University of Southern Queensland

Faculty of Engineering & Surveying

Development of the drivetrain including brakes and wheels for the Formula SAE-A vehicle

A dissertation submitted by

Jeremy Little

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towards the degree of

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Abstract

This project is part of the USQ Motorsport team entry in its first Formula SAE-A competition. The project aim is to design the drivetrain, brakes and wheels of the car which was the focus of developed in this project. The design must also be optimized to provide for a competitive racing car.

The investigation into these areas starts with research into the possible systems available to be adapted to the racing car. The interfaces between each area of the racing car means working with other members of the team to decide on viable options is required. Teamwork is therefore essential for the success of the overall project to complete the construction and perform well at the Formula SAE-A competition.

A design is developed for the drivetrain and detailed analysis of the design is conducted. The main component of the drivetrain is the centre shaft which is analysed to find the stresses and fatigue life. Associated components of the drivetrain such as rear sprocket, chain, bearings, half shafts and chain tensioner are also designed to complete the drivetrain assembly.

The brakes and wheels have also started with background research followed by a detailed selection process to find the best option. Consideration is also required as to how all the components will fit together to provide a car that is well designed and has good performance characteristics.

The finial construction and procurement of the components making up the drivetrain, brakes and wheels was on track for completion to enter the Formula SAE-A competition.

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Date

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Chapter 1

Introduction

1.1 Introduction

This year the University of Southern Queensland (USQ) is entering a competition organised by the Society of Automotive Engineers. The competition involves young engineers to designing, fabricating and compete with a formula type racing car against other universities. As part of the USQ Motorsport team the areas of drivetrain, brakes and wheels are under my control for development on the Formula SAE-A entry in 2004.

The investigation into these areas starts with research into the possible systems available to be adapted to the racing car. The interfaces between each area of the racing car means working with other members of the team to decide on viable options is required. Teamwork is therefore essential for the success of the overall project to complete the construction and perform well at the Formula SAE-A competition.

The outcome of this research will provide a solid foundation for future students involved in developing a racing car for future competitions. This year we have little experience into what designs are best for the develop in a racing car. But future teams will be better equipped with students having a car to test and gain experience for the selection of designs to be implemented. Also with a car constructed from this years competition will allow future teams to test new designs against the old design.

1.2 Project Aim

My project aim is to develop the drivetrain, brakes and wheels for the Formula SAE racing car. The designs must meet all Formula SAE-A rules and regulations including all safety features. The design must also be optimised to provide for a competitive racing car.

1.3 Project Objectives

The objectives of this project as outlined in Appendix A are to review literature in the areas of the drivetrain, brakes and wheels to gain a solid background into these areas. The development of conceptual designs for the implementation in a formula type racing car. Further the selection and undertaking of detailed design work on the best concepts that are within the constraints of the design criteria, time and finances will be undertaken. Considering the constraints involved an investigation into possible parts available for the use in constructing the vehicle is required. Detailed drawing and solid models of the components is required for the design and manufacturing of the components. The implementation of the designs in this years Formula SAE-A car is required for the effective completion of the cars construction ready for the competition.

1.4 Overview of the Dissertation

This dissertation is organised as follows:

Chapter 2 Starts with background information in the areas of drivetrain, brakes and wheels.

Chapter 3 Covers the drivetrain selection and development for the Formula SAE car.

Chapter 4 Concentrates on the Centre Shaft design and analysis.

Chapter 5 Concentrates on the associated drivetrain components to complete the drivetrain design and analysis.

Chapter 6 The brakes and wheels are combined with the selection and procurement processes outlined.

Chapter 7 Concludes the dissertation and suggests further work.

Appendices All the relevant SAE rules, tables, MATLAB files, solid models and detailed drawings involved in the drivetrain, brakes and wheels development are attached.

1.5 Conclusion

As part of the USQ Motorsport team entry in the 2004 Formula SAE-A competition the parts pertaining to my project are the drivetrain, brakes and wheels. The project aim is to design the components for a competitive formula type racing car. The next chapter covers the background of the competition and areas of design for this project.

Chapter 2

Background

2.1 Introduction

This project is the designing of particular sections of a racing car for competing in a competition. The background to the competition and the particular sections of the car pertaining to this project are covered to find the current state of development of drivetrain, brake and wheel designs today.

2.2 Formula SAE

The Formula SAE Challenge is a competition organized by the Society of Automotive Engineers (SAE). Three competitions are held at in different countries including the United States of America, United Kingdom and Australasia in with we will contest the Formula SAE-A Australasia series held in Melbourne for 2004. The Formula SAE competition concept is for a team of students to conceive, design, fabricate and compete with a racing car. Detailed rules are provided for the students to adhere to for the competition so the teams would be challenged in their knowledge, creativity and imagination.

The scenario given to the teams was that a manufacturing firm has engaged them in producing a prototype car for the non-professional weekend autocross racer market. The objectives of the competition are to optimise the acceleration, braking and handling qualities while trying to keep a low cost, easy to maintain and reliable racing car. Further improvements involve the aesthetics comfort and uses of common parts to improve the cars marketability.

To evaluate the vehicles overall design and performance characteristics against the other competitors a series of events will be entered a the details of these events are outlined in Appendix B. The teams will compete in a number of static and dynamic events with the following points allocation:

Static Events	Points
Presentation	75
Engineering Design	150
Cost Analysis	100
Dynamic Events	
Acceleration	75
Skid Pad	50
Autocross	150
Fuel Economy	50
Endurance	350
Total Points	1000

The first of the static events is the cost and manufacturing analysis which is an assessment of the cost considerations used to produce the car. Assessment of the team's ability to present their prototype car to a manufacturing firm will be evaluated in the presentation event. Lastly the design event evaluates the engineering effort to meet the requirements of the competition as well as assessing the teams understanding of the design.

The safety and technical requirements for competition must be satisfied before qualification to compete in the dynamic events is allowed. Comparing each entry in a series of dynamic events aims to reveal the cars performance characteristics compared to the competition. First an acceleration event, timing the cars to travel 75 metres in a straight line on pavement, then a skid-pad event is held to evaluate the cars cornering ability by making a constant radius turn at different speeds. To assess the cars manoeuvrability and handling qualities each car will run separately on a tight course. This is called the autocross event and will combine acceleration, braking and cornering into one event. Next is the hardest test, the endurance and fuel economy event which will evaluate the overall performance and reliability as well as fuel economy of the car.

2.3 Drivetrain

The engine and drivetrain makeup the powertrain of the car with the drivetrain describing the way the engines power is transmitted to the wheels. There are many different forms of drivetrains utilizing different components in various combinations. So the details of these different components that could be used in the cars drivetrain will be briefly described for later evaluation in the selection of a drivetrain design for the SAE car. The Formula SAE rules outline the restrictions the drivetrain needs to meet for the competition.

2.3.1 SAE Constraints

The following is a summary of the Formula SAE Rules in Appendix B, that the drivetrain must have:-

- Four wheels not in a straight line
- Limit on the front and rear track difference
- Only the tyres can touching the ground
- Suspension travel for at least 50.8mm
- Any drivetrain combinations can be implemented
- Scatter shields fitted to any exposed high-speed equipment for safety

2.3.2 Transmissions

The purpose of a transmission is to vary the drivetrain gearing in a way that optimizes the vehicles performance. This is achieved by keeping the engine in the rev range where the horsepower and torque are at a maximum. Transmissions come in a range of variations of three basic configurations including manual, automatic and variable.

Manual transmissions directly connect the engine and rear wheel speeds through gears. The basic operation of a manual transmission involves manually moving a shift lever to select a different gear ratio as the operator desires. To be able to change gears the spinning shaft and collar have to be rotating at the same speed as the required gear so the shaft and gear can be engaged through the dog teeth. This is achieved by double clutching or the use of synchronizers. Double clutching involves disengaging one gear and then varying the revs so the dog teeth can engage for the next gear. With synchronizers it eliminate the need to vary the revs as the collar and gear make frictional contact before engaging so both spin at the same speed. Another important feature required for a manual transmission is a clutch.

Clutches are used in manual transmissions to disengage and engage the engines output shaft enabling the vehicle to stop without having to stop the engine and it is used also in changing gears. The clutch gradually transfers the engines power to the drivetrain by increasing the friction between surfaces until they spin at the same speed. A slipping clutch that doesn't transfer all the engines power is referred to as a worn-out clutch but the art of slipping a clutch in some vehicles is desirable. The reason for this is some engines make their power high in the rev range and to keep the engine revving at peek power can be achieved by slipping the clutch.

The automatic transmission can change gear without the operator manually changing gears. The advantage of the auto is the driver doesn't have to worry about gear changes and can pay more attention on driving. Changing gears is made by a number of inputs such as vehicle speed, throttle position and manual selection, which activates clutches and band brakes through hydraulic circuits. This acts on a single planetary gearset, which is capable of producing a number of gear ratios.

The manual transmission requires a clutch where as the auto transmission requires a torque converter to transfer the engines power. Torque converters are a fluid coupling used in transferring the engines power to an automatic transmission. It acts as a centrifugal pump pushing oil through a turbine connected to the auto. The advantages include the exclusion of any manual operation and it multiplies the torque of the engine to the automatic transmission. The disadvantages is the fluid coupling, as the automatic transmission never reaches the engine speed so the resulting efficiency will never be a good as the direct coupling of the manual transmission. In some vehicles they have a lockup clutch in the torque converter so the engine and automatic rotate at the same speed.

Variable transmissions are known, as Continuously Variable Transmission (CVT). This system is infinitely variable in the "gear" ratios by changing the radius of the drive and driven pulleys. The pulleys are two cones that can move to change the radius the belt drives at providing the infinitely variable ratios.

Until recently CVT systems couldn't compete with automatic transmissions in cost, performance and durability but design advancements have made the CVT viable and are in production of some vehicles today which supply a considerable amounts of torque. The main advantage is the engine will operate at the exact optimum revs for a given power demand from the vehicle so it will optimize the efficiency. There is no need for a clutch or torque converter as the belt is loose at idle. Like the auto there is no need to change gears but the CVT provides a smooth seamless power delivery. The use of CVT in production has been widely used in the four-wheeler market and now even the automotive industry has vehicles in production using CVT systems.

2.3.3 Chain Drive

Chain drive systems transmit power from one rotating shaft to another by chain interlocking with sprockets connected to the shafts. Different types of chains and sprockets include single and multi-strand roller and inverted-tooth. Examples can be seen on pushbikes, motorcycles and engine timing chains. The biggest advantage of chain drive is not being limited by the distance between the drive and driven shafts. The main limitation being only able to transmit power in one plane and alignment is critical for its proper operation. The size of the sprockets can be changed to obtain the desired gearing. To accommodate for different size sprockets and elongation due to wear requires the need to have adjustment to the chain tension either by an idler sprocket or the distance between centres of the shafts changing.

Limitation of the chain drive include the angle of wrap of the sprockets and the need of regular inspections and maintenance of chain and sprocket wear. One advantage over the belt drive systems are when power is not being transmitted the load on the bearings and shafts is reduced whereas belt drives have a constant tension in the belt at all times.

2.3.4 Belt Drive

Belt drive connects pulleys on shafts by a belt to transmit power over any distance. Like chain drive the pulleys must be in the same plane and tension on the belt is need to accommodate for different pulley sizes. Belt tension is also critical as the belt grips the pulleys by means of friction and for this reason the belts require a tensioner that is adjustable. Multiple belts can be used with different types of cross-sections available in flat and V configurations. Another type of belt drive is toothed with uses teeth instead of friction to grip the pulley, which is very similar to chain drives. Examples include electric motors driving other devices, vehicle fan belts and timing belts.

2.3.5 Differential

The aim of the differential is to provide a final gear reduction before splitting the engine torque two ways through separate drive shafts to power the wheels. The unique advantage of the differential is allowing the wheels to spin at different speeds. This is what gave the differential its name. So when a vehicle turns a corner the driven wheels follow a different radius and the differential allows the wheels to follow their arc without the tyres slipping. There are two groups of differential the open and limited slip which will be described in more detail.

The open differential is the most popular and basic differential used in vehicles today. Figure 2.1 illustrates the main parts of the open differential, which are also shared with the limited slip differentials. The key to the open differential is it will deliver the same amount of torque to each wheel regardless of its rotational speed. The major disadvantage is when traction is a problem as the wheel with the least amount of traction will spin and the wheel with good traction will stop. So this limitation has lead to the design of the limited slip differentials.

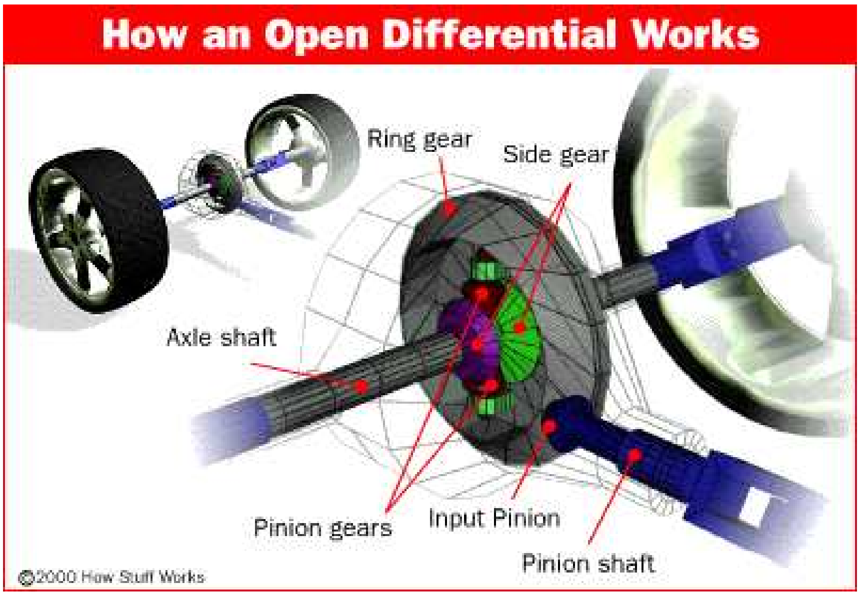

Figure 2.1: Open Differential.

Source: How Stuff Works

The limited slip differential (LSD) or positraction aims to change the torque delivery to give the wheel with traction the most torque. There are different types of LSD mechanisms used to allow normal differential operation for turning but when wheel slippage occurs more torque is given to the non-slip wheel.

The clutch-type LSD is very common and it uses the same parts as the open differential but adds a friction clutch to the assembly. These clutches are either a cone or clutch and spring pack arrangement. The clutch holds the two side gears together so for one wheel to spin freely it must first overcome the force of the clutch. With one wheel with good taction the maximum force that can be supplied to it is equal to the force to overpower the clutch. The disadvantage is found when going around corners as the clutch must be overpowered for the wheels to spin at different speeds.

The viscous coupling LSD is mainly used between the front and rear wheels of four-wheel drives. It works by housing multiple plates submerged in a high viscosity fluid and when the plates are spinning at different speeds they are forced to catch up with each other due to the shear forces created. The disadvantage is that there is no torque transferred until slippage starts. But the faster the wheels spin the more torque is transferred through the viscous coupling. Cornering is not a problem, as the slow speeds don't create much torque.

The locking LSD operates by locking the output shafts together through the side gears. Activated by a manual switch the side gears can be locked together by means of electric, pneumatic or hydraulic mechanisms. So the differential will operate as an open differential until the user decides to activate the locker. Another locker is the Torsen LSD, which is a purely mechanical device. When the difference in wheel speeds reaches a curtain level the gears in the differential lock together so the wheels turn as one.

2.4 Braking System

The braking systems for the racecars have many design attributes that are different to the average car brake designs. The main aim of the brakes is obviously to provide a means of slowing the car down. As the braking system is an important part of the overall performance and safety of the car it is very important to design a system that is more than adequate for it given application. The constraints implied by the formula SAE competition will have to be explored.

2.4.1 SAE Constraints

The following is a summary of the Formula SAE Rules Appendix B, that the brakes must have:-

- Act on all four wheels by a single control
- Brake-by-wire systems are prohibited
- Two circuits with separate fluid reserves
- A single brake acting on a LSD is acceptable
- Scatter shields for safety
- The ability to lock all four wheels at high speed

2.4.2 Types of Braking Systems

There are three basic brake designs including band, drum and disk. The brakes function is to absorb energy by converting kinetic and potential energy into frictional heat (Juvinall & Marshek 2000). They achieve this using the principles of leverage, hydraulics and friction to multiply a human input like as a foot force into a braking force to stop a rotating cylinder such as a wheel.

Band brakes are a simple mechanism consisting of a metal strap lined with a woven friction material wrapped around a rotating cylinder. The ends of the strap are connecting to a tightening device such as a lever and when the strap is tightened a braking force is applied around the cylinder. One use of the band brake is in automatic transmissions to stop the gearset drums from rotating to change gears.

Drum brakes come in two types describing whether the brake shoes are external and internal. The internal shoe types are commonly used on vehicles and work by forcing the stationary shoes out radially to contact the inner surface of a rotating drum closely located to the shoes. The drum brake has the disadvantages compared to disk brakes of using more parts and being harder to service. But they are cheaper to manufacture and provide an easy means of incorporating a handbrake. Some brakes actually have a self-actuating (or self-energizing) action by wedging the shoes into the drum. As long as it isn't self-locking this reduces the force required to apply the brakes.

Disk brakes exert a normal force on a rotating disk to provide a braking force. Disks have replaced drum brakes recently mainly due to their better heat dissipation as well as simplicity of the design, easy inspection and replacement of the pads and their self-adjusting ability. This simple design consists of a rotor, pads and caliper (Figure 2.2) is from How Stuff Works, but the cost of manufacturing disk brake components is higher than drum brakes.

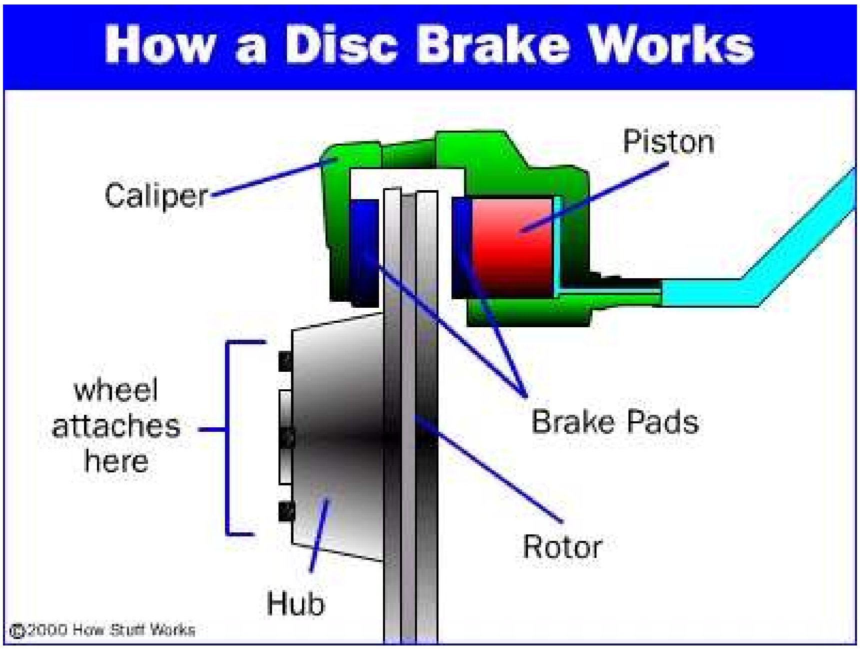

Figure 2.2: Disk brake assembly.

2.4.3 Brake Component Descriptions

The following descriptions will cover various components used in disk and drum brake systems as well as other features that can be integrated into the designs.

- Master Cylinders convert the mechanical leverage into pressurizing the hydraulic system through the displacement of small pistons. The incorporation of the brake fluid reservoir and usually both primary (front) and secondary (rear) pistons for the hydraulic circuits are housed in the one assembly.
- Hydraulic, cable and mechanical linkages are used to transmit forces to the braking system. Hydraulic and cable systems have the flexibility of taking any route over large distances compared to the mechanical linkage.
- Brake fluid is the name for a relatively incompressible fluid used in the hydraulic

circuit. There are three types of fluids the glycol-based (mineral) Dot 3 and Dot 4 and, the silicon-based (synthetic) Dot 5. The difference between these two types is the silicon-based doesn't absorb water when exposed to air but this means it then can form water pockets in the fluid system. Brake fluids have a high boiling point to prevent air bubbles forming, which is undesirable because air is compressible. Mixing of any type of brake fluids can cause corrosion inside the system and also the glycol-based fluids will corrode paint.

- Calipers transmit the fluid pressure applied at the master cylinder to a normal force by clamping the rotor between brake pads. Early models used pistons on both sides of the disk but the improved designs use either a floating caliper or disk to keep proper wear and alignment.
- Slave Cylinders transmit the fluid pressure into forcing the brake shoes in drum brakes out radially against the drum.
- Rotors are connected to the rotating member to provide a surface for the caliper to clamp in disk brakes. Rotors come in a range of patterns such as vented, slotted, drilled or solid for the given application and appearance. The reason for the patterns are to provide a cleaning method of the pads and also to help cooling.
- Emergency Brakes (handbrakes) connect to the rear brakes to provide a means of applying the brake while the driver leaves the vehicle stationary for a period. The connection is always through cables so in the event of brake failure the emergency brake can be applied.
- Power Brakes (Brake Boosters) work by using a vacuum created by the car to assist in applying the brake pedal. The vacuum can be held for several stops in the case of engine failure.
- Combination Valves have a metering valve, pressure differential switch and proportioning valve combined together and located in the brake line route. The metering valve is for drum/disk set ups allowing pressure first to the rear drum brakes for stability. The pressure differential switch detects leaks in the hydraulic circuit. Proportioning valves are for reducing the pressure to the rear brakes as they require less pressure.

- Anti-lock Brake Systems (ABS) is a means of braking safely without locking the brakes. Skidding tyres provide less traction and the vehicle can't be steered. The system monitors the vehicle, wheel and driver behavior and applies the brakes in pulses at the right amount and duration to keep the tyres turning at maximum grip.

2.4.4 Problems with Brake Systems

According to the Physics of Racing (Beckman 1991) the main problem encountered by brake systems stems from their purpose of absorbing energy and converting to heat energy. If the heat energy can't be dissipated into the surroundings fast enough the brakes will heat up to temperatures above their normal operation conditions. This firstly affects the brake pads/shoes, which are designed to work at high temperatures but only up to a point before their coefficient of friction is affected which affects the frictional force. This result is called brake fade where there is a reduction in the brakes braking ability.

If the temperature of the brake system continues to rise it will reach the point where the brake fluid boils. This results in air bubbles being created in the hydraulic circuit with is not desirable as air is compressible which means the pressure to the brakes is no longer maintained and total brake failure occurs.

2.5 Wheels

The important interface between the racing car and the track surface is made through the wheels. Vehicles handling qualities can be changed through the wheel selection as the wheels affect the suspension, steering, braking, drive and comfort of the overall design. Provided the vehicle has enough power then the traction between the tyre and track surface interface is the limiting boundary for the potential performance.

2.5.1 SAE Constraints

The following is a summary of the Formula SAE Rules Appendix B, that pertain to the wheels:-

- A minimum outside tyre diameter of 203.2mm
- Dry or wet weather tyres are allowed
- Any size or type of rims and tyres
- Same rim size is required for both dry and wet weather tyres
- Adhere to restrictions to tyre tread patterns

2.5.2 Wheel Dynamics and Terminology

The following terms describe the handling characteristics and measurements of a vehicles steering geometry, which are related to the wheels.

- Understeer describes the behavior of the front wheels loosing grip and the vehicle has a feeling of pushing the front wheels straight ahead. The problem is the front tyres don't have enough grip so there is a delay before the vehicle turns.
- Oversteer is the opposite to understeer so the front has more grip than the rear. The result is the rear slides out so the front wheels are turned to counter react this behavior.
- Caster is the change in angle the wheel makes from the straight in line direction.
- Camber describes the angle of the wheel whether it leans in or out at the top.
- Slip angle or grip angle is the difference in angle of the contact patch and tyre direction. The contact patch is the contact area of the tread on the track surface.
- Aspect Ratio is the length to width ratio of the tyre tread contact patch.

2.5.3 Wheel Rims

Wheel rims provide a means of fitting a tyre as well as clearing the brake system, tie rod ends, suspension and mounts. All rim designs have the basic measurements including the bolt pattern, diameter, width, backspace and offset to identify the geometry of the rim. The following Figure 2.3 depicts the bolt pattern measurements of various lug numbers.

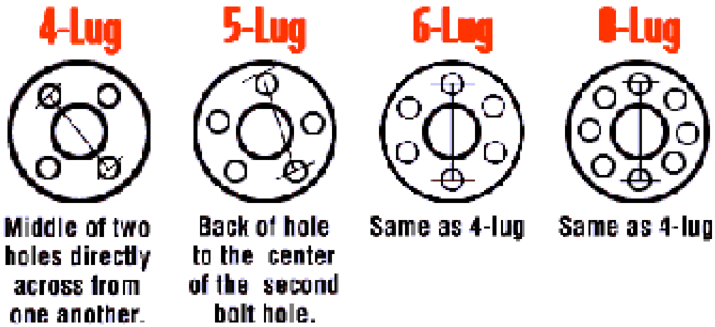

Figure 2.3: Lug pattern measurements.

Source: RS Racing Wheels and Tyres

The width and diameter of the wheel rim are the major measurements for tyre specifications. The next diagram Figure 2.4 from RS Racing shows the rim terminology and dimensions. Wheel backspace marked as the rear spacing along with the wheel offset measures the mounting flange position relative to the centreline as either positive or negative.

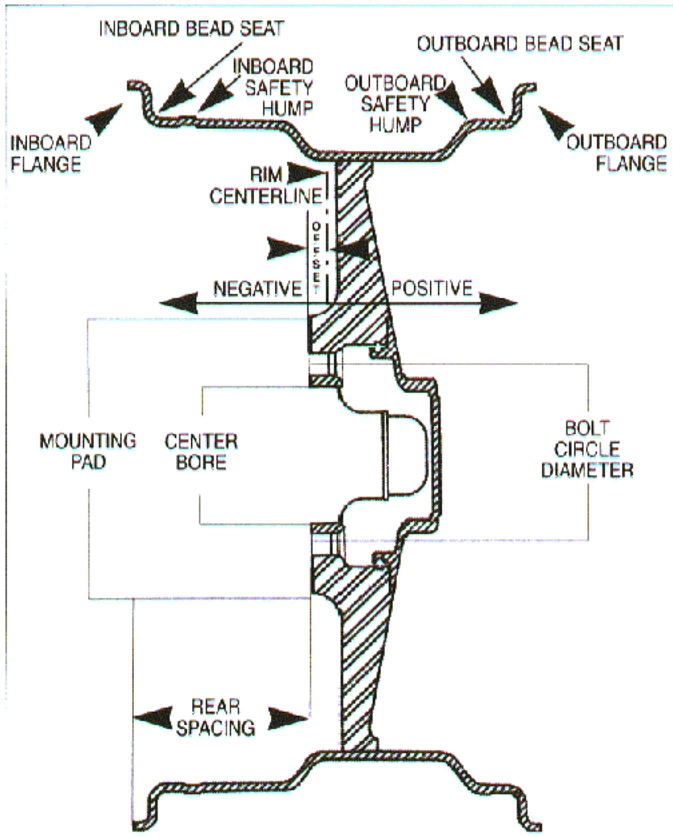

Figure 2.4: Wheel rim measurements.

