University of Southern Queensland
Faculty of Engineering and Surveying

Development of a Drivetrain System for a Formula SAE-A Race car

A dissertation submitted by

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In fulfilment of the requirements of

ENG4111 and 4112 Research Project

towards the degree of

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Abstract

This project encapsulates one of many areas that make up the USQ Motorsport vehicle for entry into its second Formula SAE-A competition. The aim of this project was to undertake the design and development of the drivetrain system, including brakes and wheels, which would optimise performance and reliability in the competition.

To begin the project many different drivetrain systems and components were researched to gain the knowledge and understanding required to select an appropriate system. Investigation into the drivetrain in the 2004 vehicle was also conducted along with the review of the competition rules and regulations. The development of the vehicle is a team project and therefore required good communication and cooperation among team members to design a successfully competitive racecar.

The most viable, best drivetrain systems were analysed for comparison. A differential was considered to the optimum option to implement into the car for future years when resources are available. The solid rear axle design chosen for this year's car was critically analysed for stress and fatigue. The remaining drivetrain components including sprockets, bearings, CV assembly and wheel hubs were all sourced and designed to complete the assembly.

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Limitations of Use

Certification

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

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

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Signature

Date

Acknowledgements

I would like to acknowledge and thank my supervisor Chris Snook and also Derek Mulder for all their guidance and assistance throughout the course of my project.

As a member of the USQ Motorsport team I would like to thank all my fellow team members involved in this years challenge:- Ray Malpress, Selvan Pather, Melinda Plank, Tony O'Neill, Cristina Poppa, Rueben Molloy, Scott Coombes, Bruce Llewellyn, Kevin Gilham, Scott Corsan, Dos, and anyone else who has helped the team to compete in this years Formula SAE-A competition.

I would like to thank the USQ workshop staff Chris Galligan and Brian Aston for their patience and assistance in manufacturing the car.

Finally I appreciate any support given by the team sponsors who helped make this project possible.

Matthew Harber

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Chapter 1

Introduction

1.1 Introduction

Last year a team of engineering students from University of Southern Queensland (USQ) entered a competition organised by the Society of Automotive Engineers (SAE) for the first time. This year a new team of students are continuing the challenge and hoping to improve on the efforts of their fellow. This competition allows a team of young engineers to design and manufacture a formula-style racing car and to compete against other universities in a number of events. As a member of the 2005 USQ Motorsport team it is my responsibility to develop the drivetrain system, with the addition of wheels, tyres and the braking system.

To begin this project investigation into the Formula SAE-A competition rules and regulations was required. The competition has strict guidelines on the safety features of the cars and other limitations to restrict the power and cost in order to challenge the imagination and ingenuity of competitors. Following this a great deal of research was done into the many different systems available to be incorporated into the drivetrain of this style of racecar. This research will not only lead to my design decisions for this years car, but should also become a foundation for future students involved in developing a race car for USQ Motorsport in future competitions.

The task of designing and manufacturing a competitive vehicle requires a team of students each responsible for specific areas of the car. As a result of this the project requires great teamwork and communication to be completed successfully, particularly in the time permitted. The team is also responsible for fundraising and sponsorship efforts in order to gain enough finances to allow the purchase of all necessary components.

1.2 Project Aim

My project aim is to undertake the design and development of a drivetrain system for a Formula SAE racing car that will optimise performance and reliability whilst minimising cost and weight. Wheels, tyres and the braking system have also been included with the drivetrain. The design must meet all Formula SAE-A rules and regulations including all safety features.

1.3 Project Objectives

The objectives of this project are to review the Formula SAE-A competition rules and scope of the Formula SAE-A competition. Then research literature relating to differentials and related drivetrain systems to gain solid knowledge and understanding of the concepts. This includes comparing and evaluating the information concerning differential performances and specifications. Select an appropriate system for use in the 2005 USQ FSAE-A challenge. Design an appropriate drive system to carry the power of the engine to the differential considering an appropriate final drive ratio. Evaluate the design of wheel axles, CV assembly and the braking system on the 2004 USQ car and redevelop to improve performance. Complete detailed drawings of the entire system and position in the car. Liaise with team members, sponsors and Faculty workshop staff in the acquisition, manufacture and assembly of the drivetrain system. Complete Design and Cost Report documentation associated with the drivetrain. Finally test the performance characteristics of the improved drivetrain design.

1.4 Overview of Dissertation

This dissertation is structured as follows:

Chapter 2 Provides some foundation knowledge to understand drivetrain systems, braking systems and wheels and tyres.

Chapter 3 Describes the selection and development process of the drivetrain for the Formula SAE car.

Chapter 4 Details the centre shaft design and analysis.

Chapter 5 Covers the differential designs considered and recommendations for future implementation.

Chapter 6 The selection and analysis of the associated drivetrain components.

Chapter 7 Details the selection of the wheels, tyres and braking system along with the analysis of a braking system for possible future use.

Chapter 8 Concludes the dissertation and suggests further work.

Appendices All of the relevant SAE regulations, detailed drawings and solid model of drivetrain components, gearing tables, cost report and project specification.

1.5 Conclusion

As a member of the USQ Motorsport team entry in the 2005 Formula SAE-A competition, I am responsible for the design and development of all drivetrain components, the braking system, wheels and tyres. The aim is to develop an appropriate system to ensure a competitive racing vehicle is achieved in the specified time. The following chapter provides information regarding the competition and the necessary knowledge for understanding my areas of design.

Chapter 2

Background

2.1 SAE Competition

To develop a drivetrain system for a formula SAE racecar one must first understand what formula SAE racing is and what the formula SAE-A competition is all about. To begin with the formula SAE competition was created so students could experience an interesting and exciting engineering project before completing their studies and entering the world as a qualified engineer. The cars are built with a team effort over a period of about one year and are taken to a host institution for judging and comparison with other competitors. The Society of Automotive Engineers developed this idea in the USA and has more recently spread to Australia, Japan and Europe.

For the purpose of the competition, the scenario given to the teams was that a manufacturing firm has engaged them to produce a prototype car intended for the non-professional weekend autocross racer. Therefore it is very important the car has high performance in terms of its acceleration, braking, and handling qualities. The car must be low in cost, easy to maintain, and reliable. In addition, the car' s marketability is enhanced by other factors such as aesthetics, comfort and use of common parts. The challenge to the design team is to design and fabricate a prototype car that best meets these goals and intents. Each design will be compared and judged with other competing designs to determine the best overall car.
The cars are judged in a series of static and dynamic events. The static events begin with a technical inspection and then focus on the engineering design and cost effectiveness of the car. The cars then compete in dynamic events such as solo

performance trials and high performance track endurance. The maximum available score in each event is listed below.

Static Events	Points
Presentation	75
Engineering Design	150
Cost Analysis	100
Dynamic Events	
Acceleration	75
Skid-Pad Event	50
Autocross Event	150
Fuel Economy Event	50
Endurance Track Event	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. Evaluation of a team's ability to present their prototype car to a manufacturing company will be evaluated in the presentation event. Finally the design event evaluates the engineering skills, innovation and effort to meet the requirements of the competition as well as rationale behind their 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 in comparison to the competition. The first of these dynamic events is an acceleration event, timing the cars to travel in a straight line over a short distance, then a skid-pad event is held to evaluate the cars cornering ability by making constant radius turns at speed. To assess the cars maneuverability 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. The final test will be the endurance and fuel economy event where the vehicles reliability and fuel economy are tested to evaluate the overall performance of the car.

2.2 Drivetrain

To develop a drivetrain system for a Formula SAE racecar one must also understand what components make up this system. The drivetrain includes a gearbox, a power transmission or transfer system i.e. chain and sprocket or drive shaft, a differential, axles, CV joints, wheels and tyres and finally the braking system. Each of these components will be briefly described for later evaluation in the selection and design of the drivetrain. The Formula SAE rules outline specific safety restrictions that must be incorporated into the design of the drivetrain.

2.2.1 SAE Constraints

The following Formula SAE rules are those relevant or directly related to the drivetrain system. The drivetrain must have:

- Four wheels not in a straight line
- Any transmission and drivetrain can be implemented
- All exposed high speed components must be guarded by scatter shields
- Only tyres touching the ground

2.2.2 Transmission

A transmission or gearbox is used to vary the gearing ratios of a drivetrain during operation, to optimise a vehicles performance. An engine produces maximum power and torque at a specific rev range; therefore by utilising a number of gears it allows a vehicle to maintain the optimal engine revolutions whilst travelling at different speeds. The three configurations of transmissions available are manual, automatic and variable.

A manual transmission allows the operator to manually select the desired gear ratio by moving a shift lever. This directly connects the engine to the differential through a set of gears. For the selected gear to engage through dogteeth the spinning shaft and collar must be rotating at the same speed as the gear. This is achieved through the use of synchronisers. Synchronisers, also known as a synchromesh unit, act as a brake or clutch by making contact before the teeth to synchronise their speed through a frictional surface. This enables the teeth to be meshed easily and without noise or damage. Without synchronisers the operator would have to vary the engine revs between gear changes to allow the teeth to engage. This is known as double clutching. An important feature required for a manual transmission is a clutch.

A clutch is used in a manual transmission to engage and disengage the engines output shaft to the transmission. The purpose for this is that an engine must be started and running at speed before a load can be applied. A clutch also allows for low or reverse gear to be selected when the vehicle is stationary and the load to be applied to the engine gradually. The clutch gradually transfers the engines torque to the drivetrain by increasing the friction between surfaces until they spin at the same speed.

As the name suggests an automatic transmission selects the gears automatically without the activation by an operator. This gives the driver the advantage of not having to worry about gear changes and can then pay more attention on driving. Gear changes are made by a number of inputs such as vehicle speed, throttle position and manual selection, which activates clutches and band brakes through hydraulic circuits. These act on a single planetary gear set, which is capable of producing a number of gear ratios. The automatic transmission does not require a manually activated clutch like the manual transmission, but instead uses a torque converter to gradually transfer the engines torque automatically. A torque converter is a fluid coupling that transfers the engines torque to the transmission without the operator's activation. It does this by pushing oil through a turbine connected to the auto, using the centrifugal motion of the fluid. Whilst it does not require any manual operation, the fluid coupling is not as efficient as the manual clutch because the automatic transmission can never reach the engine speed.

Continuously Variable Transmissions (CVT's) are a transmission with an infinite number of gear ratios. The CVT does not actually use any gears but rather achieves this by changing the radius of the drive and driven pulleys. Variable-diameter pulleys must always come in pairs to ensure the belt remains tight. The pulleys are effectively two cones facing each other with the ability to move in and away from each other. A V-belt rides in the groove between the two cones and as the cones move in or out, the belt rides higher or lower in the groove effectively varying the diameter and as a result the drive ratio. The advantages to this design are that the engine will remain in the optimum power range regardless of speed and acceleration, thereby optimizing efficiency and acceleration. There is also no need for a clutch because the belt is loose at idle. Only recently are CVT's beginning to be accepted for use in motor vehicles. This is because of the advancement in durability and the amount of torque they can handle, combined with the reduction in cost.

2.2.3 Power Transfer Systems

A power transfer system must be capable of transferring the engines torque from the output of the transmission to the input of a differential. In the majority of motor vehicles this is done with a rotating drive shaft, connected to the transmission and differential by universal joints. However there are a number of different methods that can be used to effectively transfer this power. Other methods include a chain and sprocket, belt drive and gear meshing.

2.2.3.1 Chain Drive

A chain drive transmits power between shafts by connecting them through interlocking a chain over sprockets. Different types of chains include single and multi-strand roller chains and the inverted-tooth chain. Chain drive systems are in common use pushbikes, motorcycles and engine timing. The main advantage of a chain drive is not being limited by the distance between the drive and driven shafts. However some limitations include only being capable of transmitting power in one plane and alignment is critical for its proper operation. Also this system requires regular maintenance and inspection for chain and sprocket wear. The difference in the diameter and number of teeth on the sprockets are what determines the gearing ratio. A chain tensioner or idler sprocket is used to accommodate and adjust for small changes in chain tension due to a change of sprocket size or elongation of the chain due to wear.

2.2.3.2 Belt Drive

A belt drive system connects the two shafts through a tensioned belt wrapped around two pulleys. This has similar limitations to the chain drive where the pulleys must be aligned in the same plane and requires an adjustable belt tensioner. This is important because the belt tension creates the required friction between the belt and the pulleys; consequently slippage can be another disadvantage to this system. There are wide ranges of different belts available including a toothed belt that is similar to the chain drive and has eliminated the problem of slippage.

2.2.4 Reduction box

A reduction box is a system used in drivetrain systems to significantly change the gear ratio between the input and output shafts. It consists of a set of gears inside a compact housing that rotate along parallel axis. The housing supports the rotating gears with small bearings either side, whilst keeping them immersed in oil to reduce the wear and operating temperature of each gear.

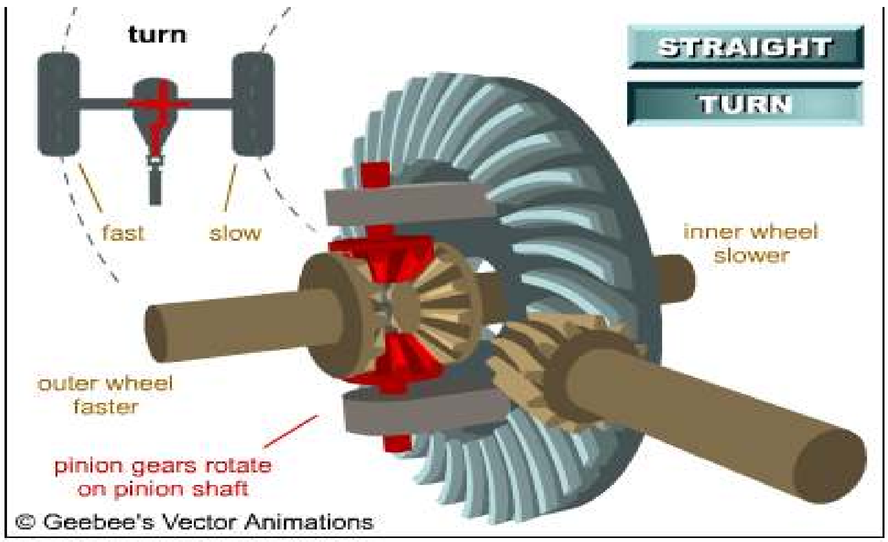
Figure 2.1: Gear reduction box

2.2.5 Differentials

The function of a differential is to transfer torque to both rear drive axles whilst also allowing them to spin at different speeds. The name came from its ability to allow both wheels to rotate at different speeds or to differentiate. The need for this differential action is because as a car turns a corner, the outside wheel must travel along a larger radius than the inside wheel over the same period of time. Velocity is given by distance / time therefore around every corner the outside wheel must travel faster than the inside wheel. This affect on wheel differentiation increases as the radius of turn decreases.

If the rear wheels were unable to differentiate as in the case of a solid rear axle or locked differential, than the outside wheel would force the inside wheel to rotate faster. This additional force to the inside wheel can have two negative affects. Either the tyre will break traction and slip or the tyre will hold traction and act against the turning motion of the car creating understeer. There are two kinds of differential available, an open and limited-slip designs, which will be described in more detail.

The open differential is the most common diff used in motor vehicles today. Figure 2.2 below clearly illustrates the gearing components that form an open differential. As the large gear rotates the pinions (red) drive both axles whilst allowing free differentiation. Therefore a noted characteristic of the diff is that will always share the torque equally between the axles regardless of rotational speed. The major disadvantage with this diff is that because it offers no resistance to differentiation, the drive force is limited to the tyre with the least amount of traction. This limitation led to the designs of limited slip differentials.



Animation courtesy Geebee's Vector Animations

Figure 2.2: Open Differential

A limited slip differential (LSD) is designed to transfer more torque to the wheel with the most traction. There are a variety of LSD's available that will allow normal differential operation when cornering but when slippage occurs more torque is transferred to the non-slip wheel. A few different types of LSD are the clutch-type, the torsen, a viscous coupling and a locking differential.

The clutch-type LSD is essentially an open differential with the addition of a friction clutch pack. It uses the same gearing components with either a cone or preloaded spring clutch pack. The clutch holds the two side gears together and differentiation can only occur when the friction of the clutch is overcome. The advantage of this over an open differential is that if one wheel completely loses traction, the other wheel with traction is supplied with a drive force equal to the force to overpower the clutch. In comparison the wheel with traction in an open diff would not receive any torque if the other wheel has no traction whatsoever. Furthermore by altering the friction can set the amount of force required to overpower the clutch.

Figure 2.3: Clutch-Type Limited Slip

A Torsen meaning 'Torque Sensing' LSD is a less common type usually found in small four-wheel drive cars. As seen in the figure below the drive axles are joined by a helical gearing arrangement. This gear design creates a torque bias ratio between the drive axles that allows the wheel with the most traction to be supplied with the majority of torque. For example a torque bias ratio of 4:1 means that the torsen can deliver to the tyre with the most traction, 4 times the torque that can be supported by the lower traction tyre. Therefore the torsen can provide much more traction force under such conditions than the open differential.

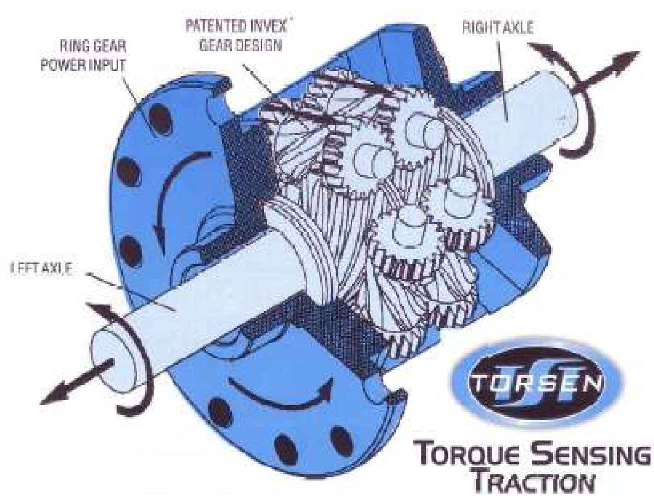


Figure 2.4: Torsen Differential

A viscous coupling is again not very common but can be found in some all-wheel vehicles to transfer torque between the front and rear set of wheels. This LSD contains two sets of plates inside a sealed housing that is filled with a thick or viscous fluid. One set of plates is connected to each output shaft, and under normal conditions both the plates and fluid will all rotate together at the same speed. However when there is a difference in velocity between shafts i.e. when one set of wheels is losing traction, than the set of plates corresponding to those wheels will also rotate faster. This causes the viscous fluid between the plates to speed up which then in turn force the other set of plates to rotate faster. This effectively transfers torque from the slipping wheels to the non-slip wheels. The disadvantage with this differential is that it cannot transfer torque until the wheels are slipping to produce a large enough difference in rotation between shafts.

A locking differential, also known as a spool, acts as though the drive shafts are connected together and does not allow any wheel differentiation. However, this locking action can be manually activated so that the operator can choose when this is to be engaged. Disengaged the differential will simply act as an open differential. Activating mechanisms can include electric, hydraulic or pneumatic systems to lock the side gears together.

2.2.6 Drive shaft joints

There are two commonly used joints throughout the drivetrain of a vehicle. These are the universal joint and the CV joint. The purpose of both joints is to allow for a change in angle of drive and small axial displacements between drive shafts. This is most common in front wheel drive cars where the drive axle comes directly out of the gearbox along a fixed axis, and the wheels must be permitted to turn when steering hence changing the angle of the drive shaft. It is also common in vehicles with independent suspension where the wheels and suspension travel up and down whilst the drive of the car is fixed to the chassis and remains stationary.

2.2.6.1 Universal joint

The most common universal to be used in a vehicle is the cross-and-yoke joint, or Hooke's join t. It consists of two yokes and a cross, with needle roller bearings carried in cups between the surfaces of the yoke and cross. A yoke is attached to both the driving and driven shafts, and connected by the cross that allows the shafts to operate at different angles. When a change in angle occurs between the shafts, the joint will impart rotation at a varying velocity, due to geometry of the joint. The rotation of the driven shaft momentarily accelerates and decelerates as the universal joint rotates. The greater the angle of drive, the greater this effect will occur. However a solution to this problem is to incorporate two universal joints into the drive system, connected by an intermediate shaft. The second joint will cause fluctuations in the rotational speed opposing that of the first joint, thus cancelling this effect out.

Figure 2.5: Universal Joints

2.2.6.2 Constant Velocity (CV) joint

There are three kinds of CV joint available. The first is the Birfield joint, typically used as the outer joint on the drive shaft, next is the Double-offset joint and the Tripod joint, both typically used as the inner joint.
The Birfield joint consists of a housing, or outer ring, an inner ring, and a caged set of balls. The housing has a short splined shaft that fits into the wheel hub, and the inner ring has an internal spline to accept the spline on the drive shaft. The balls are retained by the cage but also fit into axial grooves in the rings. This allows torque to be transmitted from the housing, through the balls, to the inner ring. These grooves in the rings only permit the balls to roll in the axial direction, which enables the joint to allow for a change in angle of the drive shaft.

