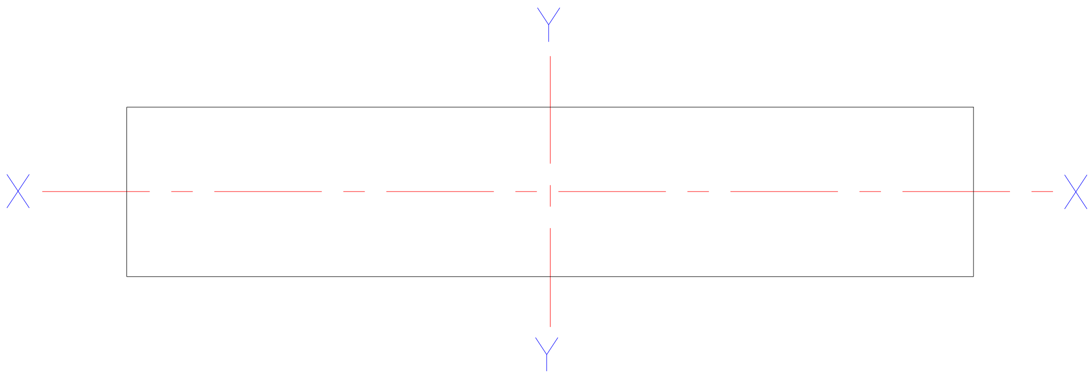


Figure 2.2: Lateral slip.

2.2.3 Ackermann Geometry

When the engine was moved to the front of the car and as the vehicles speeds increased, tiller steering proved to be ineffective. More complex steering systems had to be introduced to cope with the increased demands on stability, increase the ease of use by the operator and to make a more effective use of the available space. It was soon found that to achieve true rolling using a fixed front axle, the inner wheel had to turn at a greater angle than the outer wheel, because the inner wheel follows a smaller radius circle than the outer wheel. It was found that the axis produced by the two front wheels must intersect at a common point along the axis produced by the rear wheels. It follows that the tighter the turn, the closer this point will be to the side of the vehicle, and this point also becomes the centre about which the whole vehicle is turning. This is called the Ackermann Principle after Rudolph Ackermann, who patented a double pivot steering arrangement in 1817 for horse drawn vehicles (Heisler 1997). The original patent had the track-rod arms parallel to each other, so that they formed a rectangle when the road wheels were pointed straight ahead. In this configuration, both wheels are turned exactly the same amount. See Figure 2.3. This does not produce true rolling, except for in the straight ahead position.

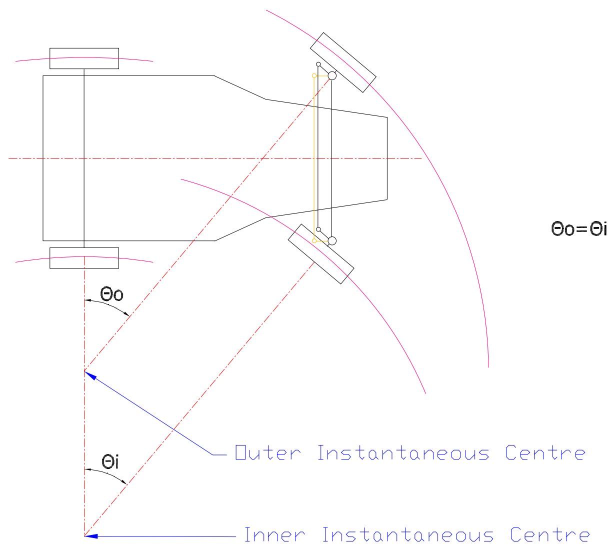


Figure 2.3: Parallel track-rod arms giving true rolling at straight ahead only.

A modification of the Ackermann linkage layout introduced in 1878 by Charles Jeantaud was to incline the track-rod arms, so that lines drawn along the track-rod arms converge and intersect near the centre of the back axle. This arrangement forms a trapezium shape at the straight ahead position and enables the inner wheel to turn at a sharper angle than the outer wheel. See Figure 2.4 and Figure 2.5.

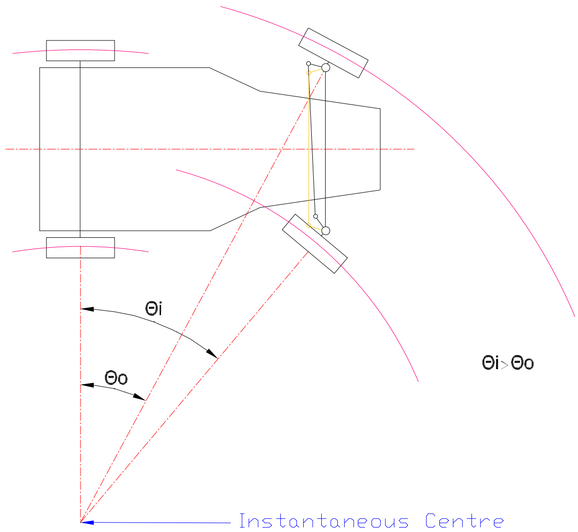


Figure 2.4: Inclined track-rod arms giving true rolling in three positions.

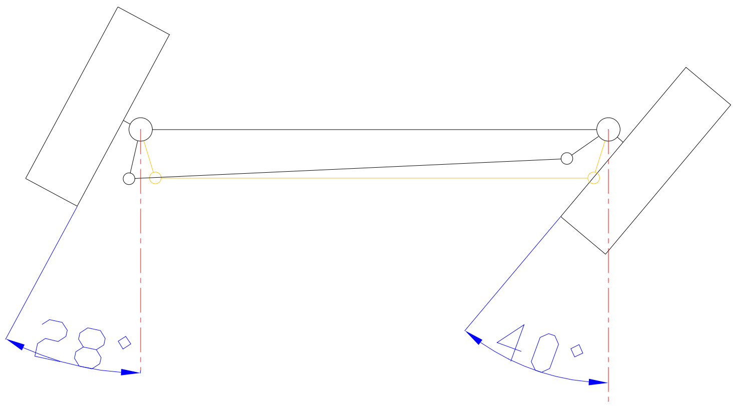


Figure 2.5: Outer wheel turns at a greater angle than the inner wheel.

This inclined track-rod layout is still referred to as an Ackermann linkage as the only modification is the inclination of the track-rods. As stated this is necessary to help achieve true rolling, but in reality true rolling is only achieved at the straight ahead position and at one other position on either side of centre. In practice the misalignment is inconsequential for turns up to fifteen degrees, (Heisler 1997) and the vehicle will usually be traveling relatively slowly for sharper turns than this, so the misalignment inconsequential. Also because of major advances in tyre technology in recent years, tyre sidewall flexibility and distortion of the tread correct some of the deviation from true rolling, and help minimize the consequences of this problem. Complicated linkages have been developed that do give true rolling in all positions but the extra weight, cost and complication has proved unnecessary.

2.2.4 Toe In and Toe Out

For free rolling conditions the wheels on each axle should be as close as possible to being parallel to each other when the vehicle is traveling straight. When the front wheels are aligned so that they converge toward the front the wheels are said to have toe in, and will be trying to roll together. See Figure 2.6.

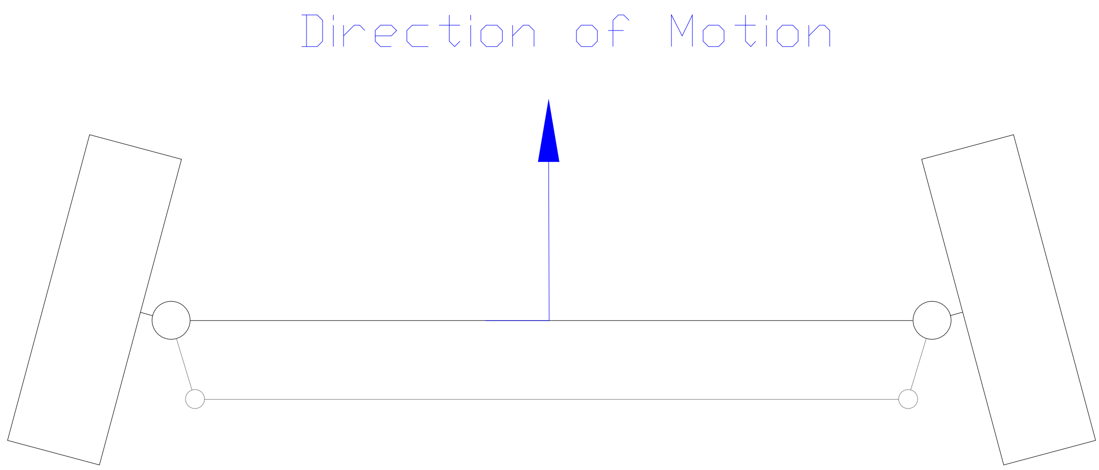


Figure 2.6: Toe-in.

Conversely if the wheels are aligned so that they converge toward the rear they are said to have toe out and will be trying to roll apart. See Figure 2.7.

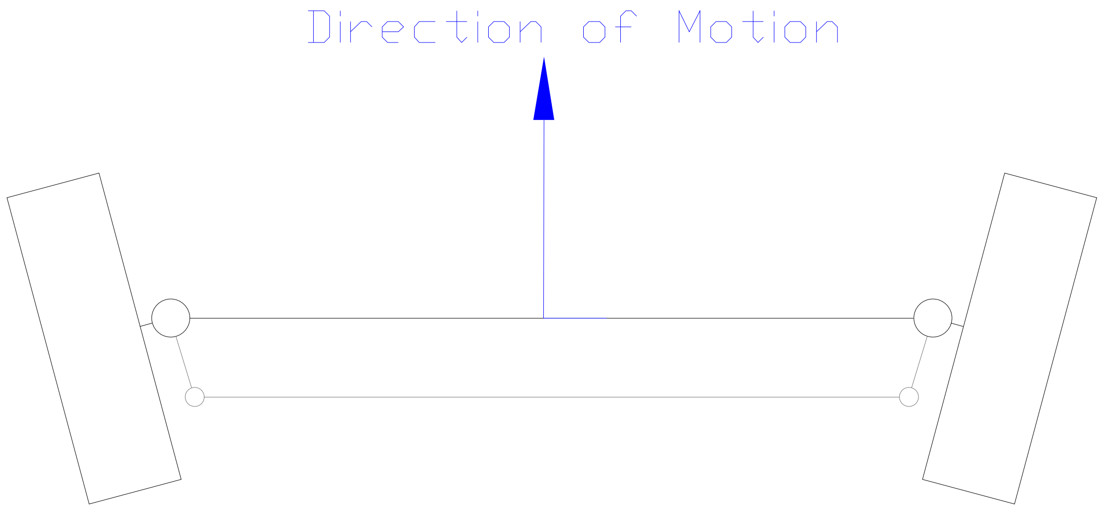


Figure 2.7: Toe-out.

The tendency to roll together or roll apart will be counteracted by the steering linkages but causes a cross-tread scrub action on both tyres resulting in excessive tyre wear, poor handling and heavier steering. With a typical rear-wheel drive car, the force caused by the reaction of the tyres on the road causes distortion in the suspension rubbers and the wheels will tend to diverge at the front. This tendency is counteracted by setting the steering with a small amount of toe-in so that the wheels run parallel for normal running.

2.2.5 Reversible and Irreversible Steering

In most steering systems the steering wheel can be turned by gripping the road wheels and turning them about their pivots. If the friction forces in the steering system are high enough to prevent this the steering is said to be irreversible.

2.2.6 Rear-wheel Steering

Rear wheel steering is not used on vehicles except where high maneuverability is a necessity on low speed vehicles, such as forklifts. It is far too unstable at high speed to be used on any modern motor vehicle.

2.2.7 Four-wheel Steering

Four wheel steering has been to improve the handling on some recent vehicles. There are a few different methods of obtaining small steering effects on the back wheels.

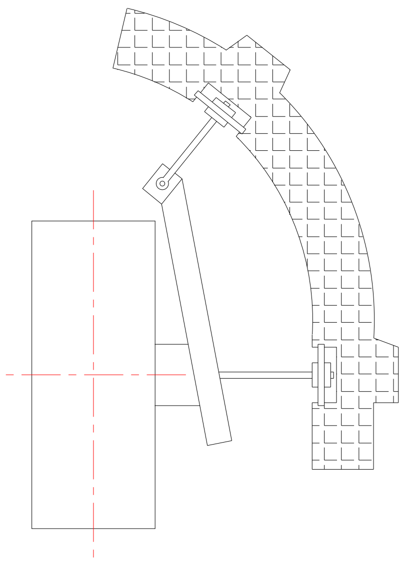


Figure 2.8: Porsche 928 system running freely.

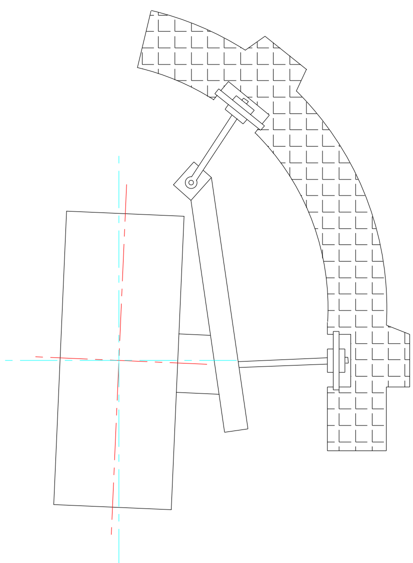


Figure 2.9: Porsche 928 system with brake drag. Note the link's deflection giving the wheel slight toe in.

In a passive rear steering system the vehicle's weight and the forces on the suspension components are used to force the outside wheel to toe in, and the inside wheel to toe out. See Figure 2.8 and Figure 2.9. This is usually achieved by making one suspension bush softer than the other so that drag forces during braking deform the softer bush more than the harder. These systems steer the rear wheels through a max of about one and one half degrees. (Garrett 2001) These systems improve cornering without adding many additional parts and adding huge extra expense.

Another four wheel steering system uses a shaft from the front steering box to a second steering box in the rear to provide the steering of the rear wheels. The design provides steering of the rear wheels in the same direction as the front until the steering wheel is turned more than approximately one third of a turn either side of centre, to improve high speed cornering. After this point the rear wheels are steered in the opposite direction to the front, to improve low speed maneuvers like parking.

Active rear wheel steering systems use hydraulic or electric actuators to steer the rear wheels. They are complex and costly but can be computer controlled, therefore the system can be optimised even while the vehicle is being driven. A good example of this is the Nissan HICAS system. It comprises an electronic control unit (ECU) which controls a solenoid valve, a hydraulic pump and a power cylinder. (Garrett 2001)

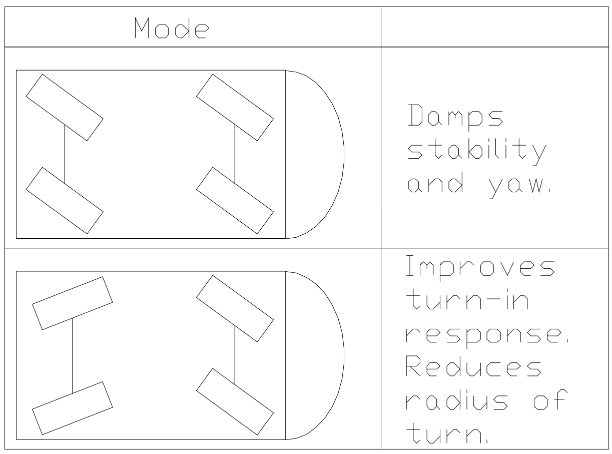


Figure 2.10: The effects of the two modes of the Nissan HICAS steering system.

The ECU receives signals from a sensor on the steering column, an electronic speedometer, clutch, gear selector, brake stop lamp, front-rear braking distribution sensors and sensors for the displacement of the suspension linkages. Super HICAS has a phase reversal system that earlier versions did not. This phase reversal allows the rear wheels to turn in the direction opposite the front as well as in the same direction to improve handling at different speeds as well as to improve the turning circle at parking speeds. See Figure 2.10

2.3 Rack and Pinion

All steering boxes are designed principally to translate the rotational movement of the steering wheel and column to a side to side linear travel of the steering arms.

Rack and pinion steering is the system predominately used in road vehicles today. A rack and pinion steering box should:

- reduce steering wheel effort by using a gear reduction;
- reduce steering response directness;
- enable the wheels to be turned through approximately thirty-five degrees in each direction from centre;
- tend to be semi-irreversible to dampen light wheel wobbles, but still allow some road shock to be transmitted back to the steering wheel.

The pinion is connected by the steering column to the steering wheel. When the steering wheel is rotated, it turns the pinion which is meshed with the mating rack teeth. See Figure 2.11.

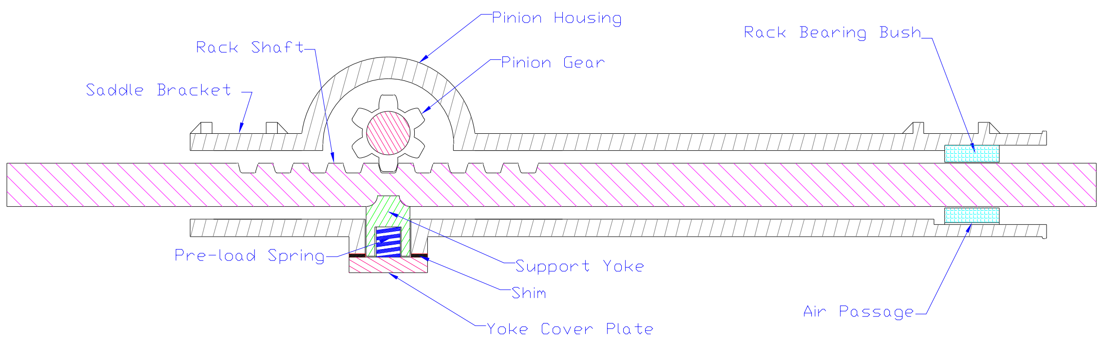


Figure 2.11: Rack and pinion steering box.

This pinion rotation is converted to linear movement by the rack which is supported at one end by a plain bush bearing and at the other end by an adjustable half bearing support yoke opposite the pinion gear. This is adjusted so that it pushes the rack into mesh with the pinion gear and minimizes backlash between the two gears. The circular pitch of the pinion must equal the linear pitch of the rack for correct operation. This linear movement is relayed through the tie-rod to the track rod arms and stub axles to the road wheels, which then causes the vehicle to turn the corner. See Figure 2.12.

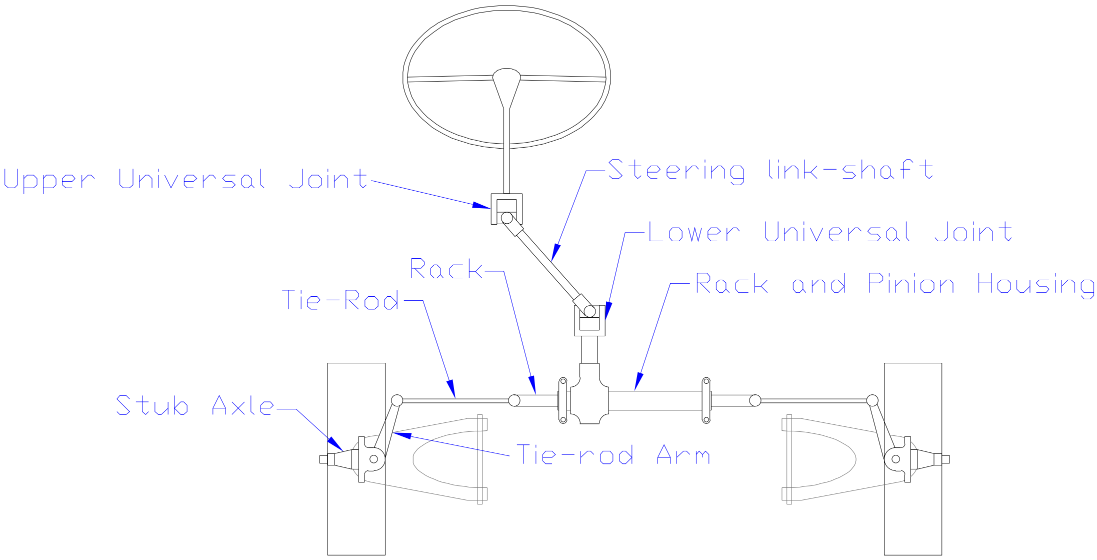


Figure 2.12: Rack and pinion steering linkage.

Early pinion gears were simple straight cut spur gears but these have been replaced by helical-toothed pinions. This is because straight cut teeth will mesh with only one pair of teeth in contact at any one time. Uneven movement of the rack results from this arrangement as the steering load is transferred from one pair of teeth to the next.

Helically cut teeth eliminate this problem as more than one pair of teeth is in contact at any one time. Helical pinion teeth:

- take higher loads;
- are quieter; and
- are smoother.

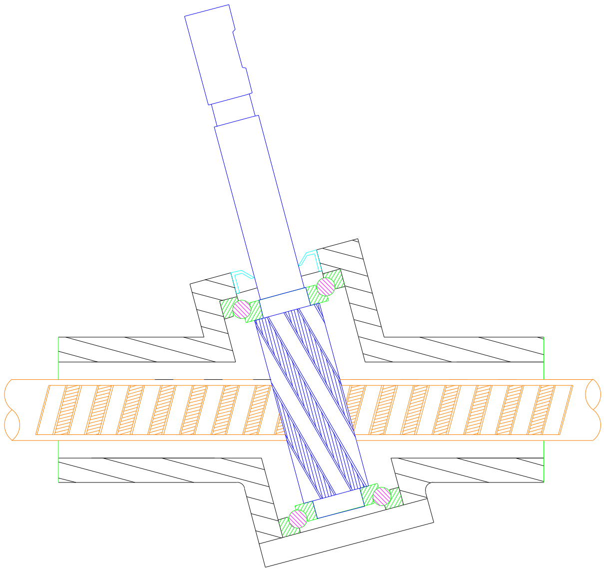


Figure 2.13: Helical rack and pinion steering gear with tilted pinion axis.

The pinion axis is usually tilted from the perpendicular to the rack, as this increases the effective pinion pitch-radius allowing fewer and stronger pinion teeth to be used. See Figure 2.13. This means larger gear-ratio reductions are possible for a given rack travel. It also increases friction which helps reduce the amount of road shock which is transmitted back to the steering wheel and therefore to the driver.

Rack and pinion steering's advantages are:

- it is light compared to other systems;
- it's cost is less than other systems;
- it takes up a small amount of space by comparison to other systems; and
- it provides good steering response.

It's disadvantage is that unless power steering is used, adding to the complexity, it is only efficient on small, light vehicles.

2.3.1 Power Rack and Pinion

Rack and pinion was originally only used in small light vehicles because the steering was too heavy or the reduction ratio needed was too high which resulted in too many turns of the steering wheel for a given road wheel movement. This problem has been minimized with the introduction of power rack and pinion steering systems. The most common method in use at this time employs a hydraulic pump which is driven by the vehicle's engine. The pressure created is applied through a control valve to either side of a piston on the rack to assist in moving the rack in the appropriate direction. See Figure 2.14.

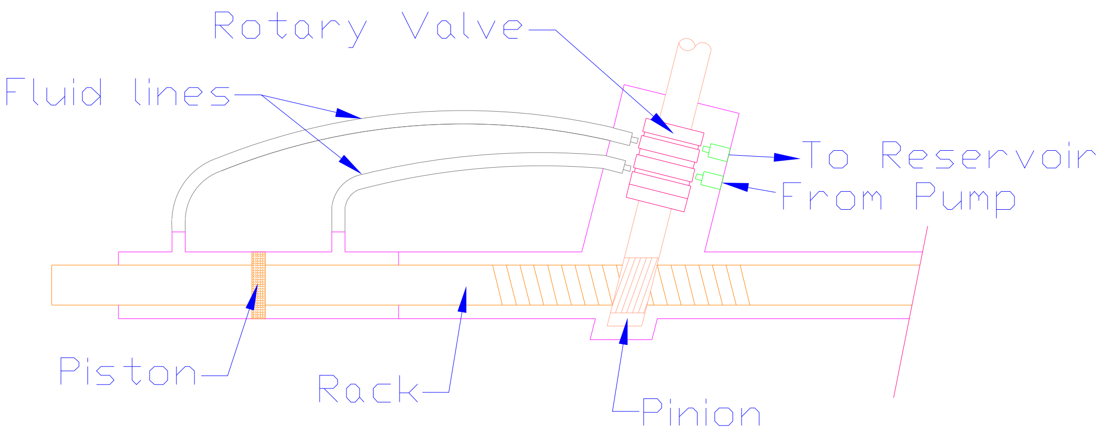


Figure 2.14: Power assisted rack and pinion steering box.

Power steering decreases the effort needed to turn the steering wheel and also allows lower gear ratio reductions to be employed therefore making the steering more direct. A variation in common use, is to add electronic control to the hydraulic pressure system, thereby allowing variable assistance to be used. The system is usually set to increase assistance at low speed to reduce steering effort and to reduce assistance at higher speeds to allow better feedback to the driver.

Another variation of rack and pinion system is to have a variable-ratio rack. The rack and pinion gear-set has a different tooth pitch in the centre compared to the outside. This allows the steering to be more responsive when small steering angles are used and reduces driver effort at larger steering angles.