2.5.4 Tyre Designs

It is convenient to know the different tyre sizing systems to specify a tyre geometry. The use of SI and imperial units are used in many ways for identification of tyre sizes and the following list shows the systems used in the past to the present (Yokohama current May 2004).

- Numeric Sizing System was used until the late 1960's with measurements in inches for example 7.75 -14. The first number is the approximate section width then the rim diameter follows. If the last digit is a zero in the width designation then the aspect ratio is approximately 92 percent and if it's the number five then it is approximately 82 percent, which was considered "low profile".
- Alpha-Numeric Sizing System is a load-based system, which was introduced world wide in 1968. For example FR70-14, with the first letter indicating the load rating

and the second letter for the type of construction. The number indicated the aspect ratio followed by the rim diameter in inches.

- P-Metric Sizing System represents a passenger metric system, which was introduced in 1976. For example P215/70R14 with the "P" standing for passenger car designation, then the section width in millimetres followed by the aspect ratio. The next letter indicated the type of construction then the rim size in inches follows.

- ISO Metric Sizing System is the same as the numeric sizing system converted to metric measurements. When aspect ratio's started to change they were then added like the p-metric system. The addition of service descriptions indicating the load index number and speed rating symbol are added to the end for example 185/60R14 82H.

- Millimetric Sizing System uses the ISO metric system but with the rim diameter in millimetres for example 210/65R 365.

2.5.5 Different Types of Tyres

The manufacturing process for different tyre constructions is the same process. Basically a machine cuts and wraps the layers called plies together to form what is called a green tyre with no tread or markings present. Next is the curing machine, which moulds the tread, markings and bonds the components together, which is call vulcanising.

Tyres can be divided into different types of constructions including the bias ply, radial ply and combinations of the two called belted bias. The contents of this section were sourest from RS Racing and How Stuff Works.

The radial ply tyre describes the different construction setting it apart by the body cords running directly across the tread and under the tread is a belt over wrap made of steel or fibreglass mesh. Therefore the characteristics of radials are better tread contact due to the stiff tread and movements taken up by sidewall distortion. As a result less warning is given before the tyre reaches the breakaway point at which slippage occurs. The pressure can be changed to alter sidewall rollover. Radials operate a lower slip angles compared to bias ply translating to better transient response for the driver. RS Racing recommends radials are good where transient handling is a concern and adequate negative camber is available.

Bias ply tyres differ to the radial in the construction with the body cords criss-crossing across the tyre between angles of 32 to 40 degrees and there are fewer plies. The result is a lighter tyre with stiff walls and a flexible tread. Tyre pressures are critical due to the effect of the tread flexibility. The driver has more feedback from the tyre behavior and can drive at the tyres limit. The larger slip angles means the driver has to 'lead' the corner more. RS Racing recommends bias tyres when negative camber is limited but only with a wide enough rim.

Belted Bias Tyres are combination between the previous two tyres giving the belted bias tyre, which is called a racing radial tyre. The combination provides good feedback like the bias ply and a higher breakaway traction like the radial ply.

Tyre tread patterns are made up of some basic shapes in different patterns. The tread design improves the traction, handling and durability as well as ride comfort, noise level and fuel efficiency for the given application. The basic shapes include blocks, ribs, sipes and dimples for achieving the design characteristics.

One important aspect of tyre design is the void ratio, which is a measure of the amount of grooves in the tyre. The advantage of a high void ratio is the ability to handle wet weather by dispersing the water in front of the wheel so contact with the track surface is maintained.

2.6 Conclusion

This concludes the background to the Formula SAE-A competition, drivetrain, brakes and wheels. The selection of designs and components can now be made with the basic knowledge gained from this research. The next chapter covers the drivetrain selection and development for the SAE car and starts the analysis of the loading situation.

Chapter 3

Drivetrain Selection and Development

3.1 Introduction

The major part of this project is the drivetrain design. In developing a drivetrain for the Formula SAE-A racing car I first started with the design criteria and then developed conceptual designs and then chose a design to be developed in the racecar. All the drivetrain development decisions up to the detailed analysis are summarised to show the reasons for implementing the specific design. To conduct a detail analysis the gearing and loading of the shaft are detailed to provide the necessary design information for all the drivetrain components.

3.2 Design Criteria

The design criteria for the drivetrain system has a number of different goals and limitations to consider in evaluating the best final design. These aspects cover the Formula SAE rules, interfaces between components, performance characteristics, reliability, cost and the teams overall aim.

The Formula SAE rules, which were discussed earlier generally, give some guidelines to the layout and aspects that need to be accommodated by the design. But the rules don't limit the design selection in any way so free range was given in choosing a drivetrain design. But obviously the drivetrain is not the only component of the cars design and must work together with the other areas of the car such as the brake, suspension, chassis and engine systems. So the interfaces between each of the components needs coordinating by the designers to ensure that the drivetrain accommodates for all the required features, space and mounting requirements.

The aim of the drivetrain is to transmit the torque of the engine to the wheels in the most efficient and easiest way possible to maximize the cars traction across a broad range of maneuvers. A proper appreciation of the dynamic tests was required to fully understand the desired car performance characteristics. The overall performance of the car will be optimised by each component such as the drivetrain working together to complement the advantages of each design. Another aspect of improving the performance is reducing the weight of each component, which will improve the power to weight ratio.

The cars reliability is another important aspect of the design with the failure of any component resulting in a possible DNF (Did Not Finish) and consequently zero points allocated for the event. So the service life of the drivetrain should be evaluated to ensure that the car doesn't fail during the competition. The design has to use parts that are readily available so in the event of component failure new parts can be acquired from local sources. Also the replacement and maintenance of the various drivetrain components needs to be addressed.

The cars cost is also a major factor in the designing of the drivetrain. The car will be scored on the costing report so it is important to consider the cost incurred by each design. With this being the USQ Motorsport team's first entry in the competition our budget is limited so the cost becomes one of the most important design criteria. With added sponsorship a wider selection of alternative designs and products could be evaluated. Utilising existing components acquired during the development of the car was therefore a high priority. The question of whether the higher cost of components was worth the performance or reliability advantage was continually addressed whereas in a racing team this is usually the last consideration when performance is the main priority.

In summary the design criteria for the drivetrain was mostly governed by the USQ Motorsport team aim. Which is to design a car with high reliability and low cost for our first entry in the formula SAE competition and ensure the car successfully completes every event.

3.3 Drivetrain Design Selection

The drivetrain design is open to any type of system of transmitting the power from the engine to the rear wheels. So designs including combinations of chain, belt, differentials and CVT were initially considered and their relative merits noted. Different conceptual designs are conceived to for implementing in a rear wheel drive independent suspension car. All the particulars of each component is covered in the drivetrain background.

The first design option is a chain drive system with a solid drive axle. Examples include small racing karts and four-wheelers. The advantages are:-

- Direct drive for good straight line acceleration
- Compared to belts less tension in the chain when coasting
- Also no power loss through slippage
- Cost effective
- Light weight
- Simple and reliable design
- Ease of maintenance and replacement

The disadvantages are:-

- Cornering ability affected

- Tyres required to slip when cornering
- Lubrication of chain required
- Chain tensioning required
- Noise of chain and sprocket operation

The second option is using a toothed belt drive system to transmit power to a solid rear axle. The advantages are:-

- Direct drive for good straight line acceleration
- Quiet and smooth operation
- Cost effective
- Light weight
- Simple design
- Ease of maintenance and replacement

The disadvantages are:-

- Slippage possible
- Cornering ability affected
- Tyres require slipping when cornering
- Tensioning required
- Initial tension required for belt/s causing extra forces on shaft and bearings
- Harder to implement

The third option is implementing a locking differential the patented TORSEN. A special version is available especially for the Formula SAE vehicles called the university special (Toyoda-Koki 2004). Originally the differential come out of a Audi Quattro. Other options include the differentials out of a Mazda Miata or Toyota Rav4/Lexus RX300. The engines power will be transmitted to the differential via a chain drive system. This design is a popular choice for SAE cars. The advantages are:-

- Direct drive for good straight line acceleration
- Cornering ability improved as differential allows different tyre speeds when cornering
- Small variety of options with possibility of components purchased from wreckers

The disadvantages are:-

- Higher cost
- Required chain drive from motorcycle engine
- Torque Bias requires setting up to be of full benefit
- Complex design with more components
- Chain tensioning required
- Weight
- Maintenance required
- Oil leakages possible

The forth option is a Continually Variable Transmission (CVT). The idea behind the design is for two pulleys to be used one mounted to the crankshaft which is variable and the other to the rear axle. The variable pulley alters the radius of the belt position according to the engine speed. This system can be implemented on a solid axle or to a differential. Some examples in use today are modern four-wheelers and the Audi A4. The advantages are:-

- Different engine choices possible

- Theoretically supplies maximum torque at all times
- Quiet and smooth operation
- No gear changes required making it simple to operate
- Infinitely variable gear ratios within limits
- Like a chain drive tension is only provided when under acceleration

The disadvantages are:-

- High cost
- Getting pulley variability right
- Complex front pulley design
- Losses in pulley friction
- Weight

Considering the design criteria specified a drivetrain design was selected. The chain drive and solid rear axle was chosen based mainly on the simplicity, reliability and low cost. The motorcycle engine also uses a chain drive so implementing the design doesn't require any changes to the engine. This decision was a compromise on the car cornering ability which could be improved by using a differential option. With changes to the rear track being less than the front and the right suspension setup the inside tyre should be able to slip in tight corners enabling effective cornering maneuvers. Changes to the implementation of the chain drive were made as the design was developed.

3.4 Drivetrain Development

The drivetrain development started with the selection of a simple straight axle and a one-piece swingarm suspension arrangement for the rear assembly. Then consultations with the suspension designer were undertaken to discuss how the swingarm suspension system with a solid axle would work together. This design was very similar to the racing four-wheelers. The main point of discussion included the mounting of the axle to the swingarm and still accommodating for chain adjustment. As the design was being developed our appreciation of the handling characteristics required by the car were becoming more apparent. The adverse handling effects that our current design was going to exhibit was becoming a concern.

The team was concerned about the initial design of a swingarm suspension and solid axle and how it would affect the handling of the car. The team came to the conclusion that the suspension at the rear had to be independent the same as the front for both ends to really work together. The benefit of implementing a differential in the car was also raised at the same time but the solid axle was retained. The reasoning behind this was the car would still have acceptable performance characteristics for our first entry. Also the time left to implement a differential in the car was not sufficient and our finances would not allow the purchase of a differential at that stage. Changes to the rear track reducing it to less than the front and using stiff spring rates at the rear would help the rear wheels to slip when cornering.

With the independent suspension system at the rear being implemented it changed the configuration of the rear axle. The axle no longer carried any axial or vertical loads from the tyres, as the suspension would wholly support the weight of the car both statically and dynamically. The new requirement now for the drive is to accommodate for vertical movement of the axle drive ends. The use of universal or CV joints was therefore required.

The first solution, which jumped to mind was utilising the front CV shafts out of small passenger cars. The CV joint is chosen for its ability to cater for axial displacement and the constant velocity at any angle of rotation. So with some thought to how these shafts would be implemented and perform in the design were assessed. The drivetrain design concept now centred around designing a centre shaft which links both sides of the drive through CV joints and half shafts. The main concern with implementing the CV shafts was how they would attach to the centre shaft. It is much easier to machine an external spline so the CV joints had to have internal spines and this was the only condition other than weight and cost in the procurement of some CV shafts to implement.

How the CV shafts would perform handling the torque and suspension travel of our situation was assessed. The torque produced by small car engines was more than the torque produced by our 600cc engine so the size and material properties of the half shafts would be more than adequate. The vertical jounce and rebound required by the suspension travel was well within the range of the CV drive shafts even if the half shafts were shortened. The CV joints could also cater for different shaft orientations without any problems.

With the experience of one of our team members the drive shafts used in the Ford Telstar were selected because they used an internal splines on the CV joints. Another advantage was the wheel stud pattern was the same as the rims we had earlier acquired. Two left hand drive shaft assemblies were purchased from City Auto Wreckers. The components included the main drive shaft, bearing block, upper CV, half shaft and lower CV, which cost $44 per assembly. Also the front hub assembly was purchased so the internals could be used in the rear uprights and they cost $33 each.

At this stage the acquired FZR 600 engine was mounted in the spaceframe and the rear end of the frame constructed with its size based on accommodating the rear sprocket diameter. All the components were measured and preliminary sketches of the drivetrain assembly drawn. Different positions of the drivetrain assembly in the spaceframe were tried with consideration to the vertical height and space requirements. The final positioning of the CV joints and bearing housings lead to the calculation of the centre shaft length. Also using the track dimension specified by the team the half shaft lengths were found.

The design now featured a centre shaft driving the wheels via CV joints and half shafts. The drive was provided by chain and sprockets with the sprocket mounted to the shaft by a flange. The rear wheels required a brake assembly so one of the front brake rotors off the FZR was to be mounted to the shaft and operated on by the FZR's rear caliper. Another feature the drivetrain needed was chain adjustment and method chosen will affect the brake caliper mounting. One option of adjusting is to move the axle but this means the caliper needs to move also. The second option is use a chain tensioner on the slack side of the chain and this method was chosen.

This was the development process for implementing a drivetrain system in the SAE car. With the preliminary drivetrain configuration developed the design of each individual component require for the drivetrain assembly could be undertaken.

3.5 Final Gearing

The final gearing is required to analyse the loading of the shaft and the gearing specified would affect the performance of the car in its acceleration and top speed. To determine the required gearing ratio of the car, research into the top speed required in the competition revealed a speed of 105km/h from the brief technical specification (Society of Automotive Engineers 2004*b*). Further study into previous entrants to the SAE competition revealed a common top speed of about 130km/h.

The question of whether the engine could still reach its maximum rpm with the restrictor was a concern. So for the initial predictions it was determined that the engines rpm would not be affected and to aim for 130km/h (36.11m/s) top speed so if the rpm was affected there was a little safety margin. To determine the required final gear ratio a few quick hand calculations using the bikes maximum speed were conducted. Research revealed the top speed of the standard FZR 600 is 229km/h (63.61m/s) with 45/15 gearing (Bikez.com Motorcycle Encyclopaedia current June 2004). The bikes rear wheel measures 625mm in diameter and the car wheels are 552mm in diameter. The following formula from (Meriam & Kraige 1998) equation number (2/11) was used.

$$\nu = \omega r \tag{3.1}$$

where: ν = velocity (m/s)

ω = angular velocity (rad/s)

r = radius (m)

Rearranging and finding the angular velocity of the bikes rear wheel, R:

$$\omega_R = \frac{\nu_R}{r_R} = \frac{36.11}{0.3125} = 203.5 rad/s$$

Then the front sprocket, F angular velocity is found with standard gearing ratio 45/15:

$$\omega_F = \left(\frac{45}{15}\right) 203.5 = 610.66 rad/s$$

The desired angular velocity of the car wheels, C is found:

$$\omega_C = \frac{\nu_C}{r_C} = \frac{36.11}{0.2765} = 130.6 rad/s$$

Now the required final gearing ratio will be close to the following with t = number of teeth:

$$\frac{\omega_F}{\omega_C} = \frac{t_C}{t_F} = \frac{610.65}{130.6} = 4.676$$

This allowed the front and rear sprocket size combination be determined. With an off the shelf selection of 13, 14 or 15 tooth front sprockets then a suitable rear sprocket size was required to obtain the gear ratio. A 60-tooth rear sprocket with a 13-tooth front gives a 4.62 gear ratio so this gearing was chosen.

Next the top speeds in each gear at the maximum design rpm (11000rpm) were determined using the service data (Yamaha Motor 2004) for the 1993 FZR600. See Appendix C for tables of the selected gear (1-6) top speeds for different secondary reduction ratios including the standard FZR 45/15, 60/13, 60/14 and 60/13 ratios. The following gear ratios were used in the calculations:

- Primary Reduction Ratio = 82/48 (1.708)
- Secondary Reduction Ratios:
  - 1st = 37/13 (2.846)
  - 2nd = 37/19 (1.974)
  - 3rd = 31/20 (1.550)
  - 4th = 28/21 (1.333)
  - 5th = 31/26 (1.192)
  - 6th = 30/27 (1.111)

With the secondary reduction ratios evaluated the option of running a range of front sprockets to change the top speeds could be achieved depending on the track layout.

3.6 Axle Loading

To design the drivetrain components the loads exerted on the axle are required. Assessing the situation revealed that the only significant load exerted on the axle would be transmitted by the chain drive. The resulting forces would be a torque and a force pulling the axle forward. The masses of the drivetrain components were considered negligible to simplify the loading. Now three different approaches to determining the maximum torque on the rear axle were explored including the engine performance, tyre friction and the predicted acceleration to determine the actual torque.

3.6.1 Engine Performance

Initially the engine performance was used to find the maximum torque induced in the rear axle. The engine is a 1993 model Yamaha FZR 600 and has unrestricted power and torque figures quoted at approximately 60.1KW at 10600rpm and 60.33Nm at 8500rpm respectively measured at the engines crankshaft. To determine the amount of torque transferred to the rear axle of the car a number of factors needed to be considered. Firstly these figures are quoted on a new engine and our engine has traveled over 60000km so this will affect the performance. Secondly the engine will have a restricted air intake. Thirdly frictional losses would occur through the transmission and chain drive.

An estimate of the engines performance with the restrictor was based on previous SAE entrants with quoted figures of approximately 57Nm torque. This figure was judged to be an over estimate as many teams run current model engines whereas our engine is using 12-year-old technology. Also the frictional loss of the drive was predicted to be 15 percent after consultation with Brisbane Dyno Bikes. The final estimated maximum torque of the engine was 48.5Nm.

With the specifications on the FZR transmission the engine torque is multiplied by the gear reduction ratios to find the maximum torque that can be delivered to the rear axle. It starts with the primary reduction ratio multiplier then next the selected gear ratio that multiplies the torque the most, which is first gear. Then finally the final drive or secondary reduction ratio multiplies the torque. The final torque works out to be 943Nm. A MATLAB program Engine Torque Appendix D was written to quickly predict the torque at the rear axle for different engine torque predictions.

3.6.2 Tyre Friction

The maximum torque produced in the rear axle is limited by the amount of torque the tyres can transmit before slipping. Determining the force at which the tyres will slip is controlled by the normal force and coefficient of friction. Determining the coefficient of friction, μ for the Falken ZE512 road legal tyre was estimated from the range of values presented in various web sites. The values ranged from 0.6 to 1.4 with passenger car tyres as high as 0.95 so a coefficient of 1 was used in the calculations.

The normal force on the rear tyres under acceleration was initially estimated based on a 30:70 front to back weight distribution with an assumed total mass of 350kg including a 100kg driver. The calculations for the torque are as follows.

Newton's second law equation (1/1) (Meriam & Kraige 1998).

$$F = mg \tag{3.2}$$

where: F = force (N)

m = mass (kg)

g = acceleration due to gravity (m/s^2)

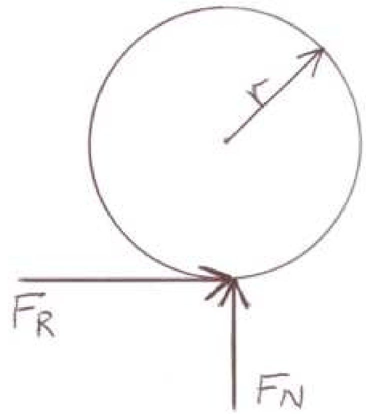

Figure 3.1: Free Body Diagram of tyre.

The normal force on the rear tyres.

$$F_N = mg = 350\left(\frac{70}{100}\right)9.81 = 2403.45N$$

Frictional force provided by rear tyres if $\mu = 1$.

$$F_R = F_N\mu = 2403.45 \times 1 = 2403.45N$$

The moment created by the rear wheel is found by equation (6/5) (Meriam & Kraige 1998).

$$\Sigma M = \Sigma m\bar{a}d \tag{3.3}$$

Expressed in the form of torque.

$$T = Fr \tag{3.4}$$

where: T = torque (Nm)
F = force (N)
r = radius (m)

Total torque produced in shaft.

$$T = F_R r = 2403.45 \times 0.2765 = 664.5Nm \tag{3.5}$$

So these initial calculations produced a torque being delivered to the rear axle of 664.5Nm. Later on when the centre of mass of the car was predicted the estimated weight distribution was checked. This was achieved by summing all the forces and moments but this required an acceleration prediction. A prediction of 0-100km/h in 3 second was chosen which is an absolute upper limit of the car's performance.

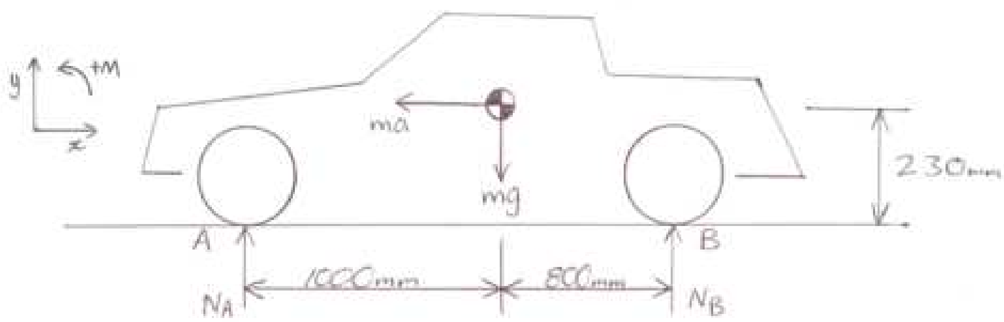

Figure 3.2: Free Body Diagram of accelerating Car.

where: N = Normal force (N)

m = mass (kg)

a = acceleration of car (m/s^2)

g = gravity (m/s^2)

The mass is still 350kg, and the acceleration is:

$$a = \frac{27.77}{3} = 9.256m/s^2$$

Using the previous Equation 3.3 the sum of moments about A gives the normal force on the rear wheels:

$$\begin{aligned} 1.8N_B - 350 \times 9.81 \times 1 &= 350 \times 9.256 \times 0.23 \\ N_B &= 2321.48N \end{aligned}$$

The percentage of weight distribution on the rear wheel under acceleration is then found.

$$Percentage = \frac{N_B}{mg} \times 100 = \frac{2321.48}{3433.5} \times 100 = 68\%$$

This produced a 32:68 front to back weight distribution, which is less than the initial estimation of 30:70. The torque produced in the rear axle was base on the initial estimated weight distribution giving 664.5Nm. The previous calculations to find the torque were also repeated in a MATLAB program Tyre Torque Appendix D so if any values change the torque could be quickly found.

3.6.3 Acceleration

Another method of determining the torque in the rear axle is by using the acceleration of the car. This method has the limitation of firstly assuming constant acceleration over the distance timed and secondly predicting the acceleration that our car will achieve is unknown. So for these reasons the acceleration of the car is only a check to see if the torque found previously is a reasonable estimate. The acceleration figures from previous SAE entrants were readily available so they are listed in Table 3.1.

Table 3.1: Acceleration figures of SAE cars.

Team	Speed	Time(sec)	Mass(kg)	Acceleration(m/s^2)
Cornell	0-60mph	3.2	216.5	8.38
Buckeyes	0-60mph	3.4	unknown	7.89
Queens	0-60mph	3.9	236	6.88
Toronto	0-100km/h	2.9	212.5	9.26
Wollongong	0-100km/h	3.6	210	7.72

From these results a desirable time to accelerate from zero to 100km/h would be about 3 seconds corresponding to an acceleration of $9.259m/s^2$. If our car were capable of this accelerating the torque produced in the rear axle is found to be 627.23Nm. Comparing this torque value and the potential torque supplied by the tyres reveals they are very close so this is a good check to confirm the calculated torque value.

3.7 Conclusion

The development of the preliminary design of the drivetrain assembly is documented to show the reasons behind each design decision. The specifying of the finial gearing allowed the calculation of torque produced in the axle. The first method of assuming all the power of the engine reaches the wheels is an extreme case where only if the tyres had infinite grip could this ever occur. But tyres are limited in the amount of grip they have on the surface. A tyre will slip at a certain amount of torque limited by the friction force between the tyre and asphalt. This second method, based on the tyre friction is the most valid prediction and is confirmed by the acceleration estimate to be in the vicinity of the right torque. All the information required to start a detailed analysis of each drivetrain component can be undertaken. This is described in the next chapter.

Chapter 4

Centre Shaft Design and Analysis

4.1 Introduction

This chapter covers the design and analysis of the centre shaft with two designs being assessed. The first is the prototype centre shaft manufactured for the engine dyno testing the second is a improved design after analyising the prototype shaft. The process of designing the centre shaft is outlined along with all the information required to manufacture the shaft. Detailed stress analysis, fatigue life and deflection calculations are performed on the two designs.

4.2 Centre Shaft Design

The centre shaft design is influenced by a number of factors in its positioning and layout to accommodate all the required features. To start with, the acquired components were physically positioned in the rear spaceframe to take measurements. Initially sketches were developed before a solid model (Figure E.1) for the centre shaft configuration was created.

As the CV joints and bearing housings from the Telstar assembly were going to be utilised in the design, the existing dimensions of the shaft ends were incorporated in to the new centre shaft. The measurement of the overall shaft length was calculated from sketches of the frame geometry and considerations for the CV joint and bearing housing dimensions. The involute splines were specified so they could be reproduced in the new shaft.

Early on in the development of the centre shaft a prototype shaft was manufactured for the purpose of dyno testing the cars engine performance. The material with which the centre shaft was manufactured from was mild steel AISI 1020. At that stage the stresses, fatigue and deflections of the shaft were unknown so the diameter of the shaft was left at 38mm to accommodate the dust seal flange of the then current bearing housings. This shaft has now become the finial shaft used in the due to constraints in finance and time resources. The analysis of this shaft was conducted and the material properties that should be used are presented later in the improved design section.

4.2.1 Positioning

The positioning of the centre shaft incorporated the mounting method and location. The geometry of the rear spaceframe seen in Figure 4.1 below was set to accommodate for the suspension and sprocket size. Positioning options are represented by solid models in Appendix E.

Different mounting concepts to provide chain tensioning were considered. But the use of a chain tensioner was seen as the easiest way of providing chain tension, which meant the shaft could be fixed in position. Physical positioning of the bearing housing and CV joint assembly in the frame showed the joints needed to be located on the outside of the frame rails to keep the centre shaft level with the wheel centre. The joints are obstructed by the frame rails if mounted closer to the centre of the frame. Outside also allowed for the mounting of a bracket on the top frame rail for the bearing housing to be mounted. The only negative, which was realised later, is the bearing is a long way from the driven sprocket. Another problem to address was clearance for the chain as a frame support was in the potential chain path as seen in Figure 4.2. Keeping the shaft mounted as high as possible was a priority.

Figure 4.1: Positioning of CV joint and bearing housing in spaceframe.

Figure 4.2: Drivetrain assembly with arrow indicating the point of chain contact with frame.