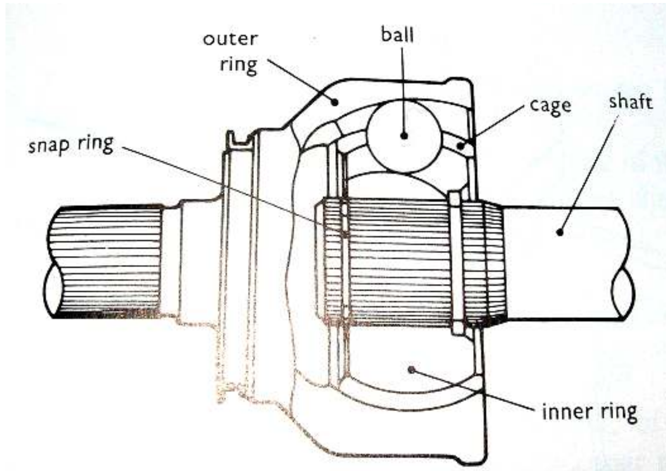


Figure 2.6: Birfield CV joint

A double-offset joint is very similar to the Birfield joint, except that the housing or outer ring is longer. This gives longer grooves for the balls and allows more axial movement to compensate for changes in shaft length. This is important because the larger the angle in the shaft, the greater the distance becomes between the joints.

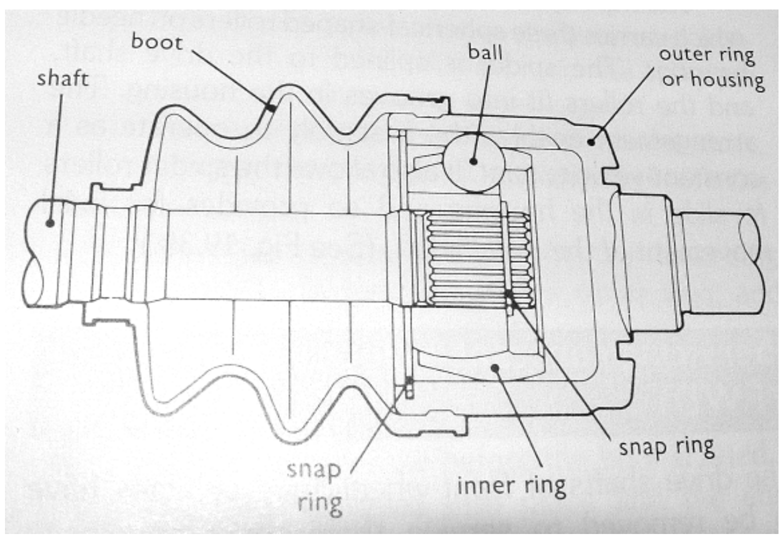


Figure 2.7: Double-offset CV joint

Tripod joints operate in a very similar manner to the Double-offset joint, except a spider carrying three spherical shaped rollers on needle bearings, has replaced the caged set of balls. The spider acts as the inner cage and is splined to fit the drive shaft, and the rollers fit the grooves in the housing.

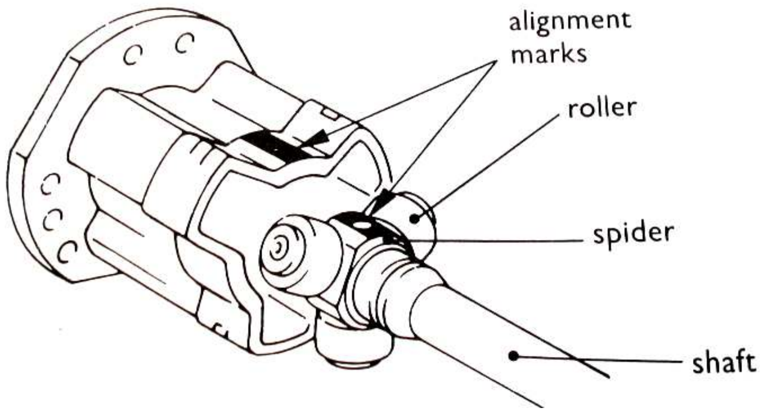


Figure 2.8: Tripod CV joint

2.3 Wheels and Tyres

Tyre selection is arguably the most important decision concerning the performance of a vehicle. The contact patches of the tyres are the single interface connecting the racing car to the surface of the track. Therefore all acceleration, braking and cornering forces performed by the car must be transferred through the tyres. Providing your engine has sufficient power then the traction between the tyres and the racetrack will become the limiting factor on potential performance.

2.3.1 SAE Constraints

The following Formula SAE rules are those relevant or directly related to the wheels and tyres. They include:

- Wheels must be 203.2mm (8.0 inches) or more in diameter
- Single retaining wheels nuts must have a device to retain nut
- Tyres are only to be cut or treaded by the manufacturer
- Dry tyres may be slicks or treaded
- Rain tyres must have a minimum tread depth of 2.4mm

2.3.2 Wheel and Tyre Selection

Before selecting a tyre one must understand how a tyre behaves and the effect of different variables such as the diameter, width, aspect ratio, stiffness, compound, tread and also tyre pressures.

Tyres perform better when a larger area of rubber is in contact with the road. One way to increase this area is to use a tyre with less void area. The void area is all the area not in contact with the ground, such as the tread cut into all road tyres. This is why most forms of motorsport use a slick tyre. Using a smooth flat tyre with no tread means no void area and optimises the contact area with the road, enhancing traction in dry conditions. However these grooves are required for traction in the wet. The grooves allow for water to disperse out from underneath the tyre maintaining the contact of rubber with the road. Without the treaded grooves the tyre would 'aquaplane' over water lying on the road, meaning a film of water would remain between the tyre and the road surface, diminishing contact with the road surface and losing traction and control over the vehicle. Therefore for safety and performance in wet conditions a treaded tyre with some void area for dispersing water is required.

Tyre Pressures are also extremely important when considering tyre performance. Lower tyre pressures will increase the contact area with the road but decreases the tyre's sidewall strength. Sidewall strength is extremely important for cornering to avoid the tyre deforming. When a sidewall deforms, the tread shifts out from under the wheel and the tyre begins to roll onto its side, not just losing grip but also the reaction time of steering inputs. At lower tyre pressures there is less reinforcement to stretch and support the sidewall. Therefore it is more important to maintain sidewall strength than it is to decrease the tyre pressures for a little extra contact area.

Another way to support the sidewall and prevent it from deforming is to increase the width of the wheel. By using a rim wider than the tyre tread, the lateral pressure on the tyre can be more directly transmitted to the wheel. The sidewall can than brace itself against the rigid wheel, minimising deformation. Sidewall deformation robs traction, overheats tyres and causes uneven wear.

It is often argued that wider tyres are better because they increase the area of rubber contacting the track surface. Whilst wider tyres are better, it is not because of the increased width and area of the contact patch, but rather the change in shape of the contact patch.

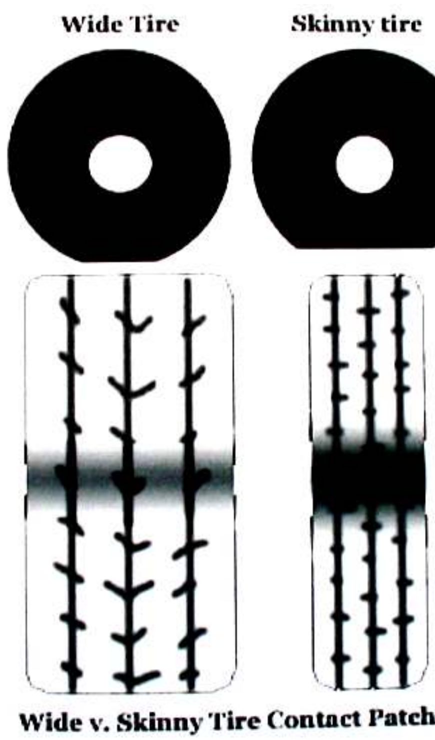


Figure 2.9: Contact patch shapes

A narrow tyre has a longer and narrower contact patch compared with a wide tyre having a short and wide contact patch. The sidewall of a tyre supports a portion of the cars weight, and because of the circular shape of a tyre, this force is greatest directly below the tyre and smallest at the front and back of the contact patch. Once a tyre is loaded with this weight the sidewall must bend a little to create the contact patch, and as seen in Figure 2.9, this happens less with a wider tyre. Instead this weight is being supported by the air pressure, which is constant over the entire contact patch. Therefore the distribution of this force evenly over the tyre's surface allows the tyre to function better, improving grip. A shorter but wider contact patch is also advantageous to the vehicles performance because it is the front and back edges of the contact patch that allows the tyres tread to deform and create slip angle.

Rotational inertia is another important consideration in high performance wheels and tyres. The greater the rotational inertia from the wheels and tyres than a greater amount of torque is required to slow or accelerate, making the car less responsive or sluggish. The further the mass of the wheels and tyres are from the axle, than the greater the rotational inertia. By using a larger diameter wheel, not only is the mass further from the axle, but also the wheel and tyre require more material for the larger surface area thereby increasing weight. Therefore by increasing or decreasing the diameter of the wheel, the rotational inertia is changing exponentially. To minimise a racing cars rotational inertia and optimise the vehicles acceleration and braking responsiveness, use lighter and smaller diameter wheels and low profile tyres. Reducing the width of the wheels will also decrease the rotational inertia however not as drastically and has adverse affects to the cornering performance as seen above.

Understeer is the term used for when the vehicles front wheels do not follow the turning circle and continues in a straight path tangent to the curve of the corner. This is also known as ‘push’ in racing because the car pushes the front wheels straight ahead and doesn’t allow them to maintain traction and turn. Conversely oversteer is a term used when the vehicles rear end does not follow the curved path of the corner and slides out, usually spinning the car. This is also known as ‘loose’ in racing terms because when the rear end loses traction it slides out away from the corner and does not tightly follow the corner.

Both of these behaviours are unfavourable because they take away from driver control and makes cornering slower since the driver has to lift the accelerator pedal to regain control. However understeer is considered more preferable than oversteer as it is easier to regain control, and less likely to cause a spin and lose valuable time.

2.4 Braking System

Braking or decelerating a car is just as important to overall performance as acceleration. Not to mention the safety aspects of using appropriate and reliable brakes, but also a car with a better braking system can be just as fast around a racetrack as a car with more torque and acceleration. It is very important that the design of the braking system be more than adequate for its given situation.

2.4.1 SAE Constraints

The following Formula SAE rules are those relevant or directly related to the braking system. The braking system must:

- Act on all 4 wheels and be operated by a single control
- Have 2 independent hydraulic circuits, each with its own fluid reserve
- Brake-by-wire systems are prohibited
- Be protected by scatter shields
- Have brake pedal over-travel switch to stop ignition and fuel pump if brake fails

2.4.2 Types of Braking Systems

Primarily there are two main types of braking systems used in cars. These are a disc brake system and a drum brake system. Drum brakes consist of a pair of stationary shoes that are forced radially outward against the inner surface of the rotating drum that is fixed to the wheel. The friction surface is the outer surface of the shoes that contact the inside of the drum and therefore slows the rotation of the wheel. Hydraulic pressure forces both the shoes outward against the drum and small tension springs pull the brake shoes back inward when the hydraulic force is released.
The disadvantages of this system are that drum brakes contain more components and are more difficult service and replace.

Figure 2.10: Drum brake assembly

The disc brake system is a far more common arrangement and can be found in use in the majority of modern vehicles. A rotating disc connected to the wheel of the vehicle is clamped by a calliper that forces a pair of brake pads against either side of the disc. This system is also powered by means of hydraulic pressure. The brake pads have a friction surface and when clamped against the disc, slow the rotation of the wheel. The advantages of this system are that the disc allows for better heat dissipation and therefore more frictional force can be applied. Also the disc brake system is a more simplistic design, and allows the brake pads to be inspected and replaced with ease.

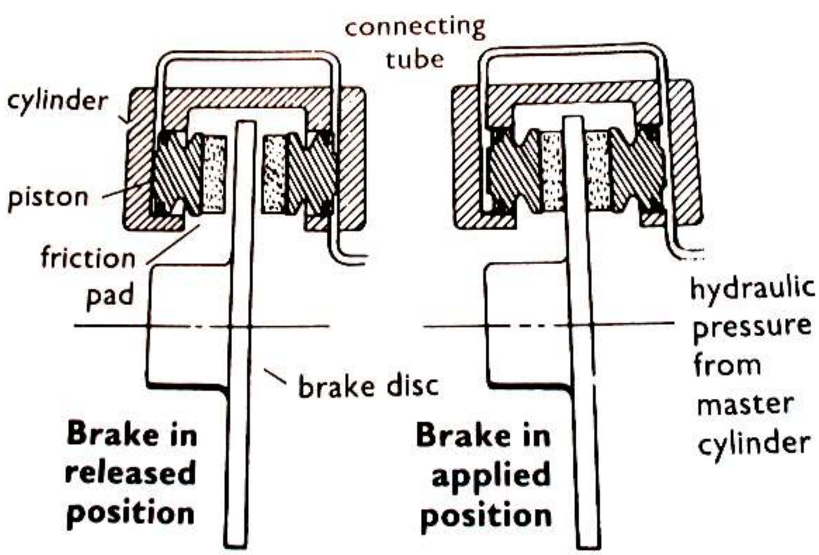


Figure 2.11: Disc brake

2.4.3 Brake Component Descriptions

Master cylinder:- is the cylinder operated by the driver in a hydraulic brake system. It contains a piston that displaces fluid to create pressure.

Calliper:- is the component that straddles the brake disc and clamps it when pressure is applied to the system. It consists of pistons that push brake pads toward either side of the rotor, which slows the rotation.

Brake pad:- has a metal back with friction material used to force against rotating components in a brake system to slow them down.

Rotor:- the brake disc in a disc brake assembly. When brake is operated the brake pads are clamped against this disc, which is attached to the wheel.

Brake fade:- is the loss of braking efficiency due to brakes overheating.

Chapter 3

Drivetrain Selection and Development

3.1 Introduction

The most significant part of this project was the selection and design of the drivetrain. To begin the process of selecting the best system for our Formula SAE-A racing car I focused on the design criteria and then produced some conceptual designs to choose from for development. Engine performance and the shafts gearing and loading information are used to perform a detailed analysis on all of the components in the drivetrain. The final decisions regarding the design and development of each drivetrain design has been summarised to show the reasoning behind the final selection for the drivetrain.

3.2 Design criteria

The design criteria for the drivetrain system consisted of the limitations and objectives set by both the Formula SAE-A competition and the USQ Motorsport team's goals. These included the Formula SAE rules and regulations, the competition's judging criter ia including cost, engineering design and performance, and the team's aims including weight, reliability and budget.
The Formula SAE rules generally set safety standards that must be accommodated into the design. These regulations do not limit the design selection of the drivetrain system however must be adhered to if to be allowed to compete in the event. The

regulations most relevant to my area have been included in Appendix B. The development of the race car is a team project and I had to liase with the other members of the team to insure the design of the drivetrain coordinated well with the rest of the car. The most significant areas of the car that the drivetrain had to adapt to were the chassis, engine and suspension. The engine determines the maximum amount of power and torque the drivetrain will have to transmit, the chassis must support all components of the car and the suspension is a critical factor in the cars performance. The rear suspension also connects and supports some drivetrain components therefore the development of both these systems required close interaction and communication.

The team's aim for this year's car was to improve the performance of the car primarily by decreasing the weight of each component where possible to improve the power to weight ratio of the vehicle. By saving a little weight in each area this will add up to make a significant difference however this had to be achieved without sacrificing reliability. This was determined to be the most cost effective way to improve the performance characteristics of the car in acceleration, braking and cornering. The cost of each design is another very important criteria that must be considered when selecting an appropriate design. Not only is cost a significant area judged at the competition but also the design must be able to be developed in accordance with the team's specific budget for the entire car.

The principal objective of this project is to develop a drivetrain system for a formula SAE racing car that will perform well in the Formula SAE-A competition held at the end of each year. To do this the car must perform well in each of the static and dynamic judging events mentioned earlier in this document. Therefore in the static events the cost and engineering design of the car must be considered and in the dynamic events the acceleration, braking and cornering performance as well as fuel economy and reliability must be considered. For the vehicle to achieve these characteristics each sub-system of the car must be designed to achieve this together, including the drivetrain.

In summary there were many important criteria to consider during the design and selection of the drivetrain system. These criteria were mainly governed by the competitions judging events and the USQ Motorsport team's goals of developing a competitive car by reducing the weight to increase performance at minimal cost.

3.3 Design Selection

The drivetrain design can be any system capable of transmitting the power from the engine through to the tyres contacting the road. The drivetrain was to be designed for a rear-wheel drive independent suspension vehicle. Thus the design options included a combination of a chain and sprocket or belt and pulley drive, a differential or solid axle and CV joints or universal joints. All of these components are described in the drivetrain background.

The first design option was the chain and sprocket drive with a solid rear axle. An example of this system can be found on racing karts. The benefits to this design are: -

- No losses through slippage
- Cheap
- Light weight
- Reliable
- Ease of maintenance and replacement
- Simple design
- Long operating life

However the disadvantages of this design are: -

- Cornering performance affected
- Tyres required to slip whilst cornering
- Chain requires tensioning
- Noise of chain and sprocket operation
- Lubrication of chain required

The second design option was a toothed-belt drive with a solid rear axle. Examples of this system are generally not found in the drivetrain of vehicles but rather used on the engine to drive the camshaft from the crankshaft. The benefits to this design are: -

- Long operating life
- Quiet operating
- Simple design
- Lightweight
- Ease of maintenance and replacement

However the disadvantages of this design are: -

- Tensioning required
- Width of belt required to transfer torque
- Tyres required to slip whilst cornering
- High cost of both belt and toothed pulleys

The third design option involves a gear reduction box to drive a solid rear axle. This design could have also been incorporated into a system with a chain drive and a differential. The benefits of this design include: -

- Reduces size of rear sprocket
- Allows optimum final drive ratio
- Allows the rear axle to moved closer to the engine, shortening wheelbase and increasing the weight distribution to the rear wheels

However the disadvantages to this design are: -

- Additional weight
- Difficulty designing housing for oil bath
- Maintenance required
- Cost of gearing

The fourth design option was a chain drive system to a Torsen differential. This limited slip differential is a popular option used in the Formula SAE competitions and a University Special Torsen differential is available to Formula SAE teams for a special discount price. Other Torsen designs are available out of a Rav4 and an Audi Quattro. The benefits to this design are: -

- Small, lightweight differential
- Increased performance characteristics
- No forced tyre slippage
- Reliability
- Long operating life

However the disadvantages to this design include: -

- Difficulty of designing housing for oil bath
- Cost
- Maintenance required
- Chain tensioning required

The final design option considered was a chain drive system to a custom or modified limited slip differential. This differential was designed by a fellow team member and acted as an open differential whilst cornering, and a locked differential when exiting a corner and accelerating. This could have been controlled via throttle use. The benefits to this design are: -

- Performance characteristics
- Design points
- No forced tyre slippage

However the disadvantages to this design are: -

- Complex to design and manufacture
- Heavy
- Unknown cost and reliability
- Chain tensioning required
- Requires actuating control

After considering the advantages and disadvantages of each system according to the design criteria, a chain drive to a solid rear axle was chosen. This was primarily based on simplicity, reliability and low cost. Although the Torsen differential appears to be the optimum option to implement into the car for increased performance, it was decided to leave this design for incorporation into cars in future years when time and financial resources are available. Whilst a solid rear axle system compromises cornering performance, the team's aim was to further develop the 2004 car for improvement and this is possible through redesign and weight reduction of the 2004 solid rear axle drivetrain system.

3.4 Drivetrain Development

The development of the drivetrain began with the selection of the chain drive to solid rear axle system. It then had to be decided how to drive the wheels from the solid rear axle with independent rear suspension. After researching similar systems two options appeared viable, CV joints or Universal joints. CV joints seemed the more common solution however the weight was a concerning factor after weighing the CV assembly used on the 2004 car out of a Ford Telstar. Therefore unless a smaller CV joint was readily available the more appealing option would have been to source and analyse a universal joint of appropriate size. However a much smaller CV assembly was sourced from a 1983 model Suzuki SS80V. This assembly only required shortening of the CV shaft to accommodate the drivetrain design. The hub flanges from the Suzuki were also utilised, as no modification was required to fit both the CV assembly and wheel stud pattern.