Car companies are now starting to use electric motors for steering assistance on rack and pinion systems. These systems are smaller and lighter than hydraulic assistance and are controlled using micro processors, so the amount of assistance is infinitely variable. Electrical assistance is much easier to control using electronics than is hydraulics, so many more variables can be taken into consideration. Speed of the vehicle, conditions of the load, rate of change of speed of rotation of the steering wheel and degree of braking or acceleration, can all be taken into account.

The advantages of electric over hydraulic assistance are that (Ludvigsen 2004):

- it increases safety at high speeds by virtue of the fact that assistance is totally independent of engine speed, so assistance is applied more sensitively;
- the kickback felt at the steering wheel by the driver can be made less because it is possible to use irreversible gearing;
- provides lower energy consumption compared to hydraulic assistance, which means lower emissions and better acceleration;
- it is more compact, has less components, weight and maintenance and has no fluid to leak or top up. Installation costs are also reduced;
- steering assistance is still provided even if the engine stops for some reason;
- it has better performance in extreme cold as there is no fluid which increases in viscosity in cold weather;
- it is much easier to design in failure warning, self diagnosis and self-protection systems. It is also much easier to have systems that can rapidly and easily change to suit different applications. User selected feel is easily possible; and
- integration with other systems like anti-lock braking systems and stability control systems is possible.

2.4 Recirculating Ball

The other steering system in common usage is recirculating ball steering. A recirculating ball steering box's biggest advantages are that it is very compact in design and has very low friction. This means it can be used in much heavier vehicles than a manual rack and pinion system. It is not well suited to front wheel drive applications because of its use of a parallelogram steering linkage which is extremely hard to fit in the small space available.

The steering column shaft is connected to a worm gear inside the steering box. The worm gear acts like a screw and moves the ball nut back and forth as the worm gear rotates either one way or the other. The ball nut is held from rotating so that it moves along the worm gear as it rotates. This movement rotates a sector gear using teeth on the side of the ball nut, which in turn moves the pitman arm which causes linear motion of the steering linkages to turn the front wheels. See Figure 2.15.

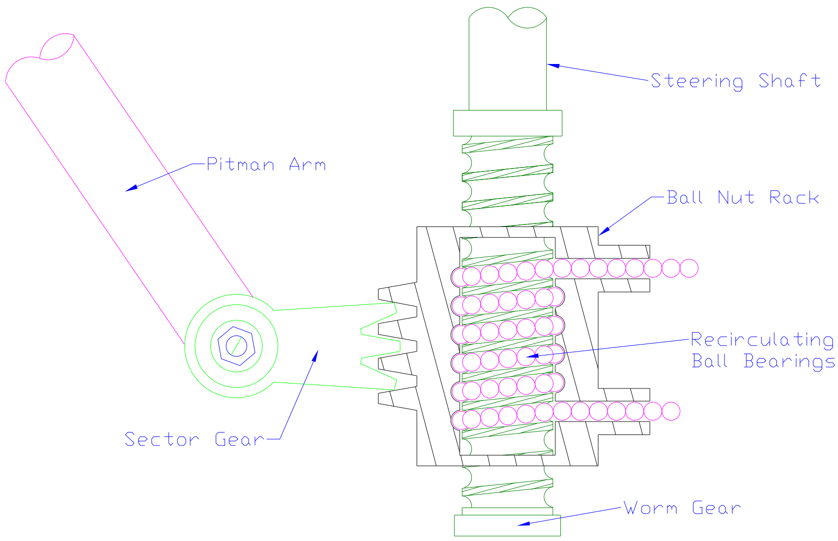


Figure 2.15: Recirculating ball steering box.

Ball bearings are placed between the ball nut and worm gear to reduce friction. They roll in a series of grooves between the ball nuts and worm gear. There is a return tube to the other end of the ball nut around the outside so the ball bearings can travel to the other end of the ball nut to be reused again. It is called a recirculating ball steering gear because of this continuous loop of balls being recirculated through the system.

The steering box is also filled with a light grease to further reduce friction.

Thrust washers or spacers are used to adjust internal clearances between all internal parts. Accuracy in setting these clearances is critical otherwise there is either free-play in the steering if set too loose, or the system will bind and have excessive wear if set too tight.

Power assisted recirculated ball steering systems work much the same as manual gears apart from the fact that they have a sliding spool valve or a rotating spool valve with torsion bar to send the hydraulic pressure to either side of the power piston. This may be located inside or outside the steering box.

There are a number of other steering systems that have been developed but are not in great use today in motor vehicles.

2.4.1 Worm and Sector

The pitman arm shaft carries a sector gear that meshes with a worm gear connected to the steering shaft. Because it only turns through an arc of about seventy degrees, only a sector of gear is needed. When the steering wheel is turned it turns the worm which rotates the sector. This in turn is connected to the pitman arm on a shaft. An adjusting nut is provided to adjust end play on the worm which rotates on tapered roller bearings. See Figure 2.16.

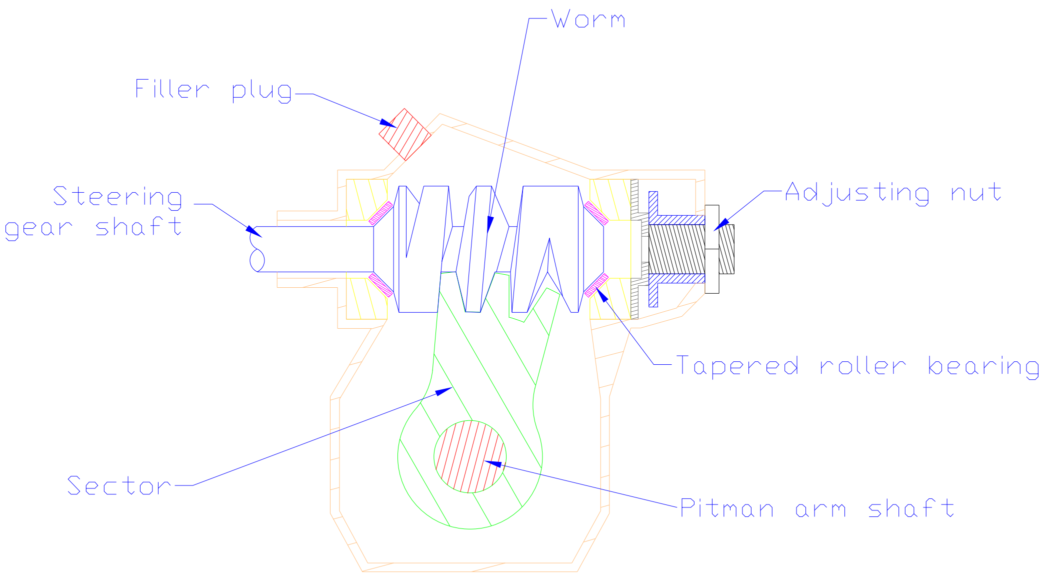


Figure 2.16: Worm and sector steering box.

2.4.2 Worm and Roller

This system is similar to the worm and sector except there is a roller in place of the sector. The roller rotates on bearings which reduces friction. When the steering wheel is moved it turns the worm which rotates the roller which causes the pitman arm to rotate at the other end of the shaft to the roller. The worm has an hourglass shape which produces good contact in all positions and also provides a variable steering ratio. See Figure 2.17.

Due to the efficiency and advancements in both the rack and pinion system and the recirculating ball system most other systems are not used or are used rarely.

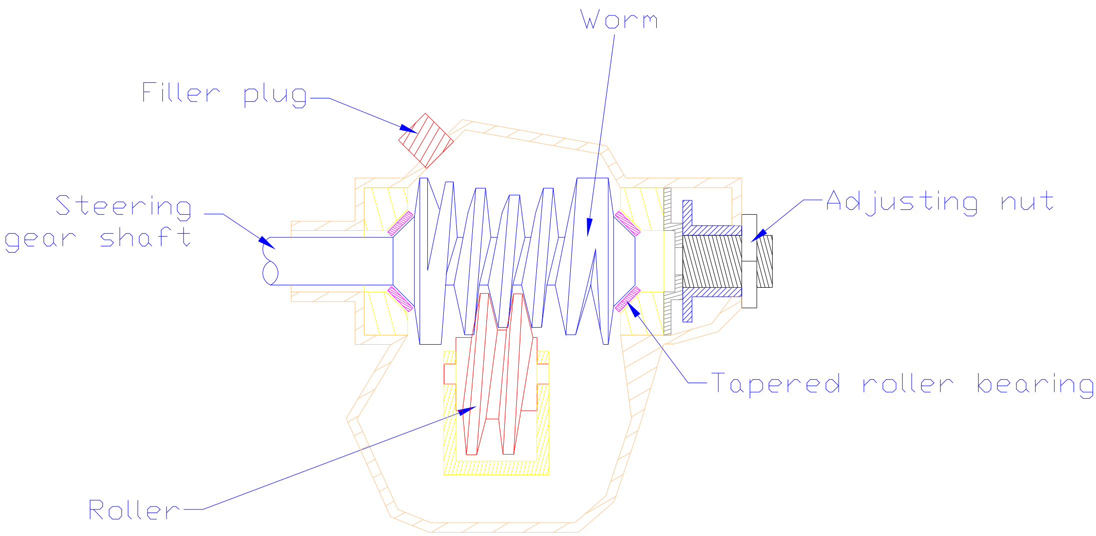


Figure 2.17: Worm and roller steering box.

2.4.3 Drive by Wire

A lot of manufacturers are currently working on drive by wire systems, in which there is no mechanical link whatsoever between the driver controls (not necessarily a steering wheel) and the front wheels. The steering is controlled by electric motors and micro-processors. The biggest problem at the moment with these systems, is the lack of driver feel being fed back to the driver because of the lack of a mechanical link between the road wheels and the driver controls.

2.5 Chapter Summary

The advantages and disadvantages of each of the common forms of steering system has been discussed in this chapter. After comparing the requirements for a Formula SAE-A race car steering system with this information, it indicates that an unassisted rack and pinion steering system would be best for our purposes. The Formula SAE-A car is light so it's steering system will not require power assistance. A relatively cheap system which takes up a small amount of space and is very responsive will achieve all our aims for the steering system.

Chapter 3

Design Constraints

3.1 Chapter Overview

The design constraints are presented briefly so that an informed decisions can be made as to what type of upright and steering system should be designed for our Formula SAE-A car.

3.2 Wheel Rim Size

Early on in the project the wheel rim size was fixed at a 330.2mm (13 inch) magnesium alloy wheel from a Datsun 1600. This fixed the total maximum height of the stub axle component at approximately 308mm. This is the internal diameter of the wheel at the point where the stub axle is located. Therefore it was decided to make the total length of the stub axle 300mm.

3.3 Suspension Restrictions

The suspension arms intersect the upright from forward and back the centre of the wheel. Because of this the maximum height of the stub axle assembly is restricted, otherwise the wheel would hit the suspension arms when the car was turned around a corner. In consultation with the suspension designer (Parmenter, 2004, personal communication), it was found that to prevent fouling between the suspension arms and the wheel at maximum lock, the suspension arms could not be place more than 240mm apart. The suspension arms end in a 10mm diameter internal diameter rod end, so a mount for this had to be incorporated into the stub axle assembly.

3.4 Brake Caliper and Brake Disk Rotor

The brake disk rotor outer diameter is 245mm but when the brake caliper was fitted onto the brake disk rotor, it measured 145mm from the centre of the brake disk rotor to the outside of the brake caliper. Also, the brake caliper is 83mm in width with the brake pads in the centre of this measurement. This places major restrictions on how far into the road wheel the brake disc rotor and caliper can be placed. This was the single biggest factor in the problem of minimising the king pin offset and inclination. If the king pin offset is too large, then it makes the car hard to turn and gives too much feedback and road shock to the driver through the steering wheel.

3.5 Wheel Bearing and Seal

The wheel bearing and seal size dictate the size and to some extent the length of the axle. There also needs to be areas machined on the axle for the seals to run.

3.6 King Pin Axis Inclination and Offset

If a line is drawn through the middle of the suspension mounting points, the angle it makes to the vertical plane is the king pin inclination angle. See Figure 3.1.

If the line is extended until it touches the ground, the distance from this point to the middle of the road wheel is the king pin offset. The distance needs to be kept small

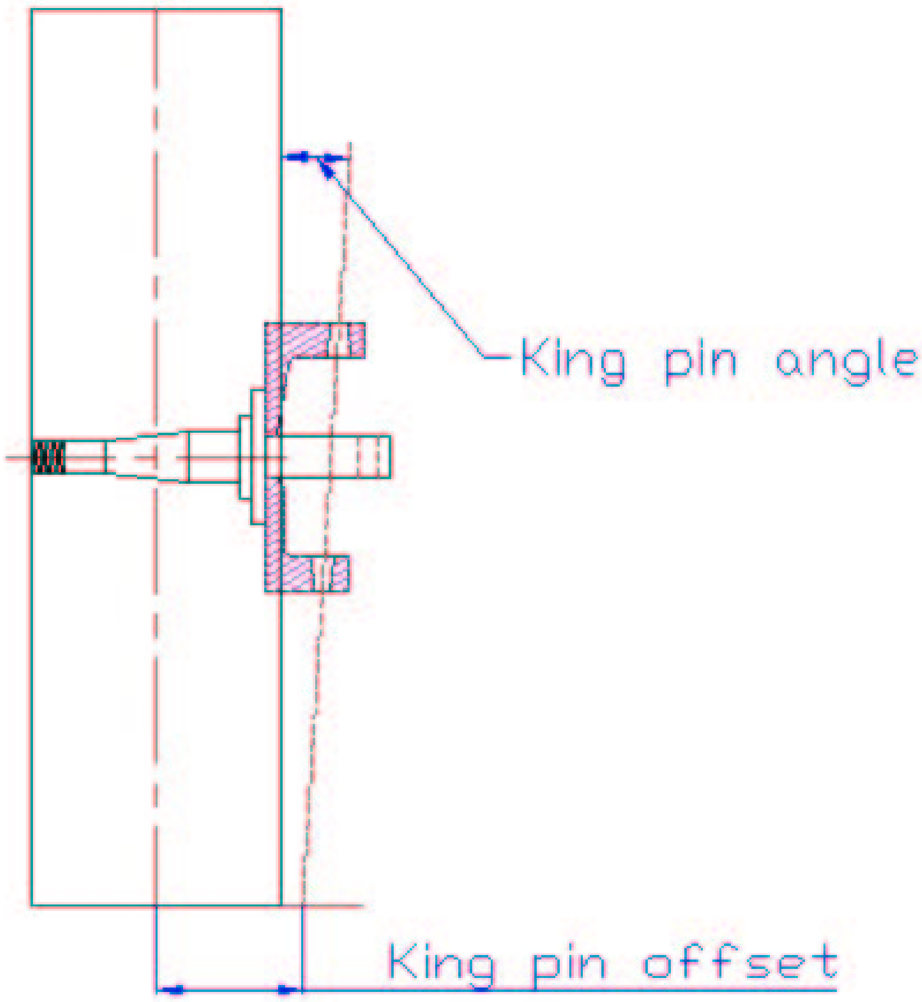


Figure 3.1: King pin angle and offset.

because if it is too large too much kickback is felt at the steering wheel if one of the road wheels hits a bump. If this distance is too small, there will be no feel in the steering system at all and the car will be very difficult to drive at the limit, as the driver will have no feedback on what is happening. It was decided to try to keep this distance to about 35mm and the final figure achieved was 36.25mm.

3.7 Material and Part Availability

Bearings, seals, rod ends, nuts and bolts are all commercially available from a variety of suppliers and are not expensive so none of these parts proved a problem to source. All bolts in the steering and suspension systems had to be high tensile as part of the rules for Formula SAE-A. After consultation with the University of Southern Queensland workshop, it was decided to manufacture the rest of the stub axle parts out of mild steel as they had suitable quantities already in stock. This kept the cost down as well. The steering rack proved more of a problem as most easily available motor vehicle steering racks have over three turns lock to lock. This is unsuitable for a Formula SAE-A race car. Also almost all standard motor vehicle racks have the pinion axis inclined. This is a major problem when trying to get the steering column connected to the pinion. It was eventually found that the Renault R5 used a centre point rack that has no pinion axis inclination. This would have been ideal but unfortunately it was found eventually that this motor vehicle was not sold in Australia.

3.8 Weight

Because this is a race car with limited power, low weight is very important. Unfortunately this has to be offset against the cost. Low weight usually means high cost as development time is extended and lighter material with the same strength usually cost more. In some areas it was decided to use a bit more material than was necessary, just to make sure the components did not fail. If more time was available, decreasing weight would be high on the list of priorities.

After discussion with the rest of the team, it was decided to make sure all the components were very strong, as we didn't want anything to break. Our highest need was for a car that worked first and foremost. It was felt that once we had a working car it would be much easier to then choose what areas needed the most improvement in the time available.

3.9 Cost

As we had a very limited budget it was necessary to keep costs to a minimum. There are strict cost controls in the Formula SAE-A rules and everything is budgeted to a strictly controlled formula. The calculated cost for the entire car should not exceed \$25000. The costing report for the steering parts is given in Appendix G.

3.10 Project Duration

Normal allocated time for a project is two semesters of an undergraduate degree course which is approximately forty weeks. This dissertation is due on the 28th October 2004.

Because the Formula SAE-A competition is scheduled from the 1st to 5th December more time is available for the completion and testing of the car. As it will take two days to travel to Melbourne for the competition, the final completion date for the car was set at the 28th November 2004.

Chapter 4

Rack and Pinion Steering Box

4.1 Chapter Overview

All the different design concepts for the steering box are presented here and the relative merits of each are evaluated.

The final concept that fulfils the requirements is selected and documented. The important features and functions are examined so the rack and pinion assembly can be integrated into the rest of the vehicle.

4.2 Rack and Pinion Design Concepts

4.2.1 Design Concept One

After consultation with various people and looking at the local wrecking yards it was initially decided that a Morris Mini rack would be our best option as a Mini rack only has approximately two and one quarter turns lock to lock. When the rack was to be ordered it was found that a quick ratio rack was available for only about $30 more. This only had one and one quarter turns lock to lock. After more consultation with experts at Road and Track in Ipswich, it was found that a Mini rack would not be suitable for our Formula SAE-A car as the pinion is offset twenty two and one half degrees from the perpendicular to the rack. It was also found that Mini racks are difficult to shorten. This concept was thus deemed unsuitable.

4.2.2 Design Concept Two

Design concept two was to manufacture a rack and pinion from scratch. This proved an unfeasible option when it was found that the USQ workshop would have difficulty manufacturing a pinion gear.

4.2.3 Design Concept Three

Design concept three was the purchase of a suitable complete rack and pinion steering box. Road and Track in Ipswich helped as much as possible and tried to supply us with a complete centre point rack with a straight pinion but did not have anything suitable for under $1200. This was much more than our budget would allow. Design concept three was unsuitable.

4.2.4 Design Concept Four

Design concept four was to buy just the straight pinion and a rack from Road and Track, and design and make our own housing to suit our car. This proposal was quashed when Road and Track could not supply the needed parts at an affordable price.

4.2.5 Design Concept Five

Road and Track's last suggestion was to use a Renault R5 rack and pinion steering box as these are centre point and have a straight pinion. After looking around we found that the nearest Renault wrecker was in Drayton. Barry, the proprietor of Peugeot and Renault Parts and Service, was most helpful but we were quickly informed that the Renault R5 was never sold in Australia, and it would be difficult if not impossible to source parts for. Design concept five was not feasible.

4.2.6 Design Concept Six

A lot of vehicles in the Peugeot and Renault wrecking yard were investigated and it was found that the Renault R10 has a small light rack with a pinion angle of only about six degrees. This is a much lower value than the other vehicles inspected at all the local wrecking yards. Barry allowed me to take one of the Renault R10 steering boxes to the USQ workshop and they informed me that it would be possible to modify the rack to suit our Formula SAE-A car. After more discussion with Barry, the steering box was donated to us. The pinion angle was measured and then the steering box was dismantled.

Figure 4.1: Renault R10 rack housing. Note: Pinion is offset to far left hand end of housing.

4.3 Final Design Concept

Design concept six was the best option so the pinion angle was measured and then the steering box was dismantled. The rack was measured and after discussion with the USQ workshop staff it was decided to modify the rack to suit our purpose. The rack gear was originally very close to one end of the rack, so the rack had to be shortened at one end and lengthened at the other by eighty millimetres. See Figure 4.2

Figure 4.2: Renault R10 rack after modification.

As can be seen in Figure 4.3, a housing was then designed and manufactured out of mild steel. This process was simplified somewhat by using the original seals and bearings. The completed rack and pinion steering box is shown in Figure B.1. The rack moves 32mm for one revolution of the pinion. With a steering arm length of 37.5mm the required turning angle of 35 degrees for the inside road wheel is attained by turning the steering wheel through an angle of 320 degrees.

Figure 4.3: Complete housing for modified rack and pinion.

4.4 Chapter Summary

Because the rack gear ration was quite small the steering arms had to be very short to get the required turning ratio at the road wheels. Values that were initially decided on for the project were:

- turning angle for the steering wheel of approximately 310 to 320 degrees;
- a maximum turning angle of 35 degrees; and
- a minimum turning circle radius of approximately 4 metres.

Actual theoretical value that were achieved are:

- turning angle for the steering wheel of 320 degrees;
- a maximum turning angle of 35 degrees for the inside road wheel; and
- a minimum turning circle of radius 3.75 metres.

With very careful design it is possible to design an accurate steering system that should perform very well in our Formula SAE-A race car.

The steering assembly has been manufactured at the USQ workshop and when the suspension system has been completed, the system will be installed and tested.

Chapter 5

Stub Axle Assembly

5.1 Chapter Overview

All the different design concepts for the stub axle are presented here and the relative merits of each are evaluated.

The final concept that fulfils the requirements is selected and documented. The important features and functions are examined so the stub axle assembly can be integrated into the rest of the vehicle.

5.2 Design Concept Evaluation Criteria

The three design concepts were evaluated using the following criteria:

- ability to minimise stress concentrations;
- low weight for strength;
- ease of manufacture;
- volume of material used;
- minimal amount scrub radius achievable; and

Figure 5.1: Typical motor vehicle stub axle design.

- location and size of heat affected zone.

5.3 Stub Axle Design Concepts

5.3.1 Design Concept One

Design concept one had conical shaped uprights for lightness and strength, and the axle was to be welded to the upright. This design most looked like a normal car stub axle assembly. This stub axle would require a separate hub which would incorporate the bearings, seals and wheel studs.

The advantages of concept one are:

- the cone shape makes the upright strongest at the points of maximum stress; and

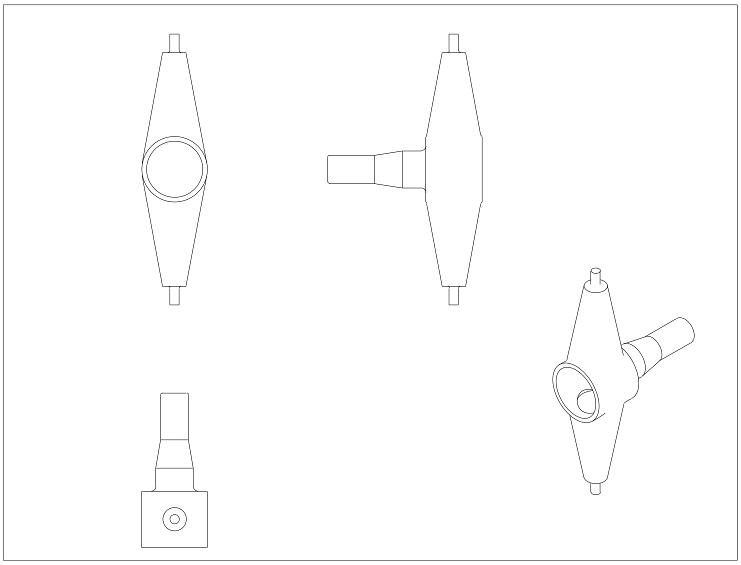


Figure 5.2: Design concept one.

- the hollowed out cone is light for its strength.

The disadvantages of concept one are:

- the cone shape is very difficult to manufacture and then weld accurately to the bearing sleeve;
- the initial volume of material required to get the conical shapes machined is quite large;
- the cone shape increases the amount of scrub radius beyond what we were trying to achieve; and
- the axle had to be welded to the upright and this would cause a large heat affected zone in the upright and axle which was deemed unsuitable.

The disadvantages of this design outweighed the advantages of the design, so concept one was deemed unfeasible.

5.3.2 Design Concept Two

Design concept two also had conical shaped uprights and the axle was removed to fix the heat affected zone problems. The axle is now incorporated into the hub and would be manufactured out of one piece of material, so no welding is necessary on the axle/hub piece.