It is desirable to keep the centre shaft positioned in the middle of the wheels so the wheelbase set the position of the shaft. But due to the chain clearance problem the shaft needed to be moved forward to increase the clearance. The angle of wrap is another consideration, which is addressed later in Chapter 5. The CV joints are designed to handle angular movement and the additional friction produced by the misalignment with the wheels is very small. Consideration was given to the suspension travel and the range of movement the CV can cater for, as the half shafts move in and out with the suspension travel. The range of movement was well within the joints capability.

4.2.2 Sprocket and Rotor Flanges

To cater for the chain drive a flange is required to mount the rear sprocket. With the engine mounted in the frame the position of the front sprocket was measured relative to the frame close to the sprocket. The problem of any misalignment errors was considered and designing the shaft to cater for a spacer or shimming at the flange was considered. However, the idea of moving the front sprocket by shimming or machining material off its spacer was considered adequate in the event of misalignment. Also the position of the rear brake rotor was physically positioned in the frame as seen in Figure 4.3 and measurements taken relative to the frame rails. The caliper governed the positioning of the rotor as to where the best mounting points were on the frame.

Figure 4.3: Positioning of potential rear brake assembly location.

Governing the diameter of the shaft flanges to mount the sprocket and rotor was the geometry of the rear brake rotor mounting surface. The rotor was off the front of the FZR 600 and a 100mm diameter flange was required for the rotor and the sprocket could be designed to the same size as well. Originally the centre shaft was going to be manufactured by machining the shaft down from one piece of material but with advice from the workshop staff the flanges were then welded to a smaller shaft, which reduces the cost immensely. The flanges have six holes for M10 bolts and have a flange for locating the sprocket and rotor on the shaft.

4.2.3 Involute Splines

The reason for selecting the CV shaft with internal splines was the ease of manufacturing the centre shaft with only external splines. The existing spline was a 26 tooth involute spline with a root diameter of 25mm and outside diameter of 26.7mm. The pressure angle and tooth profiles were unknown due to being very small to measure. To determine the dimensions the of the spline profile a section of the spline was cut off and ground smooth. Using the Mitutoyo Model PJ300 profile projector the tooth profile could be measured as seen in Figure 4.4. Repeated measurements of the tooth dimensions were conducted to find the average value and the results are tabulated in the following Table 4.1.

Table 4.1: Involute spline tooth measurements.

Description	Measurement
Tooth Land	1mm
Tooth Depth	0.94mm
Width at Root	3.16mm
Pressure Angle	45^o

From these measurements the spline's theoretical dimensions were found to be module 1, circular pitch of 3.142mm and 26 teeth. This information was found from table 1 Theoretical dimensions of splines (Standards Australia 1999) student copy of standards. It was interesting to see the wear on the tooth profile especially the rounding of the tooth profile on the drive side. But with these measurements the spline was drawn in AusCAD 2000 for machining in the CNC machine. To manufacture the spline, the shaft could be rotated in the CNC lathe to a position where a pass of the cutter will produce a 90^o angle between the teeth. A number of trial splines were manufactured to check the final size and finish before the splines on the centre shaft were machined.

Figure 4.4: View of tooth profile under profile projector.

4.2.4 Manufacturing

The manufacturing of the shaft started with flanges being welded to a 40mm bar of mild steel. The flanges were pre-drilled with the given bolt pattern before being welded to the shaft. The shaft was then mounted in the lathe to machine the final geometry. All the welding and misalignments are finished and trued in this process. The bearing seats were turned down to a light push fit class H7/k6. The splines on the ends were machined using the CNC lathe with the process outlined in the previous section.

4.2.5 Summary of Centre Shaft Design

In summary the centre shaft design now consists of the axle mounted on bearings outside the frame rails with splines for the CV joints at the ends. A solid model was created (Figure 4.5) in Pro Engineer 2001 to see the design in 3D. A detailed drawing of the centre shaft is also in Appendix F.

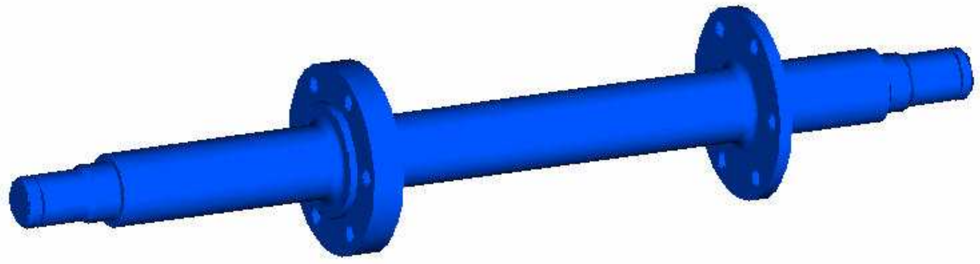

Figure 4.5: Solid model of Centre Shaft developed in Pro Engineer.

4.3 Misalignment Problems

Misalignment of the sprocket flange became apparent after the manufacturing of the prototype centre shaft. When the bearing housings and rear sprocket were attached to the centre shaft and clamped to the frame in the correct position the sprockets were misaligned. The cause of this was first thought to be a measurement or calculation error. The drawing sketches, measurements and detailed drawings were checked and the physical measurement of the engines drive sprocket checked again. No error was found in any of these areas. Using a straight edge the error was measured at 7.2mm, and wasn't half the sprocket thickness or anything similar but fortunately it could be fixed by adding a spacer, see Figure 4.6.

The only valid solution to how this misalignment had occurred was due to the engine positioning not being square in the frame. Thought was not given to the possibility of the engine not being aligned in the frame. This should have been realised before the measurements were taken off the front sprocket. When taking the measurement off the front sprocket any misalignment is not factored in. Measurements should have been made using a straight edge back to the correct position of the axle so any angular misalignment was taken into consideration.

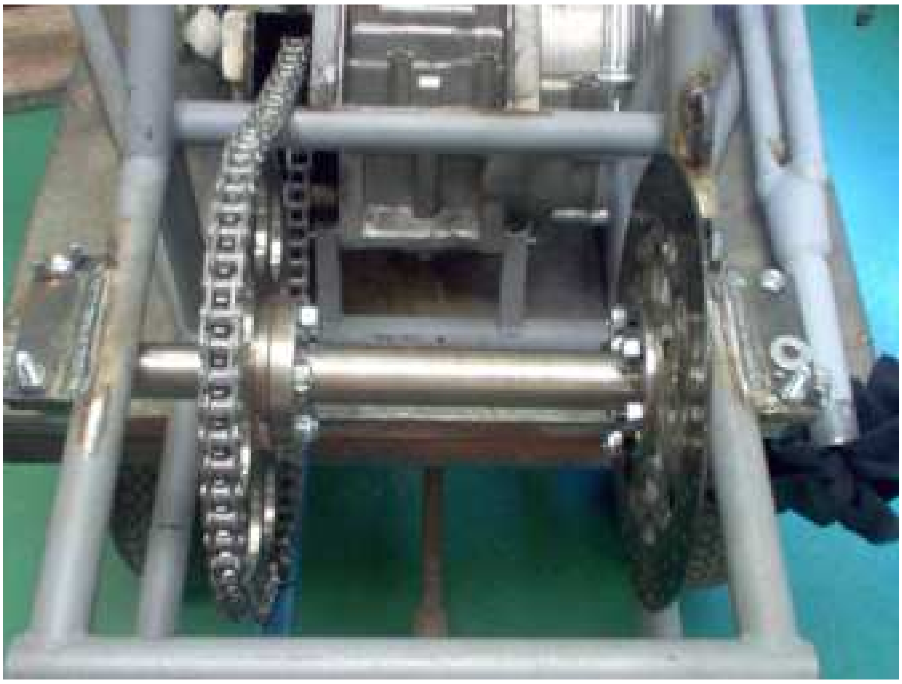

Figure 4.6: Rear assembly with spacer behind drive sprocket.

4.4 Centre Shaft Analysis

The centre shaft analysis involves calculations of the loading, stresses, fatigue and deflections. The material used in manufacturing the existing centre shaft was AISI 1020 mild steel. The yield and ultimate strength values are 331MPa and 448MPa respectively in the as rolled state from appendix C-4a (Juvinall & Marshek 2000). This was chosen for the prototype shaft but since this shaft has become the shaft being used in the SAE car a complete analysis of this shaft was conducted. To start with, the loading of the shaft is determined before the following calculations can be undertaken.

4.4.1 Shaft Loading

Loads on the shaft are only due to the chain drive and disk brake as there are no loads in supporting the cars weight due to the independent suspension. The weight of the drivetrain components are considered negligible and the most significant load on the centre shaft is the tangential force on the rear sprocket. This force produces a torque in the shaft which was found is the previous chapter (Equation 3.5) to be 664.5Nm. This torque is shared between the two tyres under straight-line acceleration but at the extreme case when the drive is on one wheel all the torque is transmitted through one side of the shaft. Due to this torque, the maximum tangential force the chain can transmit to the sprocket ($D_p = 303.33mm$) is found in the following:-

$$\begin{aligned} F &= \frac{T}{r_p} \\ &= \frac{664.5}{0.151665} \\ &= 4381.37N \end{aligned}$$

Also a bending moment is created in the shaft by this force with the chain pulling the shaft forward acting is the direction of the chain tension. The bearings hold the shaft in place so they carry the reaction forces to this loading. The initial chain tension is neglected as well as the centrifugal effects of the chain. The angular displacement at the bearings is assumed adequate enough, that no moment is produced. Finding the bearing reactions on the left and right are illustrated in Figure 4.7.

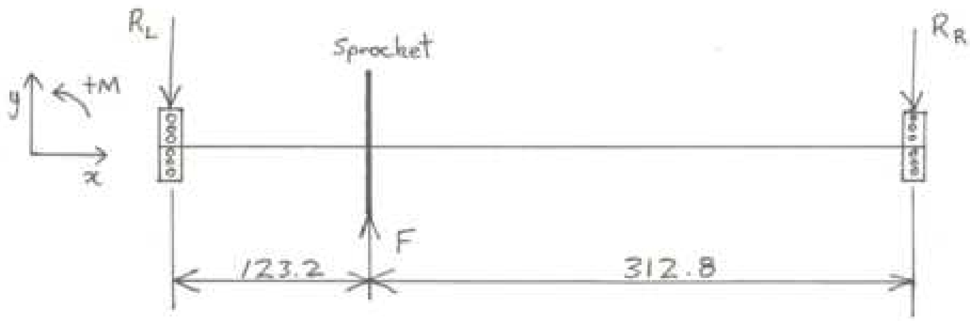

Figure 4.7: Free Body Diagram of centre shaft.

Sum of moments about the left bearing finds the right-hand bearing reaction force.

$$\begin{aligned}\sum M_L &= 0\\ 0 &= (4381.37 \times 0.1232) - (R_R \times 0.436)\\ R_R &= 1238.04N\end{aligned}$$

Sum of forces in the y-direction finds the left bearing reaction force.

$$\begin{aligned}\sum F_y &= 0\\ F &= R_L + R_R\\ R_L &= 4381.37 - 1238.04\\ &= 3143.33N\end{aligned}$$

The shear force, bending moment and torque diagrams are produced from the forces on the shaft in Figure 4.8.

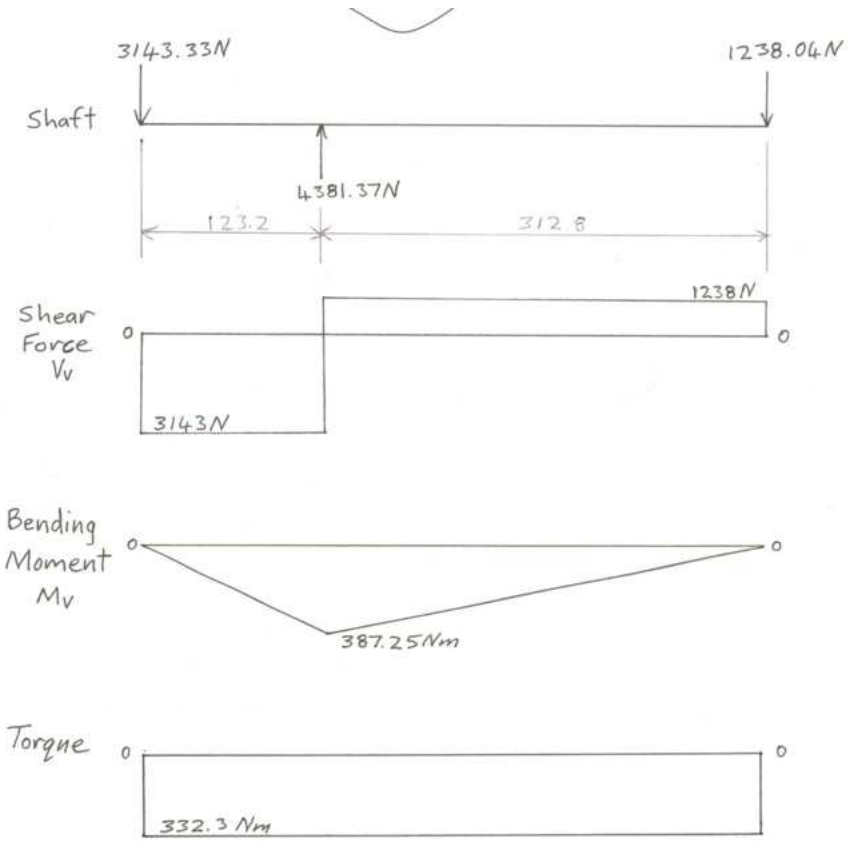

Figure 4.8: Sketches of the Shear Force, Bending Moment and Torque diagrams for the centre shaft.

4.4.2 Stress Analysis

All the suspected points shown in Figure 4.9 at which the stresses might be the greatest are analysed to check if the stesses are larger than the yield strength of the material.

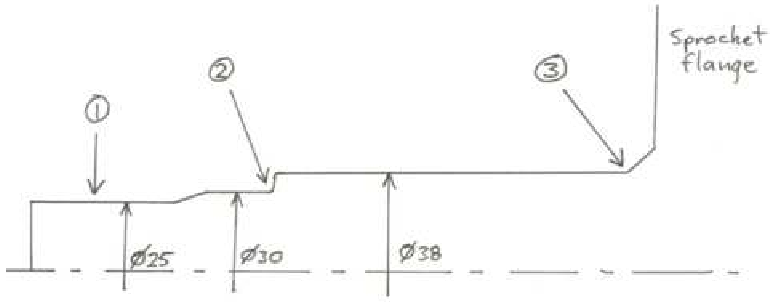

Figure 4.9: Sketch of the shaft end with critical location numbered.

The first suspected point of failure (point 1) is shearing of the involute spline. To make sure the spline doesn't fail due to shearing, it was found as a rule of thumb if the spline length of engagement is between 0.75 or 1.25 times the pitch diameter, then the shear strength of the spline will exceed that of the shaft from which it is made from. The length of the splines used for the centre shaft will make the splines exceed the strength of the parent material.

Checking for shearing of the shaft due only to the torque is undertaken next. Two different cases are assessed, first the normal loading under straight line acceleration and secondly the extreme case of all the torque being transferred to one wheel. The splines root diameter is approximately 25mm. Using equation (3.11)(Ugural 2004) for the shear stress due to torsion.

$$\tau = \frac{Tr}{J} \tag{4.1}$$

where: τ = torsional shear stress (MPa)

T = torque (Nm)

r = radius (m)

J = polar moment of inertia (m^4)

The polar moment of inertia is found from Appendix B-1 (Juvinall & Marshek 2000) for a circle where d = diameter.

$$J = \frac{\pi d^4}{32} \tag{4.2}$$

So under normal loading of 332.3Nm the shaft shear stress is:

$$\begin{aligned} \tau &= \frac{332.3 \times 0.0125}{\frac{\pi \times 0.025^4}{32}} \\ &= 109.3MPa \end{aligned}$$

Using the Maximum Distortion Energy Criterion (Ductile Materials) to predict the equivalent tensile yield strength from figure (6.10) (Juvinall & Marshek 2000).

$$\sigma_Y = \frac{\tau_{max}}{0.58} \tag{4.3}$$

where: σ_Y = yield strength (MPa)

τ_{max} = maximum shear stress (MPa)

So the equivalent tensile strength under normal loading.

$$\begin{aligned} \sigma &= \frac{109.3}{0.58} \\ &= 188MPa \end{aligned}$$

Now under the extreme loading of 664.5Nm the shaft shear stress is:

$$\begin{aligned} \tau &= \frac{664.5 \times 0.0125}{\frac{\pi \times 0.025^4}{32}} \\ &= 218.58MPa \end{aligned}$$

And the equivalent tensile strength under extreme loading.

$$\begin{aligned} \sigma &= \frac{218.58}{0.58} \\ &= 376.9MPa \end{aligned}$$

The shaft can also fail due to combined loading of shearing and torsion or bending and torsion. The following Figure 4.10 is produced from figure (3.29) of (Ugural 2004) along with the proceeding formulas.

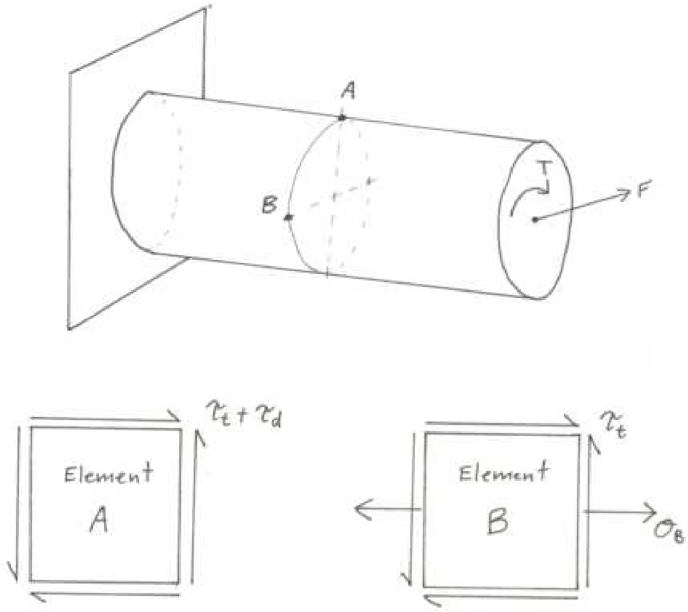

Figure 4.10: Sketch of shaft showing points of maximum shear element A and maximum bending element B.

where: τ_t = torsional shear stress

τ_d = direct shear stress

σ_b = normal stress

The Equation 4.1 for torsion is known but the new equations for shear and normal stresses due to bending are equations (3.20) and (3.17) respectively in (Ugural 2004).

$$\tau = \frac{VQ}{Ib} \tag{4.4}$$

$$\sigma = \frac{Mc}{I} \tag{4.5}$$

where: V = shearing force (N)

Q = first moment of area above neutral axis (m^3)

I = moment of inertia (m^4)

b = width of section in question (m)

c = radius for shafts (m)

Looking at element A first, it can be seen from the shear force diagram that the maximum shearing force is 3143.33N between the left-hand bearing and the sprocket. The torque is taken as the extreme case of 664.5Nm and the 30mm diameter section is the critical location (point 2). The following calculations find the induced combined stress. First the direct shear stress is found by equation 4.5.

$$\tau_d = \frac{VQ}{Ib} \tag{4.6}$$

Where:

$$\begin{aligned} Q &= A\bar{y} \\ &= \left(\frac{1}{2}\pi r^2\right)\left(\frac{4r}{3\pi}\right) \\ &= \frac{2}{3}r^3 \\ &= \frac{2}{3} \times 0.015^3 \\ &= 2.25 \times 10^{-6} m^3 \end{aligned}$$

$$\begin{aligned} b &= 2r \\ &= 0.03m \end{aligned}$$

$$\begin{aligned} I &= \frac{\pi D^4}{64} \\ &= \frac{\pi \times (0.03)^4}{64} \\ &= 3.9 \times 10^{-8} m^4 \end{aligned}$$

So the direct shear stress is then:

$$\begin{aligned} \tau_d &= \frac{3143.33 \times 2.25 \times 10^{-6}}{3.9 \times 10^{-8} \times 0.03} \\ &= 6.04MPa \end{aligned}$$

Torsional shear stress using Equation 4.1.

$$\begin{aligned} \tau_t &= \frac{Tr}{J} \\ &= \frac{664.5 \times 0.015}{\frac{\pi \times (0.03)^4}{32}} \\ &= 126.17MPa \end{aligned}$$

Now the total shear stress is adding the direct and torsional shear stress together.

$$\begin{aligned}\tau_{max} &= \tau_d + \tau_t \\ &= 6.04 + 126.17 \\ &= 132.21 MPa\end{aligned}$$

The equivalent maximum tensile strength due to shearing and torsion at the 30mm bearing surface is found using Equation 4.3.

$$\begin{aligned}\sigma &= \frac{132.21}{0.58} \\ &= 228 MPa\end{aligned}$$

Looking at element B under combined bending and torsional loads. The bending diagram in Figure 4.8 shows the maximum bending moment occurs at the base of the sprocket flange (point 3) Figure 4.9. So the bending stress due to the 387.25Nm moment is found using Equation 4.5.

$$\begin{aligned}\sigma_b &= \frac{Mc}{I} \\ &= \frac{387.25 \times\ 0.019}{\frac{\pi\times(0.038)^4}{64}} \\ &= 72.13 MPa\end{aligned}$$

Now the torsional shear stress in the 38mm shaft under maximum torque using Equation 4.1.

$$\begin{aligned}\tau_t &= \frac{Tr}{J} \\ &= \frac{664.5 \times 0.019}{\frac{\pi\times(0.038)^4}{32}} \\ &= 61.9 MPa\end{aligned}$$

The total combined stresses are found using the formulas for Mohr's Circle for biaxial stress. The principle stresses are found from equation (4.16) (Juvinall & Marshek 2000).

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\tau_{xy}^2 + \left(\frac{\sigma_x - \sigma_y}{2}\right)^2} \tag{4.7}$$

Now substituting the normal stress in the x-direction and the torsional stress into Equation 4.7.

$$\sigma_1, \sigma_2 = \frac{72.13}{2} \pm \sqrt{61.9^2 + \left(\frac{72.13}{2}\right)^2}$$

$$\sigma_1 = 107.71MPa$$
$$\sigma_2 = -35.57MPa$$

These principle stresses equate to a yield strength equivalent based on the Maximum Distortion Energy Criterion by equation (7.6)(Stress Analysis 2003).

$$\begin{aligned}\sigma_y^2 &= \sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \\ &= 107.71^2 - 107.71 \times (-35.57) + (-35.57)^2 \\ \sigma_y &= 129.2MPa\end{aligned}$$

Tabulating all the results together (Table 4.2) to see the weakest points by comparing the stresses along with the corresponding factor of safety (FOS)with respect to the yield strength of AISI 1020.

Table 4.2: Factor of Safety Summary for AISI 1020.

Point	Load Condition	Induced Stress (MPa)	FOS
1	Normal	188	1.76
1	Extreme	376.9	0.88
2	Extreme	228	1.45
3	Extreme	129.2	2.56

Summing up the results it is apparent that the root diameter of the spline is the weakest point if the loads can reach the extreme case of transmitting all the torque through one side of the shaft. Under normal operating of straight line acceleration the maximum stresses have a FOS of 1.76 before any yielding occurs. The actual stress situation will lie somewhere in between these two conditions so the safety factor is very small. Of the other shaft sections point 2 is the next critical point at the bearing surface.

4.4.3 Fatigue Life

The fatigue life of the current shaft is assessed at three locations illustrated in Figure 4.11. All the fatigue analysis was based on Juvinall and Marshek, Fundamentals of Machine Component Design 2000.

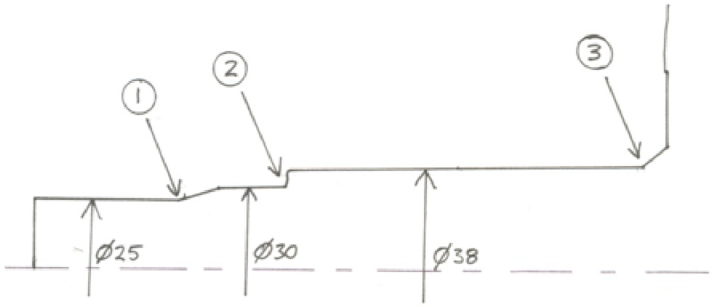

Figure 4.11: Sketch of shaft fatigue locations.

Looking at the first location the fatigue loading was to simulate accelerating and braking in a straight line with only torsional stress. Finding the braking torque assumed that the rear would make 30 percent of the total braking so the following calculations were conducted with reference to Figure 3.2.

The normal force on the rear tyres under braking using Equation 3.2.

$$F_N = mg = 350\left(\frac{30}{100}\right)9.81 = 1030N$$

Frictional force provided by rear tyres if $\mu = 1$.

$$F_R = F_N\mu = 1030 \times 1 = 1030N$$

Total torque produced in shaft using Equation 3.5.

$$T = F_R r = 1030 \times 0.2765 = 284.8Nm$$

Torques of 332.3Nm in acceleration and 284.8Nm in braking will be experienced by the shaft. The loading will be fully reversed to simulate accelerating then braking repetition. The geometry of the shaft is sketched in Figure 4.12.

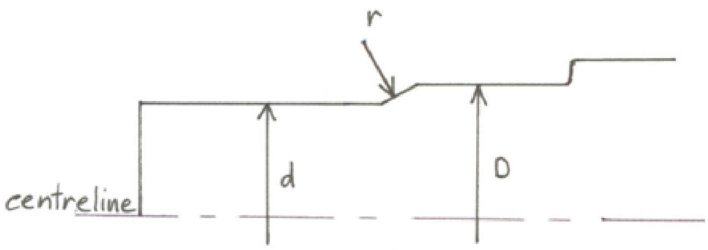

Figure 4.12: Sketch of shaft spline to bearing surface.