Once these components were sourced the centre axle design was finalised and manufactured. Details on the design and manufacture of the shaft are summarised in the following chapter. The preferred final drive ratio was then calculated and the necessary sprockets purchased. The positioning of the rear wheels is determined by the suspension, therefore having the distance of the rear axle behind the engine limited by this, additional analysis was required to confirm the sprocket sizes would allow for the required amount of chain wrap around the drive sprocket. This found that the rear sprocket intended for use only just allowed for sufficient chain wrap, so a slightly smaller rear sprocket was ordered as a precaution.

The rear axle then needed to be mounted to the chassis by its support bearings. Not only did housings have to be manufactured to hold the bearings, but also the bracket the bearing housings would be mounted to had to allow for horizontal displacement to tension the chain. It was decided to allow the entire rear axle to slide horizontally to tension the chain and remove the need for a chain tensioner. Whilst a chain tensioner would increase the amount of chain wrap on the drive sprocket, it also increases the frictional losses of the drivetrain and adds unnecessary componentry and weight to the car. Another disadvantage of a chain tensioner discovered from the 2004 car was the significant increase in operating noise during engine braking.

When the rear axle was fixed in position inside the chassis, the rear disc brake calliper had to be mounted to the chassis also allowing for the horizontal displacement of the brake rotor that was fixed to the axle. To do this slotted brackets are required to be welded to the chassis that would maintain the strength required to support the braking force. Finally the shortening of the CV shafts are to be left until last to make certain the length will be accurate. Once all of the components have been sourced and manufactured, the drivetrain will be assembled and tested.

3.5 Final Gearing

The final gearing of the drivetrain is required to calculate the loading of the shafts and the performance characteristics of the vehicle. Torque, acceleration and top speed are all major performance parameters affected by the final gear ratio. A higher gear ratio lowers a vehicles top speed but increases its torque and acceleration. Therefore the ideal gear ratio would be the highest ratio possible to allow the vehicle to reach the required top speed, thus maximising the torque and acceleration. The 2004 car featured a 15 tooth front sprocket driving a 60 tooth rear sprocket resulting in a final drive ratio of 4. This reveals a top speed capability of 151 km/hr. From researching previous entrants of the Formula SAE-A competition and a technical specification given from the Society of Automotive Engineers, the maximum speed obtainable in the competition is around 120 km/hr. However due to the tight course the average speeds are kept to around 40 – 50 km/hr increasing the benefits of better acceleration and lowering the importance of top speed.

Another factor was that last year the car primarily stayed in second gear for the majority of the course, signifying the car may have been geared a little low and wasn't maximising the cars potential performance. For these reasons it was decided to increase the final drive ratio of the drivetrain for this year. The 60-tooth driven sprocket at the rear cannot be increased in size due to availability of space; therefore the 15 tooth front sprocket is to be replaced by a smaller drive sprocket. The smallest size available from Yamaha was a 13-tooth sprocket. This would increase the final gearing from 4 to 4.615.

The top speeds in each gear at the maximum design rpm (11 000rpm) were determined using service data from (Yamaha YZF600, 1994). The maximum torque produced in each gear was also tabulated from this data.

Primary Reduction Ratio = 82/48 (1.708)

Secondary Reduction Ratios:

1^{st} = 37/13 (2.846)

2^{nd} = 37/19 (1.947)

3^{rd} = 31/20 (1.550)

4^{th} = 28/21 (1.333)

5^{th} = 31/26 (1.192)

6^{th} = 30/27 (1.111)

Appendix D displays the change in speeds and torque due to this ratio change.

The selection of the final gear ratio is a result of the right compromise between acceleration, torque and top speed. The optimum drive ratio will differ between each track layout and can be fine tuned by selecting different sized sprockets.

3.6 Axle loading

The loading of the axle must be determined to design the components of the drivetrain. After assuming the masses of each component are negligible the forces acting on the axle simplified to just the torque being transmitted by the chain through the rear sprocket. This torque also results in a force pulling the axle forward toward the engine. To determine the maximum torque applied to the rear axle a combination of three different approaches was taken. First was the study into the potential performance capabilities of the engine, then analysing the tyre friction possibly limiting the transfer of torque, and finally the predicted accelerations and cornering forces acting on the vehicle.

3.6.1 Engine Performance

To begin with the maximum torque transmitted to the rear axle was estimated using engine performance. The engine being used in USQs Formula SAE racing car is a 1994 model Yamaha YZF600 motorcycle engine, which has unrestricted power and torque figures of approximately 60 kW and 61Nm respectively measured from the crankshaft. To be able to estimate the true torque available at the rear axle as accurately as possible, there are a number of factors that need to be considered. First the Formula SAE rules require that the air intake be restricted to a narrow 20mm opening. Second the engine is not new therefore the internals of the engine have been worn and will no longer produce the same power and torque ratings. Finally the frictional losses from the transfer of torque through the gearbox and chain drive must also be considered. Assuming 15% of torque is lost through frictional forces, and at least another 5 % lost through the intake restrictor, the maximum torque will be:

$$T = (0.15 + 0.05) \times 61 = 48.8Nm$$

Therefore multiplying this by the primary and secondary gear ratios and again by the final drive ratio will give an estimate of the expected torque from the engine. First gear is used because it has the highest ratio and will deliver the most torque.

$$\begin{aligned} T_{\max} &= 48.8 \times 1.708 \times 2.846 \times 4.615 \\ &= 1094.7Nm \end{aligned}$$

3.6.2 Tyre Friction

In the majority of vehicles the maximum amount of torque to be transferred through the rear axle is limited by the traction of the tyres. The normal force on the tyre and the coefficient of friction govern the force at which a tyre will break traction and slip. The normal force on the tyre under straight-line acceleration will be increased due to weight transfer. This weight distribution is estimated to be around 30:70 front to rear. The total weight was assumed to be 350 kg including a 100 kg driver.

The coefficient of friction will vary according to tyre compounds from different manufacturers, tyre temperature and wear, the cleanliness and roughness of the tracks surface and many other conditions. Therefore after researching many internet sites the values of μ for a typical tyre on tarmac range from 0.7 to 1.3. As a result an average value for the coefficient of friction was taken as 1.

$$F = mg$$

where: F = Force (N)

m = Mass (kg)

g = acceleration due to gravity (m/s²)

The normal force on the rear tyres is:

$$F_N = mg = 350\left(\frac{70}{100}\right)\times 9.81 = 2403.5N$$

The frictional force with a coefficient of friction of 1:

$$F_R = F_N\mu = 2403.5\times 1.0 = 2403.5N$$

$$T = F_R r$$

where: T = Torque (Nm)

F_R = Frictional force (N)

r = Radius (m)

$$\begin{aligned} T &= 2403.5\times 0.2765 \\ &= 664.5Nm \end{aligned}$$

3.6.3 Vehicle Performance

A prediction of the cars acceleration capability and cornering forces was calculated to check the maximum torque to the rear axle, and to assess the amount of tyre slip from a solid rear axle verifying the benefit of a differential.

3.6.3.1 Acceleration

The straight-line acceleration of the car is unknown since it has not yet been built, however a prediction of the acceleration can be estimated from the results of FSAE teams from previous years. The majority of teams claim to be capable of accelerating from 0 to 100 km/hr in between 3 to 5 seconds.

Therefore using a desired performance of 0-100 in 4 seconds gives an acceleration of 6.94 m/s^2. This is clearly on the higher end of the scale and is most likely an overestimate, however this is preferable to an underestimate when calculating the maximum torque to provide a safety factor into the stress calculations. To be capable of this acceleration the torque produced in the rear axle would be 672Nm. This value is very close to the estimated torque value limited by the traction of the tyres.

3.6.3.2 Cornering

Torque can only be transferred through an axle if the tyre has enough traction force to support this load. During cornering the weight distribution of the car changes and the normal force acting on the inside wheel is reduced, usually resulting in loss of traction. In order to investigate when more torque is transferred to the outside wheel, the forces acting on each wheel during cornering were calculated. Analysing the different situations where wheel slip is likely to occur will also give a better comparison between advantages and disadvantages of a differential. To calculate the centripetal acceleration around different corners at different speeds a free-body diagram was drawn to aid the analysis of moments and forces at the wheels.

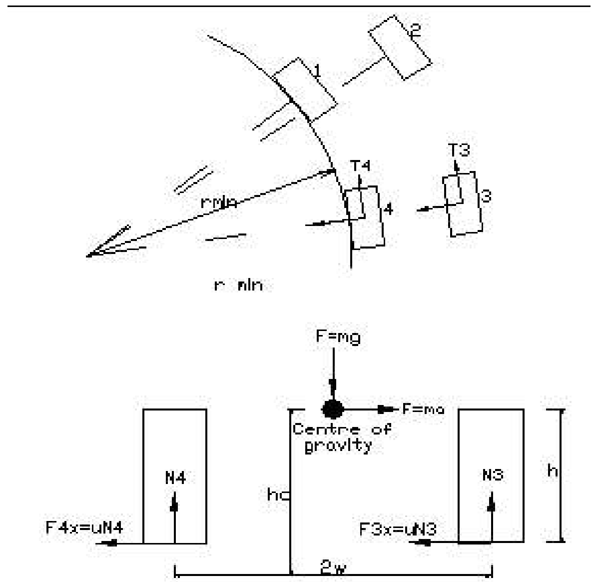


Figure 3.1: Free-body diagram of moments

Assuming a mass of 350 kg including driver, and a weight distribution of 45:55 front to rear the mass at the rear tyres is:

$$m_R = 350 \times (\frac{55}{100}) = 192.5kg$$

Therefore the normal force acting on the inner wheel around a corner is found to be:

$$\sum M_4 = -mg \times w - ma \times h_C + N_3 \times 2w = 0$$

where: M = Moment (Nm)

N_4 = Normal force on inside rear tyre (N)

N_3 = Normal force on outside rear tyre (N)

2w = Rear wheel track (m)

$$N_3 = \frac{mg \times w + ma \times h_C}{2w}$$

$$= \frac{192.5 \times 9.81 \times 0.6 + 192.5 \times a \times 0.24}{1.2}$$

$$= 944.2 + 38.5a$$

The normal force acting on the outer wheel can also be found.

$$\sum M_3 = -ma \times h_C + mg \times w - N_4 \times 2w = 0$$

$$N_4 = \frac{-ma \times h_C + mg \times w}{2w}$$

$$= \frac{-192.5 \times a \times 0.24 + 192.5 \times 9.81 \times 0.6}{1.2}$$

$$= 944.2 - 38.5a$$

The centripetal acceleration around various radii corners taken at a range of speeds has been has been calculated and used to tabulate the normal force on each tyre in each situation. Wheel slip will occur when the torque becomes greater than the traction force μN. These results have been attached in Appendix D and show each situation when the inside will lose traction.

Using two different values of μ for the coefficient of friction of both road tyres and racing slicks, the traction force was calculated and compared with the estimated torque at the tyres.

Given the range of corners we are likely to encounter during the competition from given information found in the competition specifications, and a range of speeds the corners may be attempted at, the results clearly show the circumstances at which tyre slippage will occur with a solid rear axle and therefore the benefit of a differential. This data will also be useful to determine the possibility of the extreme situation where all of the torque is transferred through one tyre, doubling the loads on that axle.

3.7 Conclusion

The selection and design of the most appropriate drivetrain system was possibly the most important aspect of this project. The design criteria for this task were very comprehensive and allowed for the best available option to be selected. The selection process following the set criteria has been summarised along with the reasoning behind the final design selection of the solid rear axle. The maximum torque produced has been estimated a number of different ways in an attempt to find the most accurate assumption. The torque estimate from the engine power can only be based on if the wheels of the car never break traction, which is clearly not the case given the vehicle characteristics. For this reason calculations estimating the tractive force of the tyres were undertaken to include this limiting factor and find a more realistic torque value.

Finally a predicted acceleration value based on the desired performance of the car was used to confirm this estimate of torque. Information concerning the probability of tyre slip occurring due to the solid rear axle whilst cornering has been tabulated to give an idea of the affect of competing without a differential. The following chapters utilize all of this information to analyse each drivetrain component.

Chapter 4

Centre Shaft Design and Analysis

4.1 Introduction

This chapter covers the design and the analysis of the centre rear axle. The design and analysis on the solid rear axle used in the 2004 car was researched and used as a base for improving the design. Detailed stress analysis, fatigue life and deflection calculations were all performed to accurately analyse the shaft. The manufacturing process is also outlined as a record of how the design was created.

4.2 Centre Shaft Design

4.2.1 Positioning

The design of the rear axle is dependant on many factors. These include engine positioning, the physical space available inside the chassis, location for supports to hold the axle in place and the CV assembly. The position of the engine determines the position of the rear sprocket flange that must align with the front sprocket, and also the distance between the front and rear sprocket centres that influences the angle of wrap of the chain. There must be enough physical space to mount the rear axle, sprocket and disc brake assembly inside the chassis frame. The horizontal positioning of the centre axle will determine the length of the wheelbase, which has a significant affect on weight distribution and performance characteristics of the car. Consideration must also be given to the location of the support bearings that are required to hold the axle in place. The support bearings will be fixed to the chassis frame and must also be as close to the sprocket flange and brake disc flange as possible to offer maximum support and reduce shaft deflection and bending stresses. The final consideration was to the CV assembly that each end of the centre shaft must drive. This influences the vertical positioning of the axle and the size of the splines either end of the shaft.

4.2.2 Flanges

Two flanges are required along the rear axle, one to support the driven rear sprocket and carry the input torque, and the other to fix the single rear brake disc to. These flanges must be strong enough to withstand the stress and fatigue of repetitive loading. The flanges feature a shoulder to locate the centre of the rotor and sprocket respectively, and were designed to match the standard bolt pattern of the brake rotor from the Yamaha motorcycle. During the manufacture of the shaft the flanges were required to be made as separate components, which were to be welded to the shaft. To ensure any distortion or other effects caused by the welding procedure were corrected, the shaft and flanges were machined and 'trued'.

There was a concern with the alignment of the rear sprocket with respect to the front sprocket. After the misalignment problems encountered last year I intended to finalise the position of the rear sprocket flange after the engine had been located in the chassis, hence eliminating any chance of alignment error. However due to the time constraints of this project the centre shaft had to be manufactured before the engine testing was complete, and the engine could be fitted into the car. In an attempt to overcome any misalignment of the rear sprocket, vigorous measuring and dimensioning of the engine and its placement into the chassis were undertaken. To allow for any inaccuracies, the rear sprocket flange was also made 3mm thicker either side, and can be machined back to the design thickness after the shaft can be aligned with the engine.

4.2.3 Splines

The inner CV joint from the Suzuki CV assembly sourced has an internal spline. This was important because an external spline is much easier and cheaper to manufacture. To measure the spline dimensions accurately the CV joint was disassembled and cleaned thoroughly, so it could then be examined under a profile projector. The projector used was a Mitutoyo PJ300 and it magnified a projection of the spline for measurements to be taken. The spline is a 22 tooth involute spline that has an outside diameter of approximately 24mm. The measurements read from the profile projector were the tooth land, tooth depth and pressure angles.

Sources of error and inaccuracies arose from the wear induced in the spline so a number of teeth were measured and then the average dimension recorded. Below is a sketch of the spline profile and the results of the spline dimensions. The spline also required a C-clip groove to be machined toward the end of the spline to locate the C-clip that would hold the CV joint onto the shaft and prevent it from sliding out.

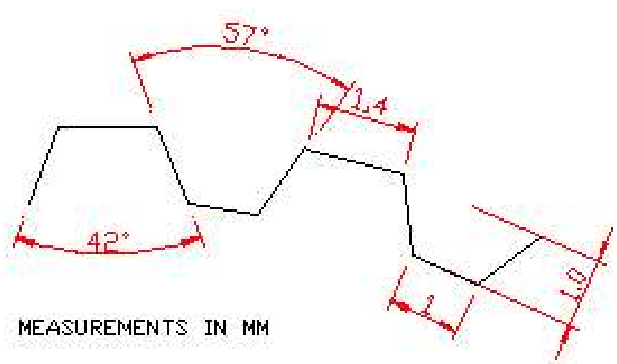


Figure 4.1: Spline profile

Table 4.1: Spline Dimensions

22 Tooth Involute Spline Dimensions	
Outer Diameter	24mm
Root Diameter	22mm
Tooth Land	1.4mm
Tooth Depth	1.0mm
Pressure Angle	21°

4.2.4 Material selection

The most significant innovation compared with last years rear axle was the use of hollow bar steel from which the centre axle was manufactured. This was chosen to meet the goal of weight conservation for the development of the 2005 car. The only consequence of this is that since the CV splines are quite small, only 24mm in diameter, so small lengths of solid bar will be welded to either end of the shaft from which the splines will be cut. The most appropriate sizes of hollow bar readily available range from 16-25mm I.D. and 32-40mm O.D. However outside diameters are not of particular importance, as the shaft will be machined during manufacture.

The internal diameter of the hollow bar was to be no smaller than the diameter of the spline, so that the solid end could fit into the hollow bar and support the weld. Using an internal diameter smaller than this would cause a serious stress concentration in the material, creating a likely point of failure. Therefore a 22mm I.D. x 36mm O.D. hollow bar was chosen to allow for the maximum diameter of the spline and the necessary wall thickness required to maintain adequate strength and shoulder the bearings.

AISI 1020 mild steel was selected primarily because of its low cost and ease to weld and machine. Hollow bar made from the higher strength alloy steels was also considered however proved to be very difficult to source. The advantage of using

higher strength alloy steel is that it would allow for smaller diameters and a thinner wall thickness to be used, therefore decreasing the weight even further.
The grade of solid steel bar required for the splines on either end of the axle was determined after a stress analysis of the splines was performed. These analyses illustrated later in this chapter determined the strength of material required to avoid deformation and failure. It was revealed that the AISI 1020 mild steel used for the rest of the centre axle would not be strong enough so alloy steel AISI 4140 should be used.

4.2.5 Manufacture

When designing a component and analysing what material it should be made from the manufacturing process must be considered. It is essential to understand that a design can appear flawless however if it cannot be made than the design is of no use. For this reason any good design must take into account the manufacturing procedure likely to be undertaken when making the component, and allow for ease of manufacture. During the design process after I had sketched my initial design I approached the staff from the USQ Engineering Workshop, who would be the professionals to make the centre axle, and discussed the procedure that would be undertaken to manufacture the shaft and what I needed to incorporate into my design to allow for this process. The most significant change made from the initial design was to allow for a stronger weld to join the solid bar material to the ends of the hollow bar.

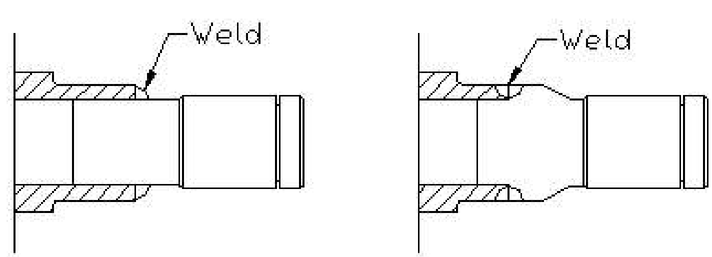


Figure 4.2: Stub end weld design

Once the design was complete the manufacture began by cutting the mild steel hollow bar material to length. Then the solid alloy steel ends were cut to length and machined to fit into and up against the hollow bar. The two flanges were then bored to fit the hollow shaft using a lathe. The alloy steel stub ends were then preheated before welding in place to ensure the strength of the weld between the two different metals and minimise any weakness of the heat-affected area. The sprocket flange was made wider than required to allow for adjustment in the alignment of the engine and drive sprocket. Once the engine is fitted into the chassis the sprocket flange can be machined back to size where appropriate. The two flanges were welded to the shaft with the stub ends, and the entire shaft was machined down to the correct dimensions and tolerances. It was critical for this process to occur after the welding to eliminate any welding effects. The splines were to be cut with the CNC machine using a modified fly-cutter, and so a few practice runs were done to ensure the quality and fit of the spline into the CV joint. The last process was to drill the flanges to suit the rear sprocket and brake rotor.

4.3 Centre Shaft Analysis

The analysis of the centre shaft involves calculations of loading, stresses, fatigue and deflection. The result of these analyses will verify the strength of material required and predict the life of the shaft. The hollow bar shaft is made from AISI 1020 mild steel, which has a yield and ultimate strength of 331MPa and 448MPa respectively in the as rolled state (Appendix C-4a, Juvinall & Marshek 2000). AISI 4140 alloy steel was chosen for the solid material to be splined either end of the shaft, because it was predicted that mild steel would not be strong enough to withstand the resulting high stress of the smaller splines. AISI 4140 has a yield and ultimate strength of 655MPa and 1020MPa respectively in the as rolled state from appendix C-4a. Before the stress and fatigue calculations can be undertaken, the loading of the shaft must be analysed.