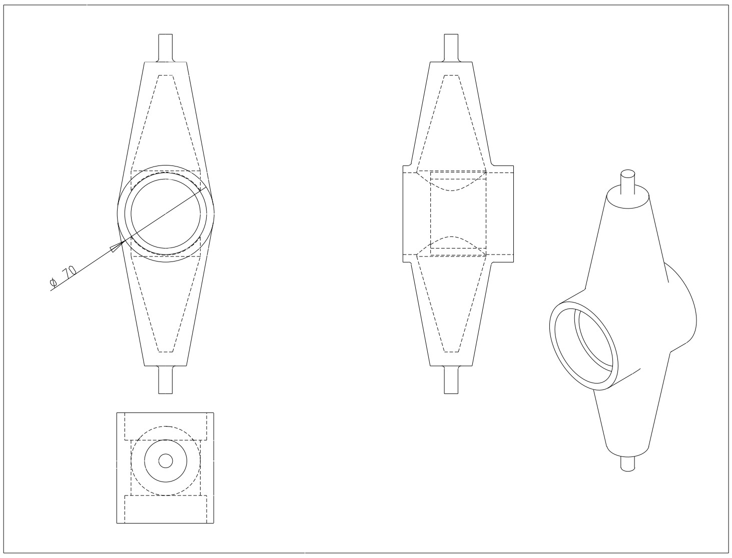


Figure 5.3: Design concept two.

The advantages of concept two are:

- the cone shape makes the upright strongest at the points of maximum stress;
- the hollowed out cone is light for its strength; and
- having the axle as a separate part removes the need to weld the axle on to it therefore reducing the heat affected zones.

The disadvantages of concept two are:

- the initial volume of material required to get the conical shapes machined is quite large;
- the cone shape is very difficult to manufacture and then weld accurately to the bearing sleeve; and

- the cone shape increases the amount of scrub radius beyond what we were trying to achieve.

The disadvantages of this design outweighed the advantages of the design, so concept two was deemed unfeasible.

5.4 Final Design Concept

The third and final design had parallel sides on the upright section to alleviate the problems with construction. It also minimizes the amount of material initially needed for the manufacture of this part.

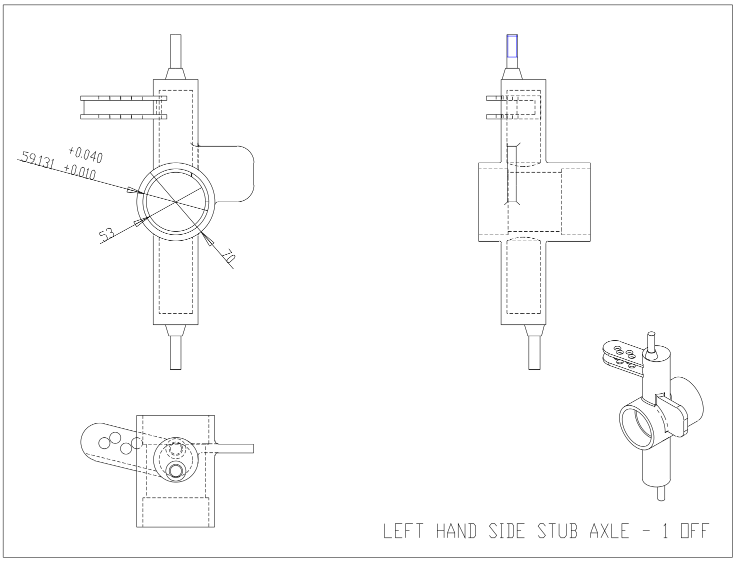


Figure 5.4: Final design concept.

Having the upright's sides parallel also decreased the amount of scrub radius needed. This was important as it was felt that the sixty-five millimetres of scrub radius with the conical uprights was too much and would make the steering too heavy. This was finally reduced to 36.25 millimetres after incorporating 4.7 degrees of kingpin inclination. The original design specifications that were decided on, aimed to get the king pin offset between 30mm and 35mm.

The hub/axle section incorporates the brake disk rotor and steering arm mounts, and space for the seals, bearings and wheel studs. It was manufactured out of one piece of material to alleviate the heat affected zones. This was important to avoid changing the material properties of the metal. The left hand side is a mirror image of the right hand side. Slots were cut out between the wheel studs to minimise weight without reducing strength. For the same reason material was removed from between the mounting hole for the brake disk rotor. The studs will be screwed in and held with 'loctite' compound and possibly a tack of weld if necessary. Through type studs proved very difficult to incorporate as the back of the axle part has the six protrusions for the mounting of the disk brake rotor and it was impossible to line up the four wheel studs into the available gaps. See Figure5.5.

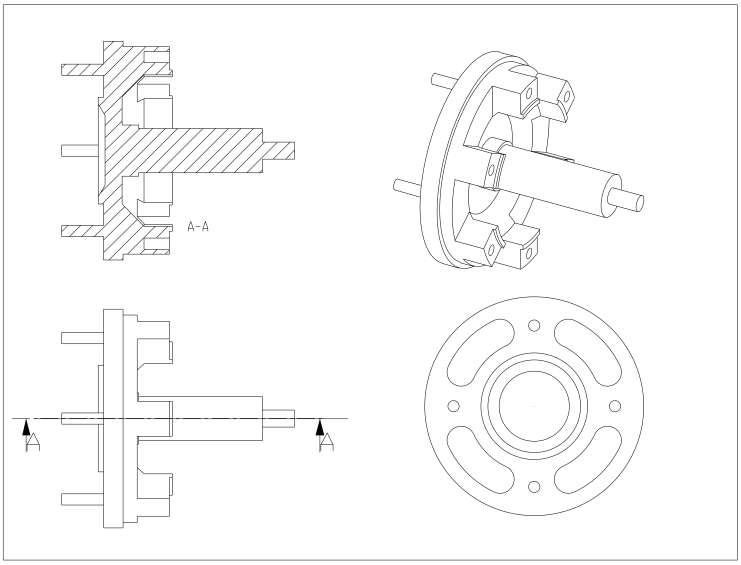


Figure 5.5: Axle proposal.

The initial design of this proposal has been changed substantially to accommodate various other components. The upright and axle were originally designed before we had any brake calipers and brake disk rotors. The sizes of these components was not finalized until the end of May when a motor bike was purchased and dismantled.

The decision was made to use the rear brake disk rotors and the front calipers off the motor bike for our front brakes. The front brakes disks off the bike were far to large to fit into our chosen wheels. The brake calipers size also necessitated an increase in the scrub radius. Once the brake calipers were drawn into the initial design, it was found that they would not fit, so the design was modified to fix the problem.

5.5 Material Selection

The criteria used in the selection of the material for the manufacture of these components:

- had to be easily machinable on the machines in the USQ workshop, as there was quite a bit of machining necessary;
- had to be relatively inexpensive;
- had to be easily available; and
- had to be relatively light for its strength.

After discussion with the USQ workshop it was found that only mild steel fitted all the criteria as it was the only material they had available at the time.

Important physical properties for mild steel are listed in Table 5.1.

Table 5.1: Physical Properties for Mild Steel

Material Property	Magnitude
Modulus of Elasticity	200GPa
Tensile Strength	455MPa
Yield Strength (tension)	250MPa
Ductility, percent elongation in 50mm	23
Poisson's Ratio	0.29

5.6 Finite Element Analysis

5.6.1 Loading for the Upright

The load for the upright was derived after working out the forces that was expected at the road wheels and then working out the moments about the bearings. This moment could then be applied at the points where the suspension was to be mounted. The aim is for the car to weigh approximately 300 kilograms and it is expected to generate about one times the force of gravity in cornering. As stated previously these forces were increased to give a suitable factor of safety. It was determined that a cornering force of two times the force of gravity and a weight force of 350 kilograms would be suitable. It has to be remembered that at no time should all the vehicle's weight be on the front wheels only.

After applying the weight force in the centre of the tyre and the cornering force across the tread surface of the tyre, and working out the moments around the wheel bearing centres, it was possible to work out the forces on the top and bottom suspension mounting points to be approximately 3500N and 10500N respectively.

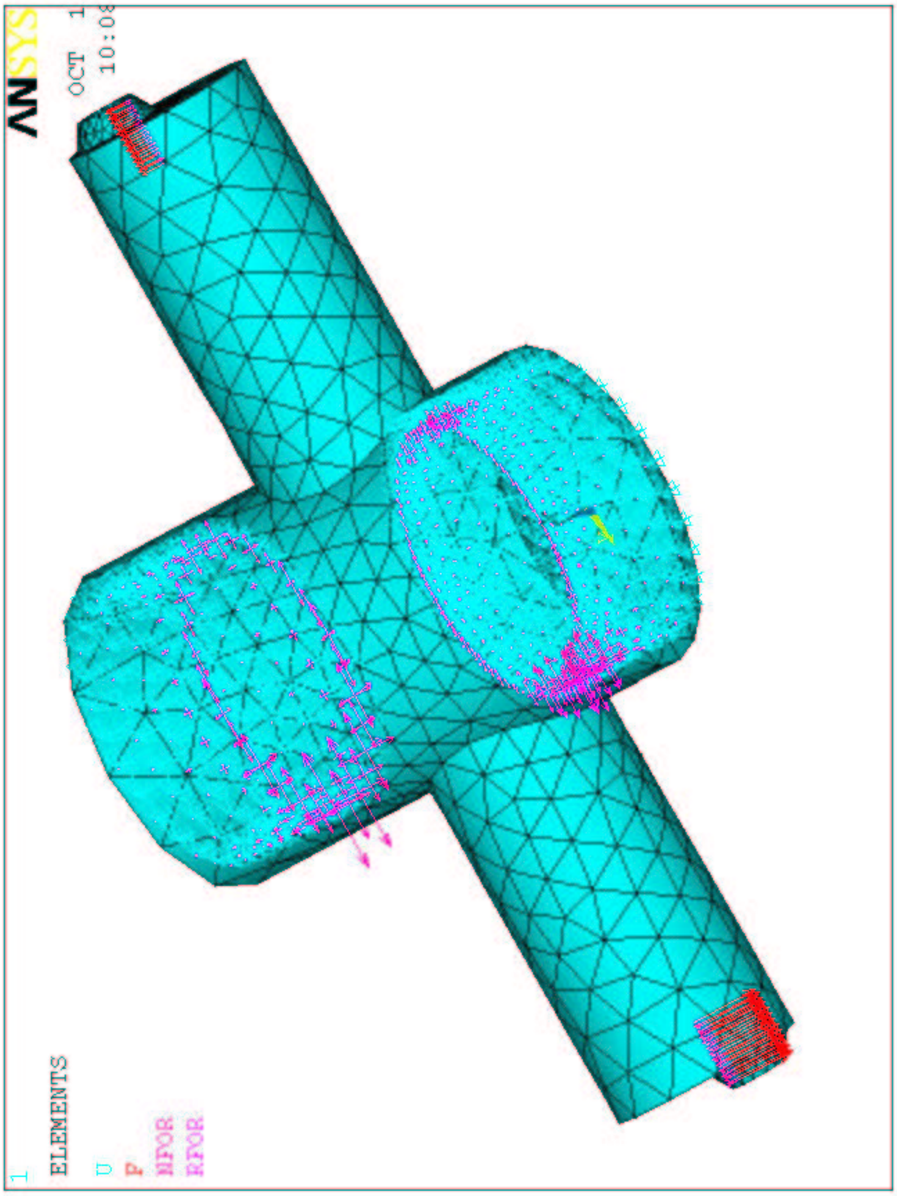


Figure 5.6: Load case for FEA on upright.

These forces were applied to the model at the suspension mounting points using thirty-six nodes to become 97.22N/node at the top and 291.67N/node at the bottom.

5.6.2 Loading for the Axle

The load for the axle was derived after working out the forces that was expected at the road wheels and then working out the moments about the bearings. This moment could then be applied at the points where the suspension was to be mounted. The aim is for the car to weigh approximately 300 kilograms and it is expected to generate about one times the force of gravity in cornering. As stated previously these forces were increased to give a suitable factor of safety. It was determined that a cornering force of two times the force of gravity and a weight force of 350 kilograms would be suitable. It has to be remembered that at no time should all the vehicle's weight be on the front wheels only.

After applying the weight force in the centre of the tyre and the cornering force across the tread surface of the tyre, and working out the moments around the wheel bearing centres, it was possible to work out the forces at the top and bottom point of the mounting surface for the road wheel. This was 12750N for each point. This force was applied at 15 nodes to give a force of 850N/node.

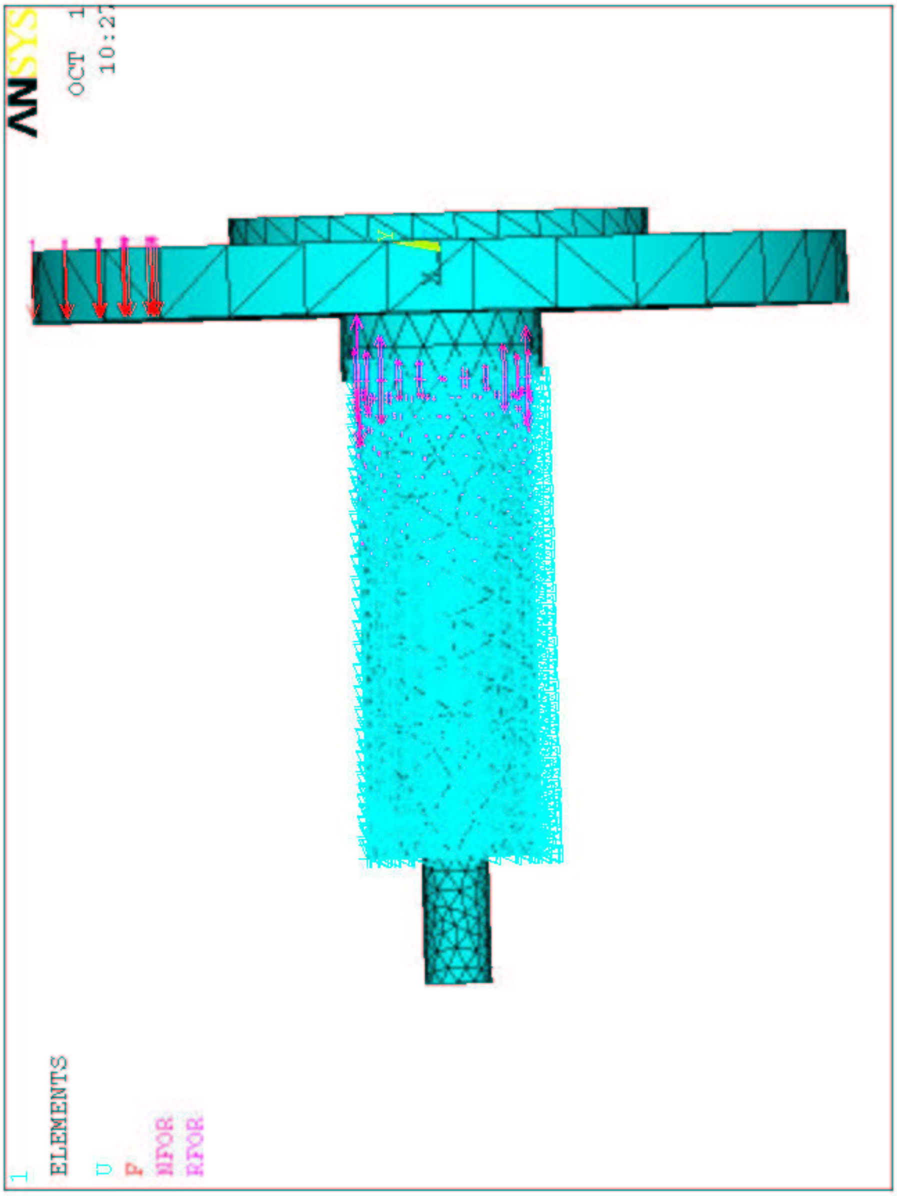


Figure 5.7: Load case for FEA on axle.

5.6.3 Finite Element Analysis Theory

Comments on the finite element are made with reference to Appendix C and D - *Finite Element Analysis Results for Upright* and *Finite Element Analysis Results for Axle.*

All comparisons made used the Von-Mises energy of distortion theory. In general, Von-

Mises energy of distortion theory is used for steel. This theory is based on the fact that any elastically stressed material undergoes a slight change in shape and/or volume. This energy is stored in the material as elastic strain energy. When the materials capacity to absorb energy is exceeded yielding (failure) has occurred. It is usual to determine an "equivalent stress" σ_e that represents the value of uniaxial tensile stress that would produce similar levels of distortion.

$$\sigma_e = \frac{\sqrt{2}}{2}[(\sigma_2 - \sigma_1)^2 + (\sigma_3 - \sigma_1)^2 + (\sigma_3 - \sigma_2)^2]^{1/2} \tag{5.1}$$

For the biaxial stress condition with σ_x,σ_y and τ_{xy} this reduces to:

$$\sigma_e = (\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2)^{1/2} \tag{5.2}$$

This equivalent stress σ_e is then compared with the shear stress at the onset of yielding in a uniaxial tensile test. (Snook 2003).

5.6.4 FEA Results for the Upright

The results for the finite element analysis of the upright are illustrated in Figure C.2. The maximum Von-Mises stress was 448MPa which is less than the maximum permissible stress (455MPa) as defined in Table 5.1.

5.6.5 FEA Results for the Axle

The results for the finite element analysis of the upright are illustrated in Figure D.2. The maximum Von-Mises stress was 394MPa which is less than the maximum permissible stress (455MPa) as defined in Table 5.1 but is still fairly close to being optimal.

5.7 Manufacture and Problems Encountered

The final design has been manufactured at the USQ workshop. There was no major problem machining the parts as the workshop staff had been shown the early concept drawings and had said that manufacture was a feasible proposition.

During assembly of the components, a couple of problems surfaced. There was not enough room between the brake disc and the upright to bolt the brake disc on with the standard bolts. This problem was rectified by countersinking the heads of cap-screws into the brake discs, so that the head of the cap-screw is flush with the edge of the brake disc.

Another problem was that the incorrect thread for the wheel studs had been cut into the axles because of incorrect specifications. It proved impossible to find any wheel nuts with a 12mm * 1.75mm thread, as the normal thread is 12mm * 1.5mm. This problem was rectified by purchasing some high tensile bolts with a long shank and the 12mm * 1.5mm thread on the end, and cutting a 12mm * 1.75mm thread on the other after machining the head of the bolts off.

A new problem surfaced when it was found that all the bolts used in the suspension and steering had to comply with Metric Grade M 8.8 specifications. This necessitated machining off the mounting posts at the top and bottom, and cutting a thread into the upright and then using the correct grade bolts to attach the rod ends.

Yet another problem surfaced when the stub axle assembly was bolted to the suspension arms. The lower spherical rod end hit on the brake disk rotor. This problem was rectified at the same time as the hole is drilled and tapped to fit the high tensile attachment bolts. The bottom hole was drilled only 6mm from the centre of the upright not 10mm as specified on the original drawings. This decreased the king pin angle from 4.7 degrees to 4 degrees and increased the king pin offset from 30.25mm to 36.25mm. This should not affect the steering adversely as it is only slightly higher than the original design specifications agreed on. Also as Formula SAE-A race car is reasonably low in weight, the steering should not be too heavy. A model of the completed assembly is shown in Figure 5.8.

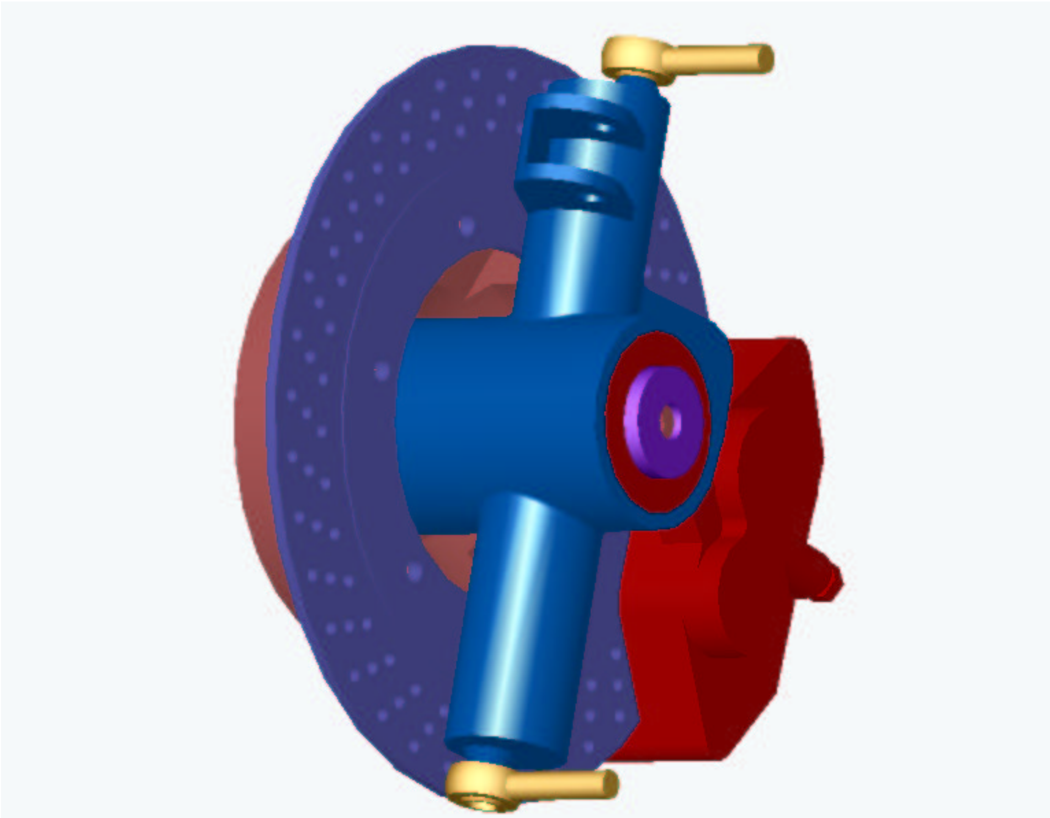


Figure 5.8: Model of final stub axle assembly.

5.8 Solid Models

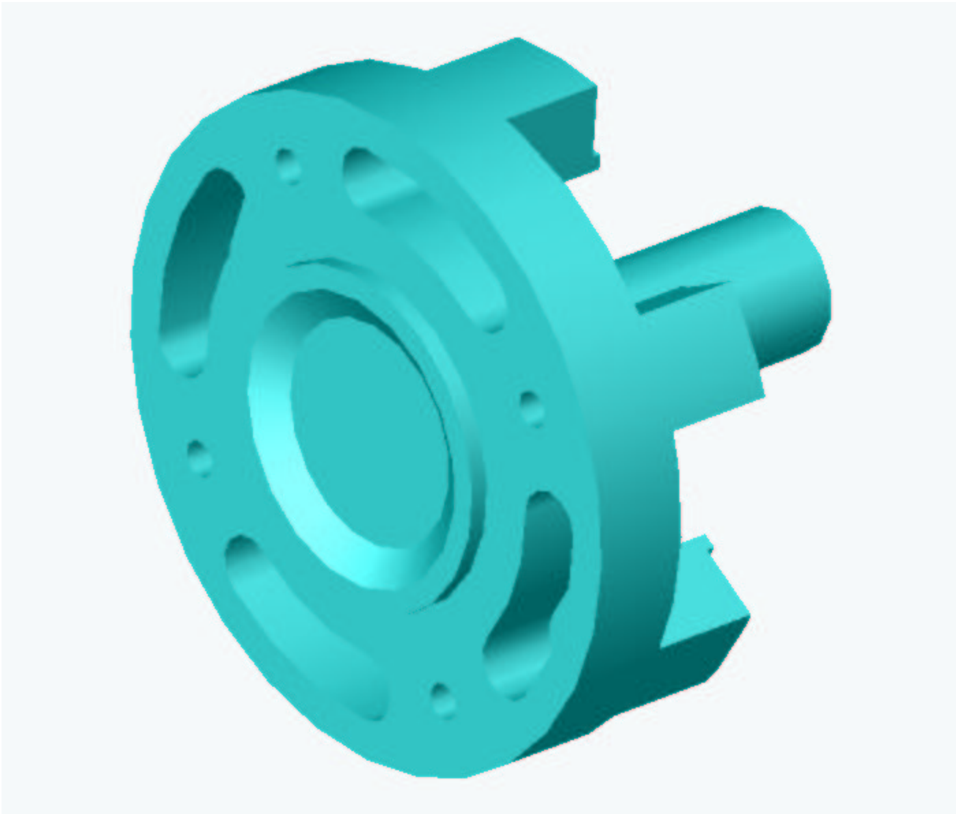


Figure 5.9: Isometric front view of final model of axle.

The front isometric view of the axle component illustrated in Figure 5.9 shows the following features:

- mating face for road wheel;
- wheel stud thread;
- locating lip for road wheel; and
- slots to minimise weight.