Using figure 4.35(c) (Juvinall & Marshek 2000) the normal torsional shear stresses are found. First under acceleration to find the maximum shear stress.

$$\begin{aligned}\tau_{nor} &= \frac{16T}{\pi d^3}\\ &= \frac{16 \times 332.3}{\pi \times 0.025^3}\\ &= 108.3MPa\end{aligned}$$

From the same figure for torsion the Kt value is found.

$$\begin{aligned}\frac{D}{d} &= \frac{30}{25} = 1.2\\ \frac{r}{d} &= \frac{2.5}{25} = 0.1\\ K_t &= 1.32\end{aligned}$$

With figure 8.24 (Juvinall & Marshek 2000) and Su=65ksi the notch sensitivity factor, q=0.81. Now the fatigue stress concentration factor, Kf is found from equation (8.2).

$$\begin{aligned}K_f &= 1 + (K_t - 1)q\\ &= 1 + (1.32 - 1)0.81\\ &= 1.259\end{aligned}$$

Equation 4.21 can be used to include the factor Kf.

$$\begin{aligned}\tau_{max} &= \tau_{nor}K_f\\ &= 108.3 \times 1.259\\ &= 136.37MPa\end{aligned}$$

Now under braking to find the minimum shear stress can be found using the same process for the maximum.

$$\tau_{nor} = \frac{16T}{\pi d^3}$$

$$
\begin{aligned}
&= \frac{16 \times -284.8}{\pi \times 0.025^3} \\
&= -92.83MPa
\end{aligned}
$$

The fatigue stress concentration factor, Kf is the same at 1.259 so the minimum shear stress as follows.

$$
\begin{aligned}
\tau_{min} &= \tau_{nor} K_f \\
&= -92.83 \times 1.259 \\
&= -116.9MPa
\end{aligned}
$$

The mean and alternating shear stress loading from figure 8.15 (Juvinall & Marshek 2000).

$$
\begin{aligned}
\tau_m &= \frac{(\tau_{max} + \tau_{min})}{2} \\
&= \frac{(136.37 + (-116.9))}{2} \\
&= 9.74MPa
\end{aligned}
$$

$$
\begin{aligned}
\tau_a &= \frac{(\tau_{max} - \tau_{min})}{2} \\
&= \frac{(136.37 - (-116.9))}{2} \\
&= 126.64MPa
\end{aligned}
$$

Finding the 10^6-cycle strength, S_n from table 8.1. where:-

$S'_n = 0.5S_u$

C_L = Load factor

C_G = Gradient factor

C_S = Surface factor Figure 8.13 (Juvinall & Marshek 2000)

$$
\begin{aligned}
S_n &= S'_n C_L C_G C_S \\
&= 0.5 \times 448 \times 0.58 \times 0.9 \times 0.8 \\
&= 93.54MPa
\end{aligned}
$$

Without drawing the Goodman Fatigue diagram it can be seen that the plotted location of the mean and alternating shear stresses will be past the infinite life line. Therefore a S-N curve (Figure 4.13) is plotted instead to predict the expected fatigue life. The following calculation for the 10^3 and 10^6 cycles are from figure (8.11) (Juvinall & Marshek 2000).

$$\begin{aligned} S_{10^3} &= 0.8S_u \\ &= 0.8 \times 448 \\ &= 358.4MPa \end{aligned}$$

$$\begin{aligned} S_n &= 0.29S_u \\ &= 0.29 \times 448 \\ &= 129.92MPa \end{aligned}$$

With the material fatigue line plotted the the 10^4 and 10^5 cycle stress limits can be found.

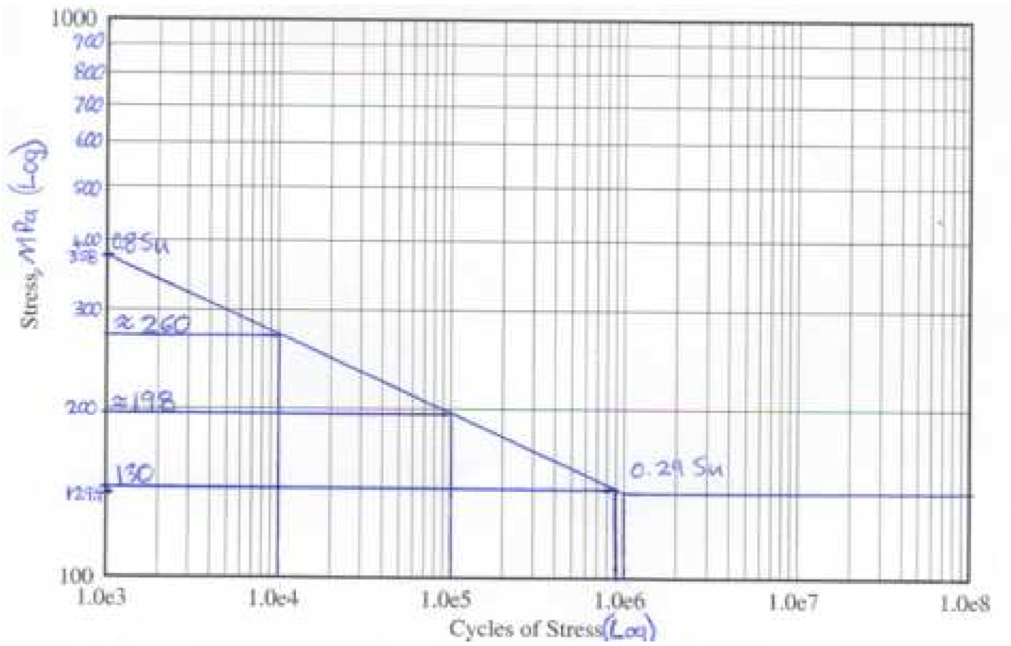

Figure 4.13: S-N Curve for fully reversed torsional loading of AISI 1020.

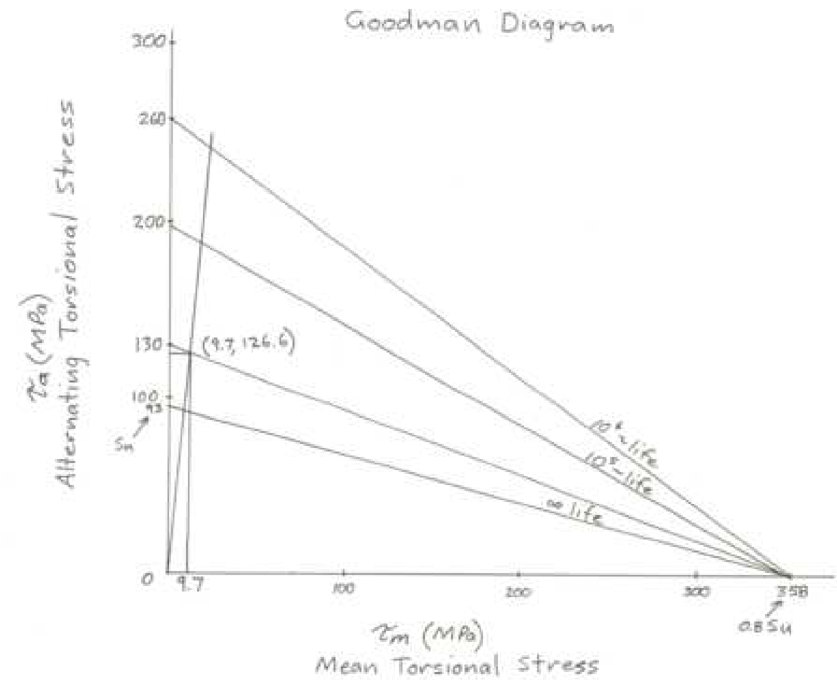

Figure 4.14: Goodman Diagram for torsional loading of AISI 1020.

Plotting the Goodman diagram (Figure 4.14) with all the cycle limits included and the alternating and mean torsional stresses of the shaft the life expectancy can be found both graphically and mathematically. Graphically it is achieved by projecting a line though the plotted point to the alternating torsional stress axis to find the fatigue strength value which is 130MPa in this case. Then this value is plotted on the S-N curve to find the expected life as shown by the bottom line of 130MPa in Figure 4.13. Mathematically it is found by the process outlined on page 3.2 (University of Southern Queensland 2001*a*) by finding the equation of the S-N line and putting the fatigue strength of the material in the equation. Both result in the same answer with the mathematical approach revealing the fatigue life at this section will be 9×10^5 cycles.

The mathematical approach.

$$N = 10^{\frac{-C}{b}} (S_f)^{\frac{1}{b}} \tag{4.8}$$

$$b = -\frac{1}{3} \log \frac{S_3}{S_n} \tag{4.9}$$

$$C = \log \frac{(S_3)^3}{S_n} \tag{4.10}$$

Where b and C are constants with:-

N = number of cycles

S_f = fatigue strength

S_3 = 10^3 cycle strength

Using these equations the expected life is found.

$$\begin{aligned} b &= -\frac{1}{3}\log\frac{358.4}{129.92} \\ &= -0.1469 \end{aligned}$$

$$\begin{aligned} C &= \log\frac{(358.4)^3}{129.92} \\ &= 2.9951 \end{aligned}$$

$$\begin{aligned} N &= 10^{\frac{-2.9951}{-0.1469}}(130)^{\frac{1}{-0.1469}} \\ &= 996504.8 \end{aligned}$$

The fatigue life at the first point is very close to infinite life.

Looking at point 2 for the current centre shaft at the 30mm to 38mm diameters. The same process was followed as the first location (point 1). Now the shaft will experience the same torques of 332.3Nm and -284.8Nm for simulating fully reversed loading. The alternating bending stress is neglected due to its small magnitude at the bearing location. The geometry of the shaft is sketched in the following Figure 4.15.

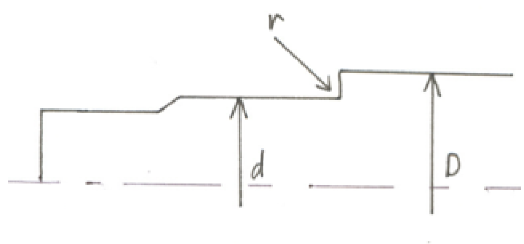

Figure 4.15: Sketch of shaft bearing surface to main shaft diameter.

The normal torsional shear stresses are found. First under acceleration to find the maximum shear stress.

$$\tau_{nor} = \frac{16T}{\pi d^3}$$

$$\begin{aligned} &= \frac{16 \times 332.3}{\pi \times 0.03^3} \\ &= 62.68MPa \end{aligned}$$

Now under braking to find the minimum shear stress.

$$\begin{aligned} \tau_{nor} &= \frac{16T}{\pi d^3} \\ &= \frac{16 \times -284.8}{\pi \times 0.03^3} \\ &= -53.72MPa \end{aligned}$$

It is apparent at this point by comparing the shear stresses of points 1 and 2 that this second point is stronger than point 1 so no further work need to be conducted. Point 3 can be analysed now but under a simpler loading due to the lack of loading data. The analysis will be of the shaft under constant maximum acceleration. This is an extreme loading case which is not experienced by the shaft all the time but rather in short bursts but the analysis will prove whether the fatigue strength of the shaft is a problem at this third point. The loading will consist of a mean torsional load of 332.3Nm and an alternating bending load of 214.37Nm with the formulas found in figure (4.35) (Juvinall & Marshek 2000).

First the mean torsional stress using figure 4.35(c).

$$\begin{aligned} \tau &= \frac{16T}{\pi d^3} \\ &= \frac{16 \times 332.3}{\pi \times 0.038^3} \\ &= 30.84MPa \end{aligned}$$

From the same figure for torsion the Kt value is found. The larger diameter is taken as the weld outside diameter.

$$\begin{aligned} \frac{D}{d} &= \frac{48}{38} = 1.263 \\ \frac{r}{d} &= \frac{5}{38} = 0.1316 \\ K_t &= 1.28 \end{aligned}$$

With figure 8.24 and Su=65ksi the notch sensitivity factor, q=0.85. Now the fatigue stress concentration factor, Kf.

$$K_f = 1 + (K_t - 1)q$$

$$\begin{aligned} &= 1 + (1.28 - 1)0.85 \\ &= 1.238 \end{aligned}$$

The modified mean shear stress is now.

$$\begin{aligned} \tau_m &= \tau K_f \\ &= 30.84 \times 1.238 \\ &= 38.18MPa \end{aligned}$$

Now the alternating bending stress is found from figure (4.35(a)).

$$\begin{aligned} \sigma &= \frac{32M}{\pi d^3} \\ &= \frac{32 \times 214.37}{\pi \times 0.038^3} \\ &= 39.79MPa \end{aligned}$$

From the same figure the Kt value is found to be 1.5. Now the notch sensitivity factor, q=0.82 so the fatigue stress concentration factor, Kf can be found.

$$\begin{aligned} K_f &= 1 + (K_t - 1)q \\ &= 1 + (1.5 - 1)0.82 \\ &= 1.41 \end{aligned}$$

The modified alternating bending stress is now.

$$\begin{aligned} \sigma_a &= \sigma \times K_f \\ &= 39.78 \times 1.41 \\ &= 56.09MPa \end{aligned}$$

Using figure (8.16) (Juvinall & Marshek 2000) for General Biaxial Loads for the equivalent mean bending stress and equivalent alternating bending stress are just the mean torsional and alternating bending stresses respectively. So now the situation can be treated as a fatigue bending situation.

The 10^6-cycle strength, S_n for bending.

$$\begin{aligned} S_n &= S'_n C_L C_G C_S \\ &= 0.5 \times 448 \times 1 \times 0.9 \times 0.8 \\ &= 161.28MPa \end{aligned}$$

Drawing the Goodman Fatigue diagram (Figure 4.16) and plotting the stress condition "operating point" reveals that point 3 under torsion and bending will not fail from fatigue. The safety factor of this point is 2.57.

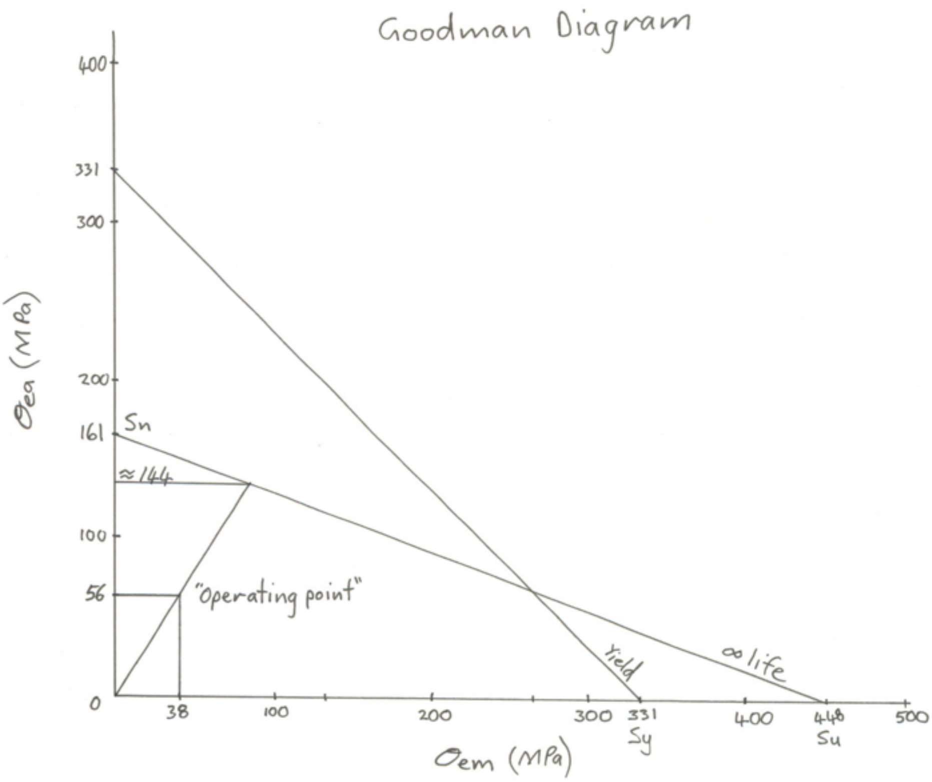

Figure 4.16: Goodman Diagram for bending loading of AISI 1020.

The conclusion of this fatigue analysis on the current centre shaft reveals the spline to bearing surface (point 1) is the weakest point which will fail first. But without the complete information of this randomly varying load conducting a concise fatigue analysis is difficult. A detail analysis would be based on the Palmgren or Miner rule of prediction the loading. But my judgement is the loading used has provided a good prediction of the fatigue life of the centre shaft made out of AISI 1020 and the shaft will not fail due to fatigue for the limited use the car will experience.

4.5 Improved Design

If time permitted a new centre shaft would be manufactured out of a new material with some slight design changes. This improved design solid model can be seen in Appendix E.9 and the detailed drawing in Appendix F.6. The centre shaft loading is the same as before but the shear force and bending moment diagrams are different.

4.5.1 Shaft Loading

Loads on the shaft are the same as before with 4381.37N force at the sprocket drive but the location of the bearings has changed so the bearing reactions on the left and right will change.

Figure 4.17: Free Body Diagram of modified centre shaft.

Sum of moments about the left bearing finds the right-hand bearing reaction force.

$$\begin{aligned}\sum M_L &= 0\\ 0 &= (4381.37 \times 0.058) - (R_R \times 0.3708)\\ R_R &= 685.33N\end{aligned}$$

Sum of forces in the y-direction finds the left bearing reaction force.

$$\begin{aligned}\sum F_y &= 0\\ F &= R_L + R_R\\ R_L &= 4381.37 - 685.33\\ &= 3696.04N\end{aligned}$$

The shear force, bending moment and torque diagrams (Figure 4.18) are then produced.

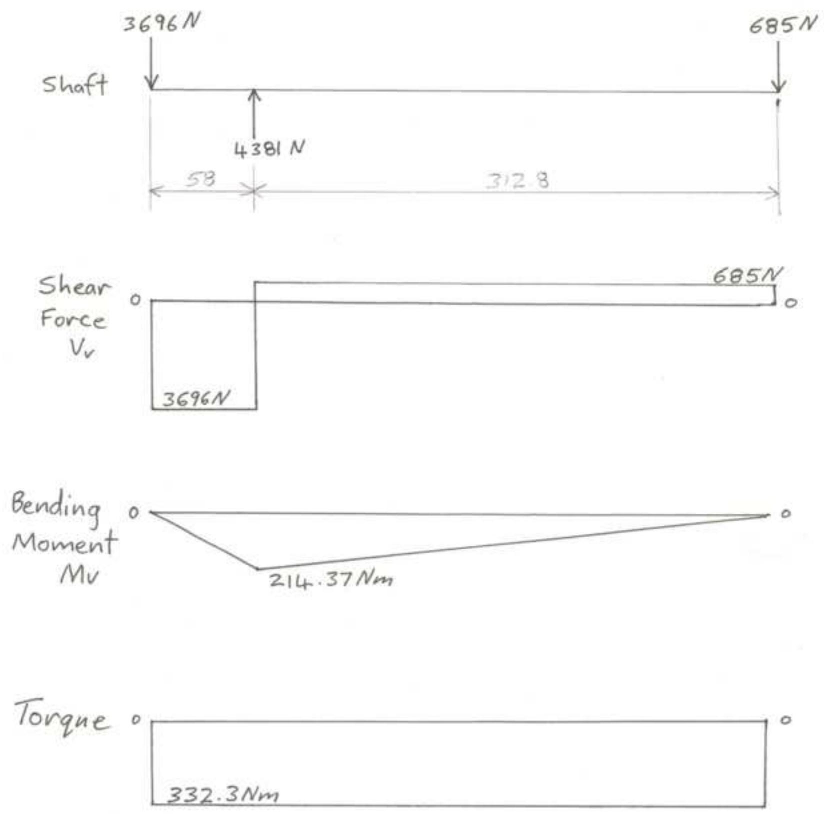

Figure 4.18: Shear Force, Bending Moment and Torque diagrams for the modified centre shaft.

4.5.2 Material Selection

The choice of using a different material was based on the fact that if all the torque available was transferred to one wheel the diameter of the spline would yield and also the fatigue life would increase. The choice of materials I looked at were the common ultrahigh-strength steels of medium-carbon low-alloy families which include AISI 4130, 4140 and 4340. The choice of the popular 4340 was based on the higher strength, ductility and toughness properties for little additional expenditure. All the required information about AISI 4340 can be found in the USQ libary's database online (American Society of Metals Handbook 2004). The yield and ultimate strength values are 861MPa and 1279MPa respectively in the as rolled state from appendix C-4a

(Juvinall & Marshek 2000).

4.5.3 Stress Analysis

The improved design has the same geometry as the previous Figure 4.9 with all the suspected points at which the stresses might be the greatest being checked. The first point of shearing of the shaft due only to torque will be the same as before with the tensile stresses of 188MPa and 376.9MPa under normal and extreme loading respectively. Again the shaft can also fail due to combined loading of shearing and torsion or bending and torsion. The following calculations refer to Figure 4.10 for the element A and B locations on a shaft under loads. Here are the elements shown again.

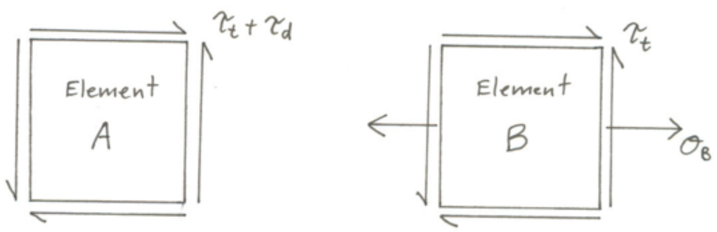

Figure 4.19: Sketch of maximum shear element A and maximum bending element B.

The shear force diagram shows a maximum shearing force of 3696.04N between the left-hand bearing and the sprocket. The torque is taken as the extreme case of 664.5Nm and the 30mm diameter section is the critical location (point 2). The following calculations find the induced combined stress represented on element A. First the direct shear stress is found by Equation 4.5.

$$\begin{aligned}
\tau_d &= \frac{VQ}{Ib} \\
\tau_d &= \frac{3696.04 \times 2.25 \times 10^{-6}}{3.9 \times 10^{-8} \times 0.03} \\
&= 7.1MPa
\end{aligned}$$

Torsional shear stress using Equation 4.1 is the same as before at 126.17MPa. The total shear stress is adding the direct and torsional shear stress together.

$$\tau_{max} = \tau_d + \tau_t$$

$$
\begin{aligned}
&= 7.1 + 126.17 \\
&= 133.27MPa
\end{aligned}
$$

The equivalent maximum tensile strength due to shearing and torsion at the 30mm bearing surface is found using Equation 4.3.

$$
\begin{aligned}
\sigma &= \frac{133.27}{0.58} \\
&= 229.8MPa
\end{aligned}
$$

Looking at element B under combined bending and torsional loads. The bending diagram in Figure 4.18 shows the maximum bending moment occurs at the base of the sprocket flange, point 3. So the bending stress due to the 214.37Nm moment is found using Equation 4.5.

$$
\begin{aligned}
\sigma_b &= \frac{Mc}{I} \\
&= \frac{214.37 \times \ 0.019}{\frac{\pi \times (0.038)^4}{64}} \\
&= 39.93MPa
\end{aligned}
$$

The torsional shear stress in the 38mm shaft under maximum torque using Equation 4.1 is the same as before at 61.9MPa. So the total combined stresses (principles) from Mohr's Circle for biaxial stresses are found from using Equation 4.7 again.

$$
\begin{aligned}
\sigma_1, \sigma_2 &= \frac{39.93}{2} \pm \sqrt{61.9^2 + \left(\frac{39.93}{2}\right)^2} \\
\sigma_1 &= 85MPa \\
\sigma_2 &= -45MPa
\end{aligned}
$$

These principle stresses equate to a yield strength equivalent based on the Maximum Distortion Energy Criterion.

$$
\begin{aligned}
\sigma_y^2 &= \sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \\
&= 85^2 - 85 \times (-45) + (-45)^2 \\
\sigma_y &= 114.4MPa
\end{aligned}
$$

Tabulating all the results together (Table 4.3) to see the weakest points by comparing the stresses along with the corresponding factor of safety (FOS) with respect to the yield strength of AISI 4340.

Table 4.3: Factor of Safety Summary for AISI 4340.

Point	Load Condition	Induced Stress (MPa)	FOS
1	Normal	188	4.58
1	Extreme	376.9	2.28
2	Extreme	229.8	3.75
3	Extreme	114.4	7.53

Summing up the results the root diameter of the spline is the weakest point again under extreme loading but the factor of safety is 2.28. According to (Juvinall & Marshek 2000) (p.263) this factor is associated with loads and stresses that can be determined which is much more acceptable than the previous AISI 1020 material.

4.5.4 Fatigue Life

The modified centre shaft will have a much greater factor of safety over the current shaft due to the higher yield and ultimate tensile strengths. The following fatigue analysis is on the first point which was seen to be the critical point for the fatigue life of the current shaft. The loading and all the geometry is the same the current shaft.

So the Kt value is 1.32 and from figure (8.24) (Juvinall & Marshek 2000) with Su=185.5ksi the notch sensitivity factor, q=0.95. Now the fatigue stress concentration factor, Kf is found using equation (8.2).

$$\begin{aligned} K_f &= 1 + (K_t - 1)q \\ &= 1 + (1.32 - 1)0.95 \\ &= 1.304 \end{aligned}$$

Under acceleration the maximum torsional stress is found.

$$\begin{aligned} \tau_{max} &= \frac{16T}{\pi d^3} K_f \\ &= \frac{16 \times 332.3}{\pi \times 0.025^3} \times 1.304 \\ &= 141.22 MPa \end{aligned}$$

Under braking the minimum torsional stress is found.

$$\begin{aligned}
\tau_{min} &= \frac{16T}{\pi d^3}K_f \\
&= \frac{16 \times -284.8}{\pi \times 0.025^3} \times 1.304 \\
&= -121.05MPa
\end{aligned}$$

Finding the mean and alternating shear stress loading.

$$\begin{aligned}
\tau_m &= \frac{(\tau_{max} + \tau_{min})}{2} \\
&= \frac{(141.22 + (-121.05))}{2} \\
&= 10.09MPa
\end{aligned}$$

$$\begin{aligned}
\tau_a &= \frac{(\tau_{max} - \tau_{min})}{2} \\
&= \frac{(141.22 - (-121.05))}{2} \\
&= 131.14MPa
\end{aligned}$$

Finding the 10^6-cycle strength, S_n.

$$\begin{aligned}
S_n &= S_n' C_L C_G C_S \\
&= 0.5 \times 1279 \times 0.58 \times 0.9 \times 0.64 \\
&= 213.64MPa
\end{aligned}$$

The Goodman diagram Figure 4.20 can now be plotted. This reveals a FOS of 2.28, which is an improvement on the AISI 1020 material FOS. The AISI 4340 material is recommended for infinite life for the centre shaft.

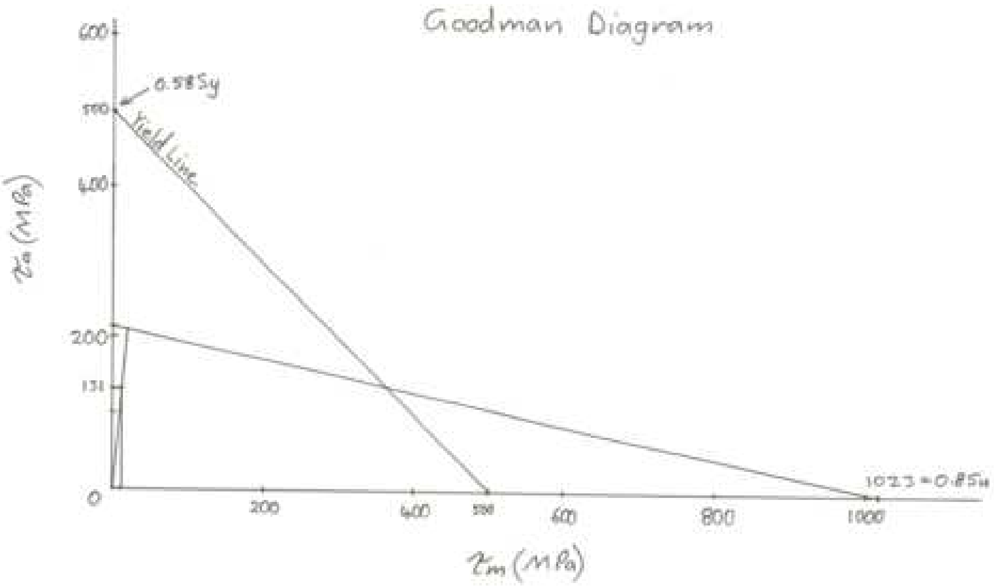

Figure 4.20: Goodman diagram for fully reversed torsional loading of AISI 4340.