4.3.1 Shaft loading

The most significant loads on the centre shaft are due to the disc brake and the chain drive, which transfers torque through the rear sprocket. The weight of the drivetrain components are considered negligible so will not be considered in the analysis of the shaft. The chain drive produces a tangential force on the rear sprocket resulting in a torque of 664.5Nm, as found in the previous chapter. Under normal operating conditions the solid rear axle will share torque equally between both rear tyres. However this analysis will also consider the extreme condition of all the torque being transferred through only one tyre. The maximum tangential force acting on the sprocket (Dp = 303.33mm) is found to be:-

$$F = \frac{T}{r_p}$$

$$F = \frac{664.5}{0.1517}$$

$$F = 4381.37N$$

The bending moment created in the shaft is from the chain attempting to pull the rear sprocket forward. The support bearings either end of the shaft act against this force and hold it in place. The initial chain tension is neglected as well as any centrifugal effects of the chain.

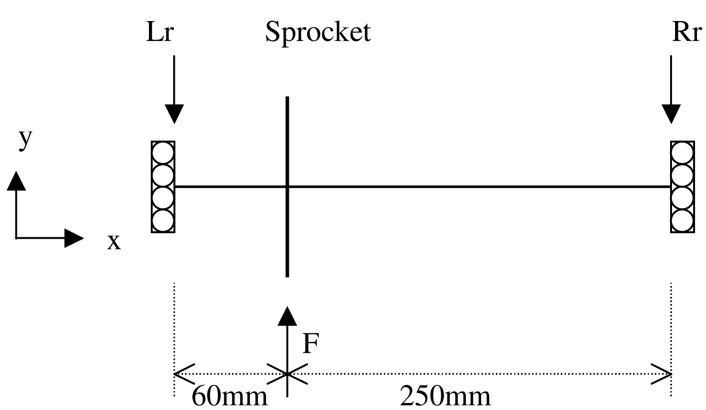


Figure 4.3: Free-body diagram of centre shaft

The sum of moments around the left bearing finds the right bearings reaction force.

$$\sum M_L = 0$$
$$0 = (4381.37 \times 0.06) - (Rr \times 0.31)$$
$$\therefore R_r = 848N$$

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

$$\sum F_Y = 0$$
$$F = R_r + L_r$$
$$\therefore L_r = 4381.37 - 848$$
$$= 3533.37N$$

The shear force, bending moment and torque diagrams are illustrated below in Figure 4.4.

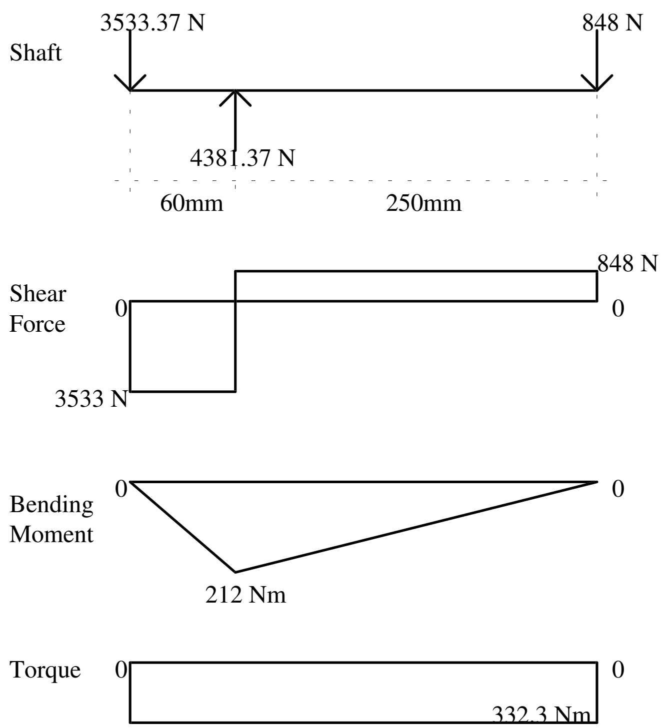


Figure 4.4: Force Diagrams

4.3.2 Stress analysis

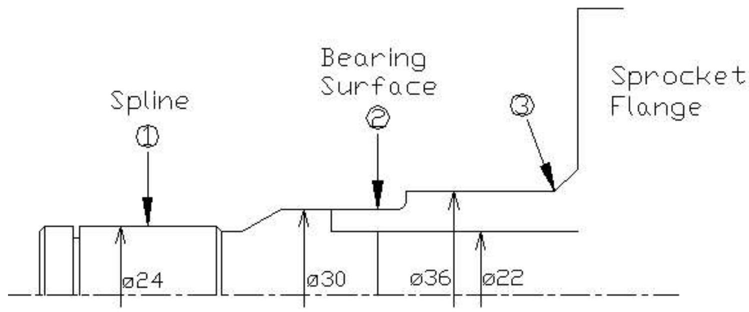


Figure 4.5: Critical locations for analysis

The first point of concern is the possible shearing of the involute spline. According to (Machinery's Handbook, 2000) if the length of engagement of the spline is between 0.75 to 1.25 times the pitch diameter, then the shear strength of the spline will exceed that of the shaft from which it is made. Since the internal spline matching the external spline we are creating already exists, then the length of the spline has already been set and was measured to be 30mm. Therefore the length of engagement of the spline is 30/24 = 1.25 times the pitch diameter and should then exceed the shear strength of the material.

Thus the next point of concern is the shearing of the shaft ends. Two conditions will be analysed to determine the shear stress due to torsion. The first is under normal loading conditions during straight-line acceleration, when torque is shared equally between both tyres. The second is an extreme case of all of the torque being transferred through one wheel. The root diameter of the spline is 22mm therefore the shear stress due to torsion can be found by using equation 3.9 (Beer & Johnston 2002):-

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

where: τ = Torsional shear stress (MPa)

T = Torque (Nm)

r = radius (m)

J = Polar moment of inertia (m^4)

The torque and radius of shaft are known, and the polar moment of inertia can be found from the following equation. For a circular cross-section where d = diameter:

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

Therefore under normal loading conditions 332.3Nm torque will create a shear stress of:

$$\tau = \frac{332.3 \times 0.011}{\frac{\pi \times 0.022^4}{32}}$$

$$= 158.94 \text{ MPa}$$

To predict the equivalent tensile yield strength from the shear stress the Maximum Distortion Energy Criterion was used. From figure 8.16 (Juvinall & Marshek 2000):

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

where: σ_Y = Yield strength (MPa)

τ_{max} = Maximum shear stress (MPa)

Therefore the equivalent tensile strength under normal loading conditions is:

$$\sigma = \frac{158.94}{0.58}$$

$$\sigma = 274 MPa$$

Next to consider extreme loading of the centre shaft, with a possible 664.5Nm of torque creating a shear stress of:

$$\tau = \frac{664.5 \times 0.011}{\frac{\pi \times 0.022^4}{32}}$$
$$\tau = 317.83 MPa$$

The equivalent tensile strength for extreme loading conditions is:

$$\sigma = \frac{317.83}{0.58}$$
$$\sigma = 548 MPa$$

The next area of concern is the 30mm bearing surface made from AISI 1020 mild steel hollow bar. This section of the shaft is subject to combined loading of shear stress with torsion and bending stress with torsion. The following formulas that predict this combined stress were sourced from (Beer & Johnston 2002).

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

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

where: V = Shearing force (N)

Q = First moment of area (m^3)

I = Moment of inertia (m^4)

b = width of section (m)

M = Bending moment (Nm)

c = radius of shaft (m)

From the shear force diagram it can be seen that the maximum shearing force occurs between the left bearing and the sprocket at a magnitude of 3533N. The critical location to be analysed is point 2, the bearing surface, assuming an extreme load of

664.5Nm of torque. To find the induced combined stress at this point the direct shear stress is found initially where:

$$
\begin{aligned}
Q &= A\bar{y} \\
&= \left(\frac{1}{2}\pi(r_o^2 - r_i^2)\right)\left(\frac{4(r_o - r_i)}{3\pi}\right) \\
&= \frac{2}{3}(r_o^3 - r_i^3) \\
&= \frac{2}{3}(0.015^3 - 0.011^3) \\
Q &= 1.36\times10^{-6}m^3
\end{aligned}
$$

$$
\begin{aligned}
I &= \frac{\pi}{4}(r_0^4 - r_i^4) \\
&= \frac{\pi}{4}(0.015^4 - 0.011^4) \\
I &= 0.28\times10^{-6}m^4
\end{aligned}
$$

Therefore the direct shear stress is:

$$
\begin{aligned}
\tau_d &= \frac{3533.37\times1.36\times10^{-6}}{0.28\times10^{-6}\times0.03} \\
&= 0.572MPa
\end{aligned}
$$

Torsional shear stress is then found.

$$
\begin{aligned}
\tau_t &= \frac{Tr}{J} \\
&= \frac{664.5\times0.03}{\frac{\pi\times(0.03)^4}{32}} \\
\tau_t &= 250.7MPa
\end{aligned}
$$

To find the total shear stress the direct and torsional shear stresses are added together.

$$\begin{aligned}\tau_{\max} &= \tau_d + \tau_t \\ &= 0.57 + 250.68 \\ \tau_{\max} &= 251.25MPa\end{aligned}$$

The equivalent maximum tensile stress due to combined shear stresses can now be found.

$$\begin{aligned}\sigma &= \frac{251.25}{0.58} \\ &= 433MPa\end{aligned}$$

Now the combined bending and torsional stresses acting on the shaft must be considered. The bending diagram in Figure 4.4 shows the maximum bending moment occurring at the location of the sprocket flange (location 3). So the bending stress due to the 212Nm moment is found.

$$\begin{aligned}\sigma_b &= \frac{Mc}{I} \\ &= \frac{212 \times 0.018}{\frac{\pi}{4}(0.018^4 - 0.011^4)} \\ \sigma_b &= 53.79MPa\end{aligned}$$

The torsional shear stress is found once again under the extreme loading condition of 664.5Nm of torque.

$$\begin{aligned}\tau_t &= \frac{Tr}{J} \\ &= \frac{664.5 \times 0.018}{\frac{\pi(0.036^4 - 0.022^4)}{32}} \\ \tau_t &= 84.29MPa\end{aligned}$$

To find the total combined stresses the formulas for Mohr's Circle for biaxial stress can be used. To begin the principal stresses must be found using equation 4.16 (Juvinall & Marshek 2000).

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

$$\sigma_1, \sigma_2 = \frac{53.79}{2} \pm \sqrt{84.29^2 + \left(\frac{53.79}{2}\right)^2}$$
$$\sigma_1 = 115.39 MPa$$
$$\sigma_2 = -62.06 MPa$$

Now using the Maximum Distortion Energy Criterion these principal stresses can be equated into a yield strength equivalent with equation (Stress Analysis 2001).

$$\sigma_y^2 = \sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2$$
$$= 115.39^2 - 115.39 \times (-62.06) + (-62.06)^2$$
$$\sigma_y = 155.97 MPa$$

All of these resultant stresses have been tabulated below for comparison to determine the most likely point of failure along the shaft. To find the factor of safety at each location the yield strength of the respective material was used.

Table 4.2: Factor of Safety Summary

Point	**Load Condition**	**Induced Stress (MPa)**	**Factor of Safety**
Material: AISI 4140 Alloy Steel			
1	Normal	274	2.39
1	Extreme	548	1.20
Material: AISI 1020 Mild Steel			
2	Extreme	433	0.76
3	Extreme	156	2.12

From the results it is evident that the root diameter of the spline is the most significant stress concentration along the shaft. Since this outcome was predicted the spline ends were made from AISI 4140 alloy steel, which can now been confirmed as the correct decision. This leaves the 30mm bearing surface made from mild steel hollow bar as the weakest section along the centre shaft. The actual loading conditions of the shaft should be close to normal conditions under straight line acceleration, however is likely to be somewhere in between the normal and extreme conditions when cornering. Therefore the theoretical factors of safety under extreme conditions for the different sections of shaft will be greater in reality.

4.4 Fatigue life

The fatigue life of the centre shaft design was determined using the formulas and tables from the Fundamentals of Machine Component Design textbook (Juvinall & Marshek 2000). Figure 4.6 below shows the location of the two stress concentrations that will be the focus of this analysis.

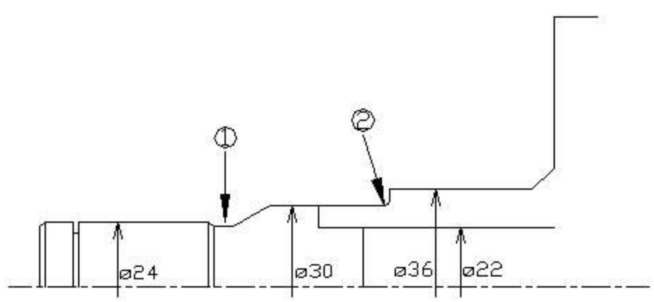


Figure 4.6: Critical fatigue locations for analysis

At location 1 the loading simulates acceleration and braking due to torsion in a straight-line. To begin the brake force must first be calculated, so by assuming the rear brakes only supply 30% of total brake force, the brake torque was found.

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

$$\text{if } \mu = 1,\ F_R = F_N \mu = 1030N$$

$$\begin{aligned} T &= F_R r \\ &= 1030 \times 0.265 \\ &= 272.95Nm \end{aligned}$$

Therefore the shaft will experience torques of 332.3Nm in acceleration and 272.95Nm in braking. This is a simulation of fully reversed loading. According to Figure 4.35(c) (Juvinall & Marshek 2000) the nominal torsional shear stresses in the shaft due to acceleration are:

$$\begin{aligned} \tau_{nom} &= \frac{16T}{\pi d^3} \\ &= \frac{16 \times 332.3}{\pi \times 0.022^3} \\ &= 158.94MPa \end{aligned}$$

From the same figure the Kt value for torsion can be found.

$$\left.\begin{aligned} \frac{D}{d} &= \frac{30}{22} = 1.364 \\ \frac{r}{d} &= \frac{2.5}{22} = 0.114 \end{aligned}\right\} \therefore K_t = 1.34$$

Now the alloy steel material used has a Su value of 148ksi, which results in a notch sensitivity factor, q=0.92, illustrated in Figure 8.24 (Juvinall & Marshek 2000). Therefore the fatigue stress concentration factor, Kf can be found from the following equation.

$$\begin{aligned} K_f &= 1 + (K_t - 1)q \\ &= 1 + (1.34 - 1) \times 0.92 \\ &= 1.313 \end{aligned}$$

Therefore by incorporating this stress concentration factor the maximum torsional shear stress can be calculated.

$$\begin{aligned} \tau_{\max} &= K_f \tau_{nom} \\ &= 1.313 \times 158.94 \\ &= 208.66 MPa \end{aligned}$$

Now the same process will be carried out to find the minimum torsional shear stress in the shaft.

$$\begin{aligned} \tau_{nom} &= \frac{16T}{\pi d^3} \\ &= \frac{16 \times -272.95}{\pi \times 0.022^3} \\ &= -130.55 MPa \end{aligned}$$

The stress concentration factor remains the same value.

$$\begin{aligned} \tau_{\min} &= K_f \tau_{nom} \\ &= 1.313 \times -130.55 \\ &= -171.39 MPa \end{aligned}$$

The mean and alternating shear stress can be determined from Figure 8.15(Juvinall & Marshek 2000).

$$\begin{aligned} \tau_m &= \frac{(\tau_{\max} - \tau_{\min})}{2} \\ &= \frac{(208.66 - (-171.39))}{2} \\ &= 18.64 MPa \end{aligned} \qquad \begin{aligned} \tau_a &= \frac{(\tau_{\max} + \tau_{\min})}{2} \\ &= \frac{(208.66 + (-171.39))}{2} \\ &= 190.03 MPa \end{aligned}$$

From Table (8.1)(Juvinall & Marshek 2000) the 10^6-cycle strength is found. The AISI 4140 alloy steel material has an ultimate strength Su = 1020MPa and a yield strength Sy = 655MPa.

$$S_n = S_n' C_L C_G C_S$$

where: $S_n' = 0.5\ S_u$

C_L = Load factor

C_G = Gradient factor

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

$$\begin{aligned} S_n &= 0.5 \times 1020 \times 0.58 \times 0.9 \times 0.7 \\ &= 186.35 MPa \end{aligned}$$

The alternating stress is larger than is value indicating that the mean and alternating stresses will not be inside the infinite life line on a Goodman Fatigue diagram. Therefore a S-N curve has been plotted along with the Goodman Fatigue diagram to predict the fatigue life of the component, shown in Figures 4.7 and 4.8 respectively. The data below are required for the plotting of the graphs.

$$S_{10^3} = 0.72\ S_u = 0.72 \times 1020 = 734.4 MPa$$

$$S_{us} = 0.8\ S_u = 0.8 \times 1020 = 816 MPa$$

$$S_{ys} = 0.58\ S_y = 0.58 \times 655 = 379.9 MPa$$

where: S_{10^3} = 10^3-cycle strength (MPa)

Sus = Ultimate shear strength (MPa)

Sys = Yield shear strength (MPa)

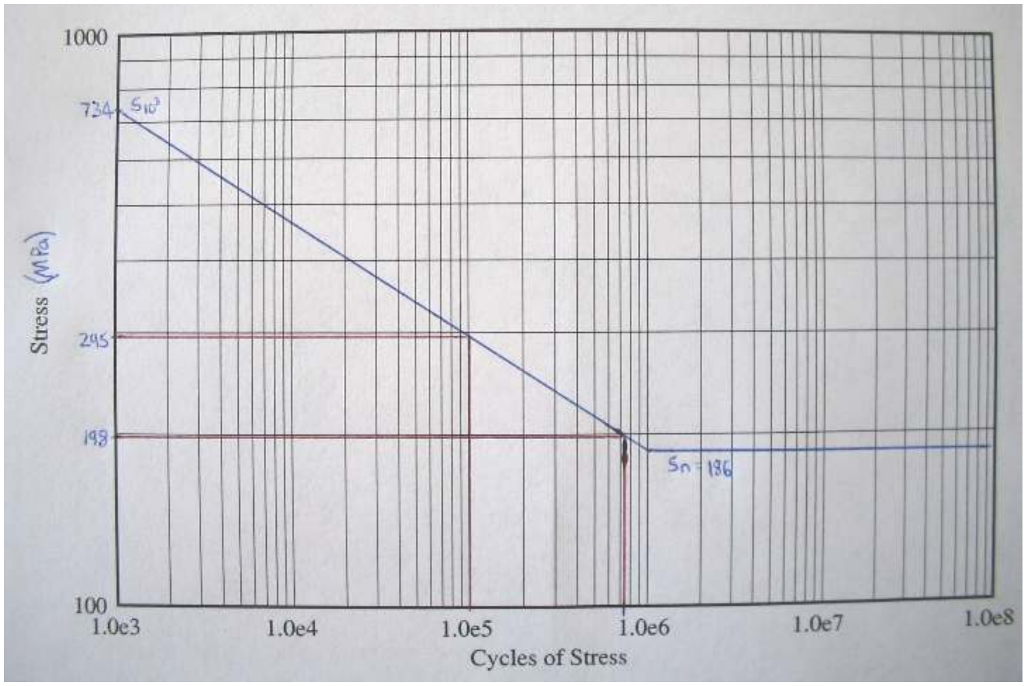


Figure 4.7: S-N Curve

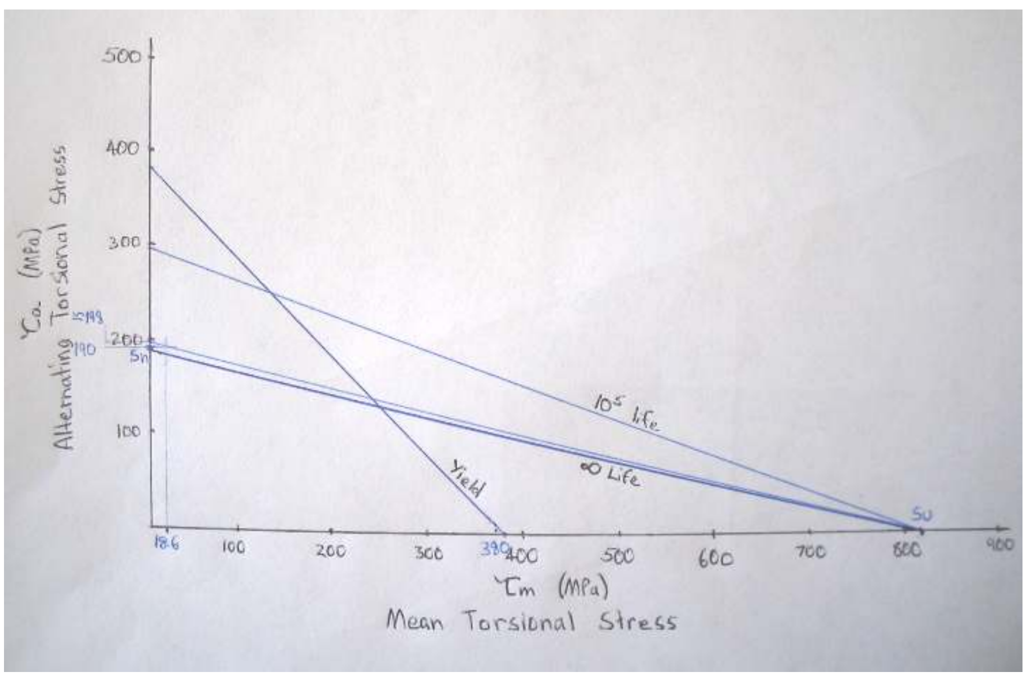


Figure 4.8: Goodman Fatigue Diagram

The resultant fatigue life of the shaft is approximately 7.3 x10^5 cycles. The mean and alternating shear stresses were plotted on the Goodman Fatigue diagram, and the fatigue strength was found to be about 198MPa. This value was then plotted against the S-N curve to determine the shafts expected fatigue life. Another approach to estimate the fatigue life is by using the numerical formulas (Design of Machine Elements 2003).

$$b = -\frac{1}{3}\log\frac{S_3}{S_n} = -\frac{1}{3}\log\frac{734}{186} = -0.1987$$

$$C = \log\frac{{S_3}^2}{S_n} = \log\frac{734^2}{186} = 3.4619$$

$$\begin{aligned} N &= 10^{-\frac{C}{b}} {S_f}^{\frac{1}{b}} \\ &= 10^{-\frac{3.4619}{-0.1987}} 198^{\frac{1}{-0.1987}} \\ &= 7.32 \times 10^5 \text{cycles} \end{aligned}$$

This confirms the result achieved on the S-N curve (figure 4.7).