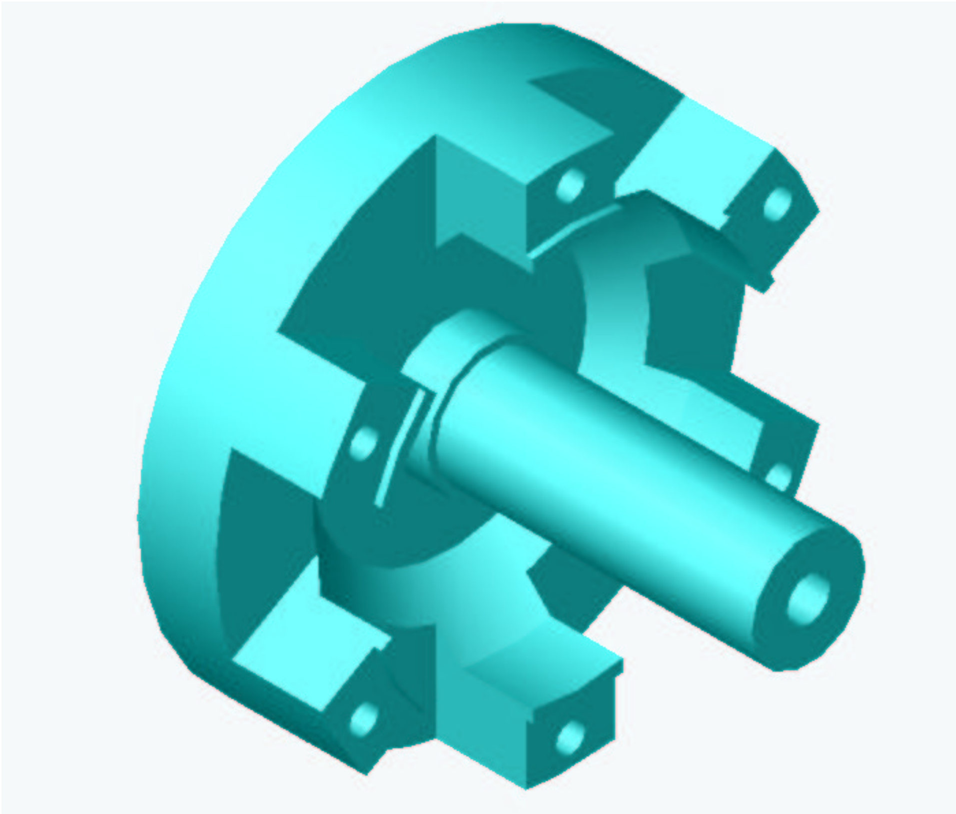


Figure 5.10: Isometric rear view of final model of axle.

The rear isometric view of the axle component illustrated in Figure 5.10 shows the following features:

- mating face and locating lip for brake disc rotor;
- seal running surface;
- inner bearing race locating surface; and
- cutouts between brake rotor mounting holes to minimise weight.

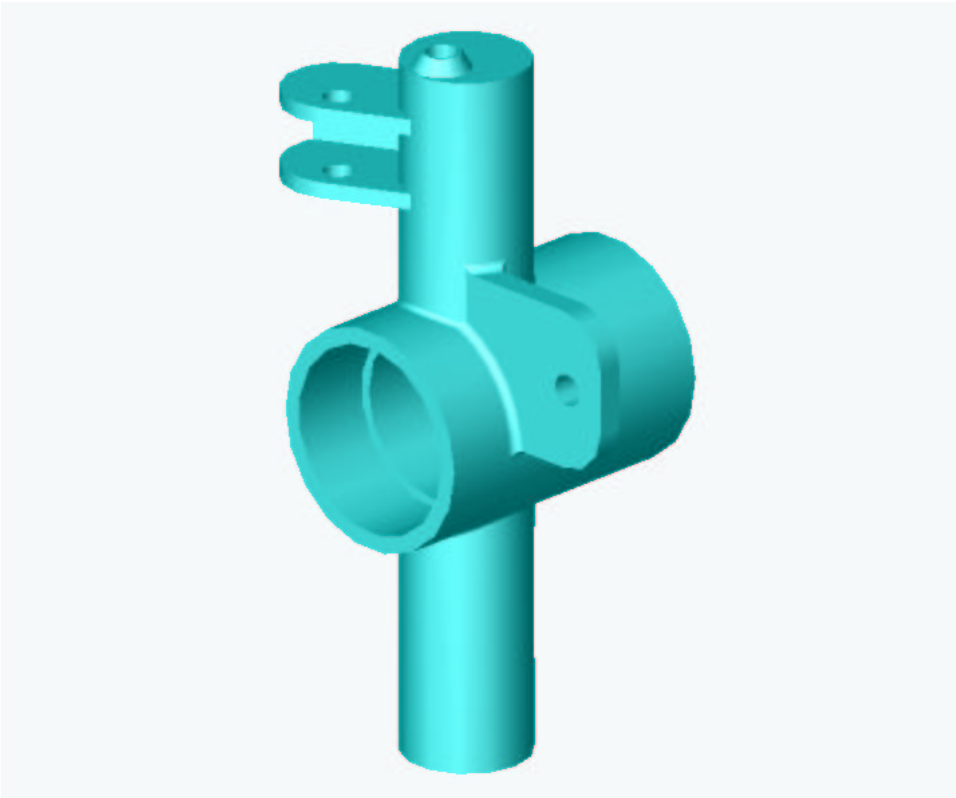


Figure 5.11: Right view of final model of left upright.

The right view of the upright component illustrated in Figure 5.11 shows the following features:

- steering arm;
- top suspension mounting point;
- one brake caliper mount; and
- inner wheel bearing outer race and seal locating surface.

Figure 5.12: Left view of final model of left upright.

The left view of the upright component illustrated in Figure 5.12 shows the following features:

- steering arm;
- bottom suspension mounting point;
- one brake caliper mount; and
- outer wheel bearing outer race and seal locating surface.

5.9 Chapter Summary

The design of the upright assembly was complicated and we had a few initial problems that were quite easily rectified. This system should give very good results in our Formula SAE-A car when testing starts.

The manufacture of all the components has been completed and will be assembled on the vehicle when the suspension system has been completed.

The original design specifications that were decided on for the project were:

- king pin offset of between 30mm and 35mm;
- a maximum turning angle of 35 degrees for the inside road wheel;
- use easily available bearing and seal components; and
- incorporate all mounts for the suspension and brake disk and caliper.

Actual specifications achieved were:

- king pin offset of between 36.25mm;
- a maximum turning angle of 35 degrees for the inside road wheel;
- design uses Holden B bearings and seals which are readily available; and
- all mounts for the suspension, brake disk rotors and calipers are incorporated into the design.

Chapter 6

Rear Upright

6.1 Chapter Overview

All the different design concepts for the rear upright are presented here and the relative merits of each are evaluated.

The final concept that fulfils the requirements is selected and documented. The important features and functions are examined so the rear upright assembly can be integrated into the rest of the vehicle.

6.2 Design Concept Evaluation Criteria

The two design concepts were evaluated using the following criteria:

- ability to minimise stress concentrations;
- low weight for strength;
- ease of manufacture;
- volume of material used;
- location and size of heat affected zone; and

- suitability to suspension design.

6.3 Upright Design Concepts

6.3.1 Design Concept One

Design concept one was very similar to the front upright apart from the differences at the wheel bearing mounting area and the fact that no king pin offset was needed.

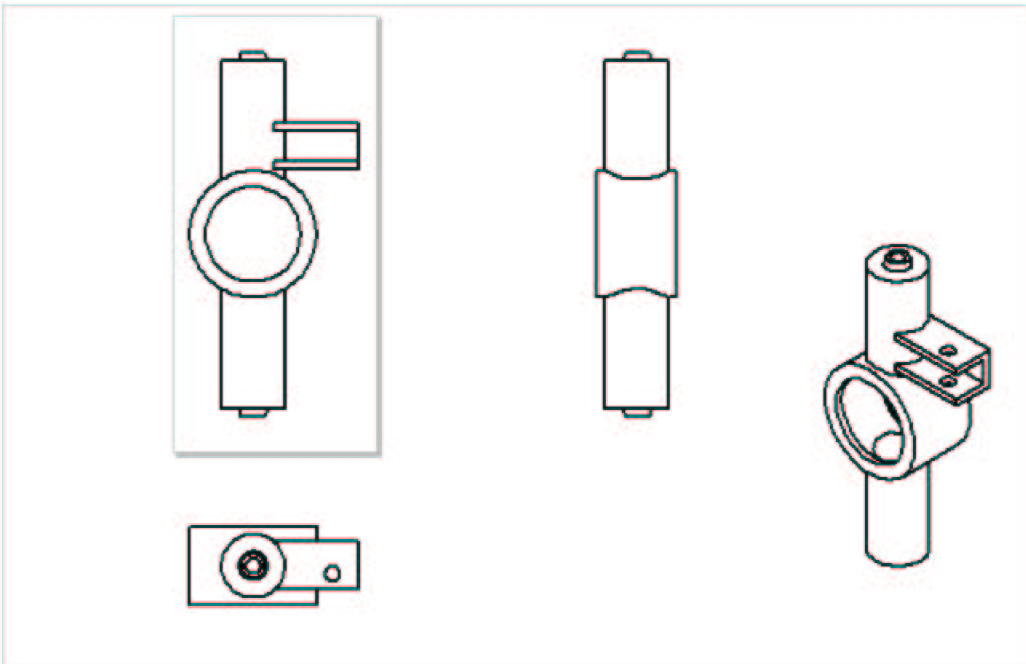


Figure 6.1: Design concept one.

The advantages of concept one are:

- the parallel sides of the upright make manufacture easier; and

- similarity to front upright for easy design, optimisation and manufacture.

The disadvantages of concept one are:

- not completely suitable to match with suspension design.

The disadvantages of this design outweighed the advantages of the design, so concept one was deemed unfeasible.

6.3.2 Design Concept Two

Design concept two was very similar to design concept one, except the top part of the upright was shortened significantly to better suit the suspension design. The suspension designer wanted to keep the suspension arms as close to parallel as possible.

The advantages of concept two are:

- the parallel sides of the upright make manufacture easier; and
- similarity to front upright for easy design, optimisation and manufacture; and
- better suits suspension design.

6.4 Final Design Concept

The second design concept was chosen as it fulfilled all our criteria. The left hand side is a mirror image of the right hand side. The design incorporates the housing for the wheel bearing which is the front wheel bearing from an AS Ford Telstar. This is because the front drive shafts out of the Telstar have been shortened to suit and are being used as the drive shafts in our vehicle. Brake caliper mounts were not necessary as the brake setup is placed inside the main chassis. The steering arm was required to stop the uprights from rotating around the vertical axis.

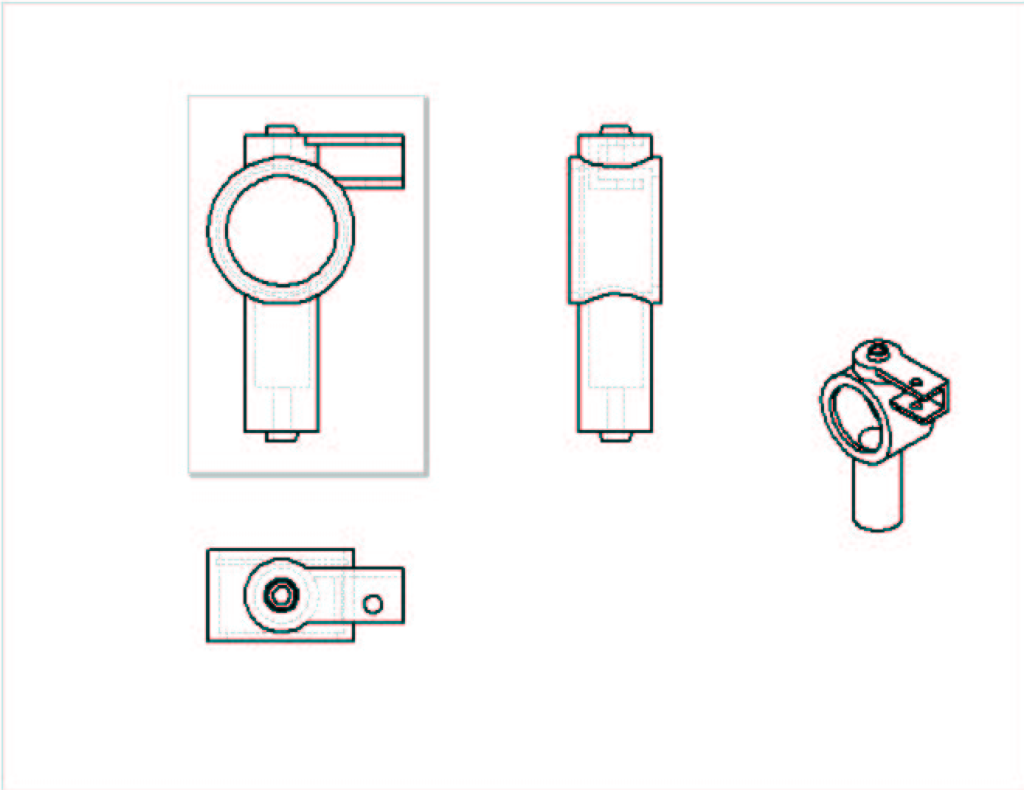


Figure 6.2: Design concept two.

6.5 Material Selection

The criteria used in the selection of the material for the manufacture of these components was the same as for the front upright:

- had to be easily machinable on the machines in the USQ workshop;
- had to be relatively inexpensive;
- had to be easily available; and
- had to be relatively light for its strength.

Again, after discussion with the USQ workshop it was found that only mild steel fitted all the criteria as it was the only material they had available at the time.

6.6 Manufacture and Problems Encountered

The final design has been manufactured at the USQ workshop. There were no major problems machining the parts as the workshop staff had been shown the early concept drawings and had said that manufacture was a feasible proposition. The drawings were also shown to Mr Chris Snook, the project supervisor, for checking before manufacture took place. The dimensions were also checked by the suspension designer to make sure they were suitable and kept the suspension arms parallel. No problems have surfaced at this point. A model of the completed assembly is shown in Figure 6.3.

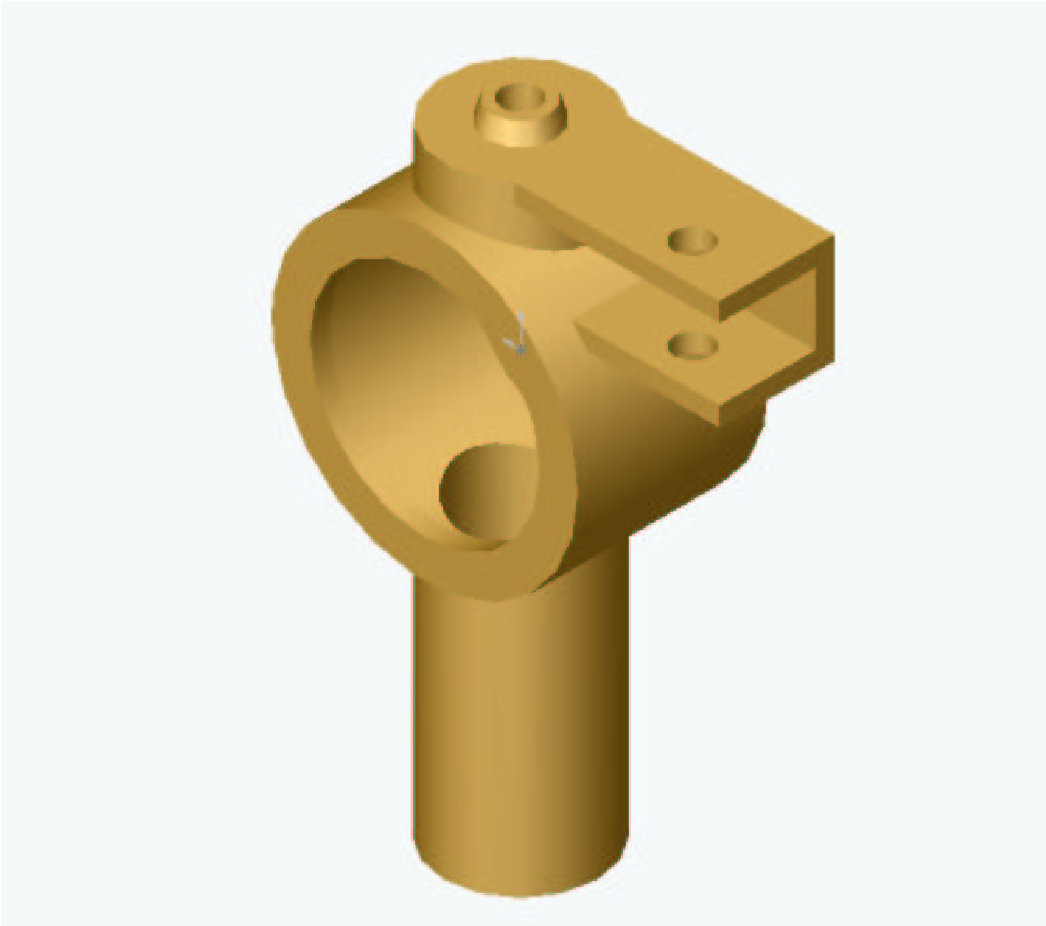


Figure 6.3: Model of the rear upright.

6.7 Chapter Summary

The design of the rear upright assembly was not as complicated as for the front as it was not necessary to design in any king pin offset or inclination. Also, as the rear brake was mounted inboard, no braking components or mounts had to be incorporated into the design. This component should help give good results in our Formula SAE-A car when testing starts.

Chapter 7

Design Integration

7.1 Chapter Overview

All the intricacies of integrating the steering system designed with the rest of the vehicle are presented here so the steering system can be integrated seamlessly.

7.2 Steering System

7.2.1 Steering Arm Angle

The angle that the steering arm is offset to get the Ackermann effect is critical for a well set up steering system. The initial value tried was about seventy-one and a half degrees. This was found after the chassis was drawn with the upright's in place at the correct place and a line was drawn through the steering arms to intersect at the centre of the rear axle. This is a very good starting point but after the wheels were turned on the model it was found that the turning centre for the car was always behind the rear axle. The only ways to correct this problem are to either move the steering rack rearward or increase the steering arm angle. The angle for the steering arms was trialled at different points in conjunction with the steering box location until a suitable location was found. The final value was set at sixty-three and a half degrees as can be seen in the production drawings in appendix E.

7.2.2 Rack and Pinion Steering Box Location

It was decided to place the steering box just above the top chassis bars at the front of the vehicle. It would have been easier to get good geometry with it below the chassis rails but there was physically no room there as this is were the drivers legs will be. This fixed the steering box location in the vertical plane and in the horizontal plane the location was found after drawing a solid model and turning the wheels in the model and checking the geometry until a suitable solution was found. This was done in conjunction with the steering arm angle as both these have a major impact on the position of the turning centre of the vehicle. The final location for the steering box has been set at 50 millimetres behind the front axle centre line. This is measured from the centre line of the rack. The results for all values trialled are set in Table 7.1. A drawing of the system with the inside wheel at an angle of 30 degrees can be seen in Figure 7.1.

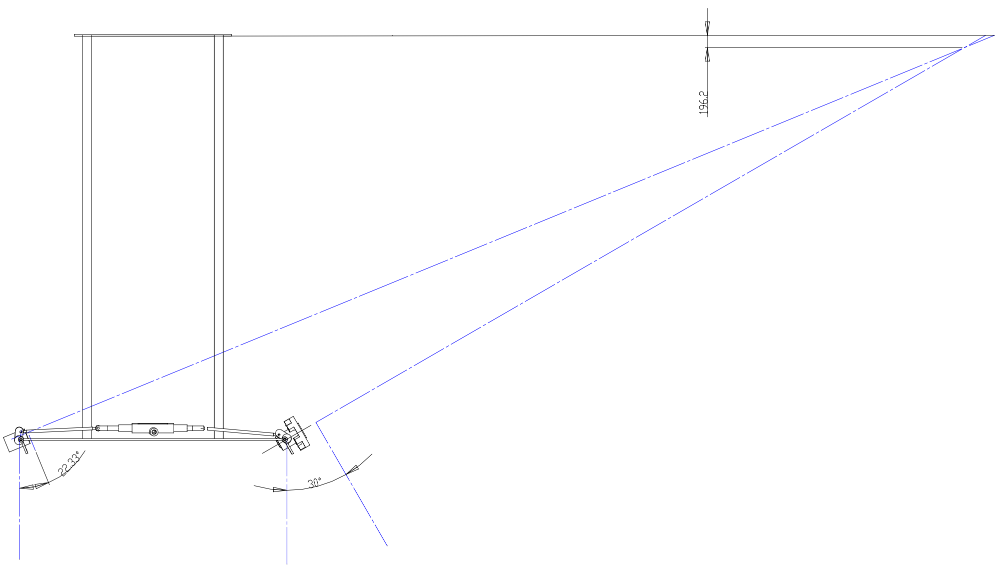


Figure 7.1: Model showing inside wheel at 30 degrees. Note also steering box location 50mm behind front axle line.

Table 7.1: Results of Steering Box and Steering Arm Angle Trials.

Distance behind front axle (mm)	Steering arm angle (degrees)	Distance from rear axle line (mm) at inside wheel turning angle		
		10 degrees	20 degrees	30 degrees
60	64	508.8 B	67.6 B	190.6 F
	63	53.6 B	111.3 F	306.8 F
55	74		1546.3 B	
	73		1232.4 B	
	71		849.4 B	
	68		478.4 B	
	66	801.1 B	311.4 B	32.7 B
	64	470.2 B	92.6 B	151.4 F
	63	27.3 B	77.6 F	266.1 F
52	63.5	490.0 B	90.2 B	156.5 F
50	62	160.9 F	150.3 F	311.1 F
	61.5	155.8 B	95.8 F	305.8 F
	62.5	144.8 F	116.6 F	276.7 F
	63	140.6 F	85.8 F	243.2 F
	64	175.5 B	69.5 B	132.0 F
	63.5	***14.1 B***	***20.8 B***	***196.2 F***
48	62	297.5 B	15.5 F	241.2 F
40	64	345.8 B	211.8 B	7.24 F
	63	517.1 B	177.5 B	69.2 F
30	63	423.4 B	259.7 B	31.4 B
	62	656.3 B	234.4 B	28.4 F

NOTE: Values for distance with letter B are behind rear axle centreline and values with letter F are forward rear axle centreline. The values in boldface italics are the final values chosen for the steering arm angle and distance from front axle centreline.

As can be seen the turning centre is very close to the rear axle centreline until after twenty degrees turning lock on the inside wheel. This should prove to be a very good setup. The final model is shown in Figure 7.2. The steering rack can be seen fifty millimetres behind the axle line and the steering arms are at a sixty-five and one half degree angle to the upright. Note: the suspension design shown is not the final design but the wheels location will be unchanged in relation to the chassis.

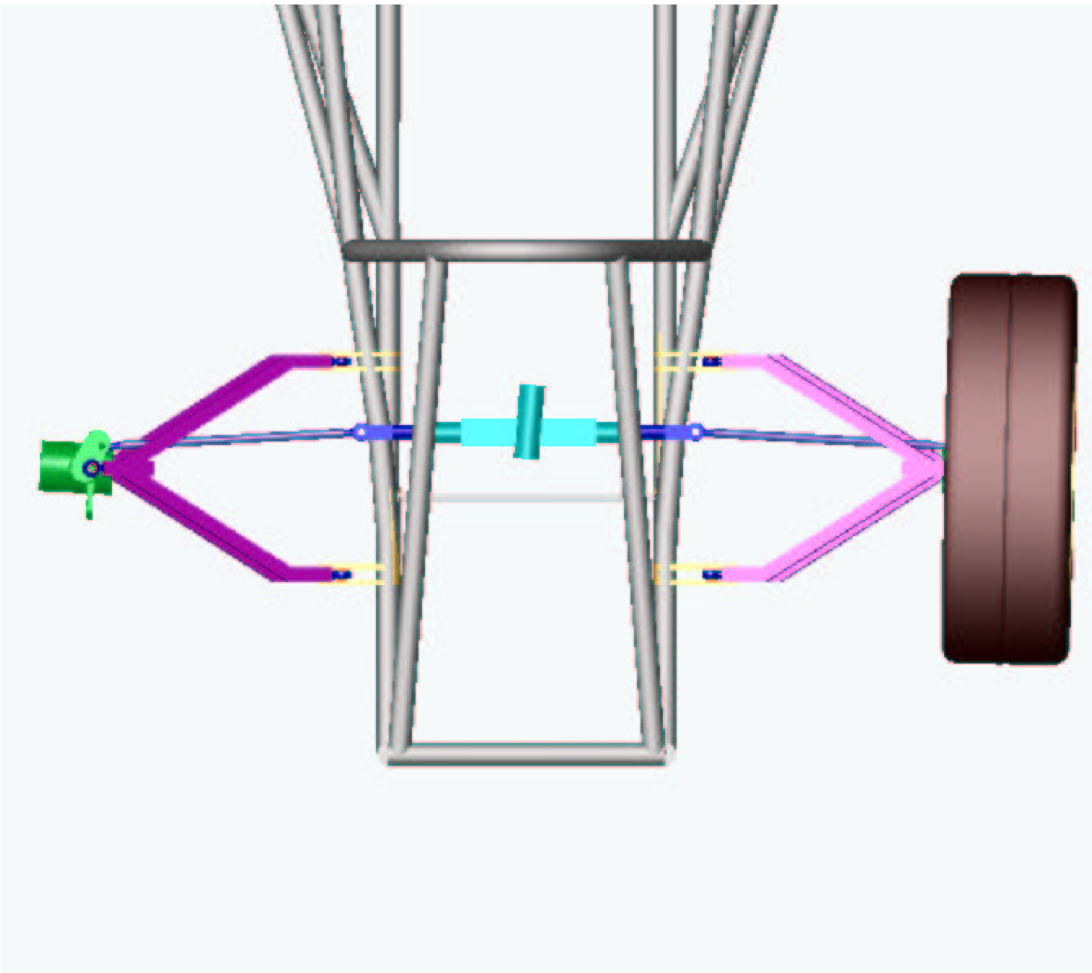


Figure 7.2: Final model of the steering system.