4.6 Deflections

Deflections of shafts are checked after the stress and fatigue analysis are conducted. The problem that will occur if the deflection of the centre shaft is to great is the angular deflection at the bearings will causes the bearings to bind. This will be further discussed in the bearing mounts section in the next chapter. The choice of material doesn't affect the deflection because all metals have same elastic modulus. So only the positioning of the bearings is left as the critical factor in reducing the deflections of the centre shaft.

To find the deflections of both the current and improved shaft designs a MATLAB program in Appendix D was written to find the effects of different bearing positions. The following Figure 4.21 and equation are from appendix D (Beer & Johnston 1992).

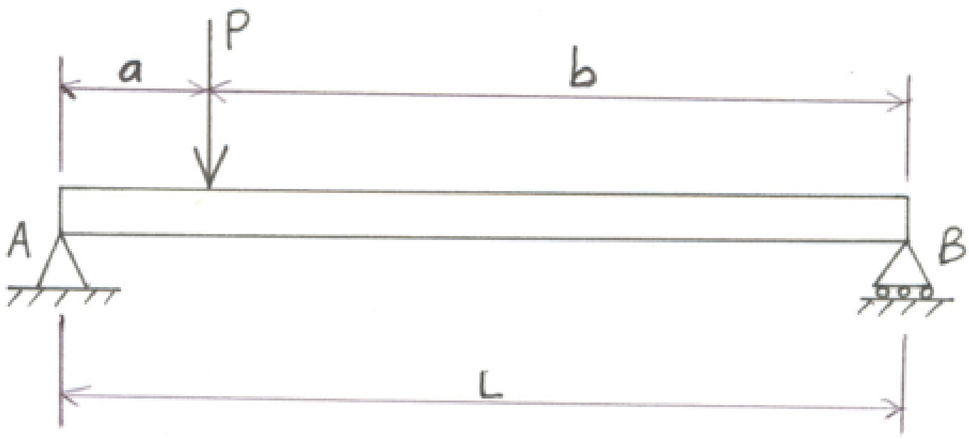

Figure 4.21: Beam deflection FBD.

$$\theta_A = \frac{Pb(L^2 - b^2)}{6LEI} \tag{4.11}$$

Where:

θ_A = Slope at end A (radians)

P = Load (N)

b = Distance from end B (mm)

L = Total Length (mm)

E = Modulus of Elasticity (GPa)

I = Moment of Inertia (mm)

Figure 4.22 shows the results of different bearing positions on the centre shaft. The red line (bottom) shows the position of the improved design and the black line (top) shows the current position of the bearing. To find out what deflections the bearings can handle bearing catalogues were consulted and revealed that deep groove ball bearings can cater for up to 10 minutes (p.180) (NTN 1990) of angular misalignment.

The results of misalignment are noise level increases, smooth running is impaired and it reduces the service life of the bearing. So for this reason the bearing on the sprocket drive was moved closer to the sprocket in the improve design. The current design is within the specified range that the bearings can handle so there will be no problems will the current design only improvement can be made.

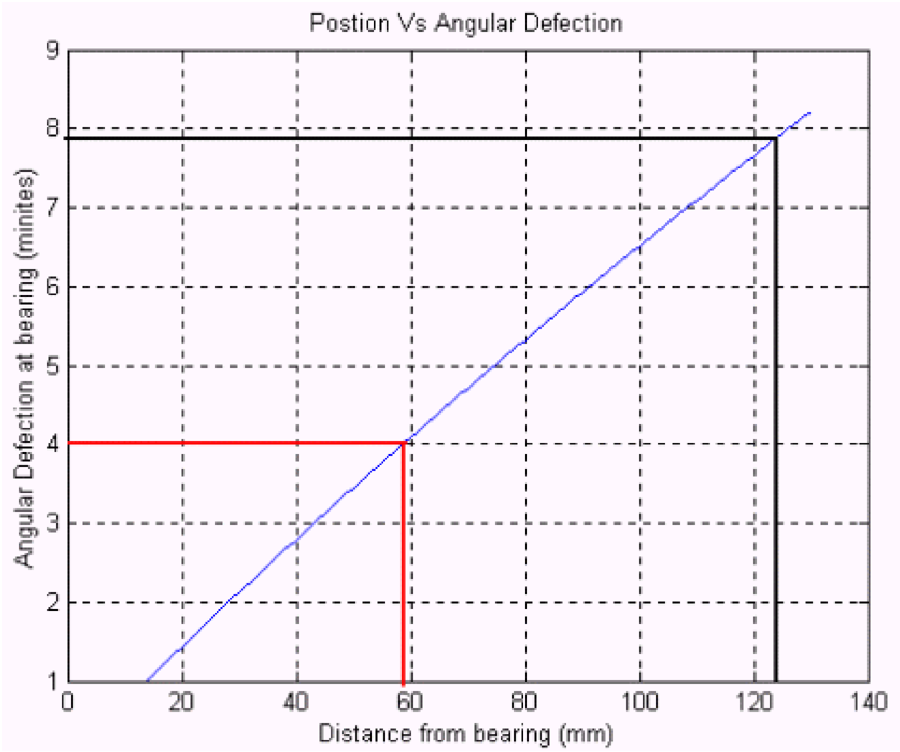

Figure 4.22: Results from MATLAB output of shaft deflection.

4.7 Conclusion

Concluding on the centre shaft design the current design is adequate and will only yield if stresses to the theoretical maximum. The improved design address the high stresses in the involute spline root diameter with a adequate FOS. Further the fatigue and deflection criteria are improved to provide a drivetrain assembly that is designed for infinite life. Using the existing centre shaft in the car is a team decision and the improved design is ready for manufacturing to be implemented in the car.

Chapter 5

Associated Drivetrain Components

5.1 Introduction

In the previous chapters the design of the centre shaft was conducted as well as all the required design information such as the final gearing, axle loading and centre shaft configuration so now the detailed design and analysis of the associated drivetrain components can begin. The associated drivetrain components, which need designing or selecting include:

- Half Shafts
- Bearing Housings
- Roller Chain
- Rear Sprocket
- Chain Tensioner

A brief outline of the design process of each of the components will be covered with areas such as the selection, procurement, modifications and manufacturing of the particular component detailed. In Appendix E and F there are solid models and detailed drawing respectively of all the drivetrain components are shown.

5.2 Half Shafts

The half shafts came from the Ford Telstar drive shaft assembly. Starting with sketches of the drivetrain assembly the lengths of the shafts were determined requiring them to be shortening and re-splining. Fortunately the middle section had enough material to machine a new spline. Similar to the centre shaft, a few trial splines were manufactured (Figure 5.1) on the CNC machine before the final involute spline was machined on the half shafts. A solid model is shown in Appendix E Figure E.3 and the detailed drawing in Appendix F Figure F.3.

The original splines were manufactured by rolling the tooth profile. By cutting new splines on a CNC machine the strength will be reduced compared to the original spline. The material used for these shafts is most likely AISI 4140 or 4340 from researching different shaft materials. So the strength of the new spline will be adequate for the torques being transmitted.

Figure 5.1: Trial splines at top with the final shorted half shaft below.

5.3 Bearing Housing

The rotating centre shaft requires to be supported by two bearing. The bearings are held in place by bearing housings which bolt to the spaceframe via some brackets welded to the outside of the frame. Originally the bearing housings and bearings out of the Telstar were going to be used but since the housings had an offset mounting position flange I decided to replace the housings with a conventional bearing housing.

The bearing housings chosen are an off the shelf part. The size of housing is governed by the bearing size used and I opted to stay with the same bearing dimensions as the Telstar shafts because they are the best sized bearings to fit over the spline with a 30mm bore. From Bearing and Power Transmission in Toowoomba the part number for the housing is P206 and retails for $12.20 and the bearings are 172-6206-2RS which retail for $18.40.

5.3.1 Bearing Loads

The bearings selected for the shaft have the same rated load capacity as the deep groove ball bearing of 62x30x16 dimensions which correlates to a 6206 bearing number (NTN 1990). A deep groove ball bearing is the best choice of bearing as there are very small axial loads and they can handle high speeds and small misalignments. Also the bearings have there own seals to keep the elements such as water and dirt out.

To find out if the bearing can handle the loading the following calculations are conducted to find the dynamic equivalent load of the bearing. The dynamic equivalent load is used instead of the static equivalent load because the bearing is rotating at high speeds under loading. Using the modified bearing situation for the greatest bearing load the left-hand bearing will have a radial load of 3696N under maximum acceleration and no axial load.

Dynamic equivalent radial load formula (6.17) (NTN 1990).

$$P_r = XF_r + YF_a \tag{5.1}$$

where: P_r = Dynamic equivalent radial load (N)

F_r = Actual radial load (N)

F_a = Actual axial load (N)

X = Radial load factor

Y = Axial load factor

Values of X and Y are given in the respective bearing tables for the bearings. Because there is no axial load the value for Y is zero and X = 1.

$$\begin{aligned} P_r &= XF_r + YF_a \\ &= 1 \times 3696 \\ &= 3696N \end{aligned}$$

Comparing the dynamic load rating of the bearing at 19500N with the dynamic equivalent radial load of 3696N the bearing is more than adequate for the given situation. Even if pronounced shock loading occurs which doubles the required dynamic load rating the bearing is still within the given range of acceptable loads.

5.3.2 Bearing Life

The basic bearing life can be found using the following equation (Pemberton & Penfold 2001).

$$L_{10} = \left(\frac{C}{P}\right)^p \tag{5.2}$$

where: L_{10} = Basic rating life in millions of cycles

C = Basic dynamic load rating

P = Equivalent dynamic bearing load

p = 3 for ball bearings

$$\begin{aligned} L_{10} &= \left(\frac{19500}{3696}\right)^3 \\ &= 146.86 \end{aligned}$$

I predicted that the average speed the car will operate at will be the top speed in 2nd gear giving a angular velocity of 86.5 rad/s. The expected life of the bearing for the given situation is:

$$\begin{aligned} L &= \frac{146.86 \times 10^6}{\left(60\left(\frac{86.5\times60}{2\pi}\right)\right)} \\ &= 2963hrs \end{aligned}$$

5.4 Roller Chain

The current chain size on the FZR 600 motorcycle we purchased was classed as a 530 motorcycle chain which corresponds to a single row ANSI Number 50 chain. Some teams used an optional 520 motorcycle size chain which is a smaller chain but with reliability of the car a major design criteria the 530 motorcycle chain was retained. The load on the chain is 4381N and the breaking load of a base chain is 22kN (table 1M)(Standards Australia 1999). A extra good quality chain was selected called the "Drag Race" from RK Chains part number 12-533-130.

This was the strongest chain available in the 530 range. Other choices include a range of sealed link types such as the o-ring and x-ring options. The choice to purchase a non-sealed chain was based on the fact that it requires more power to drive the sealed chains due to power being lost to friction between the seals and links. Also sealed chains are used when operating in harsh environments where dirt and water will make contact with the chain. The condition in which the car will operate will not require a sealed chain. The chain retails for $199 and was purchased from Toowoomba Bikes and Bits.

The length of chain supplied was 130 links and to find the required number of links the chain was installed on the sprockets and the link number noted. The chain was then cut at the required length and jointed via the joiner link. The angle of wrap was a concern as the two sprockets have only approximately 280mm between the centres and the recommended minimum is 120^o (Tsubaki current July 2004) so the wrap angle was checked. Figure 5.2 indicates the angle of wrap on the front sprocket with the following calculations finding angle of wrap.

$$\begin{aligned}\theta &= \sin^{-1}\left(\frac{113}{280}\right)\\ &= 23.8^o\end{aligned}$$

$$\begin{aligned}Wrap &= 180^o - 2\theta\\ &= 180^o - 2 \times 23.8^o\\ &= 132.4^o > 120^o\end{aligned}$$

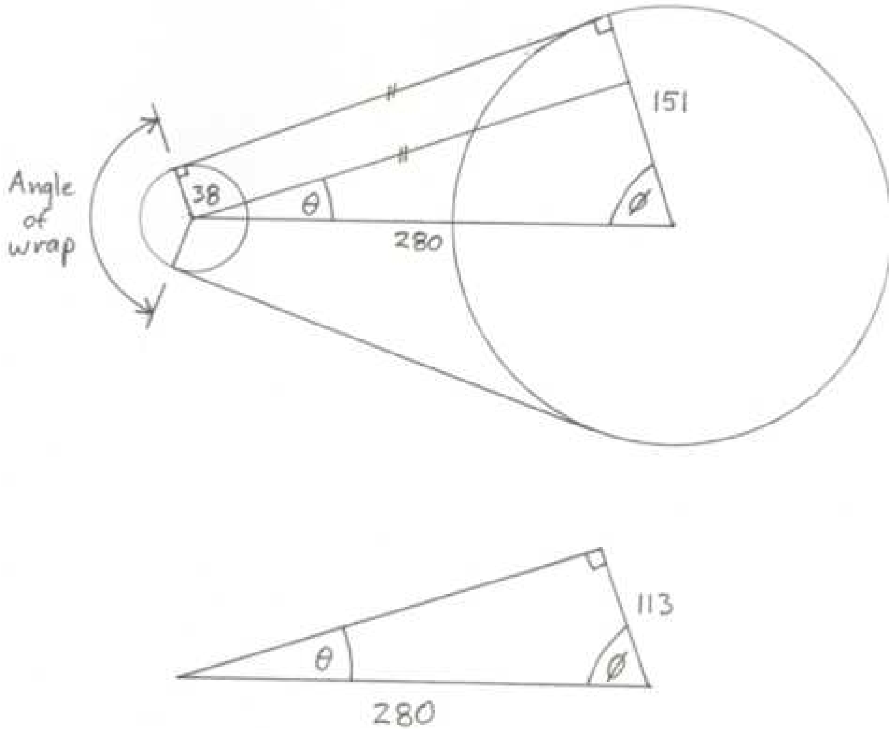

Figure 5.2: Sketch of to find angle of wrap on front sprocket.

5.5 Sprocket Design

The sprocket design started with the final drive ratio selection, which was based on finding the maximum speed required which lead to a rear sprocket size of 60 teeth. The design of the mounting pattern of holes was based on the PCD of the rear rotor pattern just to keep things the same. The reason being that it makes it easier to produce the flanges on the centre shaft if they have the same dimensions and hole pattern. A snug fit was given to the inside diameter of the sprocket to locating flange with a 0.05mm clearance. The pattern of holes to reduce the weight incorporated removing the pattern of holes from the bolting down of the sprocket to the CNC machine bed. Also the thickness of material from the bottom of the tooth profile to the patterned holes was made the same as the original motorbike sprocket. A solid model of the sprocket is in Appendix E Figure E.2.

With the rear sprocket tooth profile specified the code was written for the CNC machining of the profile. First a trial run machined out of plastic was made to check the tooth profile (Figure 5.3). When the final profile was machined the cutter made several increments closer to the final profile rather than making one cut. This was done to make the cutting easier and give a better quality surface finish with the final pass.

The material chosen to manufacture the sprocket was mild steel AISI 1020. The material used for the existing motorbike sprocket was found by using a Brinell hardness tester and a Rockwell hardness tester. The results showed a 187BHN value which corresponds to a ultimate tensile strength of 90ksi. The second test resulted in a 94HRB value which corresponds to a 207BHN. Therefore the material used in the motorbike sprocket correlates to AISI 1040 in appendix C-4a (Juvinall & Marshek 2000). Using AISI 1020 for the sprocket material will result in a slightly faster rate of wear.

Sprocket wear will be an issue if the car is subject to a lot of kilometres but this years entry won't experience much work before the competition. Measures which can be taken to keep the sprocket wear to a minimum are keeping the chain lubricated, tensioned correctly and replacing the chain before it starts to stretch. If time permits the sprocket teeth will be carburised for wear resistance. Tool Shop Engineering in Toowoomba have heat treatment facilities for carburising.

Figure 5.3: Trial plastic sprocket.

Figure 5.4: Rear Sprocket after manufacturing.

5.5.1 Tooth profile

The sprocket tooth profile was required to manufacture the rear sprocket in the CNC machine. With the decision to use a ANSI 50 chain, the tooth profile could be dimensioned using the equations (Oberg, Jones, Horton & Ryffel 1992) listed in Appendix C Figure C.2 and the diagram to go with the dimensions is Figure C.1. The dimensions were first found by hand calculations for the manufacturing of the sprocket. Later on the equations were written in MATLAB (Appendix D) for calculating the dimensions is different sprocket sizes quickly when required.

5.5.2 Chain Tensioner

The design of the chain tensioner has been forced to be reverse engineered with the design being sketched and made in the workshop before any calculations are undertaken. I know the loads on the tensioner will be from the chain trying to straighten itself out under engine braking. Under acceleration the bottom side of the chain were the tensioner is located will have less tension. The tensioner will keep adequate tension though adjustment of two threaded bolts at the top.

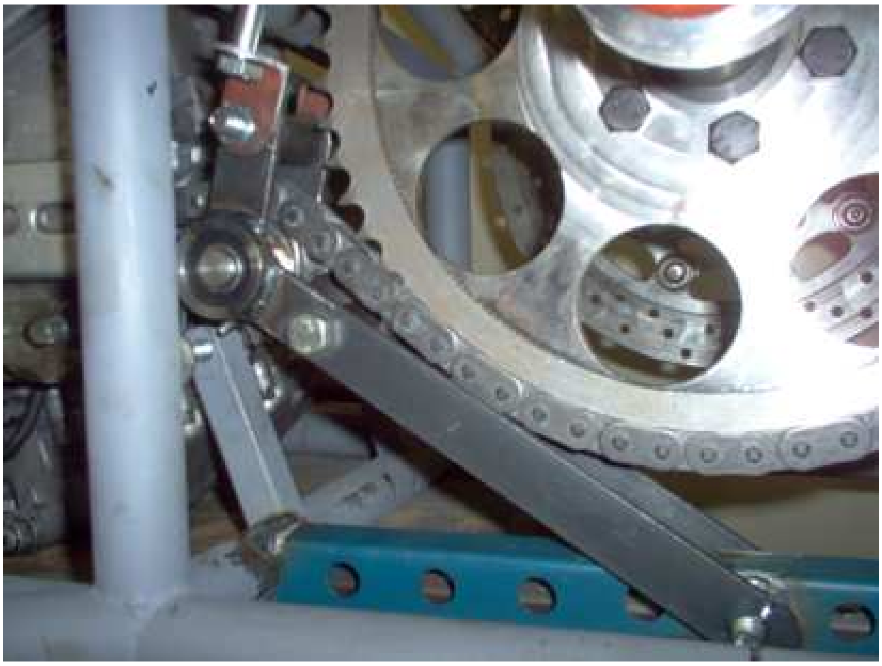

Figure 5.5: Chain Tensioner manufacture for the car.

The conceptual design of the tensioner was achieved by sketches and physical placement of parts and templates in different positions to see the affects on the chain tension. Parts purchased for the tensioners construction include two 6202 ball bearings and 12 tooth sprocket which was the smallest available. Sketches of the lever arms, shafts and housings were drawn for the workshop staff to manufacture. The final design can be seen in Figure 5.5.

5.6 Conclusion

This completes the work undertaken so far with the associated components of the drivetrain. All components have been acquired or manufactured and installed on the centre shaft and mounted in the car see Figure 5.6. The proceeding chapter will cover the areas of the braking system and wheels.

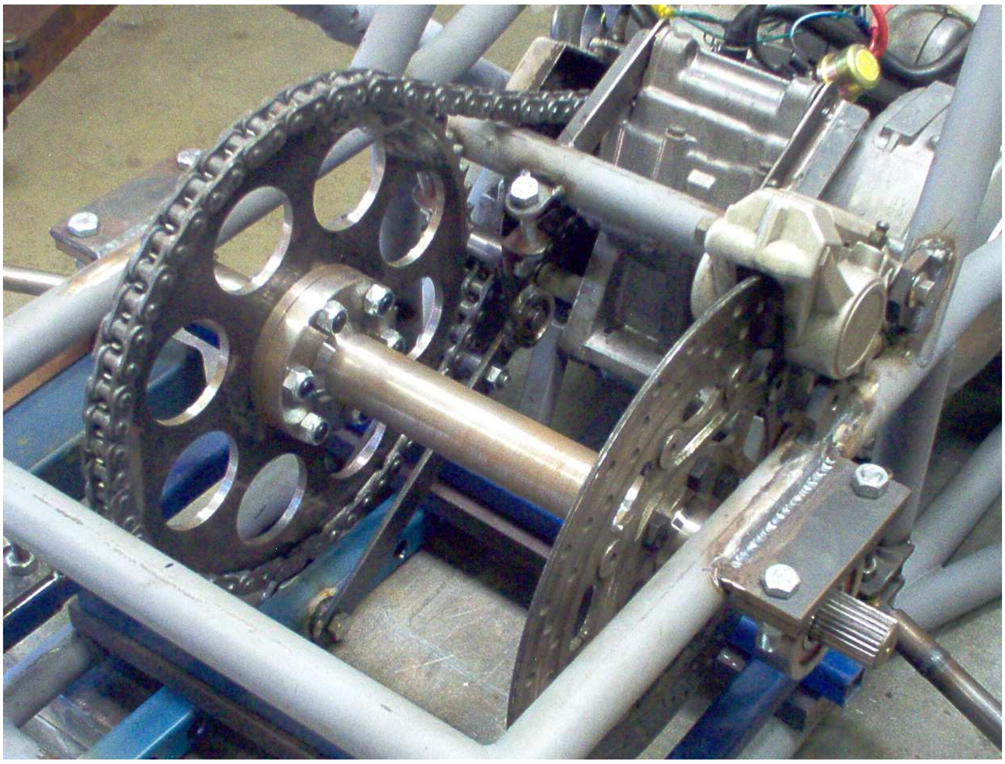

Figure 5.6: Picture of the centre shaft and accociated components assembled in the car.

Chapter 6

Braking System and Wheels

6.1 Introduction

The braking system and wheels conclude the design work covered for this project and the areas of responsibility placed in my charge. The brakes and wheels will basically be a selection process from the variety of components ready made for these applications. First the brake system will be covered followed by the wheels.

6.2 Brake System Design

6.2.1 Introduction

The braking system is the most important safety feature of the car due to it being the only means of stopping. Therefore the system should be very reliable and provide adequate stopping power. From a performance point of view the brakes should give the driver the confidence to push the car to the limits by braking harder and later.

6.2.2 Design Criteria

The design criteria on the braking system include the SAE rules, available space, performance characteristics, reliability, cost and the teams overall aim. The SAE Rules as seen in Appendix B, which pertaining to the braking system are summarised in the following:

- Act on all four wheels by a single control
- "Brake-by-wire" systems are prohibited
- Two circuits with separate fluid reserves
- A single brake acting on a LSD is acceptable
- Scatter shields for safety
- Banjo connections must be used
- The ability to lock all four wheels at high speed

The conclusion from these rules are that positive braking to all wheels is required and the system must not be operated by electric brakes. Also the system must use two separate fluid reserves for the front and rear brakes so in the event of one system failing there will still be two braking wheels available operated by the second system. For safety reasons scatter shields and banjo connection must be implemented in the design. As a final test to assess if the brakes are adequate the car must be able to lock all four wheels at high speed to pass the safety inspection for the competition.

6.2.3 Design Selection

The braking systems available include the band, drum and disk brake type. Current braking systems in use today in the automotive industry are the drum and disk systems operated by hydraulic circuits. Although racing cars use disk brakes exclusively for their advantages over the drum brake systems. The decision to use disk brakes was made due to the following advantages of the disk brake system:

- Better heat dissipation characteristics
- Simplicity of the design
- Self-adjusting ability
- Easy inspection and replacement of the pads

The main advantage of the disk brake system is the better heat dissipation characteristics. Brakes basically exert a force on a rotating member creating a friction force which produces heat that needs to be dissipated by convection cooling. If the heat is not removed then brake fade will occur decreasing the braking performance.

The option of using motorcycle disk brake systems in the design of the car brakes was implemented. Motorcycle brakes are well suited to this application because of their size, weight and performance characteristics.

6.2.4 Brake System Development

The acquisition of the Yamaha FZR 600 motorcycle for the engine meant all the bikes braking system was available to be implemented into the car. Starting with front brakes the bikes rear rotor which is a 245mm diameter cross-drilled item is use on each of the front wheels in conjunction with the front calipers off the bike which are four piston units. The calipers needed mounting modifications to fit inside the rim as seen in Figure 6.1 and 6.2 where the top mount can be seen touching the rim. A second rotor the same as the FZR rear rotor was required to be purchased from a motorcycle wrecker.

Figure 6.1: Positioning front brake assembly inside 13 inch rim.

Figure 6.2: Final assembly of front brake system inside rim.

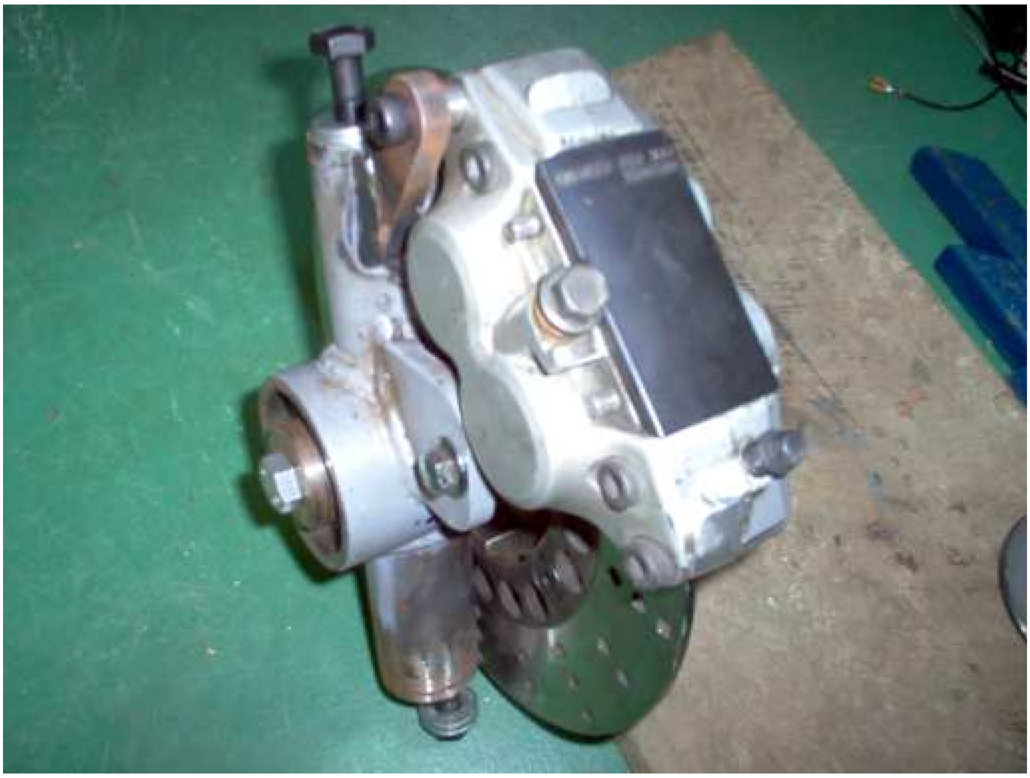

Figure 6.3: Front Caliper mounting arrangement.

Modifications to the rotors of countersinking the bolts was required for clearance of the front uprights. The calipers also required a bleed screw shortening and one of the existing mounts removing meaning a new mounting arrangement was needed. This new mount is created by using a longer bolt and a spacer as illustrated in Figure 6.3.