A fatigue analysis will now be done on the second location, the bearing shoulder made from mild steel hollow bar. The torque acting on the shaft of 332.3Nm in acceleration and -272.95Nm in braking has not changed. The alternating bending stress at the bearing will be very small and so has been neglected. The maximum shear stress is found from equation (3.25)(Beer & Johnston 2002).

$$\tau_{\max} = K\frac{Tc}{J}$$

The stress concentration factor is determined from figure 3.32 (Beer and Johnston 2002).

$$\left.\begin{aligned} \frac{r}{d} &= \frac{1}{30} = 0.033 \\ \frac{D}{d} &= \frac{36}{30} = 1.2 \end{aligned}\right\} \quad K = 1.6$$

Therefore the maximum shear stress during acceleration is:

$$\tau_{max} = 1.6 \times \frac{332.3 \times 0.015}{\frac{\pi}{2}\left(0.015^4 - 0.011^4\right)}$$
$$= 141.1MPa$$

Now the minimum shear stress caused from braking is found to be:

$$\tau_{min} = K\frac{Tc}{J}$$
$$= 1.6 \times \frac{-272.95 \times 0.015}{\frac{\pi}{2}\left(0.015^4 - 0.011^4\right)}$$
$$= -115.9MPa$$

The mean and alternating shear stresses are once again calculated and plotted against each other on a Goodman Fatigue diagram (Figure 4.9).

$$\tau_m = \frac{\left(\tau_{max} + \tau_{,min}\right)}{2} = \frac{\left(141.1 + (-115.9)\right)}{2} = 12.6MPa$$

$$\tau_a = \frac{\left(\tau_{max} - \tau_{min}\right)}{2} = \frac{\left(141.1 - (-115.9)\right)}{2} = 128.5MPa$$

Table 8.1 (Juvinall & Marshek 2000) is used to determine the 10^6-cycle strength. The hollow bar shaft is made from AISI 1020 mild steel, which has an ultimate strength Su = 448.2MPa and a yield strength of Sy = 330.9MPa.

$$S_n = S_n' C_L C_G C_S$$
$$= 0.5 \times 448.2 \times 0.58 \times 0.9 \times 0.78$$
$$= 91.24MPa$$

$$\mathrm{S}_{10^3} = 0.72 \times \mathrm{S}_\mathrm{u} = 322.7\mathrm{MPa}$$

$$\mathrm{S}_{us} = 0.8\,\mathrm{S}_\mathrm{u} = 358.5\mathrm{MPa}$$

$$S_n = 0.29S_u = 130MPa$$

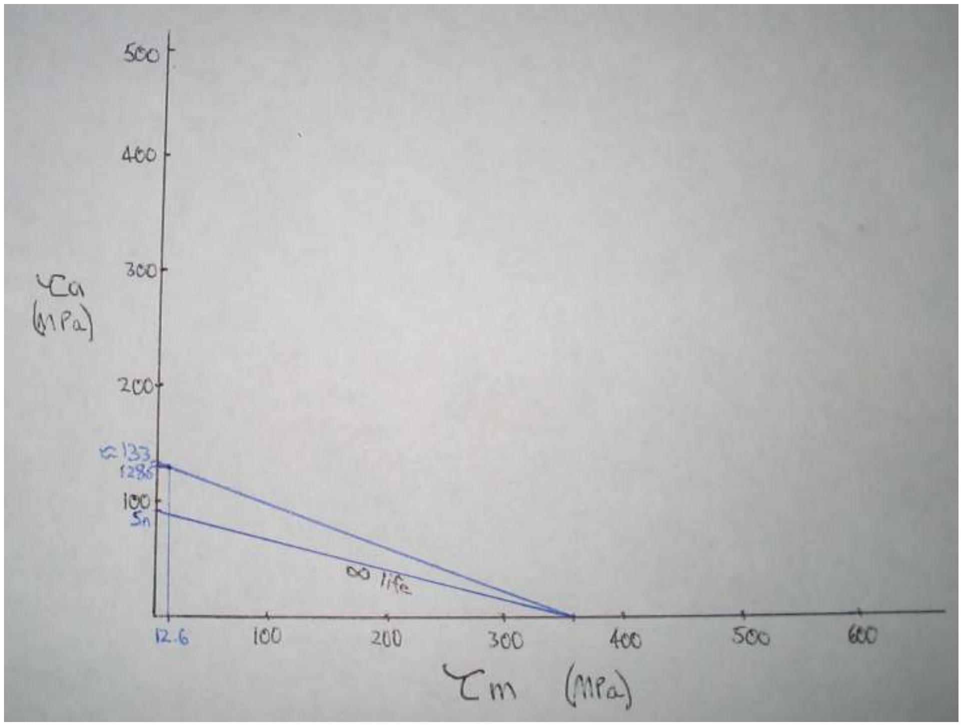


Figure 4.9: Goodman Fatigue Diagram

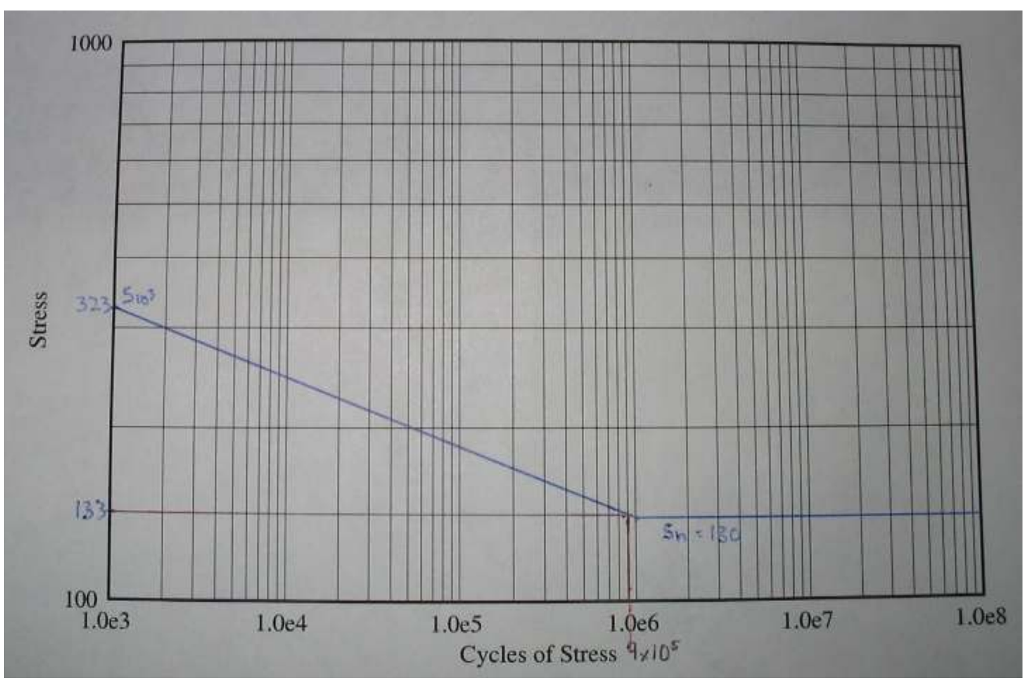


Figure 4.10: S-N Curve

4.5 Conclusions

To conclude the centre shaft design the location most likely to yield first is the bearing location made from the mild steel hollow bar. However this will not occur unless the stresses reach the theoretical maximum, which has been set as an extreme performance condition and should not occur. The fatigue and deflection analysis determined the shaft design would operate for close to infinite life. The centre shaft design was submitted and has been manufactured ready for implementation into the car. The following chapter reveals the differential designs considered for the vehicle that might replace the solid rear axle in future years.

Chapter 5

Differential Designs

5.1 Introduction

The incorporation of a differential into a Formula SAE car was discovered to be the best option when time and financial resources are available. This will reduce the difficulties of understeer or oversteer and significantly improve cornering ability. This improvement in performance should secure more competition points, particularly in the skid-pad event. The following chapter explores the possible design processes for implementing one of the following options of differentials available.

5.2 Differentials

Differentials improve the cornering ability of a vehicle because they reduce the power required when turning, and reduces the turning circle for a given steer angle. A differential will also prevent the wheels from fighting each other around a tight turn, which creates a negative yaw moment resulting in understeer. All of the mechanical differentials currently available appear to range somewhere between an open differential and a locked axle. There have been many limited slip designs that have tried to provide a compromise between low-speed turning / low drag and high torque / wheel slip operating conditions. The mechanical devices of a differential appear to be limited by the combinations of these two basic axle types. The disadvantages of both these systems, stated earlier in this document, are now clear and will therefore be used as a comparison with the following designs.

The amount of tyre slip induced for a solid rear axle is given by the ratio of the outside wheel turning radius to the inside wheel turning radius, or approximated by the ratio of the track width to corner radius.

$$V_{diff} = \frac{R_o - R_i}{R_i} = \frac{R_o}{R_i} - 1$$

Or may be approximated as:

$$V_{diff} = \frac{t}{R_{ave}}$$

where: V_{diff} = Difference in apparent road velocity, outside wheel / inside wheel

R_o = Turn radius, outside wheel

R_i = Turn radius, inside wheel

t = Track width

R_{ave} = Turn radius, measured to the centre of the axle

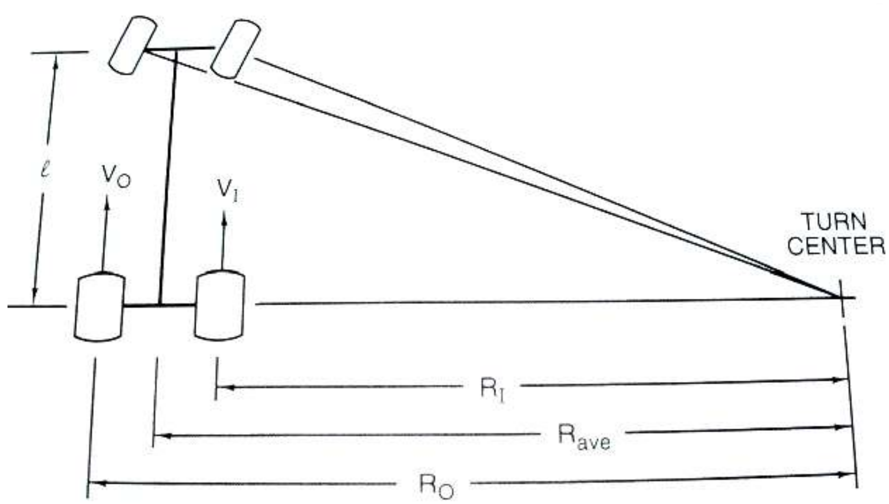


Figure 5.1: Turn radius diagram

Table 5.1 shown below lists the percentage difference in apparent road velocity of the two rear tyres as they drive around a range of different radius turns. From this data it can be clearly established that the percentage difference in road velocity or wheel rotation increases as the turn radius becomes smaller. This is logical when you consider that V_{diff} will equal zero when driving in a straight line.

Table 5.1: Wheel differentiation when cornering

Radius	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
$\mathbf{V_{diff}}$	0.433	0.325	0.260	0.217	0.186	0.163	0.144	0.130	0.118	0.108

5.3 Torsen Design

There are two versions of the Torsen differential. The original Torsen has been made available as a University Special for the Formula SAE competition and a newer version the Torsen II. The original torsen can be found in a small number of all-wheel drive vehicles such as the Audi Quattro, and is also available directly from the manufacturer as a Formula SAE special, discounted for universities. The torsen II is unavailable to be purchased from the manufacturer however can be found in a few more common vehicles such as the Toyota Rav4 and Toyota Supra.

5.3.1 University Special Torsen

The name Torsen is derived from the words 'torque sensing' can be described as a change in the operating characteristics as a function of input torque. Using high-helix angle Invex gears combined with spur gears, substitutes for the side and pinion gears used in a conventional open differential.

Torque is inputted into the differential via the housing. The small helical and spur-geared shafts will be referred to as pinions and the two larger helical gears as side gears. Two pinions are seated adjacent and parallel to each other interlocking the spur gear teeth of each, and meshing the helical gear to the side gears. There are three sets of pinion pairs spaced evenly around the side gears and are fixed to the housing by each end of the shaft. The selection of gear angles, gear surface treatments and types of bearings used together determine the torque bias ratio of the Torsen.

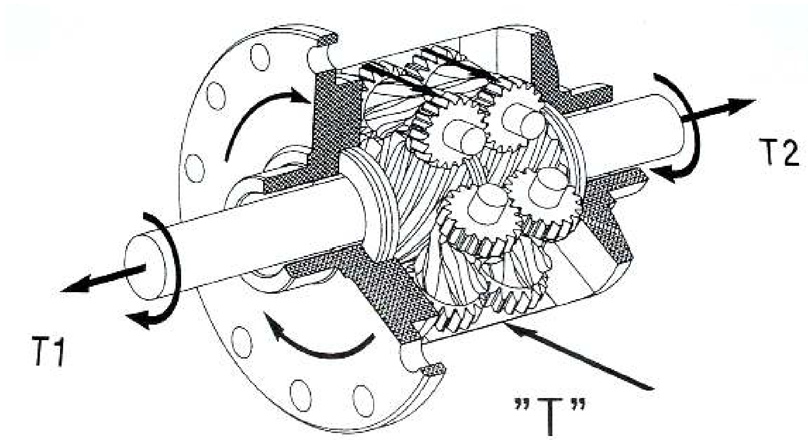


Figure 5.2: Gearing arrangement of Torsen differential

By examining the gear arrangement, it can be seen that when both axles rotate at the same speed when driving in a straight line, the pinions will not rotate and the side gears will share the torque equally to each axle. However when turning a corner, as the outside wheel rotates faster or the inside wheel slower, it forces one of the pinions in each pair to rotate the other and transfer torque across to the other side gear and axle.

Similarly due to the centripetal acceleration and weight transfer around a corner, if the inside wheel begins to break traction and spin faster, the side gear from the inside axle forces the pinion to rotate, in turn rotating the other pinion in the pair through the spur gears, and transfers more torque to the other side gear and outer axle. The torque bias ratio is what determines the amount of torque capable of being transferred from one axle to the other.

5.3.2 Torsen II

The more recent design of the Torsen II operates similarly to the original Torsen differential, however the gearing arrangement has been modified to place all of the gears on parallel shafts. This arrangement reduces friction and lowers the torque bias ratio allowing it to be tuned over a range of about 1.8:1 to 3:1.

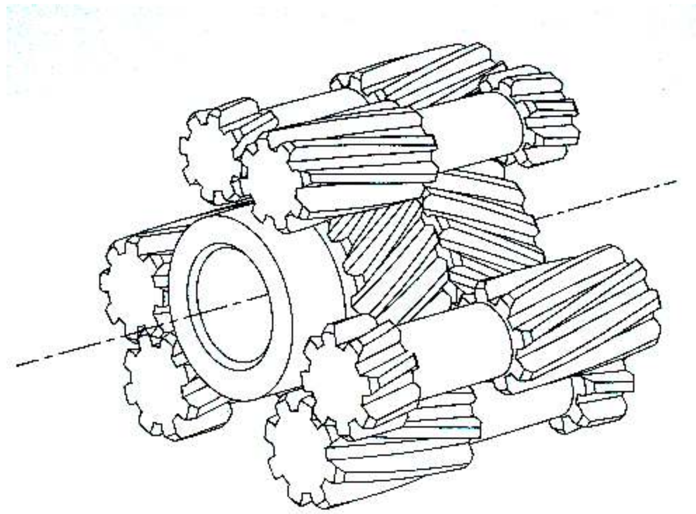


Figure 5.3: Gearing arrangement of Torsen II

5.3.3 Performance characteristics

The Torque Bias Ratio, or TBR, represents the "locking effect" of the differential. It indicates how much more torque is sent to the high traction wheel or axle, than is sent to the wheel with less traction. An example of this is illustrated in Figure 5.3.1 below, demonstrating that a differential with a TBR of 4:1 transfers 4 times the available torque that the lower traction wheel can maintain. This results in 80% of the total available torque going to the higher traction, slower spinning wheel.

Figure 5.4: Torque biasing action

The behaviour of the torque biasing ratio will improve a vehicles tractive effort on low traction surfaces such as during wet conditions. These same principles can also be adapted to a racing vehicle, when the availability of traction becomes limited due to corner speed and weight transfer. A graph displaying the tractive force capable of an open differential compared with two Torsen differentials of different torque bias ratios has been included for comparison. Assuming one wheel maintains a surface coefficient of 1, and the availability of traction to the other wheel changes, the total tractive effort capable from each differential is shown below.

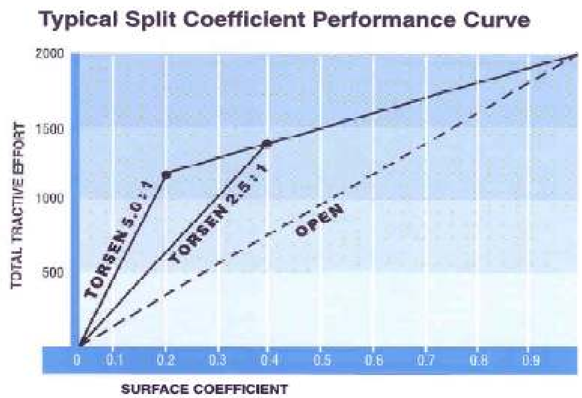


Figure 5.5: Performance curve

5.3.4 Development into drivetrain

The implementation of a Torsen differential into a Formula SAE vehicle will require careful consideration be given to many different areas. These include the input of torque into the differential, the design of a sealed housing to allow the differential to operate in an oil bath, and the mounting of support bearings to secure the differential to the chassis. The rear sprocket of a chain drive system can be attached directly, or by a flange mounted to the gear housing to input the torque into the differential. An example of an appropriate flange can be seen in Figure 5.2.

The housing for the oil bath need only be a thin, lightweight design that is capable of being sealed to contain the oil. The housing will most likely be machined from a block of aluminium, with a sidewall thickness of only about 3 to 4 millimetres. A preliminary sketch for a possible design has been included below in Figure 5.6.