7.2.3 Tie-Rod Length

After the steering box location and the steering arm location were finalised it was a simple matter to find the tie-rod length. This length was finalised from the solid model at 345mm from the centre of one spherical rod end to the centre of the other spherical rod end.

7.3 Testing

It is planned to test the geometry of the steering system and find if the solid model and the real components give the same results. The method to be used is:

- mount laser pointing devices to each front wheel perpendicular to the wheels;
- turn the steering wheel at an angle from straight ahead;
- place a moveable white-board or similar at the intersection point of the two laser beams;
- measure the distance fore or aft of the rear axle of the intersection point;
- repeat this for various other steering angles between zero and thirty-five; and
- compare the results with the results from the solid model.

Once the car is operational, different settings of toe-in and toe-out need to be tested to find the optimum setting. Because this is a rear drive car it is expected a small amount of toe-in will be ideal. There will be a large amount of adjustment available, as the tie-rods incorporate a left hand threaded spherical rod end on one end and a right hand threaded spherical rod end on the other. To adjust toe-in will be a simple matter of loosening the lock nuts on the spherical rod ends and turning the tie-rod arm the required amount and then doing up the locknuts. Camber and castor will be adjustable on the suspension arms so a good setup should be found if enough testing is carried out.

7.4 Chapter Summary

This part of the project proved to be very time consuming as the values for steering arm angle and steering box position were generated three times because of necessary changes made to the suspension. It was considered though, that this was time well spent because if these values are incorrect it will mean the car will steer and handle extremely badly. As stated previously, a good handling car is an absolute necessity in the Formula SAE-A competition as power for all cars is quite even because of the 20mm restrictor in the intake system.

The model shows that from zero degrees turning angle to until twenty degrees turning angle of the inside wheel, the turning centre stays very close to the back axle line.

Chapter 8

Technical Specifications

8.1 Chapter Overview

So that the designed parts could be manufactured, technical specifications were produced. These include:

- production drawings;
- welding specifications;
- finish; and
- weight.

8.2 Production Drawings

The production drawings were automatically generated from the solid models and can be found in Appendix E - *Production Drawings*. A description of the drawings and the drawing number are tabulated in Table 8.1.

Table 8.1: Summary of Production Drawings.

Drawing Number	Description of Drawing
1	Front Axle End Cap
2	Front Axle/Hub
3	Left Hand Front Upright
4	Right Hand Front Upright
5	Rack Housing
6	Rack Housing Cover
7	Pinion Bush
8	Threaded ends for Tie Rods
9	Left Hand Rear Upright
10	Right Hand Rear Upright
11	Front Tie Rod Assembly

8.3 Welding Specifications

Minimal welding was required during the manufacture of the uprights, rack and rack housing. No welding was required for the manufacture of the axle, axle end cap, rack housing cover and pinion bush. All the welding connections were performed between mild steel.

The weld sizes are specified on the production drawings in Appendix E - *Production Drawings*.

8.4 Finish

All the parts will be painted with enamel undercoat and enamel topcoat. This is necessary not only to improve the look of the parts but also to minimise corrosion.

8.5 Weight

The weight of all the components was determined using ProEngineer. The weights of the parts are as follows in Table 8.2:

Table 8.2: Summary of Weight of Individual Components.

Component	Weight (KG)
Axle End Cap	0.042
Axle/Hub	3.004
Left Hand Front Upright	1.93
Right Hand Front Upright	1.93
Rack Housing	1.271
Rack Housing Cover	0.041
Pinion Bush	0.005
Threaded Ends for Tie Rods	0.116
Left Hand Rear Upright	1.144
Right Hand Rear Upright	1.144
Tie Rod Assemblies	0.148

Chapter 9

Conclusion

9.1 Achievement of Objectives

The front stub axle assembly and steering rack have been constructed at this stage. The placement of the rack and the steering arm angle have been worked out and the rear uprights have been designed and are ready for manufacture. The complete manufacture and testing of the final design has been hampered because of delays in the suspension design. It is hoped that this will be completed shortly. The majority of the tasks listed in the project specification have been achieved. A complete test of all components is planned when the vehicle's construction is completed.

Engineering principles show that all components have adequate strength.

9.2 Recommendations

All components are expected to be tested when the vehicle's construction is completed. This is expected to be soon. Lighter and more specialised materials could be used in the design to good effect if time and money permit. Further research needs to be carried out in this area to maximise the strength to weight ratio.

9.3 Concluding Remarks

This project has been both challenging and rewarding. Mistakes have been made but much has been learned from these. When testing does take place, it will be extremely interesting to see how accurately modelling is replicated by the final product. Testing procedures and evaluation criteria need to be derived before the testing actually takes place.

The author is particularly keen to see this project completed and tested because of a keen interest in motor vehicles.

References

Crouse, W. H. & Anglin, D. L. (1955), *Automotive Chassis and Body*, Automotive Technology Series, fifth edn, Mcgraw-Hill, United States of America. This is a full BOOK entry.

Demeis, R. (2003), 'Bmw opts for hybrid steering system', *Design News* **58**, 46+. This is a full ARTICLE entry.

Dixon, J. C. (1996), *Tyres, Suspension and Handling*, second edn, Arnold - A member of the Hodder Headline Group, 338 Euston Road London NW1 3BH U.K. This is a full BOOK entry.

Ertas, A. & Jones, J. C. (1996), *The Engineering Design Process*, second edn, John Wiley and Sons, Inc, United States of America. This is a full BOOK entry.

Garrett, T. (2001), *The Motor Vehicle*, thirteenth edn, Butterworth Heinemann, Stanford, California. This is a full BOOK entry.

Heisler, H. (1997), *Vehicle and Engine Technology*, Vol. 1, Edward Arnold, 338 Euston Road, London, Great Britain. This is a full BOOK entry.

Jones, B. (2004), *Suspension and Steering Systems Operation*, World Wide Web, http://www.autoshop-online.com/auto101/susp.html. current 28th March 2004.

Ludvigsen, K. (2004), 'Changing directions', *Motor Trend* **56**, 72+. This is a full ARTICLE entry.

Nice, K. (2004), *How Car Steering Works*, World Wide Web, http://www.howstuffworks.com/How Car Steering Works.htm. current 28th March 2004.

Snook, C. (2003), *Computational Mechanics in Design*, first edn, Distance Education Centre, The University of Southern Queensland, Toowoomba, Queensland. This is a full BOOK entry.

Appendix A

Project Specification

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR:	**Leslie Wayne RAYNER**
TOPIC:	Steering Design for SAE-A Racer
SUPERVISOR:	Mr. Chris Snook
SPONSORSHIP:	Faculty of Engineering and Surveying

PROJECT AIM: To provide a suitable steering system for a Formula SAE-A racer, to enable it to be driven successfully at the Formula SAE-A competition.

PROGRAMME: **Issue A, 22 March 2004**

1. Research information on currently used automotive steering systems and all their variations.
2. Research the existing rules and restrictions for Formula SAE-A racing car steering design.
3. Critically evaluate existing alternatives for steering design.
4. Develop preliminary design of the chosen steering system.
5. Integrate this design with the overall vehicle design.
6. Analyse and optimize design of the steering system.

As time and resources permit:

7. Manufacture and install prototype into Formula SAE-A racer and evaluate.
8. Test and obtain feedback from drivers and modify as needed.

AGREED: ______________ (student) ______________ (supervisor)

(dated) 23 / 3 / 04

Appendix B

Photographs

The following are photographs of the completed steering components.

Figure B.1: Completed rack and pinion steering box.

Figure B.2: Front axle and upright.

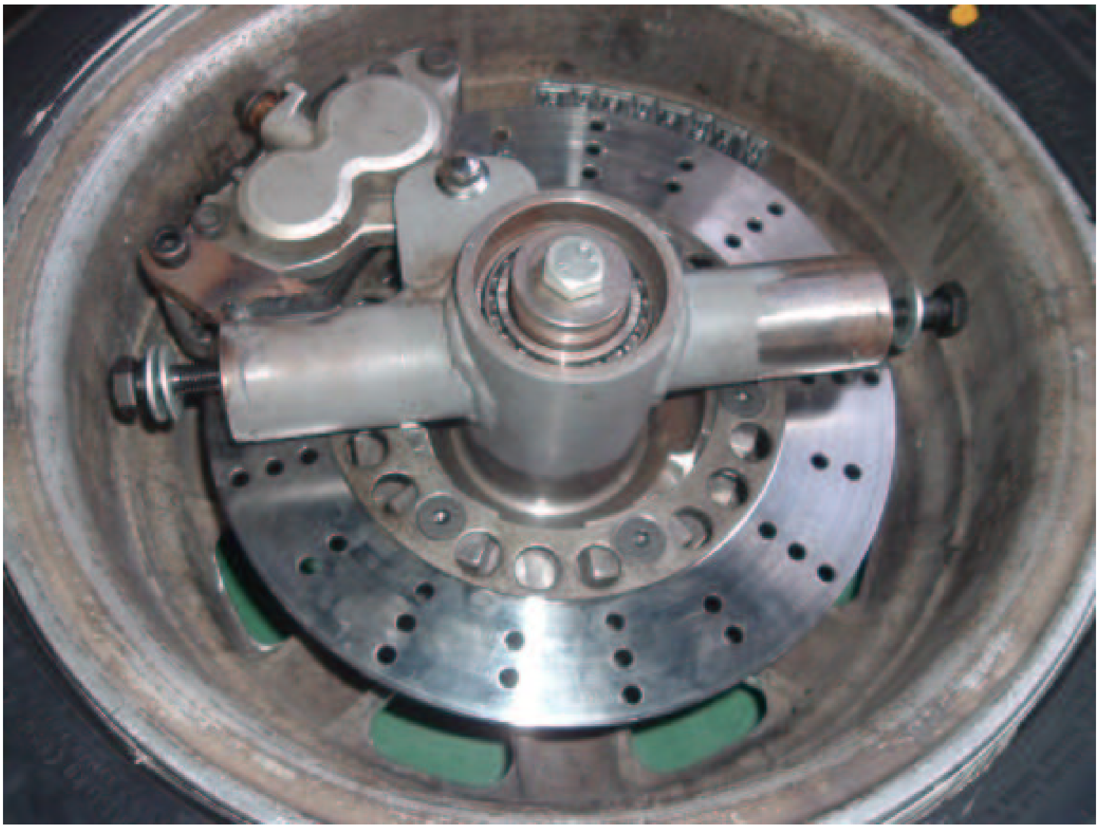


Figure B.3: Front upright mounted in road wheel.

Figure B.4: Rear upright with axle.

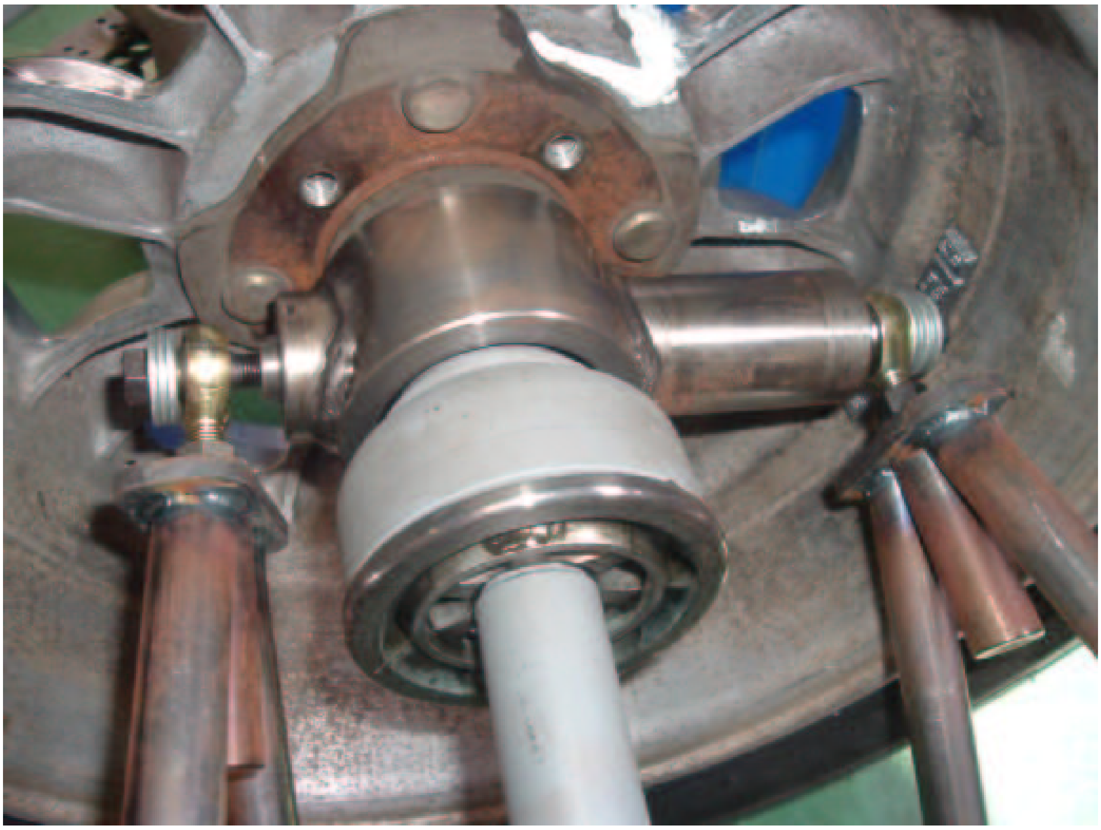


Figure B.5: Rear upright mounted in road wheel and suspension arms.

Appendix C

Finite Element Analysis Results for Upright

Results for finite element analysis for the front upright are as follows:

Table C.1: FEM Model Size

Maximum Node Number	13554
Number of Defined Nodes	13554
Number of Selected Nodes	13554
Maximum DOF per Node	3
Maximum Element Number	7052
Number of Defined Elements	7052
Number of Selected Elements	7052

Table C.2: Scratch Memory Status

Requested Initial Work Space	33685500 Words	128.500MB
Initial Work Space Obtained	33685505 Words	128.500MB
ANSYS Scratch Memory Size	16777221 Words	64.000MB
ANSYS Scratch Memory Used	623900 Words	2.380MB
Available Scratch Memory	16153321 Words	61.620MB
Maximum Scratch Memory Used	2424040 Words	9.247MB
Largest Available Contiguous Block	14671068 Words	55.966MB

Table C.3: Database Status

Current Database Position	5784042 Words	22.064MB
Maximum Database Length	2147418107 Words	8191.750MB
Memory Resident Database	16777216 Words	64.000MB
Void Space In Database	0 Words	0.000MB

Table C.4: Solution Memory

Binary I/O Page Size	16384 Words	0.062MB
Buffers Per Solution File	4	
Buffer Scratch Memory	393216 Words	1.500MB
Analysis Type	0	
Available Solution Memory	15760105 Words	60.120MB
Wavefront Available	3858	

Table C.5: Display Wavefront Information

*** NOTE ***	CP= 58.030	TIME= 13:43:13
Get or calculate element order.		
** NOTE ***	CP= 58.070	TIME= 13:43:13
Calculating Wavefront.		

Table C.6: Sparse Direct Solver Usage

Number of Active Nodes	13554	
Number of Equations	40662	
Number of Terms in Equations	1457397	
SPARSE solver memory in use		36.694MB
Max. Solution Memory Space	9619011 Words	36.694MB
In Memory Database Space	16908284 Words	64.500MB
Total Space Needed	26527295 Words	101.194MB

Table C.7: File Size Estimates

EMAT File Size	0.000MB
ESAV File Size	0.000MB
FULL File Size	18.625MB
LNxx File Size	
-Sparse Solver	233.812MB
RST Geometry Size	3.562MB
RST Load Size	13.812MB
(max. per load set)	
TOTAL	269.812MB

NOTE - TOTAL DOES NOT INCLUDE SUBSTRUCTURES OR DATABASE FILE

Table C.8: ANSYS Solution Phase Run Time Estimator

Computer	SOLARIS64
Number of Master DOF	0
Analysis Type	0
Maximum Wavefront	0
R.M.S. Wavefront	0
No. of Active DOFS	38934
Stiff. Matrix Save Key	1
Number of Matrices	1
Elem. Matrix Save Key	0
Est. No. of Iterations	1

Table C.9: Element Usage

LABEL	NUMBER	FORM TIME	RESULTS TIME	NAME
Solid92	7052	0.005	0.003	3D 10node tet struc solid

Table C.10: Scratch Memory Status

ANALYSIS PHASE	First iteration	Subsequent iterations
ELEMENT FORMULATION	32.79	32.79
ELEMENT PREPARATION	8.11	8.11
SPARSE DIRECT SOLUTION	0.00	0.00
ELEMENT RESULTS	24.59	24.59
NODAL LOAD CALCULATIONS	8.20	8.20
SUBTOTAL TIMES (SEC)	73.69	73.70
EST. TOTAL TIME (SEC)	73.69	Based on 1 estimated iteration

NOTE: Sparse solver times are approximate. Variations by a factor of about two in either direction are possible.

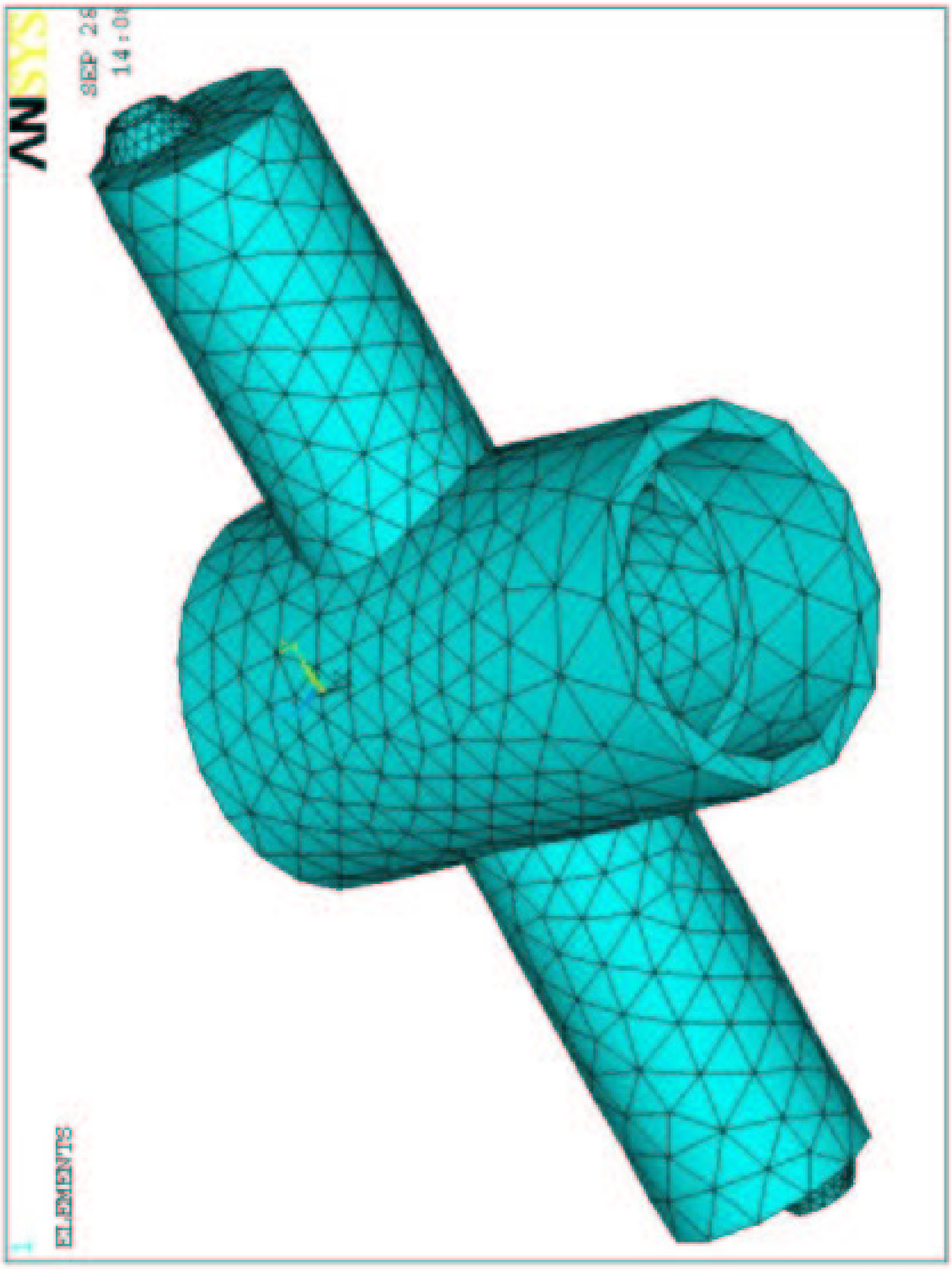


Figure C.1: Finite element mesh for upright using Ansys.

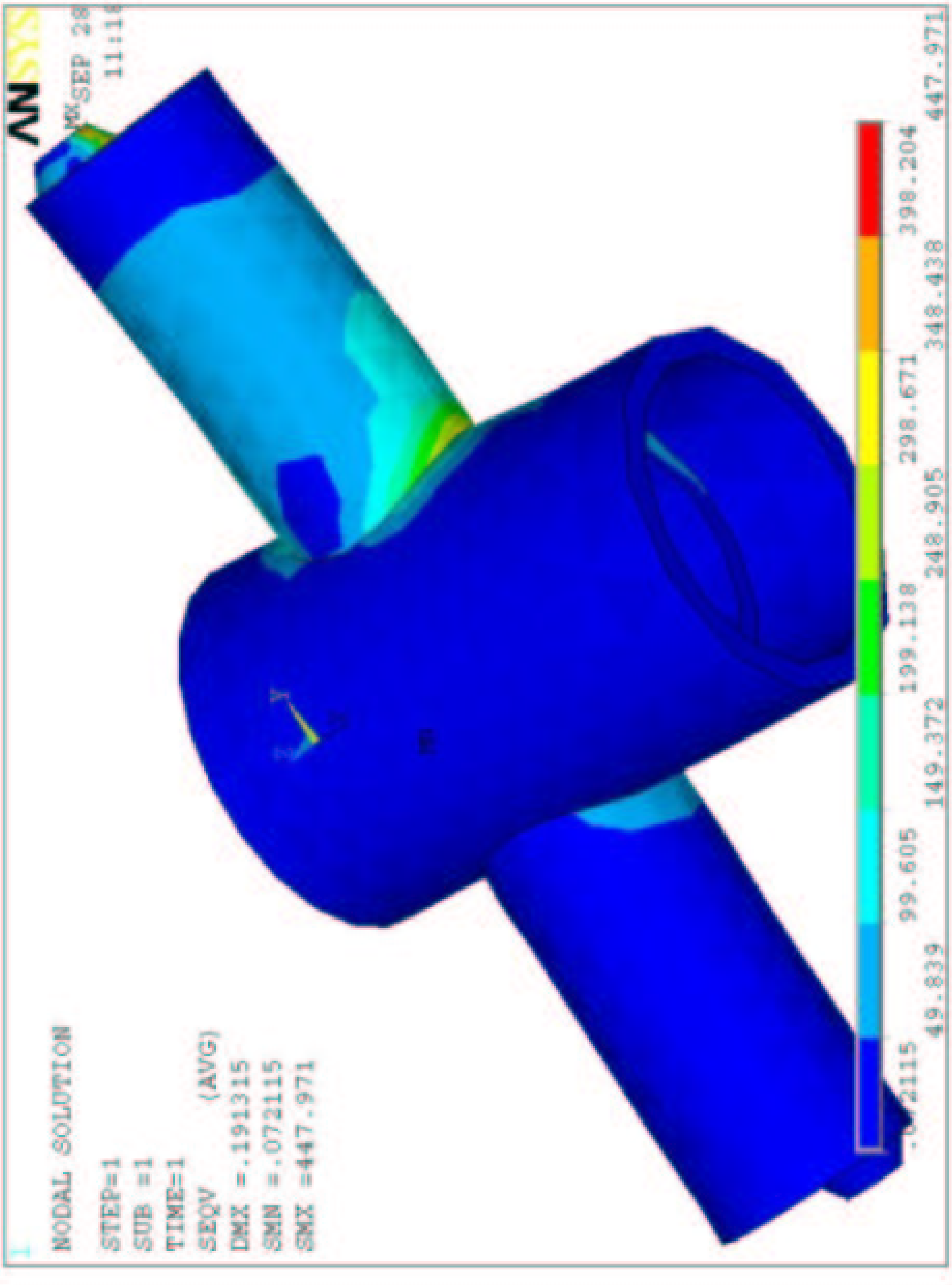


Figure C.2: Ansys results for upright.