For the rear one of the front rotors off the bike is mounted to the rear shaft and the rear caliper off the bike operates on the rear rotor. Mounting brackets for the caliper were located directly on top of the spaceframe rails so the force acts directly on a support member as apposed to using the welds strength. Figure 6.4 shows the rear brake assembly.

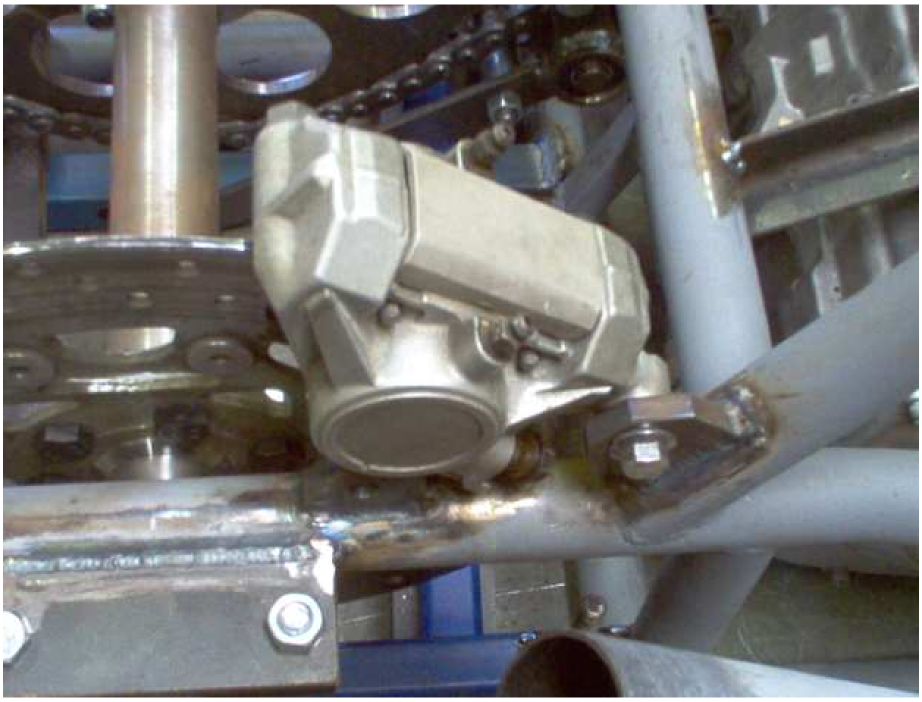

Figure 6.4: Rear Caliper mounting arrangement.

The remaining components required to be specified for the braking system include the master cylinders and brake lines. Two master cylinders were used to satisfy the SAE rules of keeping two separate fluid reserves. To find the right size master cylinders for the given application advice was sort from brake experts as to what could be acquired for the least amount of incurred cost. With the advice a master cylinder brand, PBR part number P6440 was chosen, which has a 19.05mm bore diameter. Considerations as to the end fittings of the master cylinder was required to make sure that the pedal box designer has the right fittings for their design.

Brake lines are required to transfer the brake fluid from the master cylinders to the caliper. This work is still in progress with designs based on flexible braided lines with banjo connections running to the front brake calipers from a T-ee piece. For the rear brake a steel line with flexible braided end will be used. With the required lengths specified and the bends for the steel piece supplied the lines can be constructed by Brakeland Toowoomba ready for installation at a reasonable price. Prices will be subject to the specific lengths but braid lines with banjo connection on the ends up to 1m long cost $58 and steel lines subject to the complexity of the bends cost $25 per metre.

6.2.5 Brake System Analysis

Analysis of the braking system to find the force required to be exerted at the master cylinders was required for the pedal designer. To start with stopping distances that other SAE teams designed to were studied which revealed a stopping distance of 33m from 100km/h. From this and the mass of the car with a driver at 350kg can be used to to determine to braking forces and required master cylinder force.

Using formula (2.4) (Pemberton & Penfold 2001) assuming constant acceleration.

$$\nu^2 = \nu_o^2 + 2as \tag{6.1}$$

where: ν = finial velocity (m/s)

ν = initial velocity (m/s)

a = acceleration (m/s^2)

s = distance (m)

Rearranging to find the acceleration Equation 6.1.

$$\begin{aligned} a &= \frac{\nu^2 - \nu_o^2}{2s} \\ &= \frac{0 - 27.7\dot{7}^2}{2 \times 33} \\ &= -11.69m/s^2 \end{aligned}$$

The braking distribution between the front and rear is assumed to be 70% front and 30% rear. Using Equation 3.2 the total braking force is found.

$$\begin{aligned} F &= 350 \times -11.69 \\ &= -4091.5N \end{aligned}$$

The front contribution.

$$\begin{aligned} F_F &= -4091.5 \times 0.7 \\ &= 2864.05N \end{aligned}$$

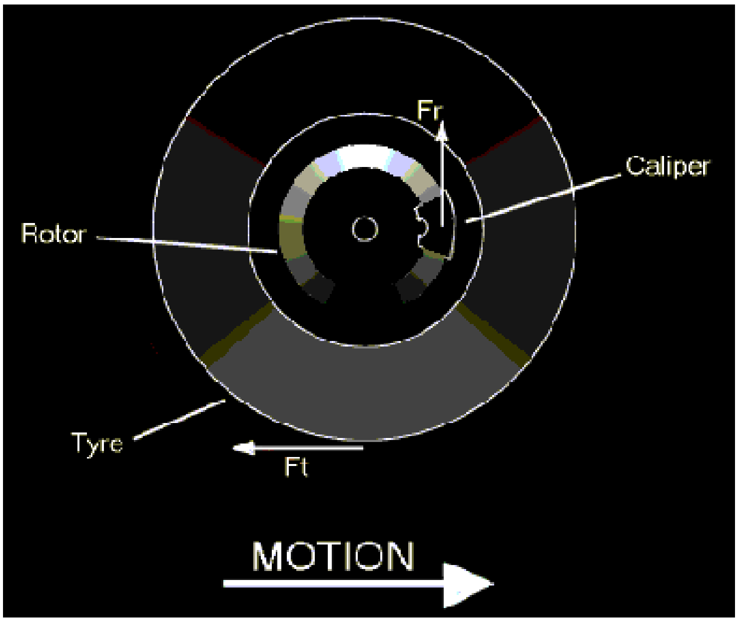

Figure 6.5: Diagram of braking forces on tyre.

Sum of moments around the centre of one front wheel using Equation 3.4. Each front tyre of 0.2765m diameter and will provide 1432N of braking force.

$$\begin{aligned}\sum M &= Fd \\ T_B &= F_F r \\ &= 1432 \times 0.2765 \\ &= 395.95Nm\end{aligned}$$

Equation (18.3) (Juvinall & Marshek 2000) is used to find the torque capacity of the caliper as a function of axial clamping force.

$$T = \frac{2Ff(r_o^3 - r_i^3)}{3(r_o^2 - r_i^2)}N \tag{6.2}$$

where: T = Torque (Nm)

r_o = Outside radius of rotor (m)

r_i = Inside radius of rotor (m)

F = Axial clamping force (N)

f = Dynamic friction coefficient

N = Number of friction interfaces

The normal force exerted by the caliper on the rotor is found by rearranging Equation 6.2.

where: r_o = 0.1225m

r_i = 0.089m

f = 0.3 for sintered metal Table (18.1) (Juvinall & Marshek 2000)

N = 2 (two pads)

$$\begin{aligned} F &= \frac{T_B}{\frac{2}{3}f\left(\frac{r_o^3-r_i^3}{r_o^2-r_i^2}\right)N} \\ &= \frac{395.95}{\frac{2}{3}\times 0.3\times\frac{0.1225^3-0.089^3}{0.1225^2-0.089^2}}\times 2 \\ &= 6188.6N \end{aligned}$$

The master cylinder force required is found using equation (3.9) (Fox & McDonald 1998).

$$P = \frac{F}{A} = constant \tag{6.3}$$

where: P = Pressure (MPa)

F = Force (N)

A = Area (mm^2)

Rearranging Equation 6.3 to find the force required on the master cylinder, subscript M. The caliper, subscript C has twin pistons on either side of the rotor of 30mm and 34mm diameters.

$$\begin{aligned} \frac{F_M}{A_M} &= \frac{F_C}{A_C} \\ F_M &= \frac{F_C}{A_C}\times A_M \\ &= \frac{6188.6}{2(\pi\times 15^2+\pi\times 17^2)}\times\pi\times 9.525^2 \\ &= 546N \end{aligned}$$

The same calculations are undertaken for the rear brakes with the rear contribution given as:

$$\begin{aligned} F_R &= -4091.5 \times 0.3 \\ &= -1227.45N \end{aligned}$$

The sum of moments around the centre of one rear wheel using Equation 3.4. Each front tyre of 0.2765m diameter and will provide 1227.45N of braking force.

$$\begin{aligned} \sum M &= Fd \\ T_B &= F_R r \\ &= 1227.45 \times 0.2765 \\ &= 339.39Nm \end{aligned}$$

The normal force exerted by the caliper on the rotor is found by rearranging Equation 6.2.

where: r_o = 0.1487m

r_i = 0.1147m

f = 0.3 for sintered metal Table 18.1 (Juvinall & Marshek 2000)

N = 2 (two pads)

$$\begin{aligned} F &= \frac{T_B}{\frac{2}{3} f \left(\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \right) N} \\ &= \frac{339.39}{\frac{2}{3} \times 0.3 \times \frac{0.1487^3 - 0.1147^3}{0.1487^2 - 0.1147^2}} \times 2 \\ &= 4271.27N \end{aligned}$$

The master cylinder force required is found by rearranging Equation 6.3 with the diameter of the twin piston caliper being 42.5mm.

$$\begin{aligned} F_M &= \frac{F_C}{A_C} \times A_M \\ &= \frac{4271.27}{2(\pi \times 21.25^2)} \times \pi \times 9.525^2 \\ &= 429N \end{aligned}$$

The results are the front and rear master cylinders require 546N and 429N respectively or a total of 975N to pressurize the hydraulic circuit enough to stop the car in 33m from 100km/h. This is only a theoretical value for the given situation. In reality the brakes will require more force to be exerted on them so the driver can lock the brakes if required. The finer adjustment of the braking performance will come down to a balance between pedal force and feel which needs to be adjusted by the pedal leverage system. As long as the brakes are supplied with more force than these estimates then the main limitation on the brakes performance will then be the heat dissertation capacity.

6.3 Wheels

6.3.1 Introduction

The wheels are the means by which the car makes contact with the asphalt and therefore plays a pivotal role in the handling performance of the car. The wheel is comprised of two parts the rim and the tyre and both parts require careful selection to obtain the best performance characteristics.

6.3.2 Selection Criteria

The selection criteria of the tyre and rim arrangement should be performance based especially on a racing car. But our team this year had a limited budget so the cost becomes a critical selection consideration. There are also a few SAE rules which are specified in Appendix B for the wheels which are summarised in the following:

- A minimum outside tyre diameter of 203.2mm
- Dry or wet weather tyres are allowed
- Any size or type of rims and tyres
- Same rim size is required for both dry and wet weather tyres
- Adhere to restrictions to tyre tread patterns

The second point about dry and wet weather tyres become one of the most important selection criteria. If the track is declared wet at the competition then all the teams must run a grooved tyre to handle the water on the track. Addition to this is both the dry and wet weather tyres must fit the same rim size. Other points to adhere to are the minimum outside tyre diameter and the tyre tread patterns which concerns the modification of the tread pattern.

Tyres come in a large range of types all designed for specific applications. With our application the main objective was to find the lowest profile tyre for the given rim size. Also the performance of the tyre is critical to the overall performance of the car so a tyre that would provide the optimum grip must be chosen.

For the rim the main objective was to supply enough room for the suspension and brake assembly as well as providing a wide range of tyre selection. Also keeping the weight of the rim as light as possible keeps the weight of the car down. One extra selection criteria which is not performance based is the appearance of the rim, which is basically trying to make the car look fast while standing still.

6.3.3 Selection

The first design decision in the selection process was the size of rim to uses that would provide the size requirements and space needed for the suspension, brakes and tyres options. The selection of a 13 inch rim was made on the basis of the available space provided and the large selection of tyres available. Next is simply selecting a specific rim that light and looks appealing out of the endless range of patterns available, which comes down to personal choice and cost. But with a limited budget the cost was very important and this lead to some 13x6 rims being purchased secondhand for only $100. To improve the rims appearance the rims were polished which improves the appearance as seen in Figure 6.6.

Figure 6.6: Polished rims used for the car.

The tyre selection ideally is to purchase both dry and wet weather tyres. Research into the lowest profile tyre available for 13 inch rim revealed that a range between 488mm and 520mm could be purchased in slicks. But the range of slicks imported to Australia is limited to what the large racing competitions use. One competition is the Formula Ford which runs specific sizes and the front tyres are 6/21.0-13 having a diameter of 528mm and tread width of 152mm. The tyre type is a Avon 7317MX with ACB10 semi-slick compound with cross ply construction and the importer is Gordon Leven Motorsport Tyres in NSW. These tyres would provide a perfect solution if our budget allows otherwise a secondhand set of these tyres could be obtained. Also the carcase could be retreaded through a company called Big Tyre in Toowoomba to whatever tread pattern and compound specified.

The cost is a major selection consideration and with slicks priced at over $220 each so other options were considered. Street legal tyres were priced and best option was the Falken ZE512 (Figure 6.7) a directional tyre in a 185/60-13 size costing $145 each retail priced from Quick Fit Tyres Toowoomba. The decision to purchase these tyres was made on the performance for the given price. These tyres have deep grooves for the wet weather so they will serve as a good performing tyre in all conditions. If the teams budget allows the purchasing of the Avon 7317MX tyres new or secondhand will be arranged.

Figure 6.7: Tread pattern of the Falken ZE512 directional tyre purchased for the car.

6.4 Conclusion

The final conclusion of this chapter is the brake system selected uses very light motorcycle components, which will perform well as long as they are supplied with the correct force so then the main limitation on the brakes performance will be the heat dissertation capacity. Further work is required to complete the brake system with the brake lines and then testing to confirm that the car can lock all four wheels at high speed. The braking system has been selected while keeping with the teams aim of producing an inexpensive car which was achieved by utilising existing components such as the parts off the motorcycle.

The wheels have also kept the teams aim with the purchasing of the Falken ZE512 tyres and secondhand rims. Slicks can be purchased at a later date when the car is finished and using the available funds left over. The best choice will be a set of secondhand front Formula Ford tyres that still have some tread left. The conclusion of this project is covered in the next chapter.

Chapter 7

Conclusions and Further Work

7.1 Introduction

This chapter concludes the project work by summarizing the final conclusion of the drivetrain, brakes and wheels. Further work required to complete the work started in this project is discussed as to how this will be undertaken.

7.2 Project Work Summary

The project aim was to develop the drivetrain, brakes and wheels for the Formula SAE racing car. The designs must meet all Formula SAE rules and regulations including all safety features. The design must also be optimized to provide for a competitive racing car. These aims have been achieved by the designing of a drivetrain assembly, the selection of brake components and procurement of wheels for the car. The design criteria has governed the development of these areas which is covered in chapter 3. The summary of the areas covered in this project will be discussed in the proceeding sections.

7.2.1 Summary of Drivetrain Development

To starting with a review of literature on the current drivetrain systems developed provided a solid background to this area. From this background conceptual designs were developed for the specific application of a Formula SAE car and from these options a drivetrain design was selected. The design was a chain driven solid rear axle with CV joints and half shafts completing the connection to the driven wheels.

To proceed onto analysing the design the loading situation of the axle was assessed. First the final gearing of the car was determined by finding the required top speed for the car at the competition. Based on the FZR's performance the cars capable speeds in each gear were found which showed that a 60/13 option was the optimum final gearing. With the gearing determined the torque developed in the rear axle was assessed by three approaches. The best approach was based on the torque that the tyres can withstand before slipping. Loading of the axle was then able to be found for assessing each of the drivetrain components.

A detailed analysis of the centre shaft by looking at the stresses and fatigue life was conducted which has been suffice for the analysis. This work has replaced the FEA analysis for the first iteration of the centre shaft. The stress and fatigue loading situations were conducted at the extreme cases to confirm that the shaft will not fail. The current centre shaft design is adequate for the application of competing in the competition. A improved design is specified, which addresses the high stresses in the shaft and incorporates an adequate FOS. Using the existing centre shaft in the car is a team decision and the improved design is ready for manufacturing and implemented in the car if our resources permit.

Other components of the drivetrain assembly which required manufacturing or purchasing include the rear sprocket, chain, half shafts, bearing housings and chain tensioner. The rear sprocket tooth profile was determined by the ANSI 50 chain size and was made on the CNC machine. The half shafts came from a Telstar because they only had external splines making it easier to shortening and re-splining. The bearings selected where influenced by the shaft diameter and the necessary calculation conducted to assess the load rating and life of the bearings was undertaken. Finally the chain tensioner has just been started to complete the drivetrain assembly. The final construction of the main drivetrain assembly at the time of writing is shown in Figure 7.1.

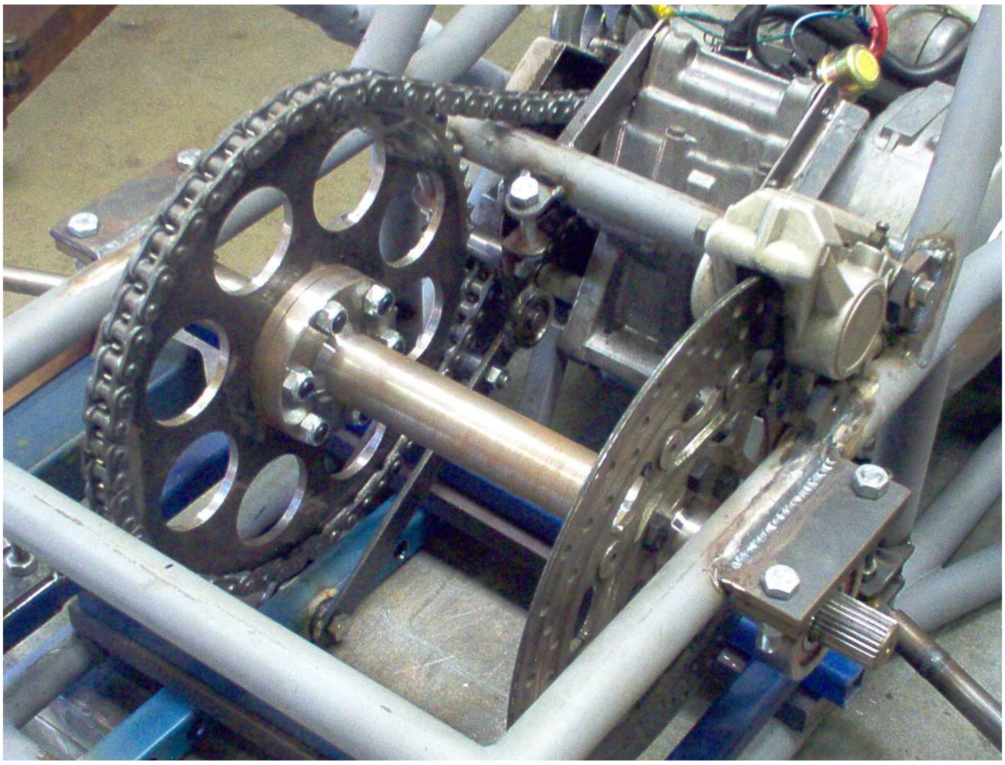

Figure 7.1: Photo of main drivetrain assembly.

7.2.2 Summary of Brakes and Wheels

The brakes and wheels also involved a literature review to establish a solid background into the current developments of these areas. The final specification of the brakes and wheels has been completed with the acquisition of the components and implementation very close to completion.

The brake system selected uses very light motorcycle components, which will perform well. The braking system has been selected while keeping with the teams aim of producing an inexpensive car utilising existing components such as the parts off the motorcycle. The front rotors are the rear disk off the FZR which are 245mm in diameter. There rear disk is also off the FZR having a diameter of 297mm. The calipers are also used from the bike. The master cylinders chosen are from PBR, part number P6440 which have a 19.05mm bore. All the brake lines will consist of steel lines with flexible braided steel lines at the ends.

The wheels have also kept the teams aim with the purchasing of the Falken ZE512 tyres in a 185/60-13 size. Slicks can be purchased at a later date when the car is finished construction and using the left over funds. The best choice will be a set of secondhand front Formula Ford tyres. These front tyres are 6/21.0-13 having a diameter of 528mm and tread width of 152mm and the tyre type is a Avon 7317MX with ACB10 semi-slick compound with cross ply construction.

7.2.3 Final Design

The final design of the drivetrain, brakes and wheels has satisfied the project aim and objectives to the point where the car is nearly finished construction. The performance and reliability of the system will be assessed in future testing of the car. But the implementation of the designs into the car has appeared to work well. The final construction of the rear end of the car at the time of writing is shown in Figure 7.2.

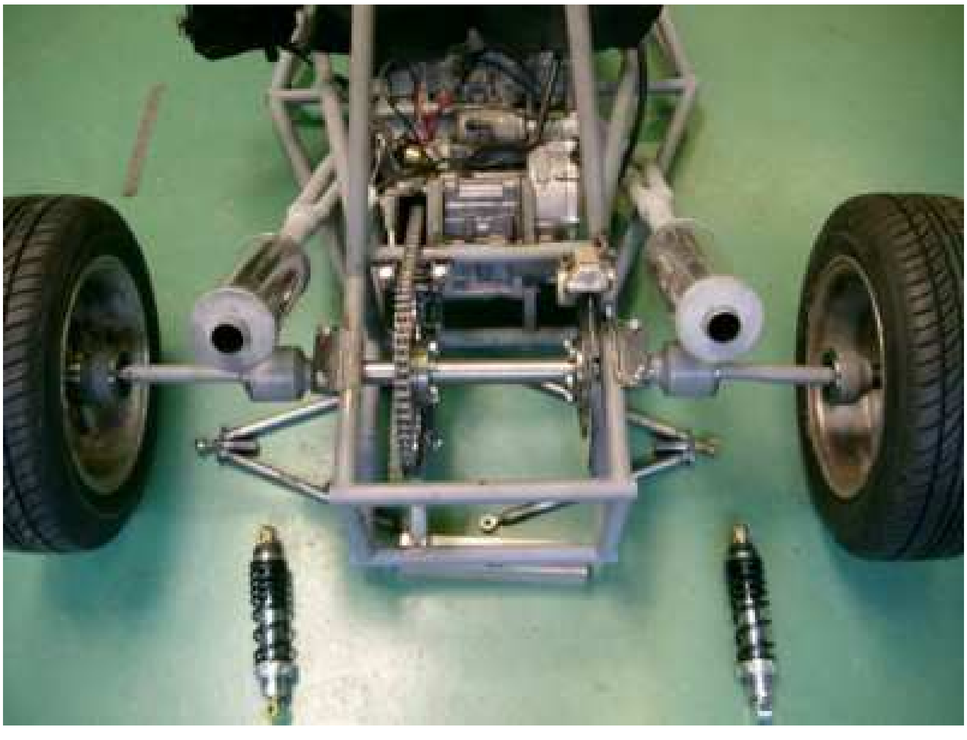

Figure 7.2: The state of completion of the rear assembly.

7.3 Further Work

Further work is required to complete the car ready for the SAE competition. If time permitted a new centre shaft would be manufactured out of a new material with some slight design changes as shown in the detailed drawing Figure F.6. Also the safety shields require construction to the SAE specification. The brake system requires instillation of the brake lines and testing to confirm that the car can lock all four wheels at high speed. If funds permit a additional set of wheels should be purchased. When all this work is completed the testing of the performance, handling and reliability of the cars components will give an indication of the success of the designs implemented.

Further work is required to analyse the centre shaft with finite element analysis (FEA) to further improve the initial design. This would be part of the design iteration by testing of the centre shaft to show the stresses at the splined ends for a better understanding of the situation. Also changes in the diameter in the mid section of the centre shaft could be modeled to find the minimum size shaft capable of transmitting the torque to the rear wheel.

Actual testing is a major part of a product development once the initial calculations of the design has been conducted. Testing of the car is the only way to be sure of the components performance and reliability. After a testing period the drivetrain assembly should be inspected for cracks and wear on the shaft splines and sprocket teeth for example to assess the designs reliability. The effectiveness of the drivetrain, brake and wheel performance will be shown by testing as well, so then further improvements can be made to the design. The braking system will rely heavily on testing to know what alteration are needed to improve their performance in power, feel, repeatability and balance between front and rear.

Work which could be undertaken as a project in future Formula SAE-A entries is the implementation of a differential. Also the merits of implementing a CVT drivetrain assembly in an SAE car could reveal some benefits to the performance of the car.

7.4 Conclusion

The design and selection of the drivetrain, brakes and wheels has been achieved. The implementation of the designs into the car has been successful so fare in the construction of the car with the implementation appears to work well judging from the construction so far. Further work in construction and testing is required before a complete assessment of the designs performance is known and further work is needed before entering the Formula SAE competition.

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Appendix A

Project Specification

University of Southern Queensland

Faculty of Engineering and Surveying

ENG 4111/4112 Research Project
PROJECT SPECIFICATION ISSUE A

FOR: Jeremy Edward LITTLE

TOPIC: DEVELOPMENT OF THE DRIVETRAIN INCLUDING BRAKES AND WHEELS FOR THE FORMULA SAE VEHICLE.

SUPERVISOR: Selvan PATHER
Peter PENFOLD

PROJECT AIM:

This project aims to develop the drivetrain system for the Formula SAE racing car. The design must meet all Formulae SAE rules and regulations including all safety features. The design must also be optimised to provide for a competitive racing car.

PROGRAMME:

1. Review literature in the areas of drivetrain, brakes and wheels.
2. Develop conceptual designs for the drivetrain, brakes and wheels for the specific use of the vehicle.
3. Select and undertake detail design of the best concept (from above) that is within the constraints of the design criteria, time and finances.
4. Investigate possible parts available for the use in constructing the formula SAE vehicle.
5. Undertake preliminary FEA analysis on specific drivetrain components.

6. Prepare 3D solid models and working drawings of the final design.

   If time permits:

7. Implement designs on the formula SAE vehicle.

8. Compare theoretical and actual performances and behaviors of the drivetrain, brakes and wheels.

AGREED:

____________________________(Student)

__________________________(Supervisor)

Dated / /

Appendix B

Relevant SAE Rules 2004

B.1 Drivetrain

3.1 General Design Requirements

3.1.2 Wheelbase and Vehicle Configuration - The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the center of ground contact of the front and rear tyres with the wheels pointed straight ahead. The vehicle must have four wheels that are not in a straight line.

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

3.2 Chassis Rules

3.2.1 Ground Clearance - Ground Clearance must be sufficient to prevent any portion of the car (other than tyres) from touching the ground during track events.

3.2.3 Suspension - The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate unsafe handling.

3.5 Powertrain

3.5.1.3 Transmission and Drive - Any transmission and drivetrain may be used.

3.5.1.4 Drive Train Shields and Guards - Exposed high-speed equipment,such as torque converters, clutches, belt drives and clutch drive, must be fitted with scatter shields to protect drivers, bystanders, fuel lines and safety equipment (such as brake lines) from flying debris in case of failure. Scatter shields protecting chains or belts must not be made of perforated material.