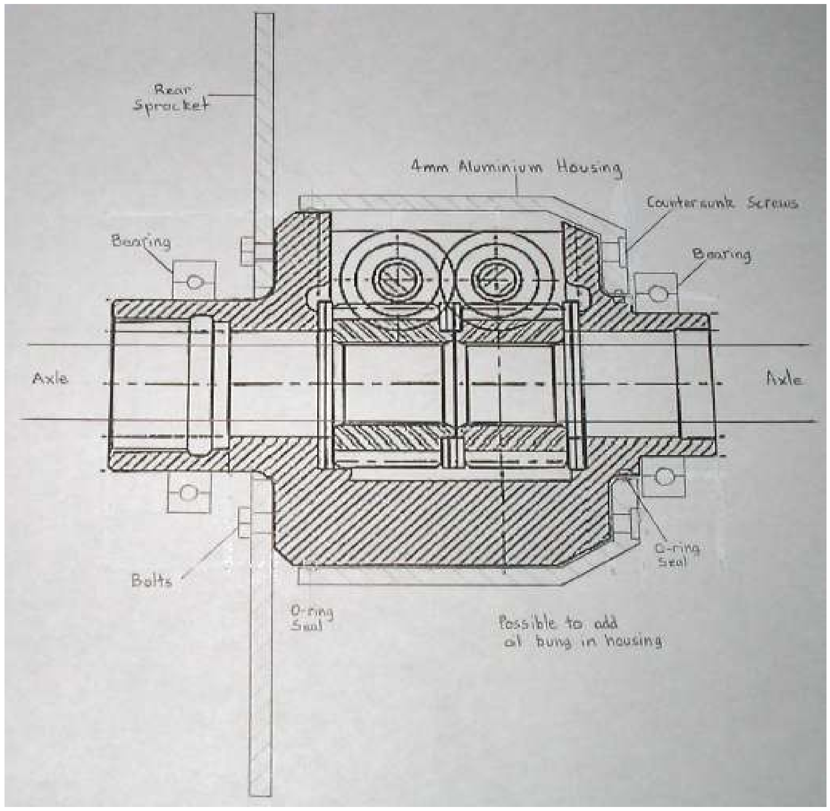


Figure 5.6: Conceptual design of Torsen housing

This design requires two grooves to be machined into the gear housing to accommodate for the o-ring seals that will keep the outer housing from leaking. Another two seals will be required to prevent oil from escaping out around the axles. The cup-shaped aluminium housing will be fitted over the gear housing from one side and secured to it by a series of bolts.

This design however requires the housing to rotate with the differential unit, which is not ideal for several reasons. To begin with the housing requires a bunghole to check and drain the oil in the differential. If this bunghole rotates as the car moves, then to check the oil the car must be either pushed forward and back to position the hole, or jacked up so the diff can be rotated to the correct position. The housing would also have to be made completely symmetrical to ensure it would be balanced when rotating, and does not cause vibration to occur. It is difficult to know whether or not this preliminary design will successfully seal the Torsen from only a drawing of the differential. If the differential was purchased, then once it arrived the effectiveness and practicality of each design idea for a housing unit and support bearings would become much more conclusive.

5.3.5 Cost Analysis

To consider the purchase of a Torsen differential a complete cost analysis of the system must be prepared. Without a final design of the system completed, the following cost analysis is purely an estimate of what resources may be required to implement the Torsen into a Formula SAE car. The estimate will be based on the preliminary design sketched above in Figure 5.6. This will also assist the evaluation of the benefits versus cost of the Torsen for comparison against other options.
All material and labour costs used are specified in the costing tables of the FSAE rules and regulations. All machine and labour times are estimated from a professional's judgement.

Table 5.2: Cost analysis of Torsen

Material

Qty.	Description	Volume	Weight	$ / Unit	Cost
1	Torsen Differential			$ 630.00	$ 630.00
1	Aluminium block – 110mm diameter x 100mm		2.66kg	$ 4.40	$ 4.40
				Subtotal	$634.40

Process Labour

Qty.	Amount	Unit	Manning	Description	$ / Unit	Cost
1	3	hrs	1	CNC Machine housing	$70.00	$210.00
1	6	holes	1	Drilling holes	$0.35	$2.10
2	0.5	hrs	1	CNC machine seal grooves	$70.00	$35.00
2	6	holes	1	Drill holes	$0.35	$4.20
2	6	holes	1	Tap holes	$0.35	$4.20
					Subtotal	$255.50
					Total	$889.90

Torsen II can be sourced from an auto-parts recycler for around $500, however will cost over $4000 for the cost report. This differential will also require a housing for the gears which will also be difficult and time consuming to design and manufacture.

5.4 Modified Limited Slip Differential

This design incorporates the behaviour of both a locked and an open differential similarly to the common limited slip differentials available. However instead of using springs to preload the clutch plates as in the majority of limited slip differentials, an actuating control arm determines the engagement and disengagement of the clutch and hence, the locking and unlocking action of the differential.

The differential consists of an open differential housing unit, containing the pinions and side gears, along with a small clutch pack operated by a diaphragm spring and thrust bearing. The differential will be driven by a chain and sprocket and so a rear sprocket flange will attach to the pinion-housing unit, where the crown gear is normally located. The clutch pack is the typical arrangement with alternate clutch plates splined to the side gear axle or housing unit respectively. When the clutch is disengaged the differential performs as an open differential with the housing driving the pinions, which drive the side gears. However when the clutch is engaged the clutch plates are pressed together and the frictional surfaces force the side gear to rotate with the housing, due to the splines. This in turn prevents the pinions from rotating and force the other side gear to rotate at the same velocity. Therefore essentially converting the open differential into a locked differential. Disengaging the clutch will release the clamping force against the clutch plates, and thus permit the side gears to rotate independent of the housing, by the pinion gear input.

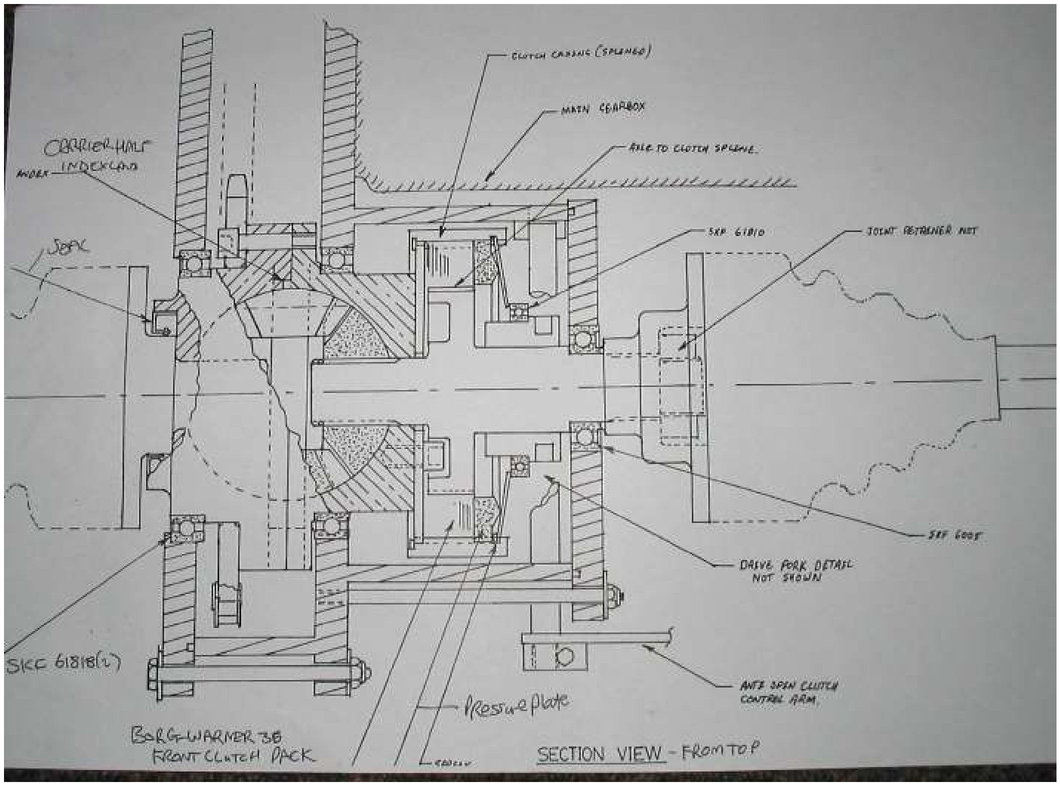


Figure 5.7: Modified limited-slip design

There are a number of different methods by which the clutch control arm may be actuated. An electric solenoid or pneumatic actuator is perhaps the best method.

5.4.1 Performance characteristics

The performance characteristics of this differential are relatively unknown. This design has been greatly modified from the more common limited slip designs for the purpose of improving the control over the locked and open differential behaviour, and fine-tuning these characteristics for better performance. However whilst it is not possible to determine the performance of this differential until it has been implemented and tested, we are able to predict the ability of the differential based on the known characteristics of a locked differential and an open differential and predicting when these actions will be engaged.

The following graphs were obtained from (fdsfdsf) and clearly show the characteristic behaviour of an open differential and a locked axle.

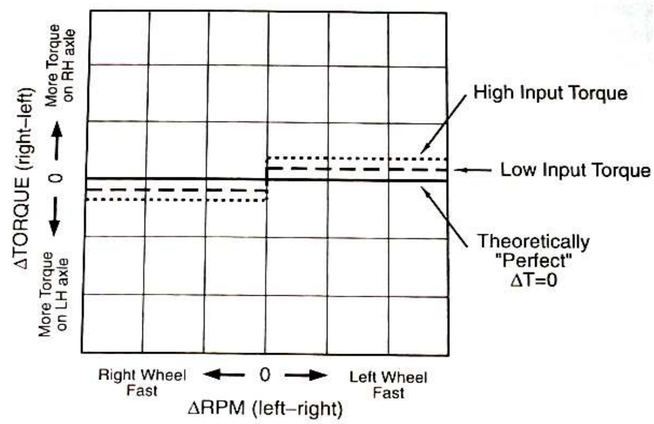


Figure 5.8: Open Differential Characteristics

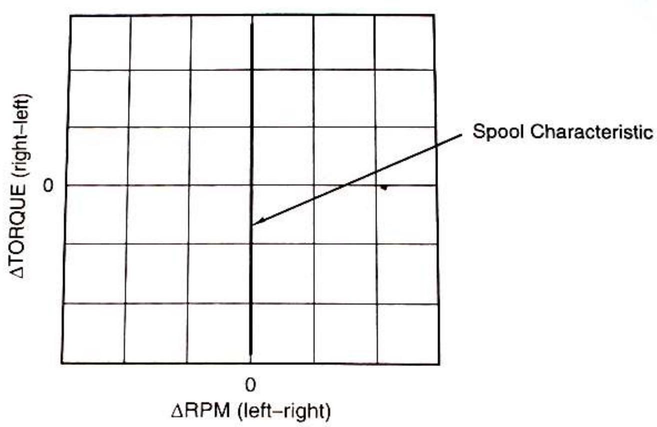


Figure 5.9: Locked Axle Characteristic

From these graphs the predicted behaviour of the modified limited slip differential can be plotted to give an idea of what can be expected in comparison.

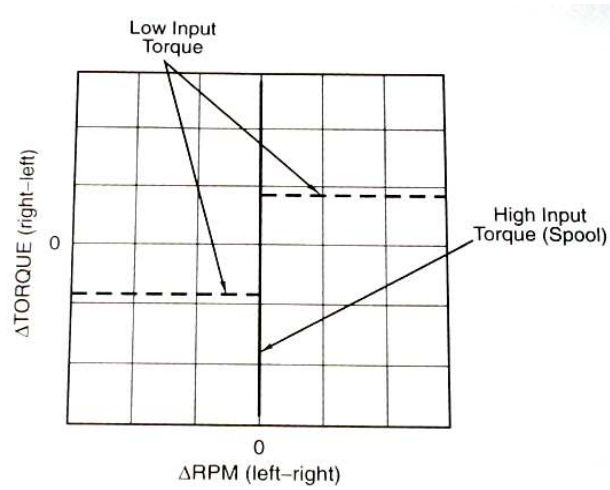


Figure 5.10: Modified Limited Slip Differential Characteristics

The success of this design to significantly improve the performance characteristics of the car, will be primarily determined by the effectiveness of the differential control arm to actuate the clutch at the most appropriate time.

5.4.2 How to develop into drivetrain

The clutch pack and required bearings are items that can be purchased new. An open differential centre complete with housing and gears will have to be sourced from an appropriate small vehicle. To modify the open differential to be chain driven a flange is required and possibly a rear sprocket may have to be made or modified to suit. The housing is required to fit all gearing and clutch assembly as well as the chain and sprocket. This housing will maintain oil and will therefore need to be sealed, including up to the gearbox casing to cover the front sprocket. Therefore a complete differential housing will need to be made and bearing and seal surfaces machined. The sketch of the initial design in Figure 5.7 shows the housing to be made from mild steel 12mm plate. Whilst this may be a cost affective design the weight is of concern, so this may be have to be reconsidered before manufacture.

The clutch is operated by a thrust bearing essentially the same as any common manual car clutch, so for the control arm to engage the clutch it must be mechanically actuated. It is obviously not practical to operate the clutch via a foot pedal; therefore this will have to be controlled electrically via the onboard computer. For the computer to actuate a mechanical lever will require some kind of electric solenoid or actuator. One method of controlling this is to install a sensor to advise the computer of a specific manifold pressure for example, which will then activate the solenoid and actuate the clutch.

To carry the torque to the wheels, drive axles will need to be made or modified to fit the CV joints, which can be located and supported directly up against the output of the differential. The right side axle is also required to be modified and splined to fit the clutch plates. The final considerations to be made are that the differential housing will need an oil bunghole to check and drain the oil. The pinion housing sourced from an open differential may also be remade in aluminium to keep weight to a minimum. It may also be plausible for an aluminium rear sprocket to be used, since the chain and sprocket drive will be submersed in oil, less wear will be expected to occur.

5.5 Conclusion

To further improve the performance characteristics of the car will require the implementation of a differential. From the predicted behaviour of the available differential designs, and the estimated time and manufacturing costs, the best option appears to be the purchase of a University Special Torsen differential. The addition of a differential should not affect the remaining components that make up the drivetrain. The following chapter explores the development of these components including the CV assembly, hub flanges and sprockets.

Chapter 6

Associated Drivetrain Components

6.1 Introduction

This chapter focuses on the design of the remaining drivetrain components including the CV assembly and chain drive. Whilst a solid rear axle has been selected for this years design, all of the following components can still be used with the implementation of a differential. The design criteria and selection of each of these components has been described in the sections below.

6.2 CV joint assembly

A CV joint assembly consists of an inner and outer CV joint that is connected by a shaft with splines either end. A Birfield joint was used as the outer joint on the drive shaft, and a Double-offset joint used as the inner joint. This is a common arrangement used in many front wheel drive cars today. There were two essential criteria used for the selection of the CV assembly. The first was that the joints must be as lightweight as possible. A typical CV joint, with the thick steel housing and solid balls, is quite heavy. The 2004 CV assembly weighed approximately 6.5 kg each side. The most significant improvement to be made from the CV assembly was to source a smaller assembly and hence reduce the weight.

The second of the selection criteria was that the inner Double-offset joint was to be driven by an internal spline. The inner joint is driven via the centre axle, which must be splined accordingly to match the existing joint. Male splines are much easier to cut and therefore less time consuming and expensive, also may not have had the facilities to cut female spline.

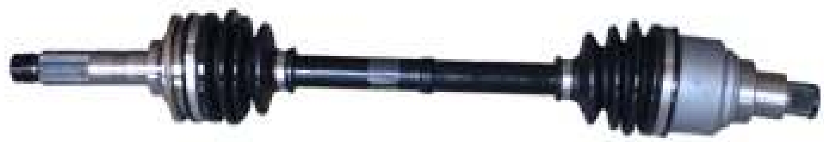


Figure 6.1: CV Assembly

After considerable investigation into small vehicles ranging from small passenger cars to ATVs, and the valuable knowledge from the professionals at Toowoomba CV joint centre, an appropriate CV assembly was found. The vehicle they were sourced from was a 1983 Suzuki SS80V. Two right hand side CV assemblies were sourced because the CV shaft on the left hand assembly was much longer. Due to the shortening modifications to be made to the CV shaft, it is of no consequence which side the CV assemblies came from. The necessary components were all sourced from a local auto parts recycler. The CV assemblies are available to be purchased new or reconditioned, however the components acquired from the auto parts recycler were quite serviceable to be used and were purchased at very low cost.

Figure 6.2: CV joints purchased

The CV assemblies were disassembled and all components cleaned up for inspection. This was to ensure all of the components were still in good condition, and to allow the splines to be viewed and measured under a profile projector. A CV assembly was also weighed for comparison and revealed to be approximately 3.2kg each, half the weight of the previously used CV assemblies. Therefore together with the new centre shaft the goal of minimising the weight of the drivetrain has been successful. Before the CV assembly can be reassembled a modification is required to the CV shaft.

6.2.1 CV shaft modification

The standard Suzuki SS80V CV shaft is required to be shortened to fit into the drivetrain of the FSAE car. There are two methods by which this can be achieved. These include cutting a middle section of the shaft out and welding the two ends back together, or cutting just one end off the shaft and reproducing the spline. Welding of the shaft is not preferred as is likely to result in failure due to the stress concentration in the already small shaft. Thus the method of cutting a section of the shaft out and joining either splined end back together cannot be used. This just leaves the method of cutting one end off the shaft to the required length and cutting a new spline on that particular end.

A profile projector was once again used to measure the profile of this 19 tooth involute spline. To accurately view the spline under the projector, a slice was cut from one end of the CV shaft. Since one end of the shaft has to be cut to shorten it to the required length, we were able to cut through one of the splines without jeopardising further use of the shaft. Caution had to be taken to ensure the edge of the metal was not burred from when the spline was cut through. Even the slightest burr when magnified under the projector would result in an inaccurate reading. The results of the spline profile are shown in Figure 6.3 and table 6.1 below. A C-clip groove was also required to be machined into the end of this spline, to ensure the shaft cannot slide out of the CV joint.

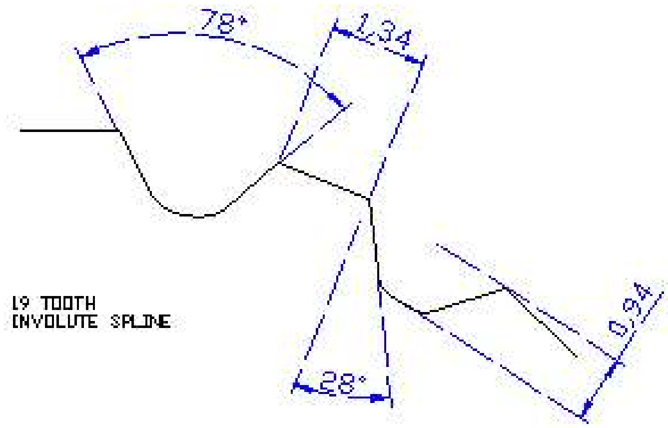


Figure 6.3: Spline profile

Table 6.1: Spline Dimensions

19 Tooth Involute Spline Dimensions	
Outer Diameter	22mm
Root Diameter	20mm
Tooth Land	1.34mm
Tooth Depth	0.94mm
Pressure Angle	30°

Analysis of the modified shaft is not necessary since the properties and section dimensions have not been altered. The shaft was initially designed to withstand greater loads in the small motor vehicle, so will be of sufficient strength to operate in a FSAE car. From research of shafts the material is most likely to be an alloy steel or a case hardened steel.

6.3 Rear hub flanges

The hub flanges are the component that the wheels are attached to and are driven by the outer CV joint. The hub flange contains an internal spline matching to the external spline on the outer CV joint and is held against the CV joint by a retaining nut. Therefore the hub flange from the Suzuki SS80V was sourced with the CV assembly to ensure the correct match could be found. The reason the hubs were selected via a match to the CV joint and not to match the wheel stud pattern, was because modifying the studs to match the stud pattern of the wheels is much easier and cost effective than attempting to machine a new internal spline to match the CV joint. After the Suzuki hub flanges were obtained the wheels stud PCD was measured and was found to be a perfect match. This meant that no modification to the hub flanges was required to be incorporated into the new drivetrain system.

By obtaining hub flanges that required no modification it has reduced valuable time and financial resources that can now be focused toward other areas of the car. To assemble the hub flanges into the vehicle, bearings are positioned on the cylindrical surface around the internal spline, which is then fitted into the rear upright assembly of the suspension. The external spline from the outer CV joint is then located into the hub flange and then secured by a retaining nut. The wheel is then simply located by the studs and bolted to the hub flange.

Figure 6.4: Outer CV and hub fitted to wheel

6.4 Gear Reduction Box

A gear reduction box was strongly considered as an option for increasing the final drive ratio of the vehicle. This option will not be implemented into the drivetrain, however the research performed will be included in this report for possible consideration in future years. By increasing the final drive ratio the top speed of the vehicle is reduced however torque and acceleration are increased. The speeds reached around the competition racing circuit are kept quite low for safety reasons, averaging at around 50 km/hr, so it is more advantageous to maximise acceleration than to have a higher top speed that may never be reached.

Another advantage to the gear reduction system would be to increase the final drive ratio high enough so the vehicle would operate mostly in 2nd, 3rd and 4th gears during competition. This would increase the torque available through each of these gears and eliminate the use of first gear. The longest and slowest gear change is from first to second gear due to having to shift over the neutral position. Therefore by starting from second gear will also eliminate the possibility of accidentally selecting neutral when accelerating out of a slow corner, thus preventing driver error and valuable lap time.