Appendix D

Finite Element Analysis Results for Axle

Results for finite element analysis for the front axle are as follows:

Table D.1: FEM Model Size

Maximum Node Number	6814
Number of Defined Nodes	6814
Number of Selected Nodes	6814
Maximum DOF per Node	3
Maximum Element Number	4001
Number of Defined Elements	4001
Number of Selected Elements	4001

Table D.2: Scratch Memory Status

Requested Initial Work Space	33685500 Words	128.500MB
Initial Work Space Obtained	33685505 Words	128.500MB
ANSYS Scratch Memory Size	16777221 Words	4.000MB
ANSYS Scratch Memory Used	708680 Words	2.703MB
Available Scratch Memory	16068541 Words	61.297MB
Maximum Scratch Memory Used	31272964 Words	119.297MB
Largest Available Contiguous Block	13402752 Words	51.127MB

Table D.3: Database Status

Current Database Position	3506243 Words	13.375MB
Maximum Database Length	2147418107 Words	8191.750MB
Memory Resident Database	16777216 Words	64.000MB
Void Space In Database	628931 Words	2.399MB

Table D.4: Solution Memory

Binary I/O Page Size	16384 Words	0.062MB
Buffers Per Solution File	4	
Buffer Scratch Memory	393216 Words	1.500MB
Analysis Type	0	
Available Solution Memory	15675325 Words	59.797MB
Wavefront Available	3891	

Table D.5: Display Wavefront Information

*** NOTE ***	CP= 133.800	TIME= 14:03:57
Get or calculate element order.		
** NOTE ***	CP= 133.830	TIME= 14:03:57
Calculating Wavefront.		

Table D.6: Sparse Direct Solver Usage

Number of Active Nodes	6814	
Number of Equations	20442	
Number of Terms in Equations	782757	
SPARSE solver memory in use		65.219MB
Max. Solution Memory Space	17096730 Words	65.219MB
In Memory Database Space	16908284 Words	64.500MB
Total Space Needed	34005014 Words	129.719MB

Table D.7: File Size Estimates

EMAT File Size	0.000MB
ESAV File Size	0.000MB
FULL File Size	9.875MB
LNxx File Size	
-Sparse Solver	125.625MB
RST Geometry Size	2.000MB
RST Load Size	7.875MB
(max. per load set)	
TOTAL	145.375MB

NOTE - TOTAL DOES NOT INCLUDE SUBSTRUCTURES OR DATABASE FILE

Table D.8: ANSYS Solution Phase Run Time Estimator

Computer	SOLARIS64
Number of Master DOF	0
Analysis Type	0
Maximum Wavefront	0
R.M.S. Wavefront	0
No. of Active DOFS	18654
Stiff. Matrix Save Key	1
Number of Matrices	1
Elem. Matrix Save Key	0
Est. No. of Iterations	1

Table D.9: Element Usage

LABEL	NUMBER	FORM TIME	RESULTS TIME	NAME
Solid92	4001	0.005	0.003	3D 10node tet struc solid

Table D.10: Scratch Memory Status

ANALYSIS PHASE	First iteration	Subsequent iterations
ELEMENT FORMULATION	18.60	18.60
ELEMENT PREPARATION	3.89	3.89
SPARSE DIRECT SOLUTION	0.00	0.00
ELEMENT RESULTS	13.95	13.95
NODAL LOAD CALCULATIONS	4.65	4.65
SUBTOTAL TIMES (SEC)	41.10	41.10
EST. TOTAL TIME (SEC)	41.10	Based on 1 estimated iteration

NOTE: Sparse solver times are approximate. Variations by a factor of about two in either direction are possible.

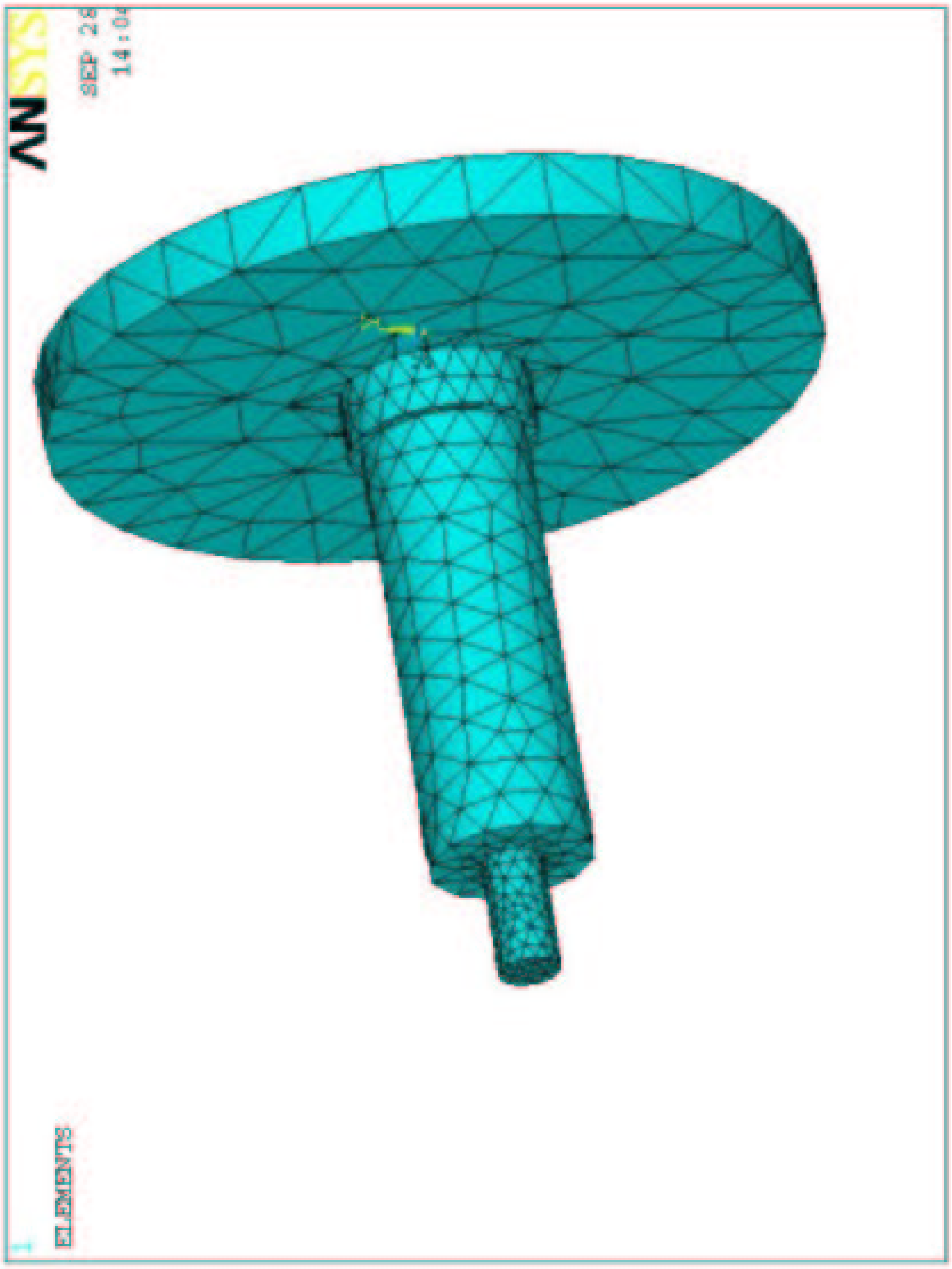


Figure D.1: Finite element mesh for axle using Ansys.

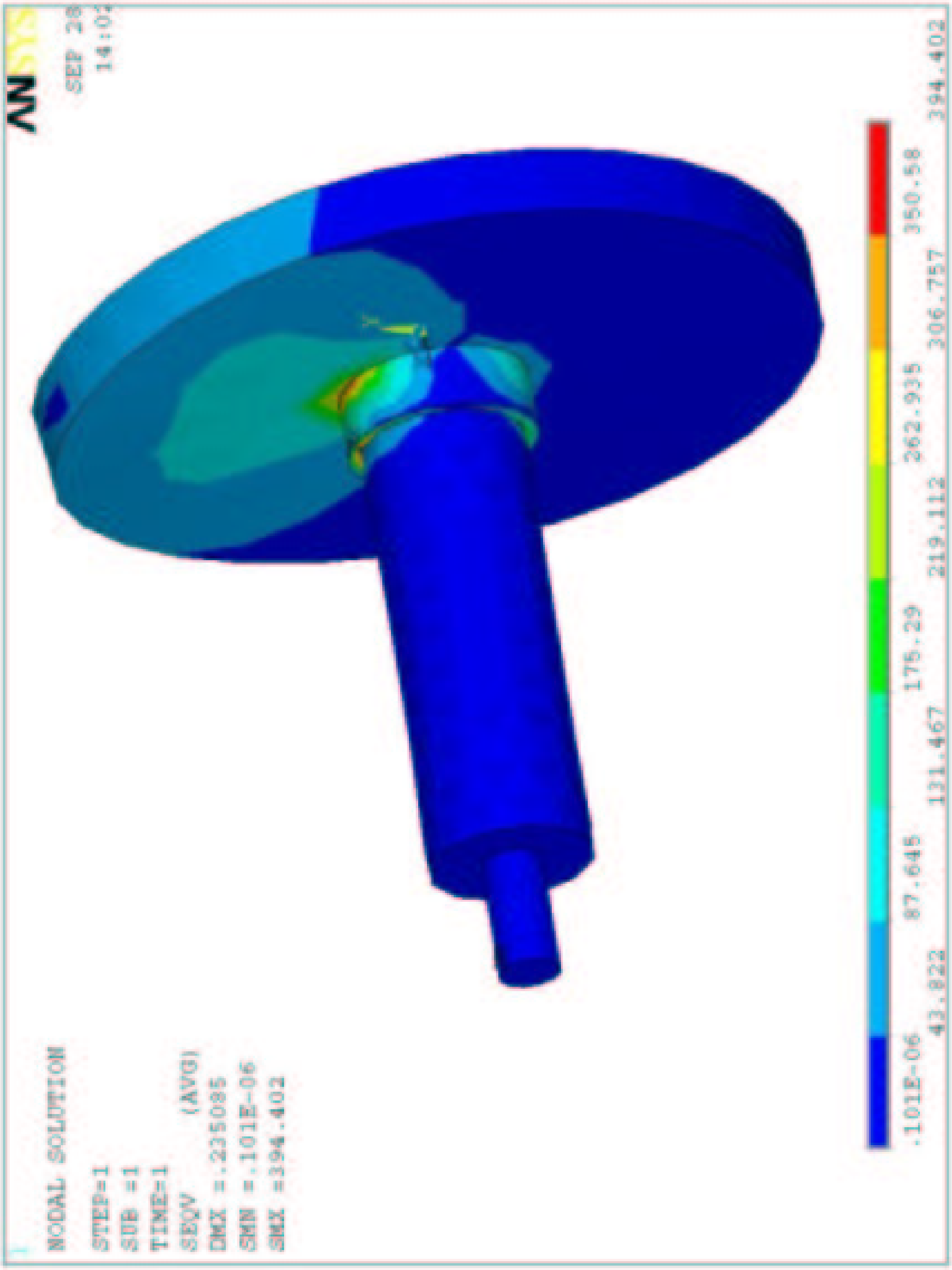
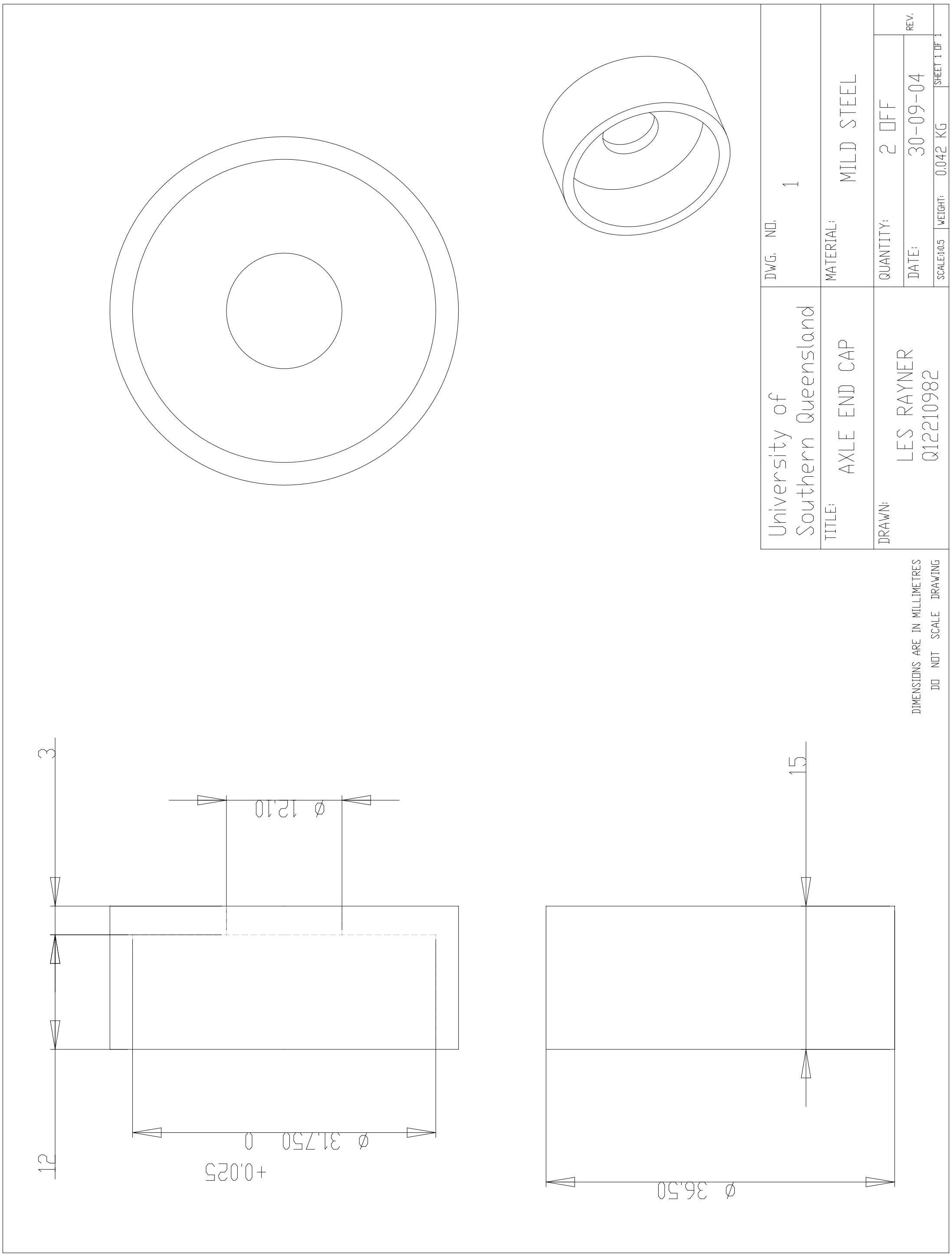
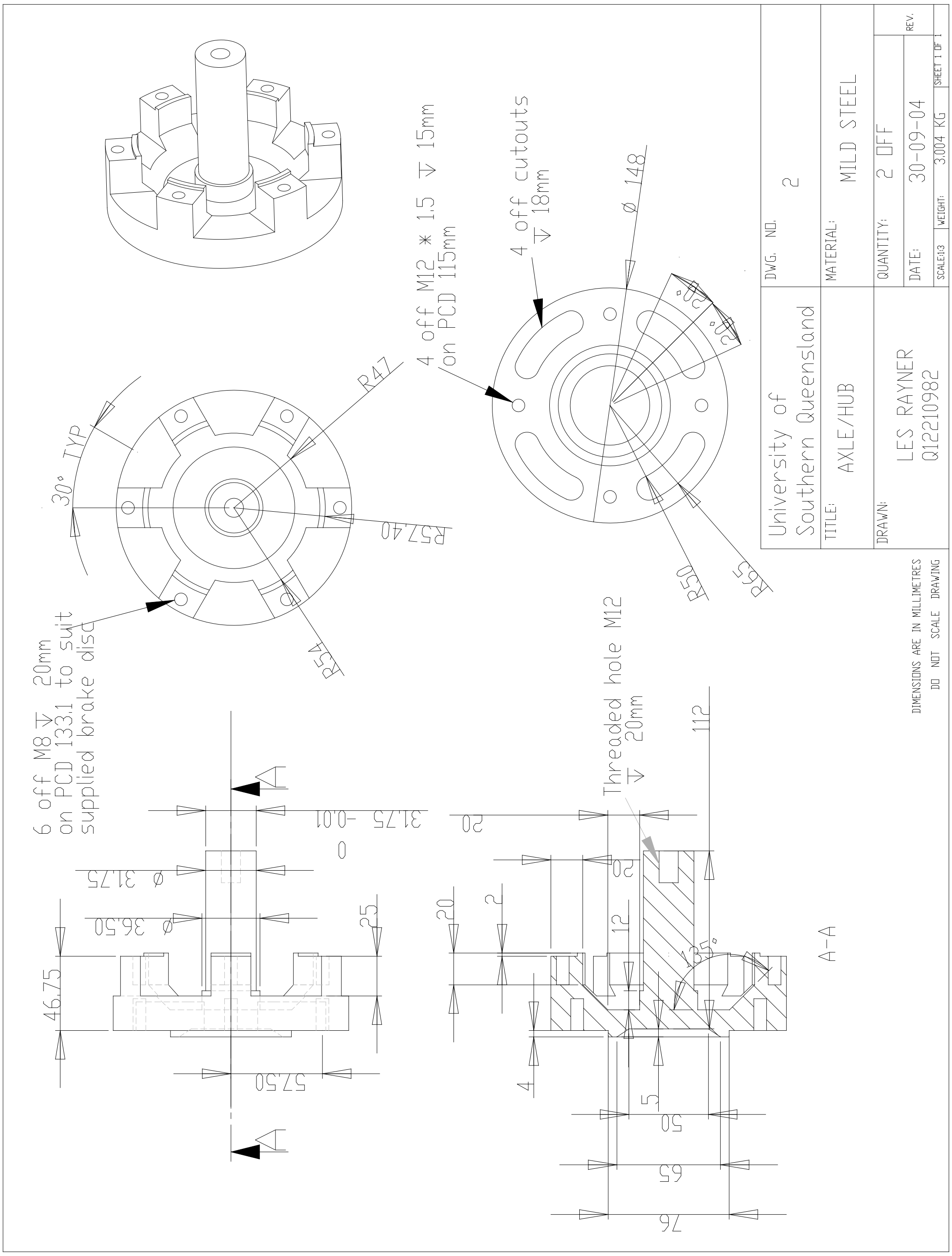


Figure D.2: Ansys results for axle.

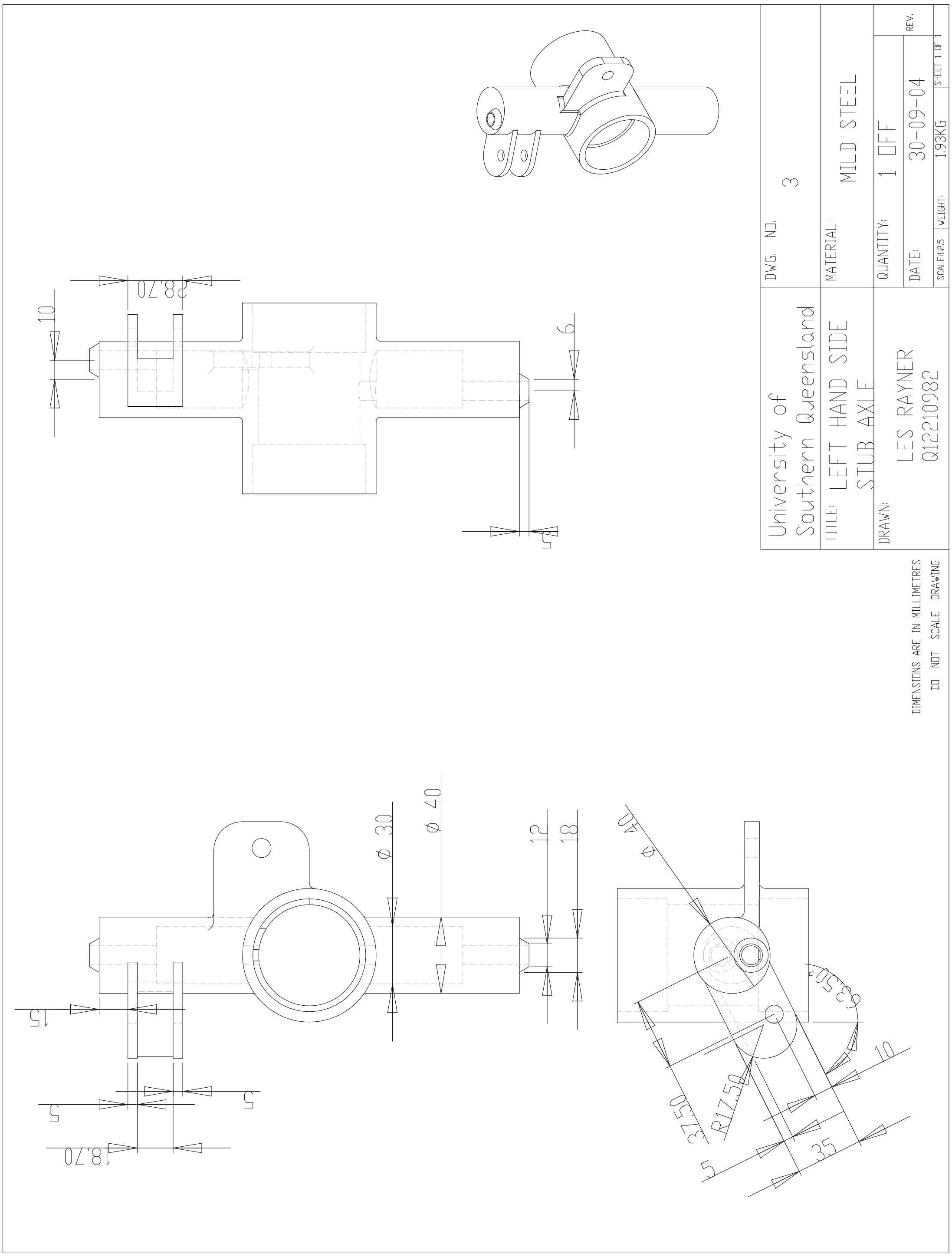
Appendix E

Production Drawings

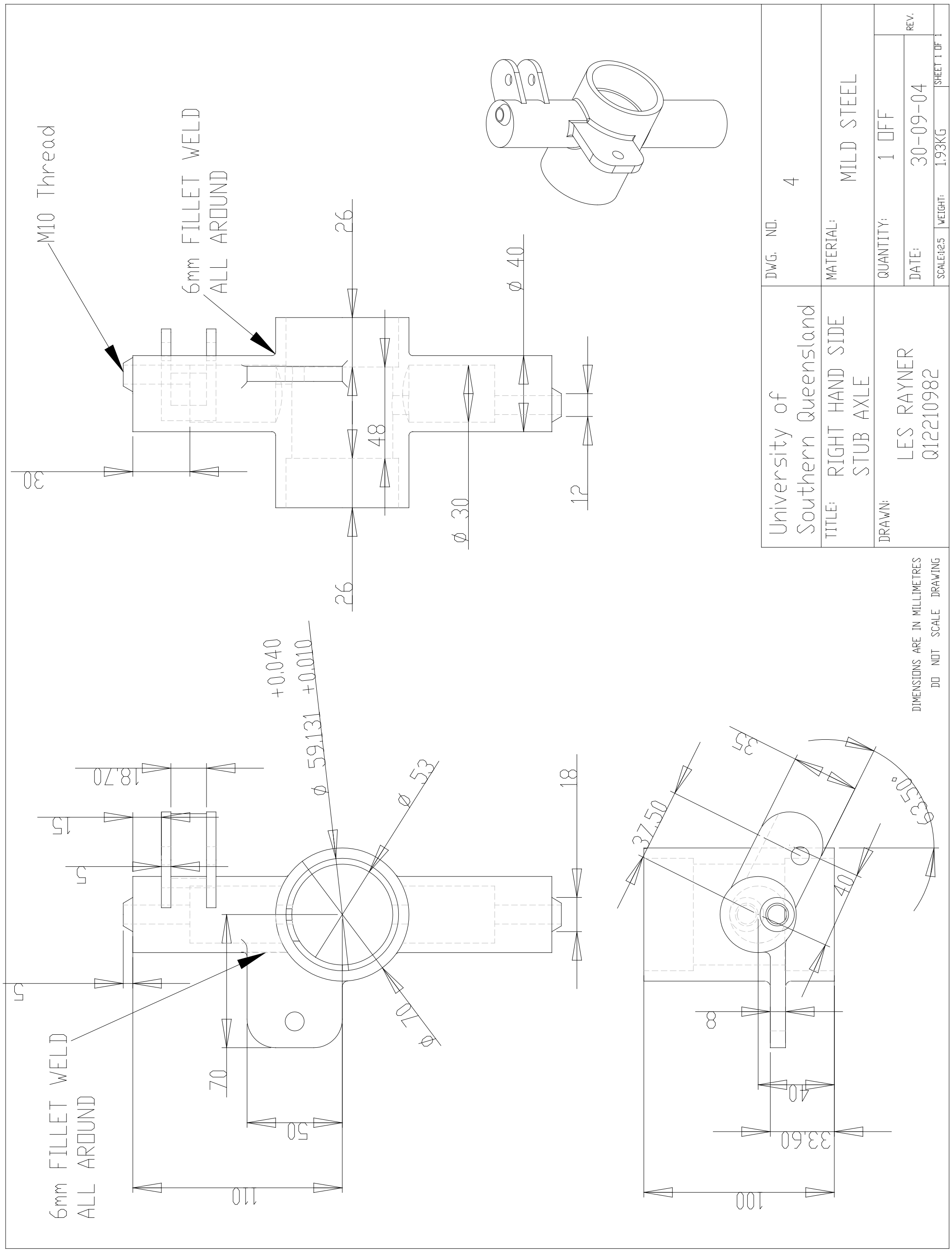

University of
Southern Queensland
TITLE: AXLE END CAP
DRAWN: LES RAYNER
Q12210982
DWG. NO. 1
MATERIAL: MILD STEEL
QUANTITY: 2 OFF
DATE: 30-09-04
REV.
SCALE:1:0.5
WEIGHT: 0.042 KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
3
12
ø 12.10
ø 31.750 +0.025 0
15
ø 36.50