(A) Chain drive - Scatter shields protecting chains must be made of at least 2.66 mm (0.105 inch) mild steel (no alternatives are allowed), and have a minimum width equal to three (3) times the width of the chain.

(B) Belt drive - Scatter shields protecting belts must be made from at least 3.0 mm (0.120 inch) Aluminum Alloy 6061-T6, and have a minimum width that is equal to the belt width plus 35% on each side of the belt (1.7 times the width of the belt).

(C) Attachment Fasteners - All fasteners attaching scatter shields and guards must be a minimum 6mm grade M8.8 (1/4 inch SAE grade 5). Attached shields and guards must be mounted so that they remain laterally aligned with the chain or belt under all conditions.

(D) Finger Protection - Guards for finger protection may be made of lighter material.

B.2 Brakes

3.2.5 Brake Systems - The car must be equipped with a braking system that acts on all four wheels and is operated by a single control. It shall have two independent hydraulic circuits such that in the case of a leak or failure at any point in the system, effective braking power shall be maintained on at least two wheels. Each hydraulic circuit shall have its own fluid reserve, either by the use of separate reservoirs or by the use of a dammed, OEM-style reservoir. "Brake-by-wire" systems are prohibited. A single brake acting on a limited-slip differential is acceptable. The braking system must be protected with scatter shields from failure of the drive train or from minor collisions. Unarmored plastic brake lines are prohibited.

4.2 Safety and Technical Inspection

Part 3 Brake and Noise Tests - Each vehicle must demonstrate its ability to lock all wheels after an acceleration run. Vehicle noise will be tested by the specified method. (Rule 3.5.5.3)

B.3 Wheels and Tyres

3.2.2 Wheels and Tyres - The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter. Vehicles may have two types of tyres as follows.

a) Dry Tyres - The tyres on the vehicle when it is presented for technical inspection are defined as its "Dry Tyres". The dry tyres may be any size or type. They may be slicks or treaded.

b) Where teams have two types of tyres, the Dry tyres must be installed on the vehicle for Technical Inspection and the Wet or Rain tyres also presented for acceptance. The Wet tyres must fit on the same wheel/rim size as the Dry tyres and be of equal or greater OD.

c) Rain Tyres - Rain tyres may be any size or type of treaded or grooved tyre provided:

(i) The tread pattern or grooves were moulded in by the tyre manufacturer, or were cut by the tyre manufacturer or his appointed agent. Any grooves that have been cut must have documentary proof that it was done in accordance with these rules.

(ii) The tread pattern and grooves are subject to approval at technical inspection. A groove depth of at least 2.4 mm is expected.

Note: Hand cutting, grooving or modification of the tyres by the teams is specifically prohibited.

d) Within each tyre set, the tyre or wheel type, compound or size may not be changed after the static judging has begun. Tyre warmers are not allowed. No traction enhancers may be applied to the tyres after the static judging is begun.

B.4 Safety Rules

4.2.1 Objective - The objective of safety and technical inspection is to determine if the vehicle meets the FSAE design and safety requirements and if, considered as a whole, it satisfies the intent of the Rules. For purposes of interpretation and inspection the violation of the intent of a rule is considered a violation of the rule itself.

4.2.3 Inspection Process Part 3 - Brake and Noise Tests - Each vehicle must demonstrate its ability to lock all wheels after an acceleration run. Vehicle noise will be tested by the specified method. (Rule 3.5.5.3)

3.7.2 Fasteners

3.7.2.1 Grade Requirements - All bolts utilized in the steering, braking, safety harness and suspension systems must meet SAE Grade 5, Metric Grade M 8.8 and/or AN/MS specifications.

3.7.2.2 Securing Fasteners - All critical bolt, nuts, and other fasteners on the steering, braking, safety harness, and suspension must be secured from unintentional loosening by the use of positive locking mechanisms. Positive locking mechanisms include: - Correctly installed safety wiring - Cotter pins - Nylon lock nuts - Prevailing torque lock nuts

Note: Lock washers and thread locking compounds, e.g. Loctite, DO NOT meet the positive locking requirement. All spherical rod ends on the steering or suspension shall be in double shear or captured by having a screw/bolt head or washer with an O.D. that is larger than spherical bearing housing I.D. Adjustable tie-rod ends must be constrained with a jam nut to prevent loosening.

3.7.3 Modifications and Repairs - Modifications to the car are not allowed after the inspection and engineering judging except as noted below. This includes modifications that affect the available gear ratios, power transfer-system, or safety. The removal of body panels for weight reduction is not allowed. Adjustments (e.g., tyre pressure, brake bias, suspension adjustments, wing angle, and chain or belt tension) are allowed to the car after the start of the performance events. Necessary repairs are allowed under the knowledge of the Faculty Advisor and the car must pass a re-inspection by the inspection judges.

B.5 Static and Dynamic Events Objectives

4.2 Safety and Technical Inspection (0 points)

4.2.1 Objective - The objective of safety and technical inspection is to determine if the vehicle meets the FSAE design and safety requirements and if, considered as a whole, it satisfies the intent of the Rules. For purposes of interpretation and inspection the violation of the intent of a rule is considered a violation of the rule itself.

4.2.3 Inspection Process Part 3 - Brake and Noise Tests - Each vehicle must demonstrate its ability to lock all wheels after an acceleration run. Vehicle noise will be tested by the specified method. (Rule 3.5.5.3)

4.3 Cost and Manufacturing Analysis Event (100 points)

4.3.1 The Concept - The objective of the Cost and Manufacturing Event is twofold: 1. To teach the participants that cost and a budget are significant factors that must be taken into account in any engineering exercise. 2. For the participants to learn and understand the manufacturing techniques and processes of some of the components that they have chosen to purchase rather than fabricate themselves.

4.4 Presentation Event (75 points)

4.4.1 Presentation Event Objective - The concept of the presentation event is to evaluate the team's ability to make a presentation to the executives of a manufacturing firm. The presentation should address the "Concept of the Competition" as described in section 1, and should convince the executives of the superiority of the team's design. The presentation judges will evaluate the organization, content, and delivery of the presentation. The team that makes the best presentation (regardless of the quality of the car) will win the event.

4.5 Design Event (150 points)

4.5.1 Design Event Objective - The concept of the design event is to evaluate the engineering effort that went into the design of the car and how the engineering meets the intent of the market. The car that illustrates the best use of engineering to meet the design goals and the best understanding of the design by the team members will win the design event.

5.4 Acceleration Event (75 points)

5.4.1 Acceleration Objective - The acceleration event evaluates the car's acceleration in a straight line on flat pavement.

5.4.2 Acceleration Procedure - The cars will accelerate from a standing start over a distance of 75 m (82 yards) on a flat surface. The foremost part of the car will be staged at 0.30 m (11.8 inches) behind the starting line. A green flag will be used to indicate the approval to begin, however, time starts only after the vehicle crosses the start line. There will be no particular order of the cars in each heat. A driver has the option to take a second run immediately after the first.

5.5 Skid-Pad Event (50 points)

5.5.1 Skid-Pad Objective - The concept of the skid-pad event is to measure the cornering ability of the car on a flat surface while making a constant-radius turn.

5.6 Autocross Event (150 points)

5.6.1 Autocross Concept The concept of the autocross event is to evaluate the car's maneuverability and handling qualities on a tight course without the hindrance of competing cars. The autocross course will combine the performance features of acceleration, braking, and cornering into one event.

5.7 Endurance and Fuel Economy Event

5.7.2 Endurance Objective (350 points) - The Endurance Event is designed to evaluate the overall performance of the car and to test the car's reliability.

5.7.3 Fuel Economy (50 points) The car's fuel economy will be measured in conjunction with the endurance event. The fuel economy under racing conditions is important in most forms of racing and also shows how well the car has been tuned for the competition.

This is a compromise event because the fuel economy score and endurance score will be calculated from the same heat. No refueling will be allowed during an endurance heat.

Appendix C

Gearinging Tables

C.1 Gearing Tables

Gearing (45/15)	Gear Ratio	Max Eng (rad/s)	Primary Gear ratio	Clutch (rad/s)	Front Sprocket (rad/s)	45/15 gearing	Rear shaft (rad/s)	Wheel Radius (552mm)	Speed (m/s)	Speed (km/h)
1st	2.846154	1151.9	1.7083	674.28	236.910	3	78.97	0.2765	21.84	78.61
2nd	1.947368	1151.9	1.7083	674.28	346.253	3	115.42	0.2765	31.91	114.89
3rd	1.55	1151.9	1.7083	674.28	435.021	3	145.01	0.2765	40.09	144.34
4th	1.333333	1151.9	1.7083	674.28	505.712	3	168.57	0.2765	46.61	167.80
5th	1.192308	1151.9	1.7083	674.28	565.528	3	188.51	0.2765	52.12	187.64
6th	1.111111	1151.9	1.7083	674.28	606.855	3	202.29	0.2765	55.93	201.35

Table C.1: Standard FZR 600 Top Speeds.

Gearing (60/15)	Gear Ratio	Max Eng (rad/s)	Primary Gear ratio	Clutch (rad/s)	Front Sprocket (rad/s)	60/15 gearing	Rear shaft (rad/s)	Wheel Radius (552mm)	Speed (m/s)	Speed (km/h)
1st	2.846154	1151.9	1.7083	674.28	236.910	4	59.23	0.2765	16.38	58.96
2nd	1.947368	1151.9	1.7083	674.28	346.253	4	86.56	0.2765	23.93	86.17
3rd	1.55	1151.9	1.7083	674.28	435.021	4	108.76	0.2765	30.07	108.26
4th	1.333333	1151.9	1.7083	674.28	505.712	4	126.43	0.2765	34.96	125.85
5th	1.192308	1151.9	1.7083	674.28	565.528	4	141.38	0.2765	39.09	140.73
6th	1.111111	1151.9	1.7083	674.28	606.855	4	151.71	0.2765	41.95	151.02

Table C.2: 60/15 final gearing option 1.

Gearing (60/14)	Gear Ratio	Max Eng (rad/s)	Primary Gear ratio	Clutch (rad/s)	Front Sprocket (rad/s)	60/14 gearing	Rear shaft (rad/s)	Wheel Radius (552mm)	Speed (m/s)	Speed (km/h)
1st	2.846154	1151.9	1.7083	674.28	236.910	4.2857	55.28	0.2765	15.28	55.02
2nd	1.947368	1151.9	1.7083	674.28	346.253	4.2857	80.79	0.2765	22.34	80.42
3rd	1.55	1151.9	1.7083	674.28	435.021	4.2857	101.51	0.2765	28.07	101.04
4th	1.333333	1151.9	1.7083	674.28	505.712	4.2857	118	0.2765	32.63	117.46
5th	1.192308	1151.9	1.7083	674.28	565.528	4.2857	131.96	0.2765	36.49	131.35
6th	1.111111	1151.9	1.7083	674.28	606.855	4.2857	141.6	0.2765	39.15	140.95

Table C.3: 60/14 final gearing option 2.

Gearing (60/13)	Gear Ratio	Max Eng (rad/s)	Primary Gear ratio	Clutch (rad/s)	Front Sprocket (rad/s)	60/13 gearing	Rear shaft (rad/s)	Wheel Radius (552mm)	Speed (m/s)	Speed (km/h)
1st	2.846154	1151.9	1.7083	674.28	236.910	4.6154	51.33	0.2765	14.19	51.09
2nd	1.947368	1151.9	1.7083	674.28	346.253	4.6154	75.02	0.2765	20.74	74.68
3rd	1.55	1151.9	1.7083	674.28	435.021	4.6154	94.26	0.2765	26.06	93.82
4th	1.333333	1151.9	1.7083	674.28	505.712	4.6154	109.57	0.2765	30.30	109.07
5th	1.192308	1151.9	1.7083	674.28	565.528	4.6154	122.53	0.2765	33.88	121.97
6th	1.111111	1151.9	1.7083	674.28	606.855	4.6154	131.49	0.2765	36.36	130.88

Table C.4: 60/13 final gearing option 3.

C.2 Tooth Profile

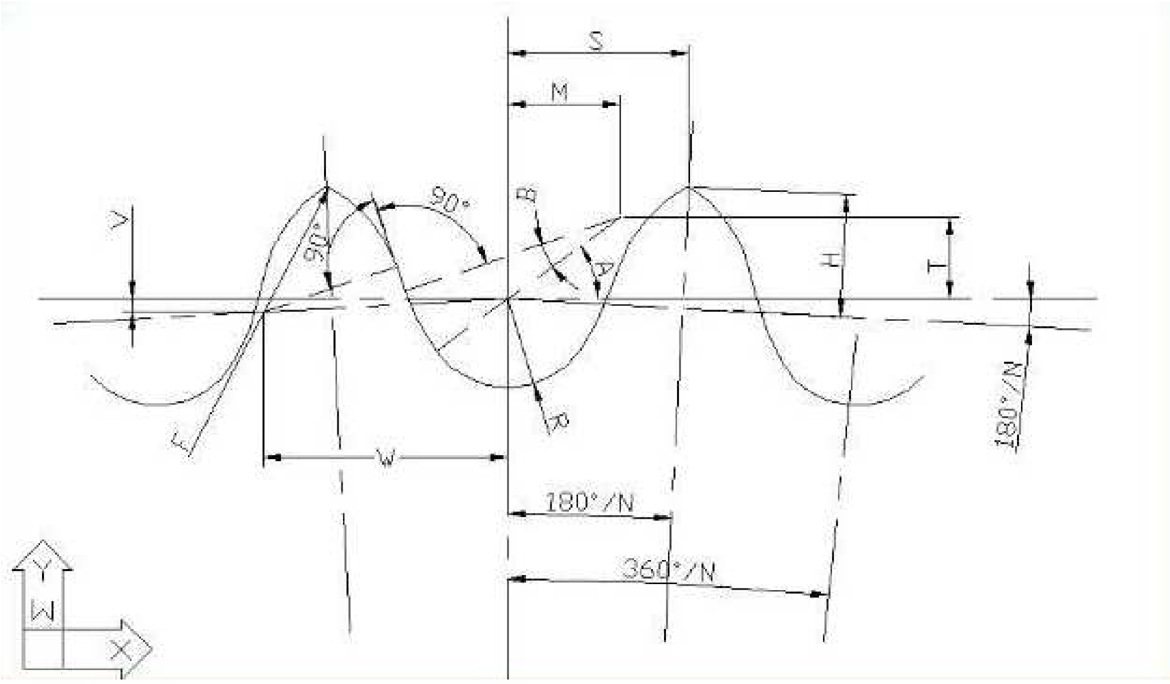

Figure C.1: ANSI 50 size sprocket tooth profile drawn in AutoCAD 2002.

Finding the sprocket tooth form for the ANSI class 50 chain the following calculations from the Machinery's Handbook, page 2350 are used. All working is in inches and degrees so final conversions are required for SI units.

Given: P = pitch; N = number of teeth

Dr = nominal roller diameter; Ds = seating curve diameter

$$Ds = 1.005Dr + 0.003$$

$$R = 0.5Ds$$

$$A = 35° + (60° / N)$$

$$B = 18° - (56° / N)$$

$$ac = 0.8Dr$$

$$M = 0.8Dr\cos(35° + (60° / N))$$

$$T = 0.8Dr\sin(35° + (60° / N))$$

$$E = 1.3025Dr + 0.0015$$

$$Chordxy = (2.605Dr + 0.003)\sin(9° - (56° / N))$$

$$yz = Dr[1.4\sin(17° - (64° / N)) - 0.8\sin(18° - (56° / N))]$$

$$ab = 1.4Dr$$

$$W = 1.4Dr\cos(180° / N)$$

$$V = 1.4Dr\sin(180° / N)$$

$$F = Dr[0.8\cos(18° - (56° / N)) + 1.4\cos(17° - (64° / N)) - 1.3025] - 0.0015$$

$$H = \sqrt{F^2 - (1.4Dr - 0.5P)^2}$$

$$S = 0.5P\cos(180° / N) + H\sin(180° / N)$$

$$OD = P[0.6 + \cot(180° / N)]$$

Figure C.2: Formulas for finding tooth profile.

Appendix D

MATLAB Scripts

D.1 Engine Torque

```
%ENGTORQUE.M Script file
%Finding torque from engine performance
%Jeremy Little 14/9/04
%
clear clc
F = 15;       %Front sprocket number of teeth
R = 60;       %Rear sprocket number of teeth
T = 48.5;       %Max predicted torque (Nm)
%Gear ratio
PR = (82/48);     %Primary Reduction
SR = (R/F);       %Secondary Reduction
First = (37/13); Second = (37/19); Third = (31/20); Forth = (28/21);
Fifth = (31/26); Sixth = (30/27);
%Torque produced in transmission shafts
Crank = T;                 %Crankshaft
Clutch = Crank*PR;         %Clutch basket/shaft
Counter = Clutch*First;    %Countershaft (front sprocket)
Axle = Counter*SR          %Rear Axle (Nm)
%EOF
```

D.2 Tyre Torque

```
%TYRETORQUE.M Script file
%Finding torque from tyre friction
%Jeremy Little 14/9/04
clear
clc
r = 0.2765;             %Radius of wheels (m)
mc = 250;               %Car mass (kg)
md = 100;               %Driver mass (kg)
m = mc+md;              %Total mass
ratio = 0.7;            %70% weight distribution
mr = m*ratio;           %Rear mass
Fn = mr*9.81;           %Weight on rear tyres, normal force
u = 1;                  %Coefficient of friction of tyres
Fr = u*Fn;              %Friction force from rear tyres
T = Fr*r                %Maximum torque limited by tyres
%EOF
```

D.3 Bearing Deflections

```
%DEFLECTIONS.M Script file
%Finding angular defections at bearing using different
%distances between bearings and sprocket flange
%Jeremy Little 28/9/04
clear
clc
%Allowable bearing angular defection
theta1=4;                        %minutes from bearing spec's
theta2=(1/60)*theta1;            %degrees
i=1;                             %counter
t=1;                             %Step interval (mm)
%Loop to see change in defections for different bearing locations
for a=(14:t:130)    %Left side allowable range (mm)
b=312.8;            %Right side fixed (mm)
L=a+b;              %Total length of shaft (mm)
P=4381.37;          %Load on shaft (N)
E=207*10^3;         %Modulus of Elasticity (MPa)
d=38;               %Diameter of shaft (mm)
I=pi*d^4/64;        %Moment of Inertia (mm^4)
%Slop at ends where bearings are located (angular defection)
%Left side bearing
thetaA1=P*b*(L^2-b^2)/(6*L*E*I); %radians
thetaA2=thetaA1*180/pi;          %degrees
thetaA3=thetaA2*60;              %minutes
%Right side bearing
thetaB1=P*a*(L^2-a^2)/(6*L*E*I); %radians
thetaB2=thetaB1*180/pi;          %degrees
thetaB3=thetaB2*60;              %minutes
%Tabulate results of left and right side bearing locations
thetaA(i,1)=a;                   %Location of bearing
thetaA(i,2)=thetaA3;             %Angular deflection of left bearing
```

```
thetaA(i,3)=thetaB3;                %Angular deflection of right bearing
i=i+1;                              %Increment counter
end
%Plot left side bearing deflections
plot(thetaA(:,1),thetaA(:,2))%
grid on %
Title('Angular Deflection of Left Bearing')%
xlabel('Distance from bearing (mm)')%
ylabel('Angular Defection at bearing (minutes)')%
pause
%Plot left & right bearing deflections
plot(thetaA(:,1),thetaA(:,2),thetaA(:,1),thetaA(:,3))%
grid on
Title('Angular Deflections at Bearings')%
xlabel('Left Bearing Location (mm)')%
ylabel('Angular Defection at Bearing (minutes)')%
legend('Left Bearing','Right Bearing')
%EOF
```

D.4 Sprocket Tooth Profile

```
%TOOTHPROFILE.M Script file
%Sprocket Tooth Form For Roller Chain Calculations
%Jeremy Little 1/10/04
clear
clc
%All Working in Inches so conversions required.
%ANSI Class 50 chain
P=0.625;            %Pitch
N=60                %Number of teeth
Dr=0.4;             %Nominal roller diameter
Ds=1.005*Dr+0.003;  %Seating curve diameter
R=0.5*Ds;           %radius
A=35+(60/N)         %degrees
B=18-(56/N)         %degrees
ac=0.8*Dr;
%convert degrees to radians in working
A1=35*pi/180;A2=60*pi/180;A3=9*pi/180;A4=28*pi/180;A5=17*pi/180;
A6=64*pi/180;A7=1.8*pi/180;A8=56*pi/180;A9=180*pi/180;A10=18*pi/180;
%
M=0.8*Dr*cos(A1+(A2/N));%
T=0.8*Dr*sin(A1+(A2/N)); %
E=1.3025*Dr+0.0015;
Chordxy=(2.605*Dr+0.003)*sin(A3-(A4/N)); %
yz=Dr*((1.4*sin(A5-(A6/N)))-(0.8*sin(A7-(A8/N)))); %
ab=1.4*Dr;%
W=1.4*Dr*cos(A9/N);%
V=1.4*Dr*sin(A9/N); %
F=Dr*(0.8*cos(A10-(A8/N))+1.4*cos(A5-(A6/N))-1.3025)-0.0015;
H=sqrt(F^2-(1.4*Dr-0.5*P)^2); S=0.5*P*cos(A9/N)+H*sin(A9/N);
OD=P*(0.6+cot(A9/N));    %When J is 0.3P
%Convert inches to millimeters
```

```
Pmm=25.4*P %

Drmm=25.4*Dr %

Dsmm=25.4*Ds %

Rmm=25.4*R %

acmm=25.4*ac%

Mmm=25.4*M %

Tmm=25.4*T %

Emm=25.4*E %

Chordxymm=25.4*Chordxy %

yzmm=25.4*yz

Wmm=25.4*W %

Vmm=25.4*V %

Fmm=25.4*F %

Hmm=25.4*H %

Smm=25.4*S %

ODmm=25.4*OD

%EOF
```

Appendix E

Solid Models

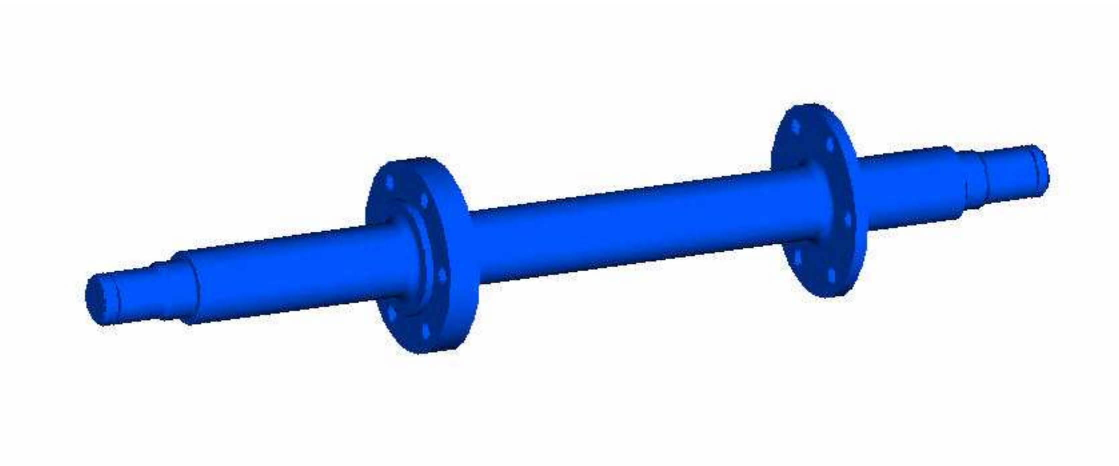

Figure E.1: Current SAE car centre shaft solid model.

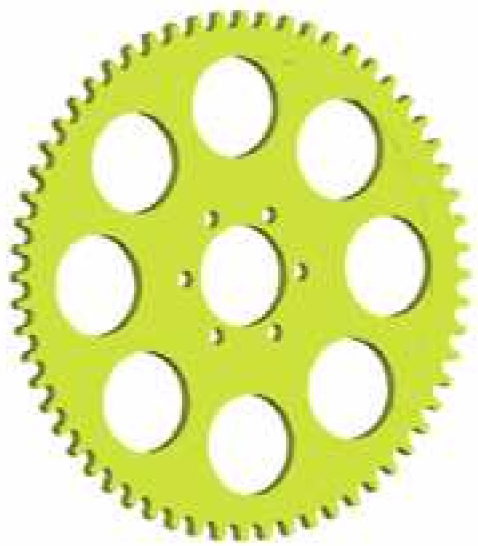

Figure E.2: Rear sprocket solid model.

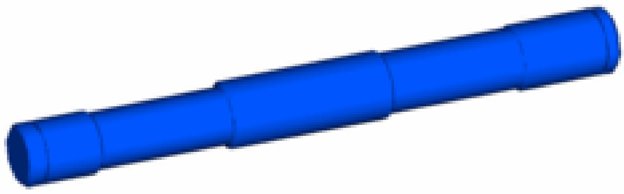

Figure E.3: Shortened half shaft solid model.

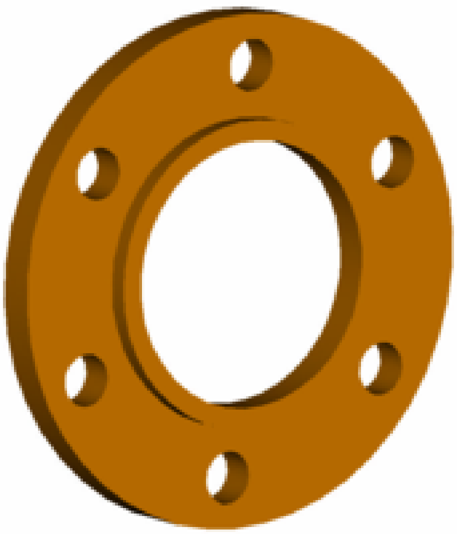

Figure E.4: Sprocket spacer solid model.

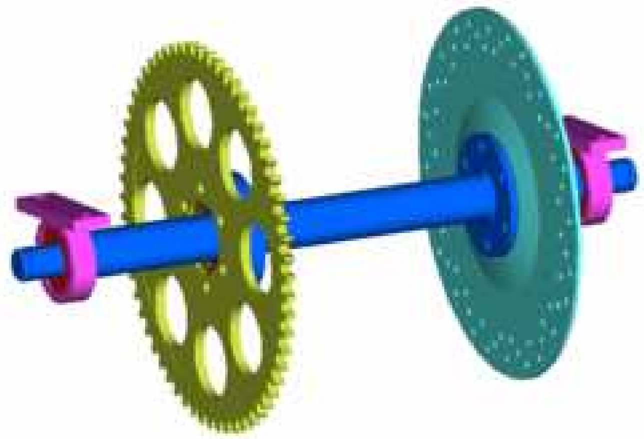

Figure E.5: Solid model of main centre shaft assembly.