Two possible gear reduction systems are shown in the sketches below. The first system consists of 4 gears sealed in a housing, two of which are joined as a compound gear. This system allows for the rear axle to be closer toward the engine, thus shortening the wheelbase and increasing the weight distribution over the rear wheels. The disadvantages of this system are that the gears must be supported within a housing that when filled with oil can become quite heavy. Also sealing the housing against the engine and either side of the gear on the rear axle may cause difficulties. The gear ratio for this would be: $ratio = \frac{d_{g1} d_{g2}}{d_{p1} d_{p2}} = \frac{N_{g1} N_{g2}}{N_{p1} N_{p2}}$, where d is diameter and N is the number of teeth.

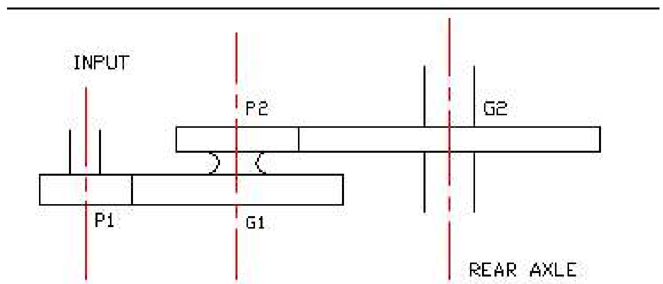


Figure 6.5: Conceptual gear train for final drive reduction

The second system incorporates a chain drive system powered by the reduction box. This system is an improvement from the initial design above as it reduces the size of housing required to decrease weight. It also eliminates the concern of sealing the housing around the rear axle.

An optimum gearing ratio was found by researching the average speeds reached at the competition by other teams, and by the results from last years USQ team. The 2004 car was noted to operate in only 2nd gear for the majority of the track. With the data on the primary ratios of each gear, and the maximum speeds likely to be reached at the competition, a table identifying the resultant speeds from increased ratios was calculated for comparison. The estimated torque available in each gear due to a change in final drive ratios was also tabulated to recognize this significant increase in torque.

Table 6.2: Gear ratio performances (Appendix D)

		Maximum Performance from Vehicle			
Gear	**Gear ratio**	**Front Sprocket (rad/s)**	**Reduction Ratio**	**Speed (km/hr)**	**Max Torque**
1st	2.846	236.9	4	59.23	943.19
2nd	1.947	346.3	4	86.58	645.26
3rd	1.55	435	4	108.75	513.69
4th	1.333	505.7	4	126.43	441.77
5th	1.192	565.5	4	141.38	395.04
6th	1.111	606.9	4	151.73	368.20

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	4.5	52.64	1061.09
2nd	1.947	346.3	4.5	76.96	725.91
3rd	1.55	435	4.5	96.67	577.90
4th	1.333	505.7	4.5	112.38	496.99
5th	1.192	565.5	4.5	125.67	444.42
6th	1.111	606.9	4.5	134.87	414.22

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	5	47.38	1178.99
2nd	1.947	346.3	5	69.26	806.57
3rd	1.55	435	5	87.00	642.11
4th	1.333	505.7	5	101.14	552.21
5th	1.192	565.5	5	113.10	493.80
6th	1.111	606.9	5	121.38	460.25

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	5.5	43.07	1296.89
2nd	1.947	346.3	5.5	62.96	887.23
3rd	1.55	435	5.5	79.09	706.32
4th	1.333	505.7	5.5	91.95	607.43
5th	1.192	565.5	5.5	102.82	543.18
6th	1.111	606.9	5.5	110.35	506.27

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	6	39.48	1414.79
2nd	1.947	346.3	6	57.72	967.88
3rd	1.55	435	6	72.50	770.53
4th	1.333	505.7	6	84.28	662.65
5th	1.192	565.5	6	94.25	592.56
6th	1.111	606.9	6	101.15	552.30

Disadvantages to a higher gearing ratio include that more gear changes would be required due to shorter gears, and higher torque may result in increased wheel spin and loss of traction. Therefore for this system to be successful driver skill must be increased, and a faster gear shifting mechanism should be implemented.

A higher final gearing ratio could also be achieved by using a smaller front sprocket, so this was chosen as a gear reduction box would have added unnecessary weight and complexity to the system. Therefore the chain and sprocket drive system was retained again this year due to its simplicity and effectiveness.

6.5 Chain and Sprocket Drive

Selection of the sprockets required the final drive ratio to be assessed and an appropriate ratio chosen. The primary focus on this was to increase the final drive ratio from the 2004 SAE vehicle. This can be done by either reducing the size of the front sprocket or increasing the size of the rear. The custom-made rear sprocket designed for last years vehicle is already quite large so increasing the size of this sprocket is not an option. Therefore this leaves the reduction of the front sprocket. The current front sprocket is the standard 15 tooth sprocket from the Yamaha motorcycle. After contacting the local Yamaha dealer it was discovered that a 13 and 14 tooth driving sprocket was available for purchase to fit the YZF600 motorcycle engine. The final drive ratios for using a 13 or 14 tooth drive sprocket with a 60 tooth driven sprocket are 4.62 and 4.29 respectively. After consideration the 13 tooth front sprocket was determined to be the optimum choice simply because it produces a higher final drive ratio, and so has been purchased.

When using two very differently sized sprockets together, particularly in close proximity, it is critically important to determine the amount of chain wrap around the smaller sprocket. This is to ensure the front sprocket has enough teeth in contact to sufficiently drive the system and operate reliably. The recommended minimum angle of wrap is 120° (hgdf) so the following calculations were used to verify our chain wrap was greater than this.

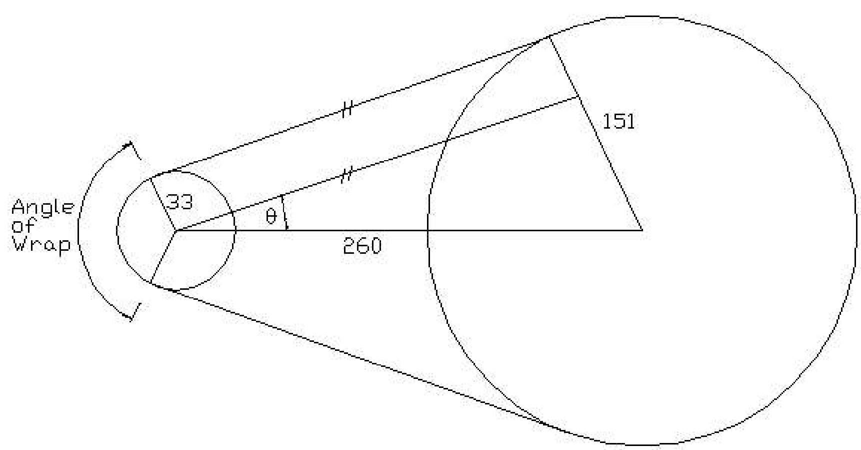


Figure 6.6: Chain wrap diagram

$$\theta = \sin^{-1}\left(\frac{(151-33)}{260}\right)$$
$$= 27^\circ$$

$$AngleofWrap = 180^\circ - 2\theta$$
$$= 180^\circ - 2\times 27^\circ$$
$$= 126^\circ > 120^\circ$$

One of the advantages of a chain drive system is that the final drive ratio can be altered and adjusted without too much difficulty by replacing one of the sprockets with the next size larger or smaller. This is advantageous in Motorsport because each racing circuit has a different set of corners and optimal speeds therefore perform better with different gear ratios. To allow for interchangeable gear ratios, a 14 tooth front sprocket as well as a 57 tooth rear sprocket were to be purchased. This results in a range complete with a 13, 14 and 15 tooth drive sprockets and 57 and 60 tooth rear sprockets for selection to be interchanged as necessary. This range of sprockets also allows for the angle of chain wrap to be increased if required.

The 530 motorcycle chain from the 2004 car will be retained for use again this year. This chain suits the 5/8 pitch sprockets and has minimal wear, so it was not necessary to source another chain. Once the rear axle is assembled into car the chain length will be determined and the required number of links will be attached. When the chain has been fixed the entire centre rear axle will be adjusted to tighten the chain.

6.6 Bearings

To support the rotating centre rear axle within the chassis requires two bearings. As discussed in a previous chapter the bearings are located either side of the sprocket and rotor flanges on the centre axle and will be fixed to the chassis. From the research conducted on bearing types, the deep groove ball bearing appears the best bearing for this particular application. This bearing was chosen because it is designed to carry radial loads, as well as being capable of supporting a small amount of axial loading. Whilst the loads on the centre axle should be purely radial, it is possible that small axial forces may act on the shaft.

A bearing housing is required in order to mount the bearings to the chassis frame. With the removal of the chain tensioner sprocket used last year, due to noise and frictional losses, the mounting of the bearing housings must be designed to allow a horizontal forward and backward movement. This will allow the entire rear axle to slide away from the engine and provide the required tension in the chain.

The bore size of the bearing to fit the centre shaft is 30 mm in diameter. Therefore by selecting the same bearing size as used in the 2004 drivetrain the bearing housings can be retained. Any cost and manufacture time that can be saved throughout this project is beneficial. The bearings are not fully housed and are exposed to the surroundings, so the bearing must be shielded either side to keep out dirt and water and allow the proper functioning of the bearing. The bearing selected for this application was a 6206-2RS, which are available for under $18 each.

6.6.1 Mounting of bearings

An additional structure member is required to be added to the chassis to support the bearing housings. The beams are located on an angle between the top and bottom chassis rails. The brackets welded to these beams will contain slotted holes to allow for the tension adjustment of the chain. The bearing housings made by the workshop last year have demonstrated to be suitable for use again this year.

6.6.2 Bearing loads

The most significant load on the bearings will be the force from the chain attempting to pull the sprocket toward the engine. The left hand bearing will carry the greatest bearing load of 3533N, as illustrated in Fig chapter 3, under maximum acceleration. The formula to find the dynamic equivalent radial load is:

$$P_r = XF_r \times YF_a$$

where: Pr = Dynamic equivalent radial load (N)

X = Radial load factor

Fr = Actual radial load (N)

Y = Axial load factor

Fa = Actual axial load (N)

However because there is no axial load, the axial load factor is zero and radial load factor is one, leaving the dynamic equivalent load factor to be 3533.37N.

From the bearing data tabulated in the SKF catalogue (rterfc), the load rating of the selected bearing is 19500N. Thus this bearing is more than adequate to support the loading of the centre axle. It is also important to consider the possibility of shock loading from the engine due to racing conditions, however the maximum load rating of the bearing is substantially greater than the predicted loading, ensuring the bearing is capable to support this as well.

6.6.3 Bearing life

The life of the bearing is an issue that also needs to be recognised. This can be found using the following formula (fdsgfdg):

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

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

C = Basic dynamic load rating

P = Equivalent dynamic bearing load

p = 3 for ball bearings

$$L_{10} = \left(\frac{19500}{3533.37}\right)^3$$
$$= 168.09$$

To estimate the life of the bearing I had to predict the average operating speed of the centre rear axle. Assuming the average speed of the car will be 50km/hr, because that is the average speed around the competition racetrack, the angular velocity of the centre axle will be 87.27rad/s. Therefore the bearing life is estimated to be:

$$L = \frac{168.09 \times 10^{6}}{\left(60\left(\frac{87.27 \times 60}{2\pi}\right)\right)}$$
$$= 3361.66 Hrs$$

As you can see due to the small load expected on the bearings and relatively low speeds the car will experience, the predicted life o the bearings will well exceed the life of the car.

6.7 Conclusion

This completes the research and selection of the drivetrain components. The goal of reducing the weight of the drivetrain has been achieved with a significant amount removed due to the new CV assembly. The CV shaft is yet to be shortened before the CV assembly is complete. The remaining components in this section have been prepared and are ready for assembly. The following chapter details the development of the wheels, tyres and braking system.

Chapter 7

Wheels, Tyres and the Braking System

7.1 Introduction

The design considerations for the wheels, tyres and braking system are all responsibilities covered in the scope of this project. The selection of the wheels and tyres are of particular importance to the performance of a vehicle, as is the braking system that must obey the many safety regulations set by the Formula SAE competition.

7.2 Wheels Selection

The most important feature of the wheels is that they must be lightweight. By decreasing the weight of the wheels, the overall weight of the vehicle and the rotational inertia both decrease and improve acceleration, braking and handling. The size of the wheels to be used is important in determining the weight and rotational inertia. By minimising the rotational inertia of the wheels, less torque will be used up trying to spin and slow the wheels and the car will become more responsive. Wheel width is also of particular importance. The width of a wheel does not affect the rotational inertia as much as wheel diameter, and the wider the tyre the larger the area of rubber contacting the road and thus more traction. A wheel can transmit more force through a tyre with a larger contact area, and because tyres tend to be a limiting factor, this means greater performance.

Magnesium wheels are commonly used in Motorsport because of the metals low density, lightweight attribute. However these wheels are very expensive to buy new, and difficult to source second-hand. Another consideration particularly relating to the front wheels is that the front braking assembly is to be located inside the rim.
The available wheel sizes considered were a 10 inch wheel and a 13 inch wheel. Price was not a major factor in this decision as tyres were available in both sizes for similar pricing, however the 10 inch wheels were going to be more difficult to find.

The 13 inch wheel was chosen because it allowed for the front brake componentry to fit inside the rim and a larger range of wheels were available in this size. Another factor was that the 2004 car used 13 inch wheels and those wheels and other componentry could be reused if required. Wheel width was not of much concern as only a small range was available for a 13 inch wheel. The wider the better so a 6 inch wide wheel was found primarily because it is the widest 13 inch wheel that can be sourced easily. Because of the advantages of using a small 10 inch wheel, the brake analysis in section 7.4.2 evaluates the possibility of using smaller front brake componentry that would fit inside a 10 inch rim.

7.3 Tyre Selection

I mentioned earlier in this document that tyres are arguably the most important component of the car. The tyres generally limit all performance characteristics of a vehicle, since all forces from the car must travel through the tyres to contact the road. Tyre selection can be divided into two variable track conditions, wet and dry.

For maximum performance in dry weather conditions a racing slick offers more traction by maximising the contact area available to the road. Racing slick tyres are also primarily made of a softer compound rubber, which is ‘stickier’ and offers a higher coefficient of friction. The compound of a tyres range from hard to soft and variably affects tyre deformation, tyre wear rate, operating temperature and traction from the coefficient of friction. A set of Hoosier 20.0 x 6.0-13 soft compound racing slicks was purchased for the car for dry weather competition.

Wet weather tyres must treaded to allow water to disperse out from under the tyre and maintain traction with the road. A wet weather tyre has not been purchased for the competition yet for we are waiting to see what the remaining resources allow.

Consideration was given to purchase a performance street tyre that would be suitable for both wet and dry conditions. This soft compound treaded tyre would eliminate the need for changing wheels when weather conditions change and would also save purchasing two sets of tyres. The reason this option was turned down was because of the amount of benefit gained from a racing slick in dry conditions. Also the treaded road tyres used in last years competition and for practice on the 2004 car are adequate enough for use if resources do not allow the purchase of a second set of tyres for wet conditions.

7.4 Brake System Design

The design of the braking system is critically important both for safety and performance. The Yamaha YZF600 motorcycle from which the engine and other components were sourced, also supplied the required brake system componentry that will be used on the SAE vehicle. The disc brake system from the motorcycle had been analysed and used on the 2004 car and had proved to be sufficient. It was intended to analyse and source a smaller, lightweight design of the disc and calliper system however due to time constraints and workload commitments, the previous system was retained.

7.4.1 Front and Rear Selection

The disc brake componentry for car was all sourced from a 1994 Yamaha YZF600 motorcycle. This motorcycle came with two large brake rotors and dual piston callipers on the front, and one smaller rotor and single piston brake calliper on the rear.

The design of the brake system on the vehicle was to use just one large brake rotor from the front of the bike, along with the single piston calliper from the rear of the bike, and mount these as the single disc brake assembly on the rear of the car. The dual piston callipers from the front of the bike will be used on the front of the car, along with two of the small rotors from the rear of the bike due to the size requirement to fit the componentry inside the front wheels. Therefore another rear brake rotor from the motorcycle had to be sourced for the front of the car.

7.4.2 Conceptual Brake Design

The braking system from the 2004 car was retained, however research was conducted into a possible brake system upgrade for the vehicle in the future, when time and resources permit.

The braking capability of the car is primarily limited by the traction of the tyres. The majority of disc brake systems have the ability to exert enough pressure on the brake rotor to lock up the wheels. Without doing an expensive redesign of a calliper and rotor, a braking system can simply be improved by decreasing its weight or allowing better heat dissipation from the rotor.

These two characteristics are the primary objectives set for the new braking system, whilst maintaining the reliability and braking capability of the system. The most obvious way to reduce the weight of a component is to reduce its size. Another major advantage to reducing the size of the disc brake componentry is the possibility of fitting the front brake assembly inside of the smaller 10 inch wheel. The other possibility of reducing weight is to redesign the components, and make them from a lighter material. This option was not considered to be viable at this point in time because of the time and expense of the task.

Therefore research into other small vehicles that use a smaller disc brake assembly was conducted to find a set of components that may be appropriate for use in a FSAE car. It was found that many quad bikes and ATV's (all terrain vehicles) were fitted with small front and rear disc brakes. Availability of these products may be the only concern with purchasing a specific component, however similarity between the different models and manufacturers should allow for an appropriate component to be sourced. Therefore a disc brake assembly from a Suzuki LT-F400 Eiger Quadrunner has been selected for analysis on the basis of its size. It consists of a 190mm diameter rotor that will fit inside a 10inch wheel, and is significantly smaller than the 245mm diameter rotors currently being used.

It will be of great advantage to continue using the same size four-piston calliper if possible. Although a smaller calliper may generally be used with the smaller rotor, a larger calliper will exert the braking force over a larger area effectively increasing the braking efficiency and decreasing stopping distances. So to avoid losing braking efficiency the YZF600 motorcycle front calliper should be retained.

If all other factors remain constant and the only change to the system is a smaller rotor, then a larger braking force must be applied by the calliper to effectively stop in the same distance. This is due to the smaller amount of surface area in contact with the brake pad during each complete revolution. To counter this a brake master cylinder with a smaller bore diameter can be used to increase the braking force exerted on the calliper with the same driver application force.

7.4.3 Braking Force Analysis

To begin the analysis of the brake system using a smaller rotor the following equation was used to determine the braking force required for the desired braking performance. Assuming the vehicle travelling at 100km/hr should be capable of stopping within a distance of 33m.

$$v^2 = v_o^2 - 2as$$

where: v = Final velocity (m/s)

v_o = Initial velocity (m/s)

a = Acceleration (m/s²)

s = distance (m)

$$a = \frac{v^2 - v_o^2}{2s}$$
$$= \frac{0 - 27.78^2}{2 \times 33}$$
$$= -11.69 m/s^2$$

The total braking force required is:

$$F = 350 \times -11.69$$
$$= -4091.5N$$

The weight transfer due to braking requires the front braking system to supply more force. Therefore assuming a braking distribution of 70:30 front to rear, the front braking force is:

$$F_F = -4091.5 \times 0.7$$
$$= 2864.05N$$

The braking torque required to decelerate a front wheel can now be calculated by the sum of moments around the wheel. The wheel diameter is 0.552m.

$$\sum M = Fd$$
$$\therefore T_B = F_F r$$
$$T_B = 1432 \times 0.276$$
$$T_B = 395.23N$$

Using equation (18.3)(Juvinall & Marshek 2000) the axial clamping force required by the calliper can be found.

$$T = \frac{2Ff\left(r_o^{\,3} - r_i^{\,3}\right)}{3\left(r_o^{\,2} - r_i^{\,2}\right)} N$$

where: T = Torque (Nm)

F = Axial clamping force (N)

f = Dynamic friction coefficient

r_o = Outside radius of rotor (m)

r_i = Inside radius of rotor (m)

N = Number of frictional surfaces

Using: r_o = 0.095m

r_i = 0.0615

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

N = 2

$$F = \frac{T_B}{\frac{2}{3} f \left(\frac{r_o^{\,3} - r_i^{\,3}}{r_o^{\,2} - r_i^{\,2}} \right) N}$$

$$= \frac{395.23}{\frac{2}{3} \times 0.3 \times \left(\frac{0.095^3 - 0.0615^3}{0.095^2 - 0.0615^2} \right) \times 2}$$

$$F = 8291.5N$$

To determine the size of the master cylinder required to apply this braking force to the calliper, assuming the same driver application force to the brake pedal, the following calculations were undertaken.

$$P = \frac{F}{A} = \text{constant}$$

where: P = Pressure (MPa)

F = Force (N)

A = Area (mm^2)

Assuming a pedal force of 546N as was the required force on the master cylinder in the 2004 vehicle:

$$\frac{F_M}{A_M} = \frac{F_C}{A_C}$$

$$A_M = \frac{F_M \times A_C}{F_C}$$

$$= \frac{546 \times 2\left(\pi \times 15^2 + \pi \times 17^2\right)}{8291.5}$$

$$= 212.67mm^2$$

$\therefore$ Master Cylinder Bore Diameter $= 16.46mm$

This is not a standard size available however 15.875mm and 17.78mm diameter master cylinders are available. The 15.875mm cylinder is the closest to size and will further reduce the required pedal force.

$$F_M = \frac{F_C \times A_M}{A_C}$$

$$= \frac{8291.5 \times \pi \times 7.9375^2}{2\left(\pi \times 15^2 + \pi \times 17^2\right)}$$

$$= 508N$$

So by sourcing 190mm rotors and using the YZF600 motorcycle front callipers the disc brake componentry will fit inside a 10inch wheel can produce the required braking performance. The only concern to note is the increased temperature at which the brakes will reach. A smaller rotor means less material to absorb and dissipate the heat. With the low speeds at the event and the low weight of the vehicles the brakes are not under extreme conditions and therefore over-heating is not likely to occur. The precautionary option may be to add a simple venting system that will redirect the flow of air over the disc brake rotor, and enhance heat dissipation.