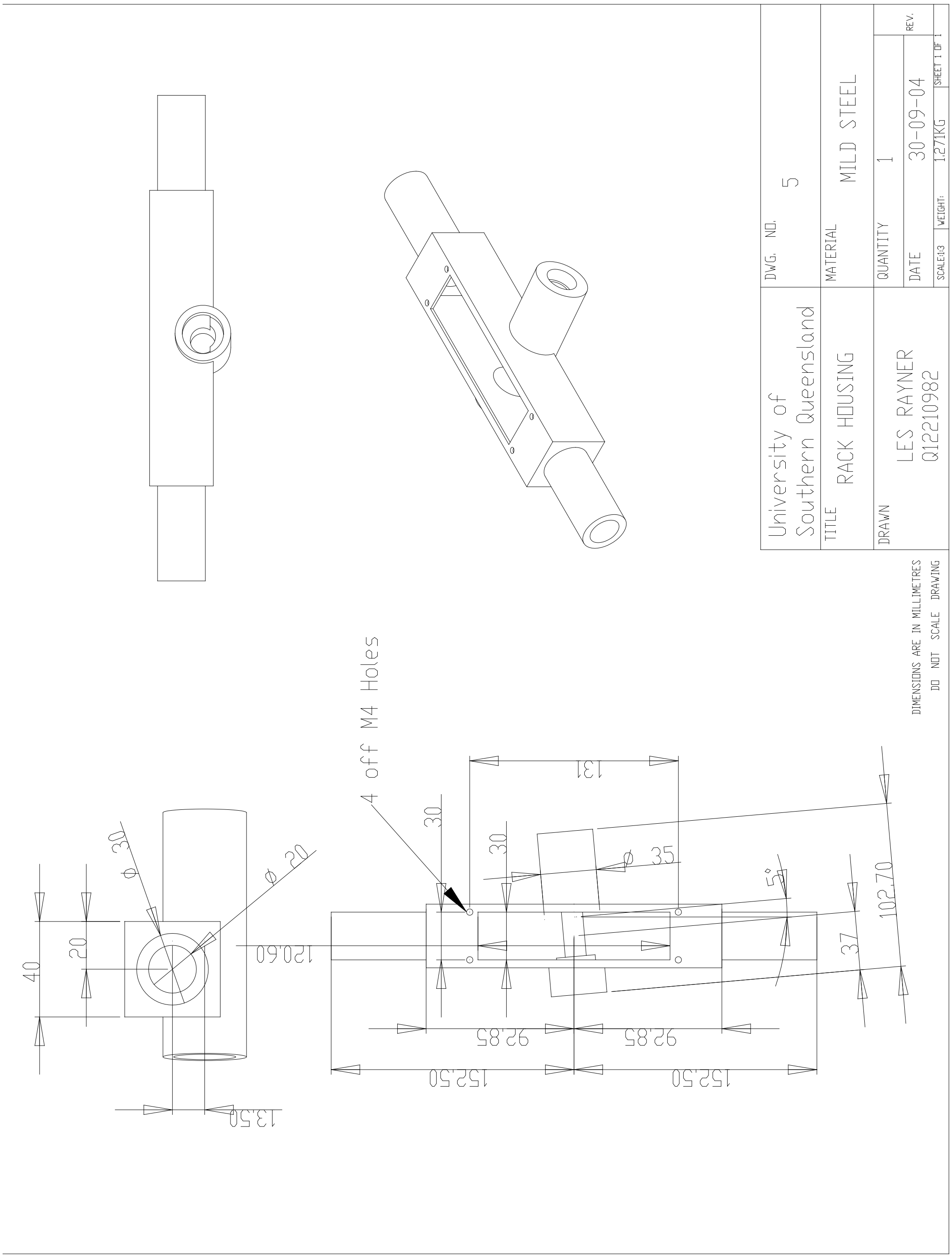


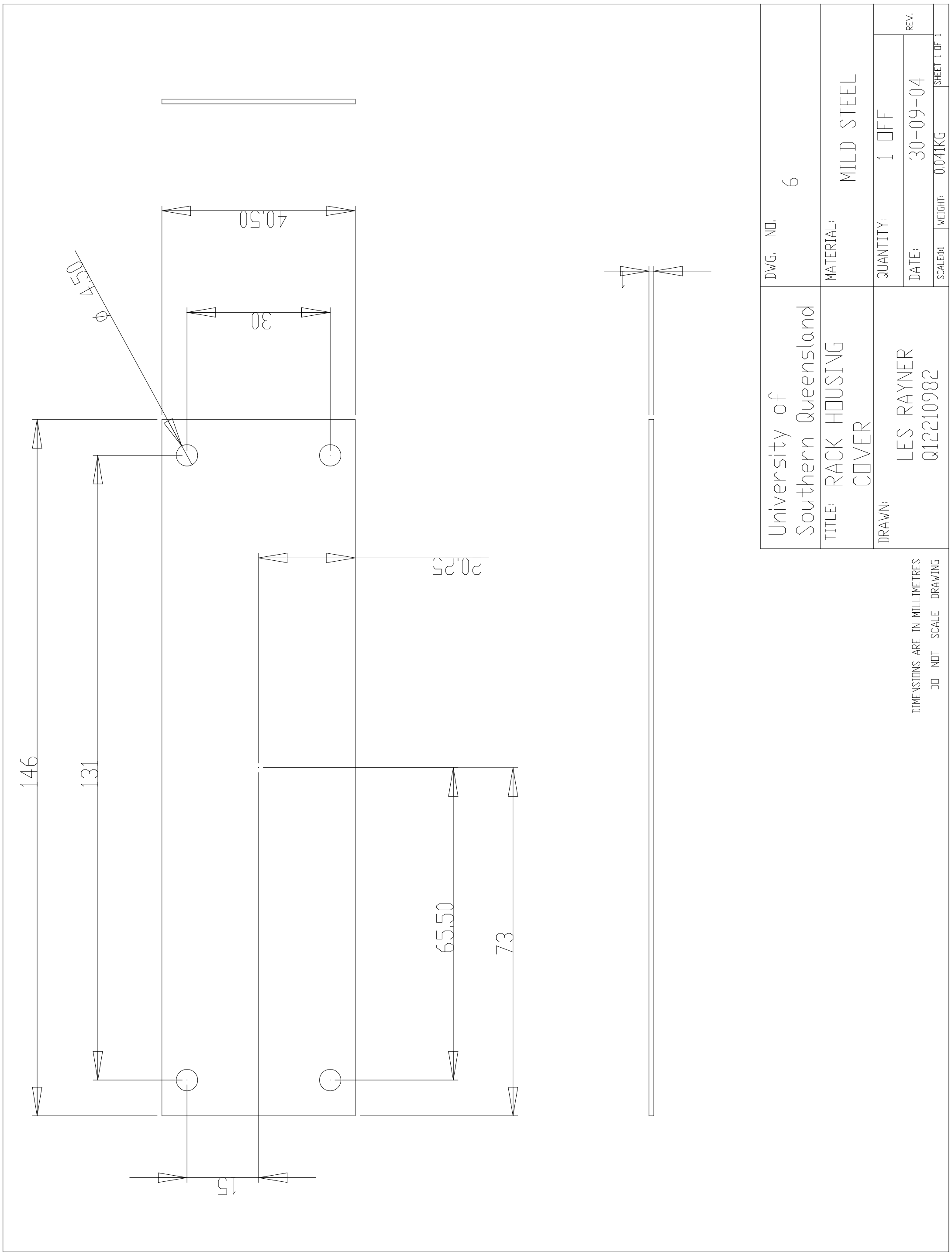
4 off M12 * 1.5 ⊽ 15mm
on PCD 115mm
4 off cutouts
⊽ 18mm
ø 148
R47
30° TYP
R57.40
R54
R50
R65
6 off M8 ⊽ 20mm
on PCD 133.1 to suit
supplied brake disc
Threaded hole M12
⊽ 20mm
A-A
University of
Southern Queensland
DWG. NO.
2
TITLE:
AXLE/HUB
MATERIAL:
MILD STEEL
DRAWN:
LES RAYNER
Q12210982
QUANTITY:
2 OFF
DATE:
30-09-04
SCALE 1:3
WEIGHT:
3.004 KG
SHEET 1 OF 1
REV.
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING

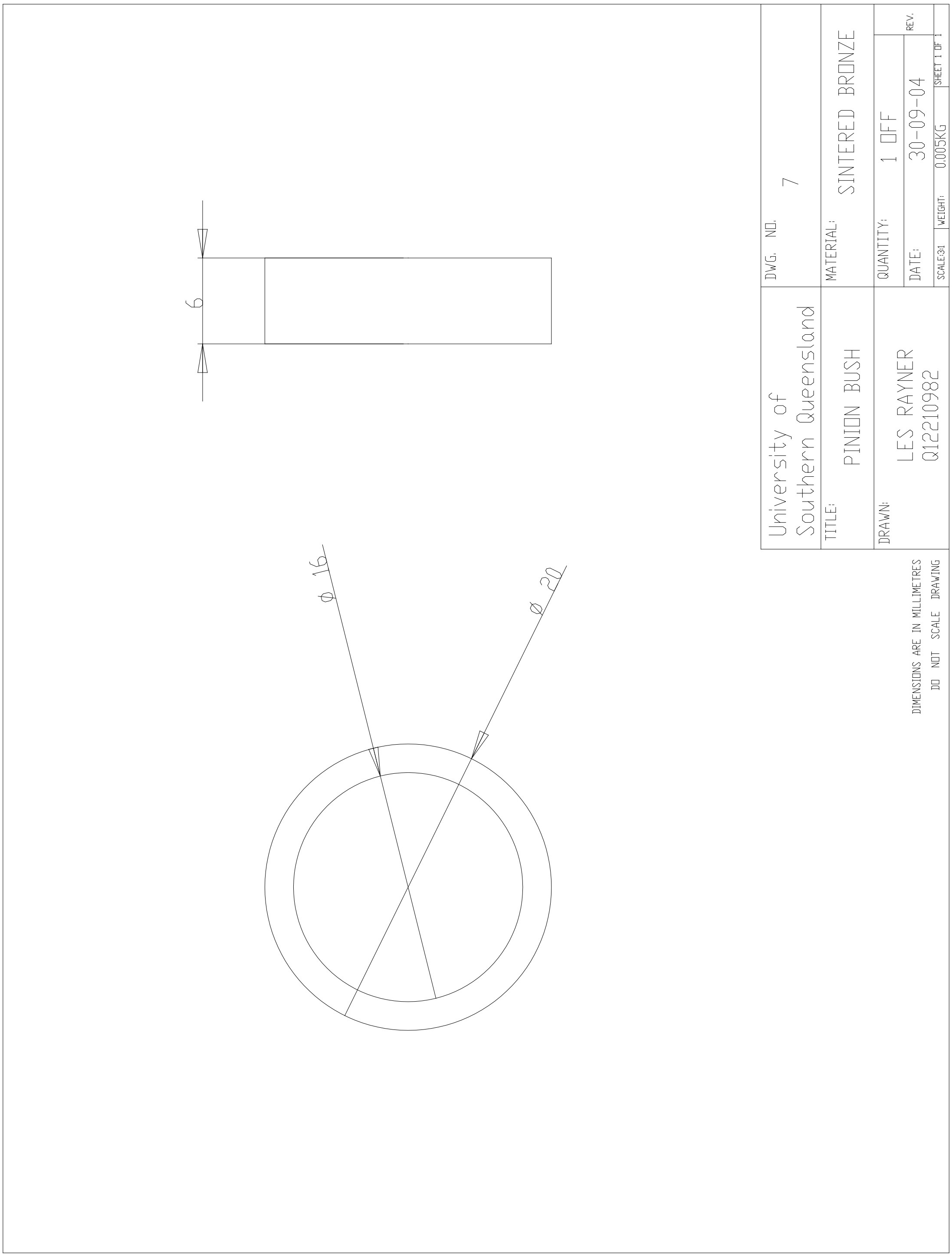


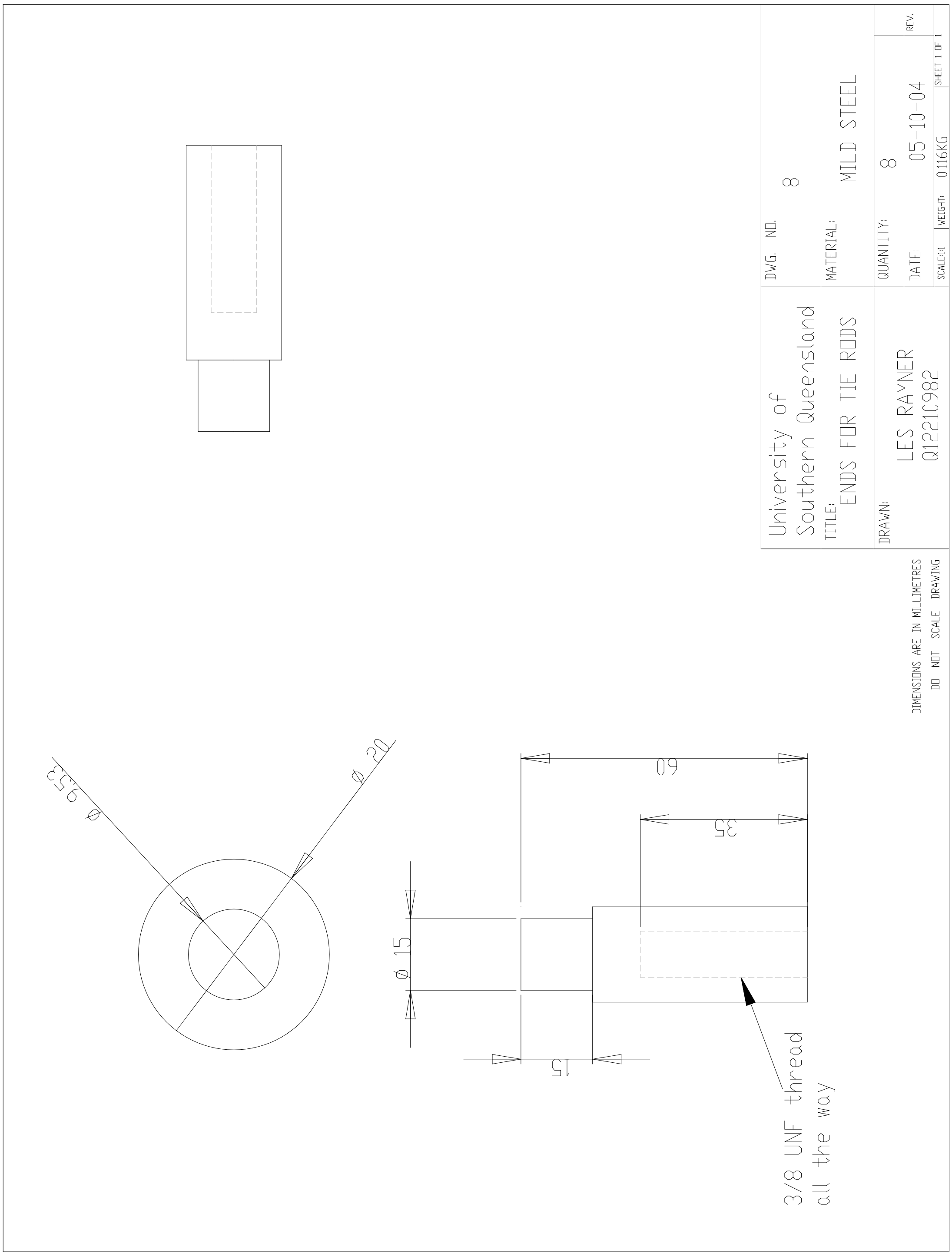
University of Southern Queensland
TITLE: LEFT HAND SIDE STUB AXLE
DRAWN: LES RAYNER Q12210982
DWG. NO. 3
MATERIAL: MILD STEEL
QUANTITY: 1 OFF
DATE: 30-09-04
SCALE:1:2.5
WEIGHT: 1.93KG
SHEET 1 OF 1
REV.
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING

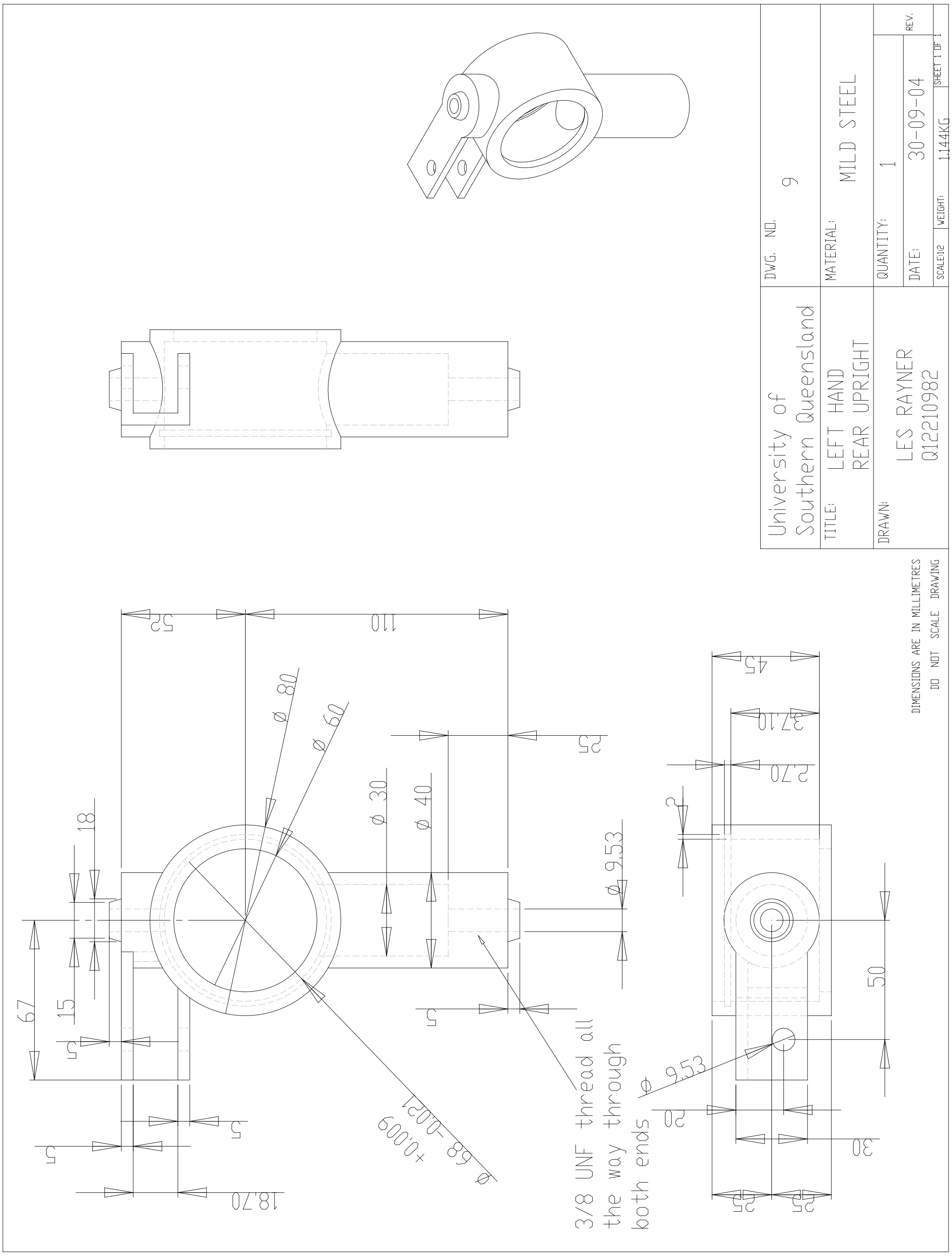

M10 Thread
6mm FILLET WELD ALL AROUND
6mm FILLET WELD ALL AROUND
University of Southern Queensland
TITLE: RIGHT HAND SIDE STUB AXLE
DRAWN: LES RAYNER Q12210982
DWG. NO. 4
MATERIAL: MILD STEEL
QUANTITY: 1 OFF
DATE: 30-09-04
SCALE 1:2.5
WEIGHT: 1.93KG
SHEET 1 OF 1
REV.
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING

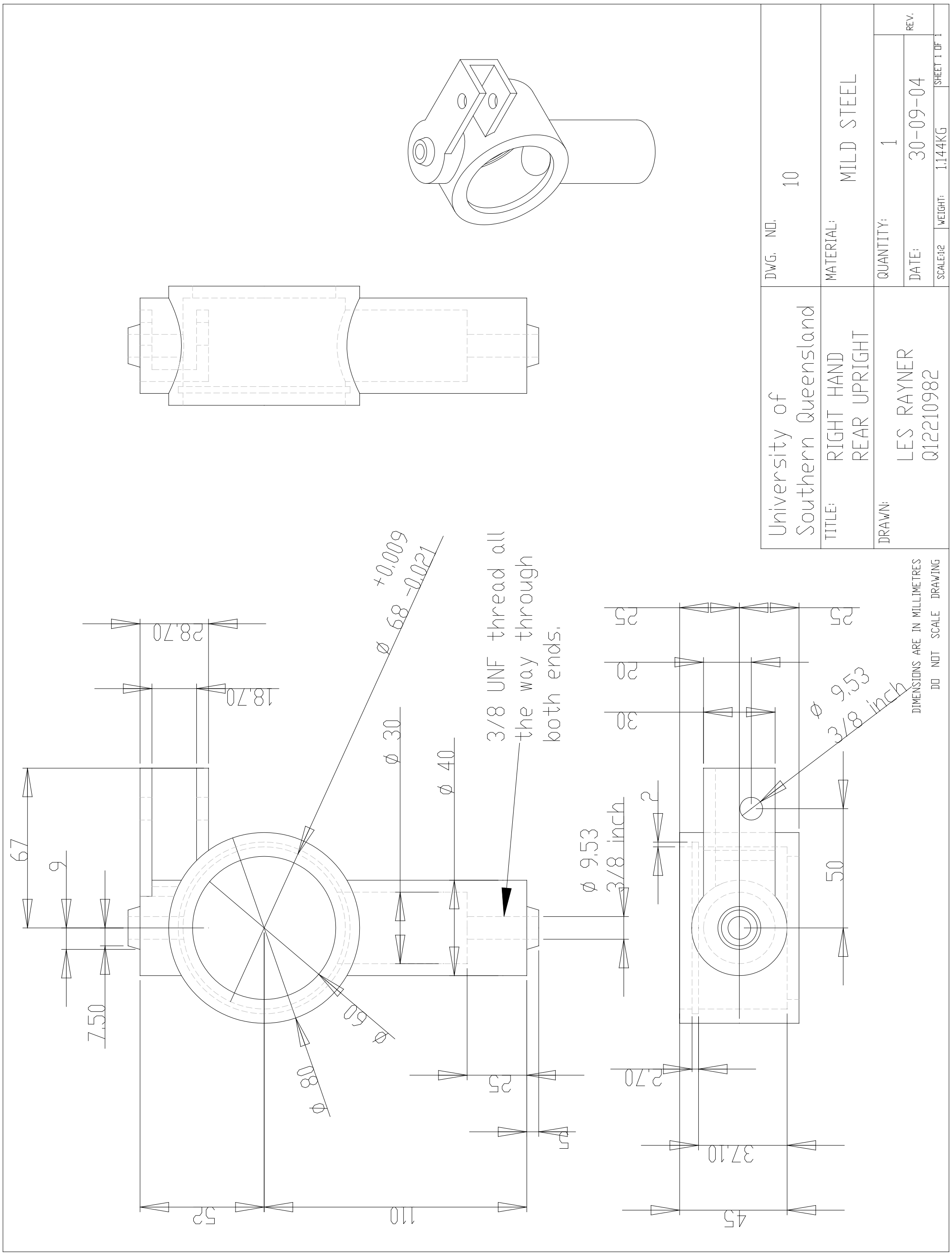

University of
Southern Queensland
TITLE
RACK HOUSING
DRAWN
LES RAYNER
Q12210982
DWG. NO.
5
MATERIAL
MILD STEEL
QUANTITY
1
DATE
30-09-04
REV.
SCALE1:3
WEIGHT:
1.271KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
4 off M4 Holes
131
30
30
ϕ 35
5°
102.70
37
120.60
92.85
92.85
152.50
152.50
ϕ 30
ϕ 20
40
20
13.50