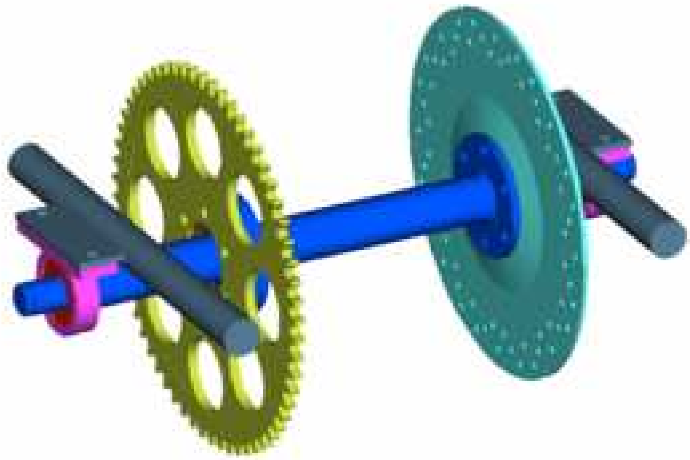

Figure E.6: Current drivetain assembly with spaceframe positioning.

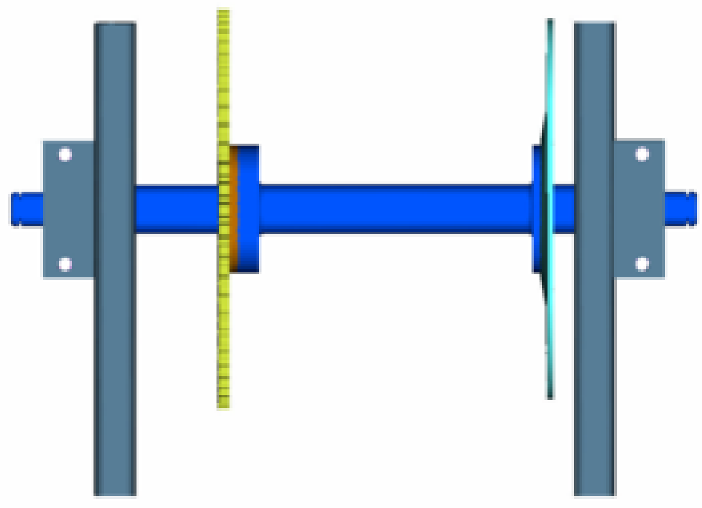

Figure E.7: Top view showing the current mounting position of the centre shaft assembly in the SAE car.

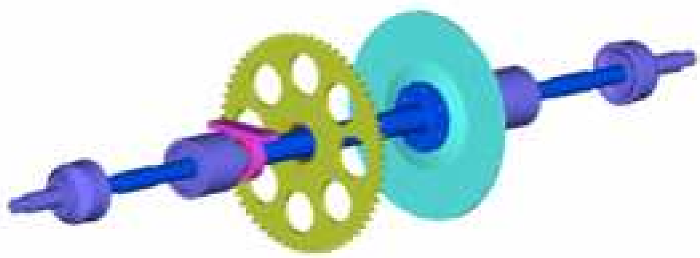

Figure E.8: Assembly from lower CV joints to the centre shaft assembly.

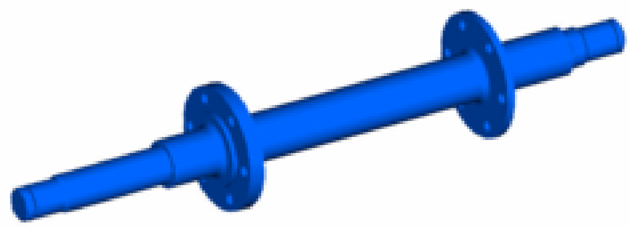

Figure E.9: Solid model of modified centre shaft.

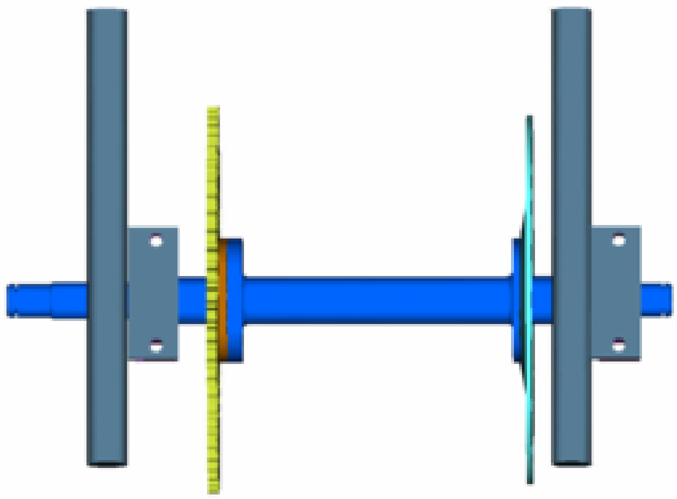

Figure E.10: Top view showing the proposed mounting position of the centre shaft assembly in the SAE car.

Appendix F

Detailed Drawings

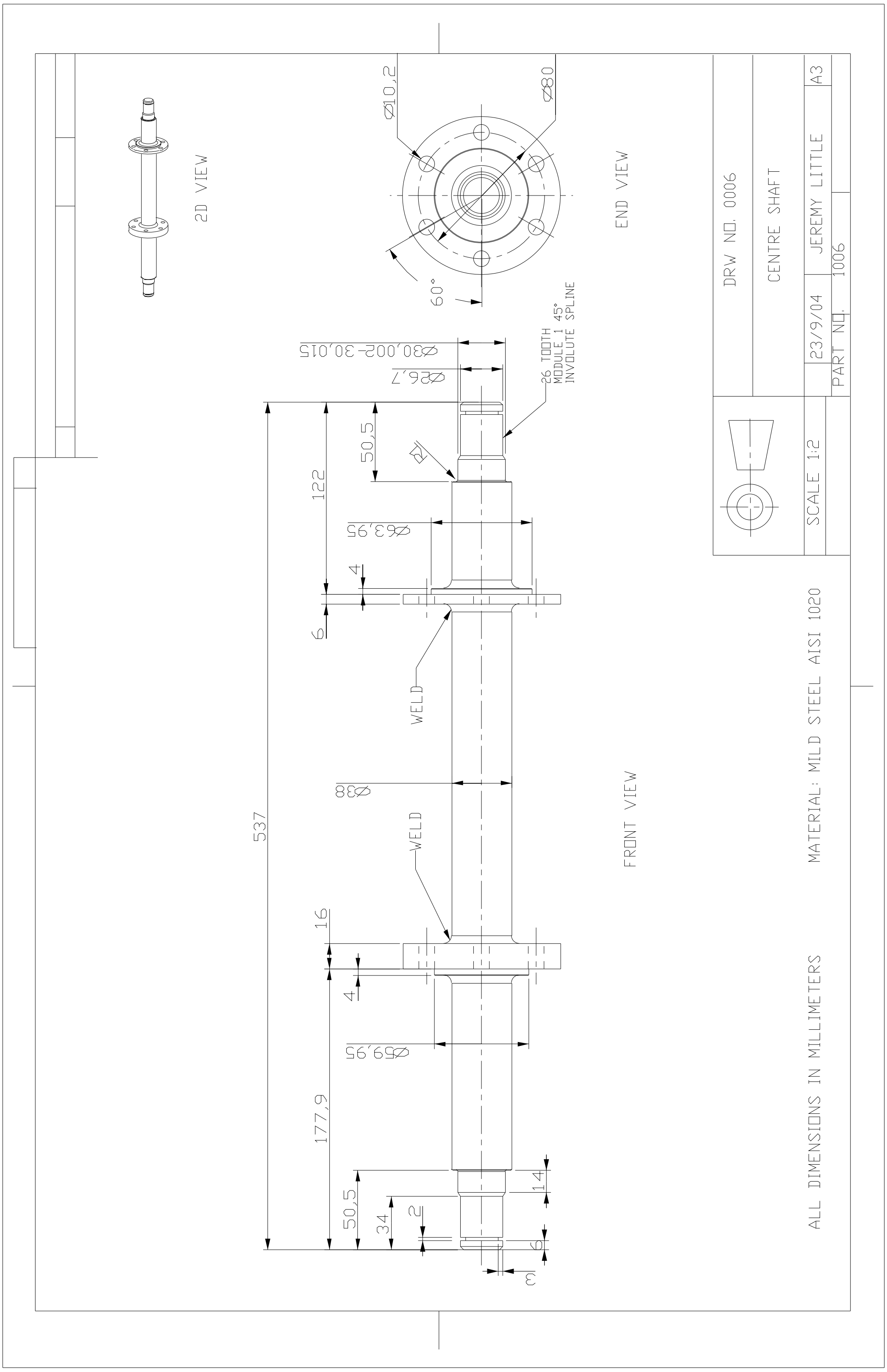

Figure F.1: Centre Shaft.

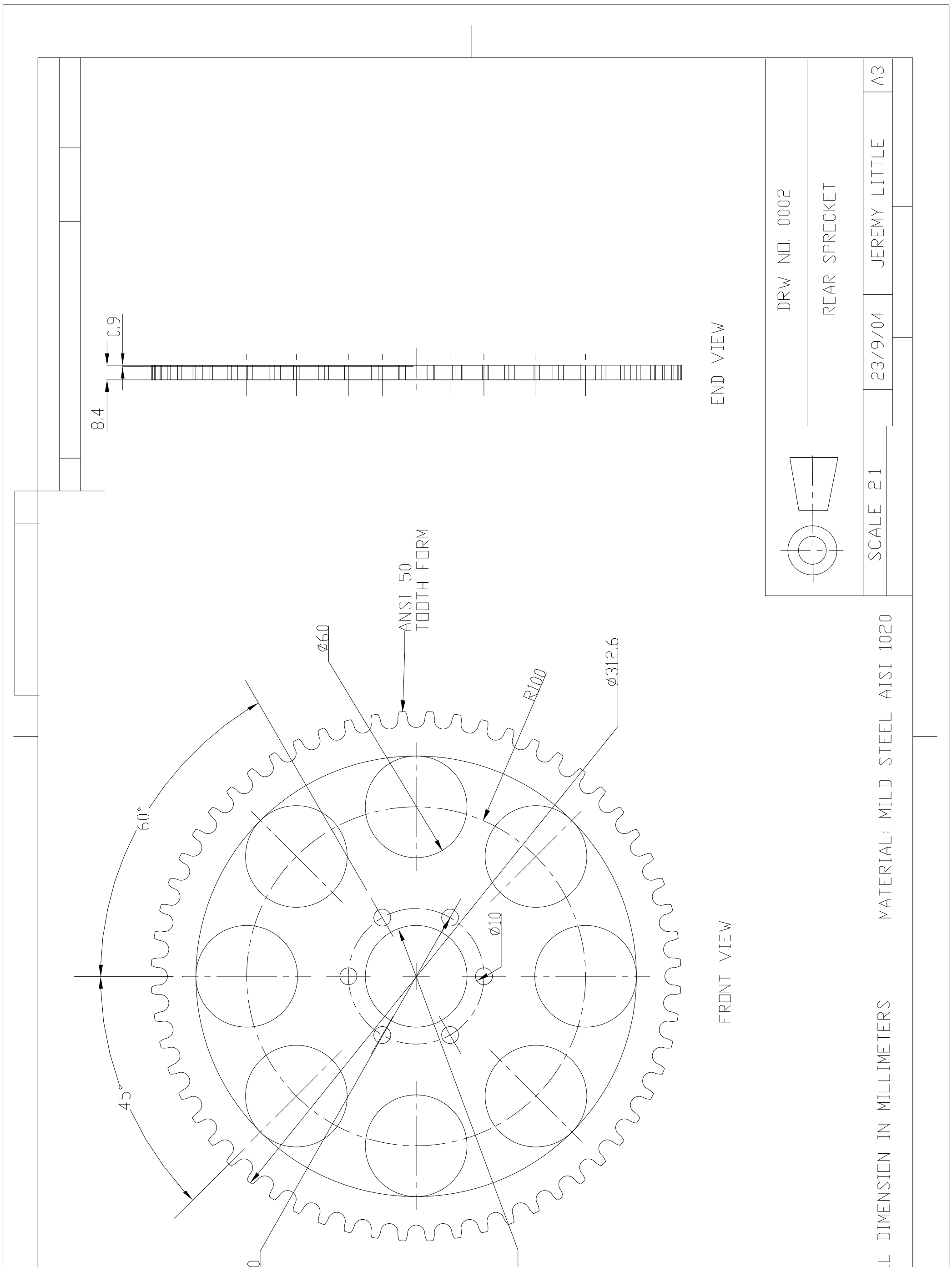

Figure F.2: Sprocket.

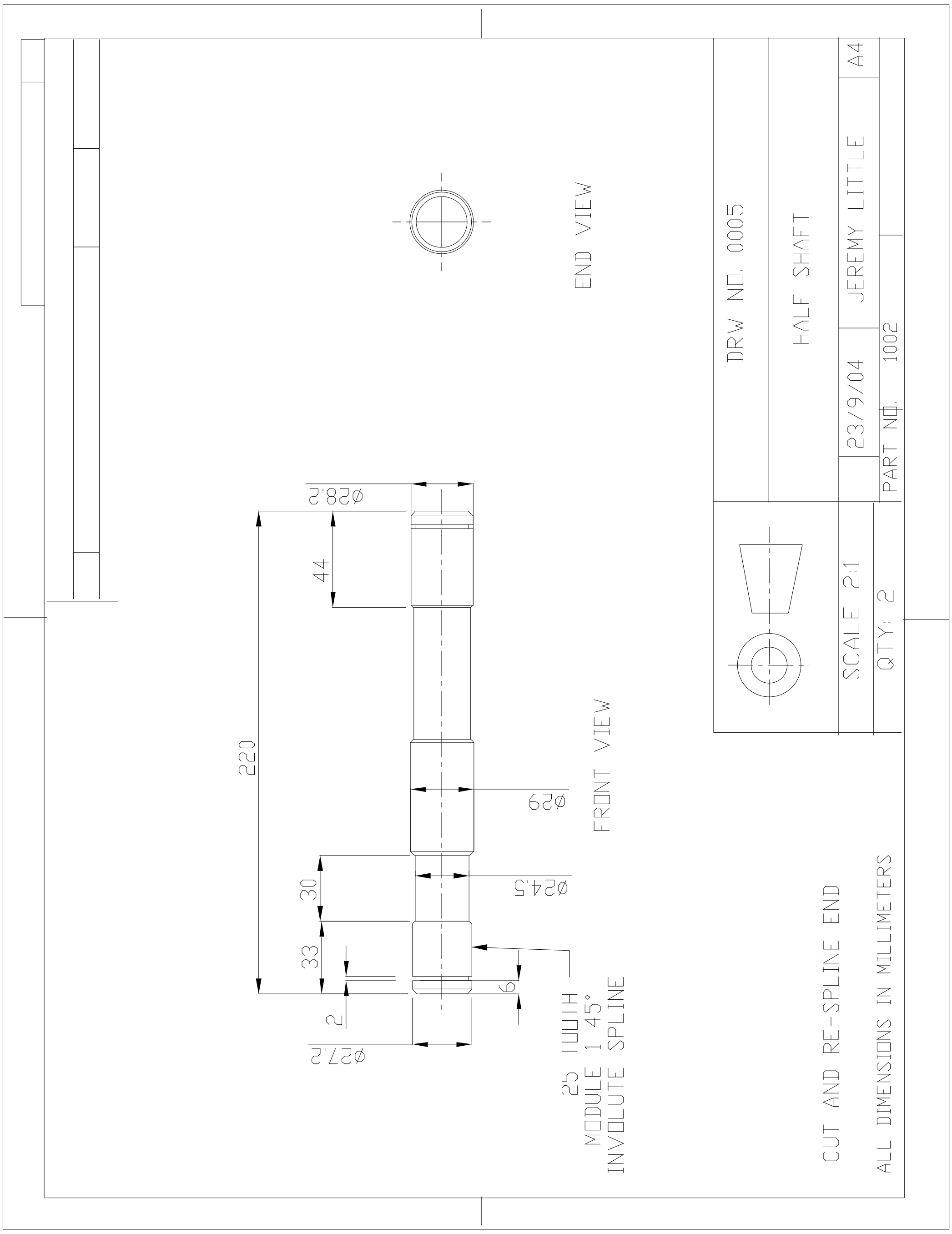

Figure F.3: Half Shafts.

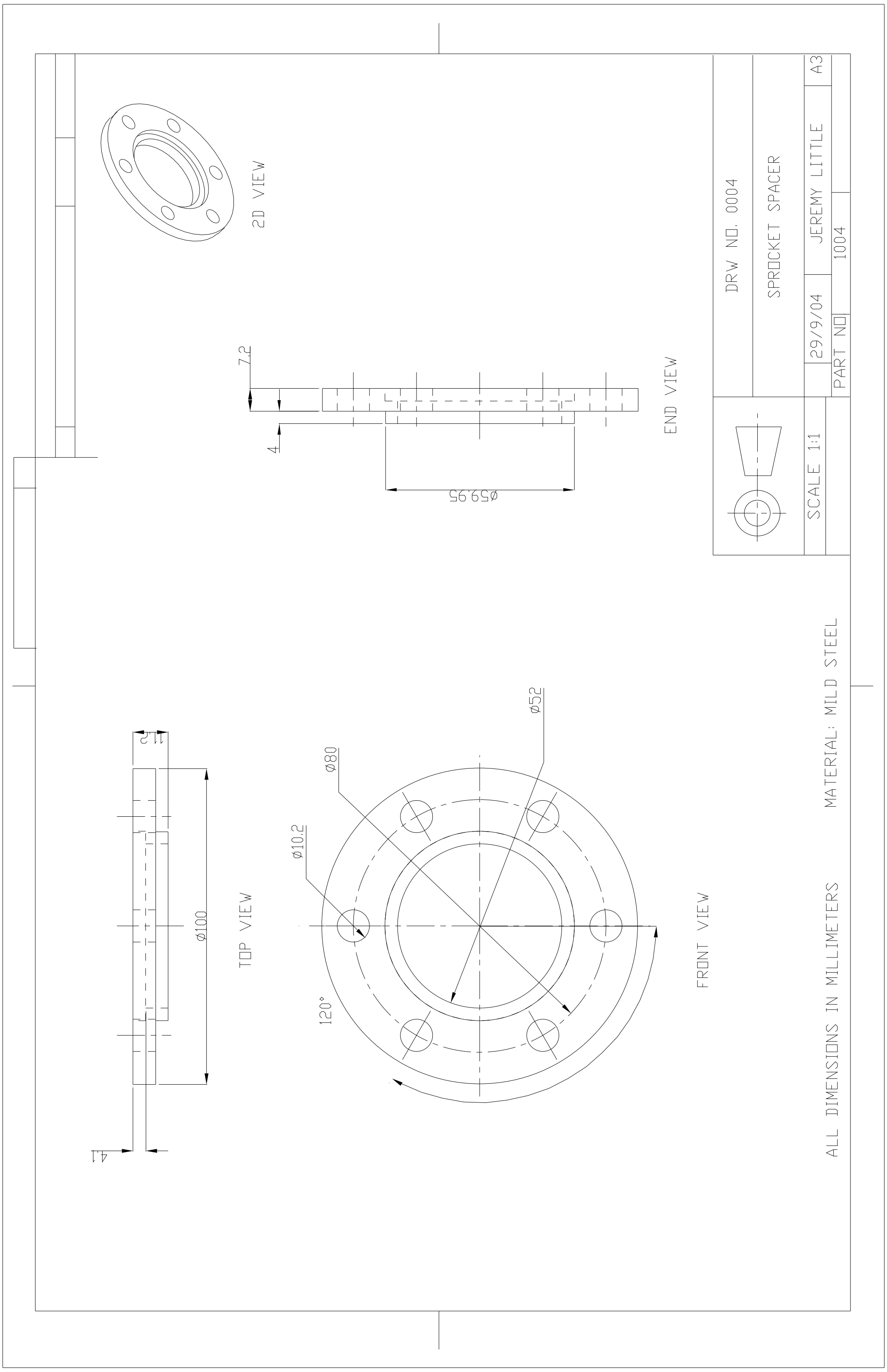

Figure F.4: Sprocket Spacer.

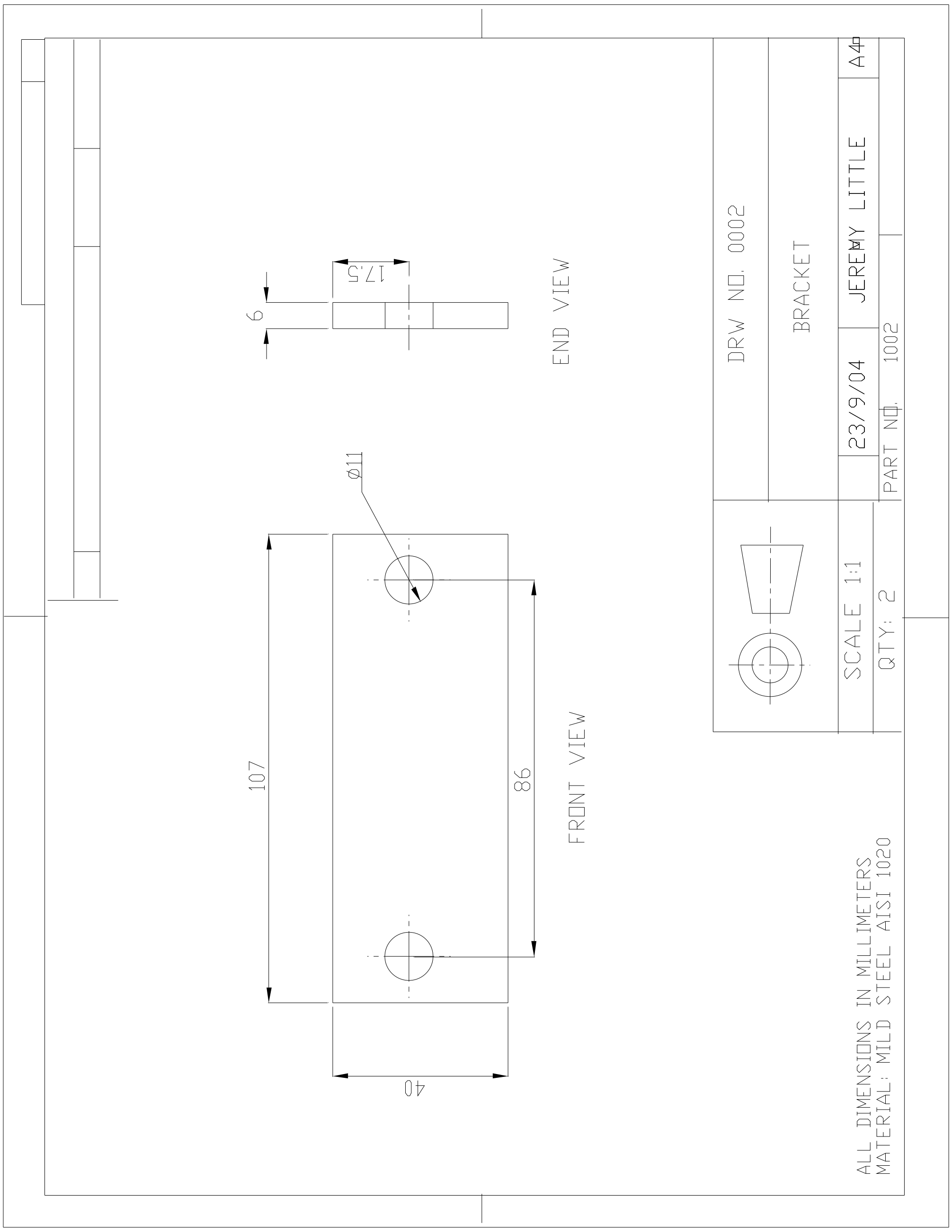

Figure F.5: Mounting Brackets.

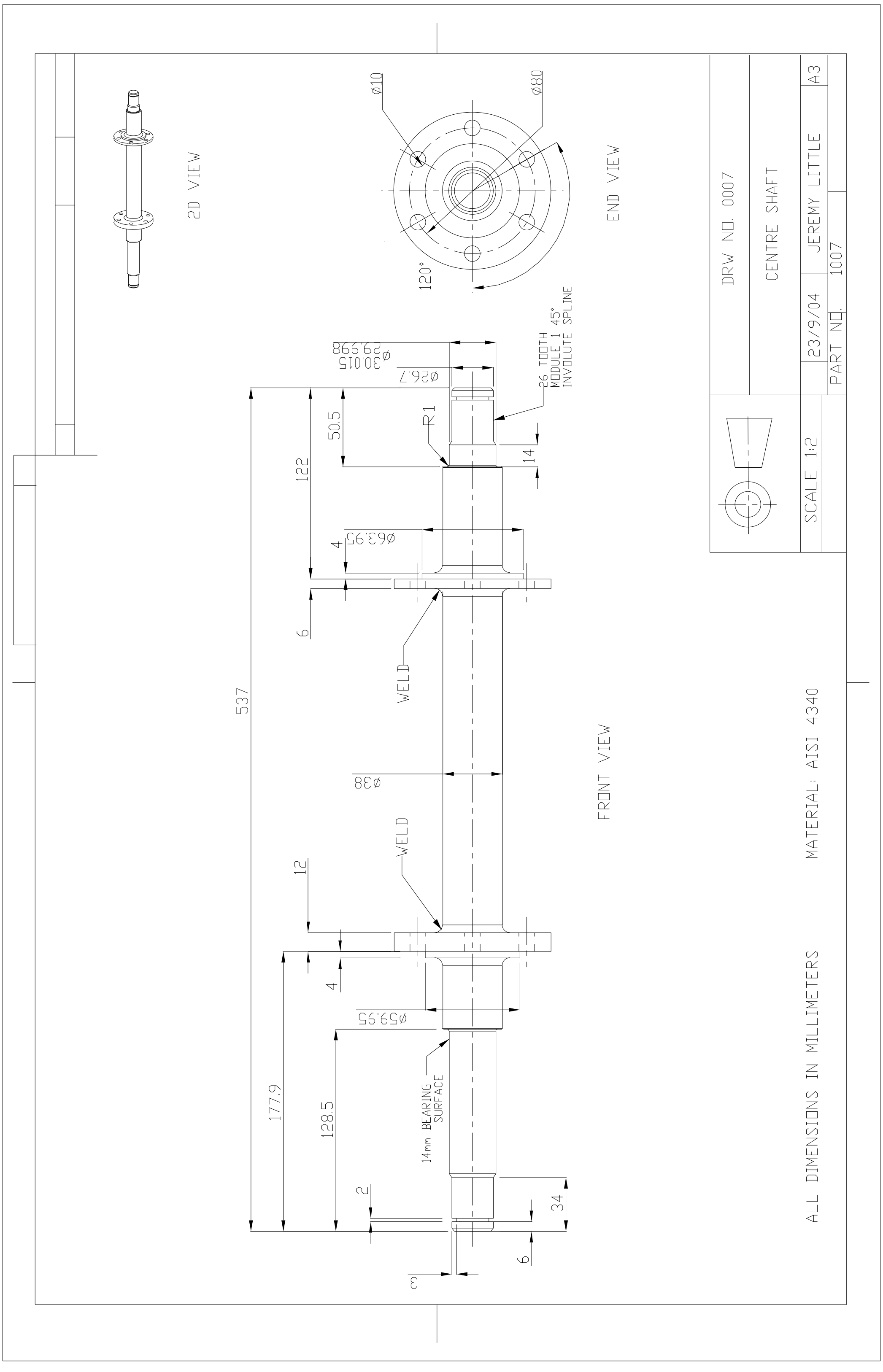

Figure F.6: New Centre Shaft.