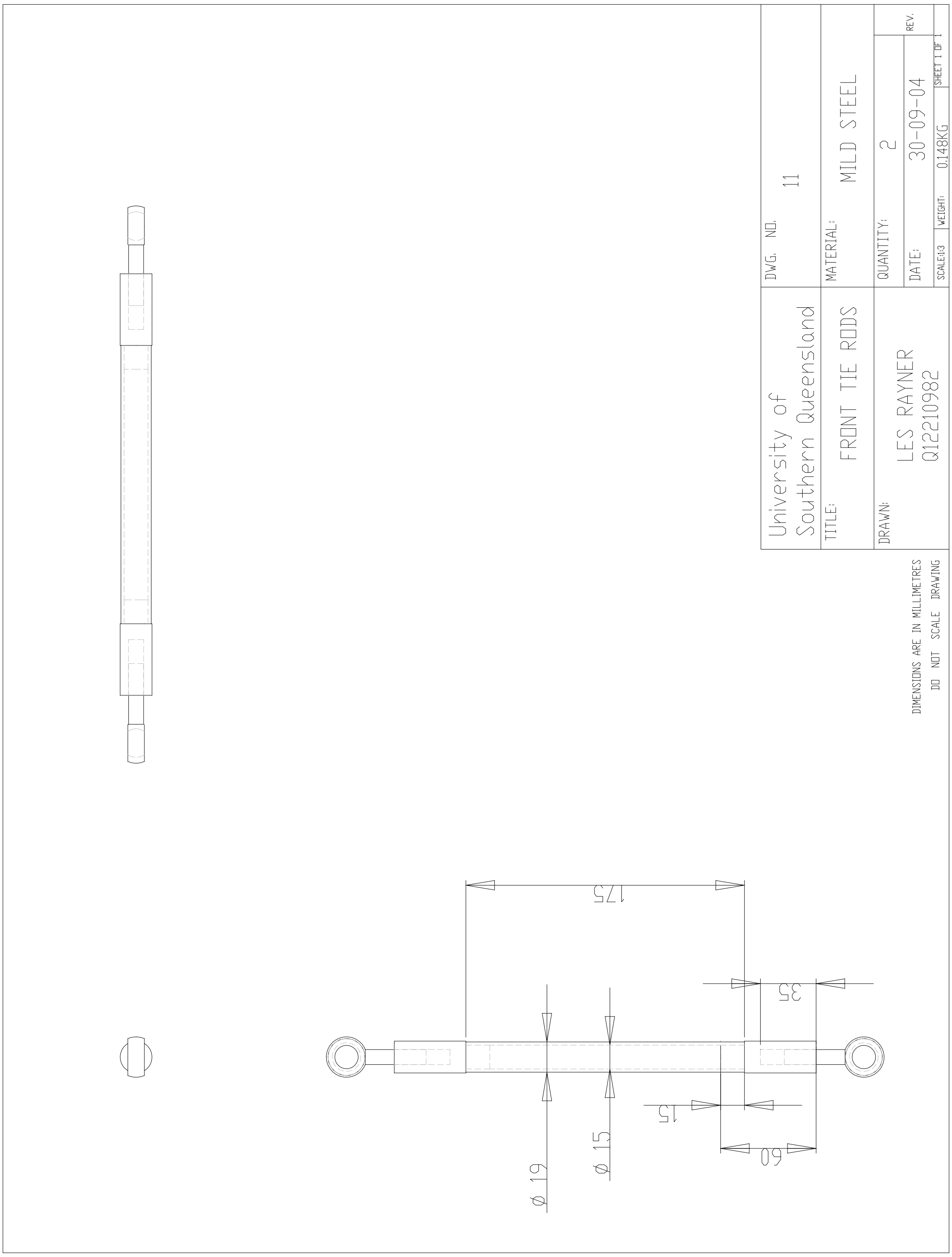

University of
Southern Queensland
TITLE: RACK HOUSING
COVER
DRAWN:
LES RAYNER
Q12210982
DWG. NO.
6
MATERIAL:
MILD STEEL
QUANTITY:
1 OFF
DATE:
30-09-04
REV.
SCALE 1:1
WEIGHT: 0.041KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
40.50
30
ϕ 4.50
20.25
146
131
65.50
73
15
1


6
ϕ 16
ϕ 20
University of
Southern Queensland
TITLE:
PINION BUSH
DRAWN:
LES RAYNER
Q12210982
DWG. NO.
7
MATERIAL:
SINTERED BRONZE
QUANTITY:
1 OFF
DATE:
30-09-04
REV.
SCALE:3:1
WEIGHT:
0.005KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING


University of
Southern Queensland
TITLE:
ENDS FOR TIE RODS
DRAWN:
LES RAYNER
Q12210982
DWG. NO.
8
MATERIAL:
MILD STEEL
QUANTITY:
8
DATE:
05-10-04
REV.
SCALE:1:1
WEIGHT:
0.116KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
ϕ 33
ϕ 20
60
35
ϕ 15
15
3/8 UNF thread
all the way


University of
Southern Queensland
TITLE: LEFT HAND
REAR UPRIGHT
DRAWN: LES RAYNER
Q12210982
DWG. NO. 9
MATERIAL: MILD STEEL
QUANTITY: 1
DATE: 30-09-04
SCALE 1:2
WEIGHT: 1.144KG
SHEET 1 OF 1
REV.
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
3/8 UNF thread all
the way through
both ends


University of Southern Queensland
TITLE: RIGHT HAND REAR UPRIGHT
DRAWN: LES RAYNER Q12210982
DWG. NO. 10
MATERIAL: MILD STEEL
QUANTITY: 1
DATE: 30-09-04
REV.
SCALE:1:2
WEIGHT: 1.144KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
3/8 UNF thread all the way through both ends.
ø 68 +0,009 -0,021
ø 30
ø 40
ø 60
ø 80
ø 9,53 3/8 inch
28,70
18,70
67
9
7,50
52
110
25
5
2,70
37,10
45
50
2
20
30


University of
Southern Queensland
TITLE:
FRONT TIE RODS
DRAWN:
LES RAYNER
Q12210982
DWG. NO.
11
MATERIAL:
MILD STEEL
QUANTITY:
2
DATE:
30-09-04
REV.
SCALE1:3
WEIGHT:
0.148KG
SHEET 1 OF 1
DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
175
35
15
60
ø 19
ø 15

Appendix F

Design Calculations

To work out the output rack load for a given input steering wheel effort it was necessary to do the following calculations.

Let: R=steering wheel radius = 300mm
r=pinion pitch-circle radius
t=number of pinion teeth = 5
p=linear or circular pitch
E=input steering-wheel effort = 2 * 20N
W=output rack load

If the pinion makes one revolution;

$$\text{input steering wheel movement Xi} = 2\pi r \tag{F.1}$$

$$\text{output rack movement Xo} \;=\; 2\pi R \;=\; tp \;=\; 32mm \tag{F.2}$$

Therefore;

$$\text{Movement ratio (MR)} \;=\; \frac{Xi}{Xo} \;=\; \frac{2\pi R}{2\pi r} \;=\; \frac{2\pi R}{tp} \tag{F.3}$$

Rearranging Equation (F.2);

$$p \;=\; \frac{Xo}{t} \;=\; \frac{32}{5} \;=\; 6.4mm \tag{F.4}$$

From Equation (F.3) we can now work out MR;

$$MR \;=\; \frac{2\pi R}{tp} \;=\; \frac{2\pi 300mm}{5 * 6.4} \;=\; 58.905 \tag{F.5}$$

Assuming no friction;

$$MR \;=\; \frac{Xi}{Xo} \;=\; \frac{W}{E} \tag{F.6}$$

Rearranging Equation (F.6);

$$E \;=\; \frac{W}{MR} \tag{F.7}$$

or

$$W \;=\; E \,*\, MR \;=\; 2 * 20N * 58.905 \;=\; 2356.2N \tag{F.8}$$

(Heisler 1997)

Analytically find the stress at the base of the front upright. Using the following values:

- F = 10500N (as found previously for the bottom of the front upright);
- L = 80mm (length);
- D = 40mm (outer diameter);
- d = 30mm (inner diameter); and
- y = 20mm (centroid).

Find the moment about the base of the upright:

$$M = F * L = 10500 * 80 = 840000 Nmm \tag{F.9}$$

Find the second moment of inertia:

$$I = \frac{\pi(D^4 - d^4)}{64} = \frac{\pi(40^4 - 30^4)}{64} = 85904.9 mm^4 \tag{F.10}$$

Find the stress at the base of the upright:

$$\sigma = \frac{M * y}{I} = \frac{840000 * 20}{85904.9} = 195.6 MPa \tag{F.11}$$

This value is lower than the value from the FEA solution of approximately 250MPa but is of a similar magnitude.

Analytically find the shear stress through the bolt for the bottom spherical rod end.

Using the following values:

- F = 10500N (as found previously for the bottom of the front upright);
- d = 10mm (diameter);

Find the area of the bolt:

$$A = \frac{\pi * d^2}{4} = \frac{\pi * 10^2}{4} = 78.54mm^2 \tag{F.12}$$

Find the shear stress in the bolt:

$$\sigma = \frac{F}{A} = \frac{10500}{78.54} = 133.7MPa \tag{F.13}$$

This is much lower than the maximum permissible shear stress for high tensile steel.

Analytically find the stress in the tie-rods.

Using the following values:

- W = 2356.2N (as found previously for the rack output load);
- d = 15mm (inside diameter of tie-rod);
- D = 19mm (outside diameter of tie-rod);
- L = 335mm (length);
- FOS = 2 (factor of safety); and
- E = 200GPa (modulus of elasticity).

From Euler's formula:

$$W = \frac{\pi^2 * E * I}{L^2} \tag{F.14}$$

We can rearrange and solve for I.

$$I = \frac{FOS * W * L^2}{\pi^2 * E} = \frac{2 * 2356.2 * 335^2}{\pi^2 * 200 * 10^9} = 2.679 * 10^{-4} m^4 \tag{F.15}$$

also;

$$I = \frac{\pi(D^4 - d^4)}{64} m^4 \tag{F.16}$$

$$2.679 * 10^{-4} = \frac{\pi(D^4 - d^4)}{64} = \frac{\pi(D^4 - 15^4)}{64} \tag{F.17}$$

Rearranging Equation (F.17);

$$D = \sqrt[4]{\frac{2.679 * 10^{-10} * 64}{\pi} + 15^4} = 15.00000064\text{mm minimum.} \tag{F.18}$$

D = 19mm so the tie-rod end is safe.

Appendix G

Cost Report

G.1 Manufacturing Costs

The following is the cost report that was prepared as part of the SAE-A competition.

Name: Steering Rack Housing

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight (g)	$/KG	Cost ($)
A	1	Mild Steel RHS 40mm*40mm*165mm long	501160	355.6	0.16308	0.05
B	2	Mild Steel Bar - Square 40mm*40mm*70mm long	112000	794	0.16308	0.26
C	1	Mild Steel Bar - round 35mm diam * 105mm	101022	716.1	0.16308	0.10
D	1	Mild Steel Plate 146*40.5*1mm thick	5849	41.5	0.16308	0.01
E	2	Mild steel bar 20mm*40mm*6mm thick	4800	34.0	0.16308	0.01

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
A	1	10	min	1	Mill slot	0.583	5.83
A	1	15	min	1	Mill slot for (C)	0.583	8.75
A	4		hole	1	Drill 4mm hole	0.35	1.40
A	4		hole	1	Tap 4mm hole	0.35	1.40
A	2	7	cm	1	Weld C to A	0.14	1.96
A	2	16	cm	1	Weld B to A	0.14	4.48
B	2	15	min	1	Lathe cut	0.583	8.75

Figure G.1: Page 1 of manufacturing cost analysis.

B	2		hole	1	Drill 20mm diam. hole	0.35	0.70
C	1		hole	1	Drill 16mm diam. hole	0.35	0.35
C	1	5	min	1	Machine 20mm diam. hole	0.583	2.92
C	1	10	hole	1	Machine 26mm diam. *18mm hole	0.583	5.83
C	1		hole	1	Tap 26mm diam hole	0.35	0.35
D	1	4	hole	1	Drill 4.5mm hole	0.35	1.40
E	2	3	cm	1	Cut 30mm	0.16	0.96
Total							44.12

The team had on hand a piece of 40mm square hollow section 165mm long (A) that had a slot milled in the top face as well as a slot for the pinion carrier (C). It also had 4 holes drilled and tapped to allow the cover to be bolted to it. The team also had on hand 2 pieces of 40mm square * 70mm long that were machined to 30mm diam for a length of 60mm. The team also had on hand a piece of 35mm diam round bar that had a 16mm hole drilled the full length. At one end a 26mm hole was machined to a depth of 18mm to accommodate the bearing preloader. The other end had a 20mm diam hole machined for the pinion bearing. The 2 pieces of machined 40mm bar were welded to the ends of the hollow section as was the 35mm machined round bar. The team also had on hand some 1mm steel plate that was had 4 * 4.5mm holes drilled in it. The pinion, pinion bearing, pinion preloader, pinion bush and modified rack from the purchased Renault R10 Rack were used to complete the part.

Name: Steering Rack and Ends

Material

Process Labour

Sub.	Qty.	Amount.	Unit	Manning	Description	$/Unit	Cost ($)
F	1	2	cm	1	Cut rack to shorten one end of rack	0.14	0.28
F	1	6.3	cm	1	Weld extension to rack	0.14	0.88
F	1		hole	1	Drill hole for 8mm thread	0.35	0.35

Figure G.2: Page 2 of manufacturing cost analysis.

F	1		hole	1	Tap 8mm hole	0.35	0.35
F	2	4	cm	1	Weld rack ends	0.14	1.12
F	2		hole	1	Drill 3/8" hole	0.35	0.70
Total							3.68

The team had been donated a Renault R10 rack that was cut, drilled and tapped at one end. The other end was lengthened by welding some of the cut off rack to it. The rack ends were manufactured out of some 20mm * 5mm mild steel bar that was available.

Name: Front Axle

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight (g)	$/KG	Cost ($)
G	2	Mild Steel Bar – round 120mm diam*150mm long	1696460	12025.7	0.16308	3.92

Process Labour

Sub.	Qty.	Amount.	Unit	Manning	Description	$/Unit	Cost ($)
G	2	240?	min	1	CNC Machining	1.167	560.16
G	8		hole	1	Tap 12mm hole	0.35	2.80
G	12		hole	1	Tap 8mm hole	0.35	4.20
G	2		hole	1	Tap 12mm hole	0.35	0.75
Total							564.01

The team had available 2 pieces of 120mm diameter round bar that was CNC machined to shape. The holes for the brake disk, the bearing preload and the road wheel studs were all tapped.

Figure G.3: Page 3 of manufacturing cost analysis.

Name: Front Upright

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight. (g)	$/KG	Cost ($)
H	2	Mild Steel Bar – round 70mm diam*100mm long	384845.1	2728.1	0.16308	0.89
H	2	Mild Steel pipe – round 40mm OD*200mm long	109955.7	779.4	0.16308	0.25
H	4	Mild Steel Bar – round 40mm OD*25mm long	31415.9	222.7	0.16308	0.15
H	2	Mild Steel flat 60mm * 50mm * 8mm thick	24000	170.1	0.16308	0.06
H	2	Mild Steel flat 25mm * 40mm * 8 mm thick	8000	56.7	0.16308	0.02
H	2	Mild Steel – bar 35mm*35mm*50mm long	61250	434.2	0.16308	0.14

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
H	4	13.8	cm	1	Weld upright to bearing housing	0.14	7.73
H	4	12.5	cm	1	Weld round to upright	0.14	7.00
H	4	45	min	1	Machine upright ends	0.583	104.94
H	2	7	cm	1	Drill 40mm holes	0.35/inch	1.93
H	2	30	min	1	CNC machine bearing housing	1.167	70.00

Figure G.4: Page 4 of manufacturing cost analysis.

H	4		hole	1	Drill hole for 10mm thread	0.35	0.70
H	4		hole	1	Tap 10mm hole	0.35	1.40
H	2	3	cm	1	Saw cut	0.16	0.96
H	2		hole	1	Drill 10mm hole	0.35	0.70
H	2		hole	1	Drill 8mm hole	0.35	0.70
H	2	45	min	1	Machine steering arm	0.583	104.94
H	2	11.0	cm	1	Weld steering arm to upright	0.14	3.08
H	2		hole	1	Drill 3/8" hole in steering arm	0.35	0.70
Total							304.78

The team had some 40mm mild steel bar available and this was machined to the correct dimensions for the ends of the uprights. Also available was some 40mm mild steel pipe and this was welded to the ends that had been made. After the 40mm holes had been cut into the 70mm bar, the upright section was inserted into this wheel bearing section and welded into place. This 70mm diam round bar was then CNC machined to shape to suit our wheel bearings. The brake caliper mount were made out of the 8mm flat and welded onto the uprights. After lining up the calipers, the 2*10mm and 2*8mm holes were drilled to mount the calipers. The steering arms were then welded on and a 3/8" hole drilled in the steering arm.

Name: Rear Upright

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight (g)	$/KG	Cost ($)
I	2	Mild Steel Bar – round 80mm diam*50mm long	251327.4	1781.6	0.16308	0.58
I	2	Mild Steel pipe – round 40mm OD*50mm long	27488.9	194.9	0.16308	0.06

Figure G.5: Page 5 of manufacturing cost analysis.

I	4	Mild Steel Bar – round 40mm OD*30mm long	37699.1	267.2	0.16308	0.17
I	2	Mild Steel – bar 30mm*30mm*55mm long	45000	319	0.16308	0.10

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
I	4	13.8	cm	1	Weld upright to bearing housing	0.14	7.73
I	4	12.5	cm	1	Weld round to upright	0.14	7.00
I	4	45	min	1	Machine upright ends	0.583	104.94
I	2	7	cm	1	Drill 40mm holes	0.35/inch	1.93
I	2	30	min	1	CNC machine bearing housing	1.167	70.00
I	4		hole	1	Drill hole for 10mm thread	0.35	0.70
I	4		hole	1	Tap 10mm hole	0.35	1.40
I	2	45	min	1	Machine steering arm	0.583	104.94
I	2	11.0	cm	1	Weld steering arm to upright	0.14	3.08
I	2		hole	1	Drill 3/8" hole in steering arm	0.35	0.70
Total							302.38

The team had some 40mm mild steel bar available and this was machined to the correct dimensions for the ends of the uprights. Also available was some 40mm mild steel pipe and this was welded to the ends that had been made. After the 40mm holes had been cut into the 70mm bar, the upright section was inserted into this wheel bearing section and welded into place. This 80mm diam round bar was then CNC machined to shape to suit our wheel bearings. The steering arms were then welded on and a 3/8" hole drilled in the steering arm.

Figure G.6: Page 6 of manufacturing cost analysis.

Name: Front Tie Rods

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight (g)	$/KG	Cost ($)
J	2	Mild Steel Pipe - round 19mmOD*15mmID*175mm long	18849.6	133.6	0.16308	0.04
J	4	Mild Steel Bar - round 20mm diam*60mm long	18692.5	132.5	0.16308	0.09

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
J	4	15	min	1	Machine ends of tie-rods	0.583	35.00
J	4		hole	1	Drill hole for 3/8 thread	0.35	1.40
J	4		hole	1	Tap 3/8" hole	0.35	1.40
J	4	6	cm	1	Weld ends to pipe	0.14	0.84
Total							37.24

The tie-rod ends were machined out of some available 20mm mild steel bar. They were tapped to suit a 3/8" spherical rod end and then welded to the 19mm OD pipe.

Figure G.7: Page 7 of manufacturing cost analysis.

Name: Rear Tie Rods

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight. (g)	$/KG	Cost ($)
J	2	Mild Steel Pipe – round 19mmOD*15mmID*150mm long	16022.1	113.6	0.16308	0.04
J	4	Mild Steel Bar – round 20mm diam*60mm long	18692.5	132.5	0.16308	0.09

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
J	4	15	min	1	Machine ends of tie-rods	0.583	35.00
J	4		hole	1	Drill hole for 3/8 thread	0.35	1.40
J	4		hole	1	Tap 3/8" hole	0.35	1.40
J	4	6	cm	1	Weld ends to pipe	0.14	0.84
Total							37.24

The tie-rod ends were machined out of some available 20mm mild steel bar. They were tapped to suit a 3/8" spherical rod end and then welded to the 19mm OD pipe.

Figure G.8: Page 8 of manufacturing cost analysis.

Name: Axle End Caps

Material: Mild Steel

Sub.	Qty.	Description	Vol. (mm^3)	Weight. (g)	$/KG	Cost ($)
K	2	Mild Steel Bar – round 40mmOD*15mm long	16022.1	113.6	0.16308	0.04

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$/Unit	Cost ($)
J	2	20	min	1	Machine axle end caps	0.583	23.32
Total							23.32

The tie-rod ends were machined out of some available 20mm mild steel bar. They were tapped to suit a 3/8" spherical rod end and then welded to the 19mm OD pipe.

Figure G.9: Page 9 of manufacturing cost analysis.

G.2 Bill Of Materials

Area or Commodity	Part Name	Description / Model # or Part #	Purchased or Manufactured (P or M)	Quantity	What You Paid	Retail Cost Each	Unit of Measure	Suppliers Name & Phone #	Total Retail Cost
Steering system	Rack	Steering rack from Renault R10	P	1	$0.00	$50.00	each	PRPS	$50.00
Steering system	Rack	Alter Steering Rack/Make new housing with brackets	M	1	-	-	each		$3.68
Steering system	Rack	M4 Bolts	P	4		$0.50	each	NBF	$2.00
Steering system	Tie Rods	Front Tie rods	M	2	-	-	each	TBS	$37.37
Steering system	Fasteners	M8 Countersunk Capscrews for Brake disks	P	12		$1.54	each	NBF	$18.48
Steering system	Fasteners	M10 bolts for attaching steering rack	P	2		$2.34	each	NBF	$4.68
Steering system	Rod Ends	3/8" spherical rod ends for front tie rods	P	4		$2.39	each	TBS	$9.56
Steering system	Brackets	Brackets for attaching rack to chassis	M	2	-	-	each		$0.97
Assembly of Steering system									$17.50
Steering system Sub-total									$144.24

Figure G.10: Page 1 of cost report.

Suspension & Shocks	Front Uprights	Front Uprights and axles	M	2	-	-	each		$874.22
Suspension & Shocks	Front Uprights	Axle end caps	M	2	-	-	each		$23.36
Suspension & Shocks	Front Uprights	Axle end cap studs	P	2		$2.50	each	NBF	$5.00
Suspension & Shocks	Fasteners	Castellated nuts for front axles	P	2		$2.00	each	NBF	$4.00
Suspension & Shocks	Rear Uprights	Rear Uprights	M	2	-	-	each		$303.29
Suspension & Shocks	Tie Rods	Rear Tie rods	M	2	-	-	each		$37.37
Suspension & Shocks	Rod Ends	3/8" spherical rod ends for rear tie rods	P	4		$18.55	each	TBS	$74.20
Suspension & Shocks	Fasteners	M10 bolts for attaching front suspension	P	4		$2.34	each	NBF	$9.36
Suspension & Shocks	Fasteners	3/8" bolts for attaching rear suspension	P	4		$2.39	each	NBF	$9.56
Assembly of Suspension Components									$35.00
Suspension & Shocks Sub-total									$1,340.36

Figure G.11: Page 2 of cost report.

Wheel, Wheel Bearings and Tyres	Bearings	Front Wheel bearings	P	4		$6.65	each	TBS	$26.60
Wheel, Wheel Bearings and Tyres	Seals	Front Wheel Bearing Seals	P	4		$3.95	each	TBS	$15.80
Wheel, Wheel Bearings and Tyres	Fasteners	M12 Bolts for wheel studs	P	8		$1.50	each	NBF	$12.00
Assembly of Wheel Bearings									$35.00
Wheel, Wheel Bearings and Tyres Subtotal									$89.40
Brake System	Fasteners	M10 bolts for attaching front brake calipers	P	2		$2.34	each	NBF	$4.68
Brake System	Fasteners	M8 bolts for attaching front brake calipers	P	2		$1.54	each	NBF	$3.08
Assembly of Brake System Components									$8.75
Brake System Subtotal									$16.51

** PRPS = Peugeot & Renault Parts & Service Drayton Toowoomba PH: 07-46301699 **
** NBF = Nut and Bolt Factory 20 Pechey St Toowoomba PH: 07 46380600 **
** TBS = Toowoomba Bearings and Seals Cnr Taylor & Eyers St Toowoomba PH: 07 46333233 **

Figure G.12: Page 3 of cost report.