The rear braking system need not be altered for the option of using 10inch wheels. The overall size of the componentry is not critical since it is mounted to the rear axle and not inside the wheel. Reducing the size of the rotor for weight conservation purposes is not very important either because there is just one rear brake required, and the rotor is only a few millimetres thick so will not make a significant change to the weight. If a differential was to be implemented and a single rear brake could no longer be used, then the option would be to simply use a similar system to the front with appropriately sized callipers. The front brakes endure the majority of the braking force and so the rear brakes are not as critical.

7.5 Conclusion

The selection of wheels and tyres suitable for the Formula SAE car has been completed. A 13 inch wheel was chosen primarily for the ease of installing the front brake componentry inside the rim. Slicks are considered critical in the performance of the vehicle and so Hoosier racing slicks have been purchased. A braking system to suit the use of 10 inch wheels was analysed for future reference to prospective competitors. This chapter finalises all tasks completed in the scope of my project. The following chapter contains a brief summary of the tasks completed and the work still to follow.

Chapter 8

Conclusions and Further Work

8.1 Introduction

This chapter concludes the work covered by my project and will summarise the completion of the development of the drivetrain, wheels and braking system. This project will not reach completion until the Formula SAE-A competition is held at the beginning of December, so a great deal of further work is required. In order to achieve the goals of this project the supplementary work will be discussed along with the necessary procedures to achieve successful completion.

8.2 Project Summary

The aim of this project was to design and develop a drivetrain system, including wheels, tyres and braking system, for a Formula SAE racecar. The selection criteria for the vehicle were determined by the rules and regulations established by the Formula SAE, the teams objectives for improving the performance of the 2004 vehicle, and optimising the performance to provide a competitive race car. The project aims have been achieved with the design and development of a drivetrain system, including further investigation into the implementation of a differential and brake system for future years. The selection of wheels, tyres and a braking system also satisfy the aims of the project with the assembly of the vehicle in time for the competition the only task still to complete.

All of the components designed and selected focus on weight reduction and increased performance whilst maintaining reliability and low cost. After thorough consideration of differentials the solid rear axle was retained again this year however the new design seen it made from mild steel hollow bar with alloy steel splines each end. This design is much lighter than the previous years axle and the bearing surfaces are closer to the flanges to reduce deflection. The final drive ratio was increased using the smallest front sprocket available to increase torque and acceleration. A much smaller and lighter CV assembly was sourced for the vehicle, reducing the weight by more than half.

The splines to match the CV joints had to be custom made by the workshop staff. An error on my behalf resulted in the wrong splines being cut on the centre shaft, however this was quickly rectified and has been remanufactured to the correct specifications.

The braking system is made from the disc brake components from the motorcycle. This is identical to last year's system with two front brakes mounted inside the front wheels, and only a single rear brake attached to the centre rear axle. A set of 13 inch diameter wheels was selected for use to allow the front disc brakes to fit inside the rim. Racing slicks have been purchased for dry weather conditions and the street tyres used on last year's vehicle may have to be used in wet conditions.

8.3 Further Work

Further work is required to complete the vehicle in time for the competition. The bearing supports must be mounted to be chassis to allow for the centre rear axle to be fixed in place. The CV shaft will not be shortened until the rear suspension is assembled so the length of the CV assembly is accurate. Once the CV shaft is modified the CV joints will be packed and booted ready to fit into the vehicle. The remaining drivetrain components will be fitted immediately after the centre shaft and CV assembly are in the car.

The offset surface on the wheels is to be machined so both wheels used with the wet and dry tyres are identical. After this the tyres can be fitted and hopefully assembled straight onto the car. The disc brake assembly should not require any modifications from the arrangement in the 2004 car. However the mounting of the rear brake calliper requires consideration to allow it to slide back with the brake rotor when the centre shaft is adjusted to tension the chain. This concludes the work to my specific area although many hours are required to complete the vehicle so I will need to become involved in making and assembling other components relevant to other areas.

For USQ Motorsport to advance the next team of students continuing this challenge will need to build from the research and design work achieved this year.
In order to significantly advance the drivetrain any further requires the implementation of a differential. I recommend that the University Torsen differential be purchased early next year to allow time to design the housing and implement it into the new vehicle for extensive testing. A differential should significantly improve the performance of the vehicle and secure more competition points.

The advantages of reducing the wheel size to 10 inches in diameter are not enormous, however if future USQ Motorsport teams choose to select this the braking analysis performed in chapter 7 reveals that an appropriate braking assembly will fit inside the smaller front wheels. I consider weight reduction to be the best opportunity to significantly improve the performance of this vehicle, and if able to reduce the wheel size than the tyre size and braking componentry decrease as well.

8.4 Conclusion

The aim of this project has been achieved so far, with the design and selection of the drivetrain, wheels and brakes completed. The construction of the car is well behind schedule however no design problems have been discovered thus far and we are confident the vehicle will be completed for the competition. My design objectives were completed though the success of the drivetrain development won't be clear until the car is completed and tested.

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Appendix A

Project Specification

University of Southern Queensland

Faculty of Engineering and Surveying

ENG 4111/4112 RESEARCH PROJECT

PROJECT SPECIFICATION

FOR: **Matthew HARBER**

TOPIC: Development of a Drivetrain System for a Formula SAE Race car

SUPERVISOR: Chris Snook

SPONSERSHIP: USQ Motorsport

PROJECT AIM: The aim of this project is to undertake the design and development of a drive train system for a formula SAE racecar that will optimise performance and reliability whilst minimising cost and weight.

PROGRAMME: **Issue A, 21st March 2005**

1. Review the FSAE competition rules and scope of the FSAE competition.
2. Research information relating to race car differentials and related drivetrain systems.
3. Compare and evaluate the information concerning differential performances and specifications.
4. Select an appropriate system for use in the 2005 USQ FSAE challenge.
5. Design an appropriate drive system to carry the power of the engine to the differential.

6. Evaluate the design of wheel axles, CV joints, uprights etc on the 2004 USQ car and redevelop to improve performance.
7. Liaise with team members, sponsors and Faculty workshop staff in the acquisition, manufacture and assembly of the drivetrain system.
8. Complete Design and Cost Report documentation associated with the drivetrain.

If time permits

9. Test performance characteristics of improved drivetrain design.

AGREED: ________________ (student) ____________________(Supervisor)

DATE: ____/____/______

Appendix B

Relevant Formula SAE-A Regulations

B.1 Drivetrain

3.5.1.1 Engine Limitations - The engine(s) used to power the car must be four-stroke piston engine(s) with a displacement not exceeding 610 cc per cycle. The engine can be modified within the restrictions of the rules. If more than one engine is used, the total displacement can not exceed 610 cc and the air for all engines must pass through a single air intake restrictor

3.5.1.2 Engine Inspection - The organizer will measure or tear down a substantial number of engines to confirm conformance to the rules. The initial measurement will be made externally with a measurement accuracy of one (1) percent. When installed to and coaxially with spark plug hole, the measurement tool has dimensions of 381 mm (15 inches) long and 30 mm (1.2 inches) diameter. Teams may choose to design in access space for this tool above each spark plug hole to reduce time should their vehicle be inspected.

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 drives, 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.

3.5.1.5 System Sealing - The engine and transmission must be sealed to prevent leakage. In addition, separate catch cans must be employed to retain fluids from any vents for the coolant system or the crankcase or engine lubrication system. Each can must have a volume of ten (10) percent of the fluid being contained or 0.9 liter (one U.S. quart), whichever is greater. Any crankcase or engine lubrication system vent lines routed to the intake system must be connected upstream of the intake system restrictor.

B.2 Wheels and Tyres

3.2.2.1 Wheels - The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter. Any wheel mounting system that uses a single retaining nut must incorporate a device to retain the nut and the wheel in the event that the nut loosens.

3.2.2.2 Tires
Vehicles may have two types of tires as follows:

(a) Dry Tires – The tires on the vehicle when it is presented for technical inspection are defined as its “Dry Tires”. The dry tires may be any size or type. They may be slicks or treaded.

(b) Rain Tires – Rain tires may be any size or type of treaded or grooved tire provided:

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

(ii) There is a minimum tread depth of 2.4 mms (3/32 inch). Note: Hand cutting, grooving or modification of the tires by the teams is specifically prohibited.

Within each tire set, the tire compound or size, or wheel type or size may not be changed after static judging has begun. Tire warmers are not allowed. No traction enhancers may be applied to the tires after the static judging has begun.

B.3 The Braking System

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 must have two independent hydraulic circuits such that in the case of a leak or failure at any point in the system, effective braking power is maintained on at least two wheels. Each hydraulic circuit must 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.

3.2.5.1 Brake Over Travel Switch - A brake pedal over-travel switch must be installed on the car. This switch must be installed so that in the event of brake system failure such that the brake pedal over travels, the switch will be activated which will stop the engine from running. This switch must kill the ignition and cut the power to any electrical fuel pumps. Repeated actuation of the switch must not restore power to these components. The switch must be implemented with analog components, and not through recourse to programmable logic controllers, engine control units, or similar functioning digital controllers.

3.2.5.2 Brake Light - The car must be equipped with a red brake light of at least 15 watts, or equivalent, clearly visible from the rear. If an LED brake light is used, it must be clearly visible in very bright sunlight. This light must be mounted between the wheel centerline and driver's shoulder level vertically and approximately on vehicle centerline laterally.

B.4 Cost Report

The Cost Report must:

(A) Reflect the actual, one-off, prototype vehicle brought to the event in terms of specification.

(B) List and cost every part on the prototype vehicle. This includes any equipment fitted on the vehicle at any time during the competition. The only exceptions are that, per Section 4.3.7 of the Rules, the cost of any finish, onboard fire suppression system, rain tires, or "stand-alone" data acquisition, video or radio system, does not need to be included in the Cost Report.

(C) Be based on the estimated costs of materials, fabrication and assembly of the car. They costs shall be calculated as defined in Section 4.3.6 of these rules.

(D) Be based on the actual manufacturing technique used on the prototype, e.g. cast parts on the prototype should be cost as cast, and fabricated parts as fabricated, etc.

The reported cost of the prototype vehicle will exclude R & D, tooling (e.g. jigs, moulds, patterns and dies), and capital expenditures (e.g. plant, machinery and tools). The prototype vehicle's calculated cost should not exceed $25,000.00. If the cost exceeds this outer boundary, it will be disqualified from the Cost Event and receive zero points for the event.

4.3.6 Cost Calculations - The costs to be entered into the Cost Report comprise of the component cost and the cost of assembling that component into a sub-assembly, an assembly or onto the vehicle.

4.3.6.1 Component Costs - Component costs consist of the material cost and the cost of the process(es) to manufacture the component. The time required to load, unload, machine, paint, fabricate, etc. is cost at $35/hour if it is not explicitly cost in the Operations Cost Table. The table given in Section 4.3.9 provides some guidelines to the cost of various operations. Do not include overhead, costs of expendables, capital costs, and depreciation of machine equipment. The components on a team's (prototype) car can be from several sources. They can be purchased new, purchased used, donated, "pirated" from a previous year's car, modified from an acquired part, or fabricated/manufactured from basic raw materials. The cost calculations must follow the following guidelines:

(A) Purchased New Part – Use the full retail cost, even if it was actually acquired with a discount.

(B) Purchased Used, Donated or "Pirated" Parts – Use the full retail cost of a "new" part of exactly the same specification.

(C) Modified (Purchase and Alter) Part – Use the full retail cost of the part as above, plus the cost of the modifications taken from the Operations Cost Table in Section 4.3.9 of the Rules.

(D) Fabricated/Manufactured Part – Use the Common Materials Cost Minimums Table from 4.3.9 to determine the cost of the basic material, bearing in mind that the weight of the material must include any scrap from cutting or machining. If the material is not listed in 4.3.9, a receipt for the material used must be provided. The cost of making the part will then be added using the rates from the Operations Cost Table of 4.3.9. The parts costs used must be the "full retail" cost of the part in question. Wholesale or discounted costs are not to be used, whether they be "educational", volume or for other reasons.

Note that in calculating the labor costs to modify or fabricate a part, the team should assume that the process has been refined and reflects the time it would take if the part in question was being fabricated on a regular basis. The Cost Judges recognize that the time taken to make parts for the team's actual prototype vehicle will have been far longer. Examples of cost calculations are given in Section 4.3.10 of these rules.

B.5 Dynamic Events

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.

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

5.5.4 Skid-Pad Layout - There will be two circles of 15.25 m (50.03 feet) diameter in a figure eight pattern. The circle centers will be separated by 18.25 m (59.88 feet), and a driving path 3.0 m (9.84 feet) in width will be marked with pylons and a chalk line just outside the pylons. The start/stop line is defined by the centers of the two (2) circles. A lap is defined as traveling around one (1) of the circles from the start/ stop line and returning to the start/stop line.

5.5.6 Skid-Pad Procedure - The cars will enter perpendicular to the figure eight and will take one full lap on the right circle to establish the turn. The next lap will be on the right circle and will be timed. Immediately following the second lap, the car will enter the left circle for the third lap. The fourth lap will be on the left circle and will be timed. Immediately upon finishing the fourth lap, the car will exit the track. The car will exit at the intersection moving in the same direction as entered. A driver has the option to take a second run immediately after the first.

5.6.1 Autocross Objective - The objective 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.

5.6.2 Autocross Procedure - There will be two Autocross-style heats, with each heat having a different driver. The car will be staged such that the front wheels are 2 m behind the starting line. The timer starts only after the car crosses the start line. There will be no particular order of the cars to run each heat but a driver has the option to take a second run immediately after the first. Two (2) timed laps will be run (weather and time permitting) by each driver and the best lap time will stand as the time for that heat. The organizer will determine the allowable windows for each heat and retains the right to adjust for weather or technical delays. Cars that have not run by the end of the heat will be disqualified for that heat.

5.6.3 Autocross Course Specifications & Speeds - The following specifications will suggest the maximum speeds that will be encountered on the course. Average speeds should be 40 km/hr (25 mph) to 48 km/hr (30 mph).

Straights: No longer than 60 m (200 feet) with hairpins at both ends (or) no longer than 45 m (150 feet) with wide turns on the ends.

Constant Turns: 23 m (75 feet) to 45 m (148 feet) diameter.

Hairpin Turns: Minimum of 9 m (29.5 feet) outside diameter (of the turn).

Slaloms: Cones in a straight line with 7.62 m (25 feet) to 12.19 m (40 feet) spacing.

Miscellaneous: Chicanes, multiple turns, decreasing radius turns, etc. The minimum track width will be 3.5 m (11.5 feet).

The length of each run will be approximately 0.805 km (1/2 mile) and the driver will complete a specified number of runs. The time required to complete each run will be recorded and the time of the best run will be used to determine the score.

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.

5.7.4 Endurance Course Specifications & Speeds - Course speeds can be estimated by the following course specifications. Average speed should be 48 km/hr (29.8 mph) to 57 km/hr (35.4 mph) with top speeds of approximately 105 km/hr (65.2 mph).

Straights : No longer than 77.0 m (252.6 feet) with hairpins at both ends (or) no longer than 61.0 m (200.1 feet) with wide turns on the ends. There will be passing zones at several locations.

Constant Turns : 30.0 m (98.4 feet) to 54.0 m (177.2 feet) diameter.

Hairpin Turns : Minimum of 9.0 m (29.5 feet) outside diameter (of the turn).

Slaloms: Cones in a straight line with 9.0 m (29.5 feet) to 15.0 m (49.2 feet) spacing.

Miscellaneous: Chicanes, multiple turns, decreasing radius turns, etc. The minimum track width will be 4.5 m (14.76 feet).

5.7.5 Endurance General Procedure - The event will be run as a single 22 km (13.66 mile) heat. Teams are not allowed to work on their vehicles during the heat. A driver change must be made during a three-minute period at the mid point of the heat. Wheel- to-wheel racing is prohibited. Passing another vehicle may only be done in an established passing zone or under control of a course marshal.

Appendix C

Cost Report

C.1 Cost Report

Name: **CV Assembly**

Material

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	2	CVs			$ 230.00	$ 460.00

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$ / Unit	Cost
A	2	2.3	cm	1	Cut shaft	$0.16	$0.37
	2	1.5	hrs		Machine Splines	$70.00	$105.00
						Total	$105.37

Name: **Front Sprocket**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	1	YZF600 13 tooth drive sprocket			$ 23.00	$ 23.00
					Total	$ 23.00

Name: **Chain**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	1	530 Motorcycle Chain			$56.00	$56.00
					Total	$56.00

Name: **Centre Axle**

Material

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	1	Mild Steel Hollow bar 22ID x 40OD		3.57kg	$0.66	$2.36
	1	Alloy steel 32mm diameter bar		0.74kg	$1.32	$0.98
					Subtotal	$3.34

Process Labour

Sub.	Qty.	Amount	Unit	Manning	Description	$ / Unit	Cost
A	1	30.4	cm	1	Cut material	$0.16	$4.86
	1	0.5	hrs	1	Lathe bore	$35.00	$17.50
	1	0.5	hrs	1	Lathe turning	$35.00	$17.50
	1	0.5	hrs	1	Lathe bore	$35.00	$17.50
	1	45	cm	1	Weld flanges & stubs	$0.14	$6.33
	1	2	hrs	1	Lathe turning	$35.00	$70.00
	1	2	hrs	1	Program CNC	$35.00	$70.00
	2	1.5	hrs	1	Machine splines	$70.00	$210.00
	2	6	holes	1	Drill holes	$0.35	$4.20
	1	2.25	hrs	1	Additional Labour	$35.00	$78.75
						Subtotal	$496.64
						Total	$499.98

Name: **Bearings**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	2	6206-2RS			$18.40	$36.80
					Total	$36.80

Name: **Hub Flanges**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	2	Suzuki SS80V hubs			$ 172.00	$ 244.00
					Total	$ 244.00

Name: **Tyres**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
A	4	20x6-13 Hoosier racing slicks			$233.60	$934.40
					Total	$934.40

Name: **Braking System**

Sub.	Qty.	Description	Volume	Weight	$ / Unit	Cost
	2	Front callipers			$600.00	$1200.00
	2	Front rotor			$250.00	$500.00
	1	Rear calliper			$466.00	$466.00
	1	Rear disc			$400.00	$400.00
	2	Master			$45.00	$90.00

		cylinders				
					Total	$2656.00

Appendix D

Performance Tables

D.1 Cornering Traction Force

D.1.1 Centripetal Acceleration

					RADIUS						
speed	**vel (m/s)**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20	**5.556**	10.29	7.72	6.17	5.14	4.41	3.86	3.43	3.09	2.81	2.57
30	**8.333**	23.15	17.36	13.89	11.57	9.92	8.68	7.72	6.94	6.31	5.79
40	**11.111**	41.15	30.86	24.69	20.58	17.64	15.43	13.72	12.35	11.22	10.29
50	**13.889**	64.30	48.23	38.58	32.15	27.56	24.11	21.43	19.29	17.54	16.08
60	**16.667**	92.59	69.44	55.56	46.30	39.68	34.72	30.86	27.78	25.25	23.15
70	**19.444**	126.03	94.52	75.62	63.01	54.01	47.26	42.01	37.81	34.37	31.51
80	**22.222**	164.61	123.46	98.77	82.30	70.55	61.73	54.87	49.38	44.89	41.15

D1.2 Resultant Normal Force

	3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
N3	1340	1241	1182	1142	1114	1093	1076	1063	1052	1043
	1835	1613	1479	1390	1326	1278	1241	1212	1187	1167
	2529	2132	1895	1736	1623	1538	1472	1420	1376	1340
	3420	2801	2430	2182	2005	1873	1769	1687	1619	1563
	4509	3618	3083	2727	2472	2281	2132	2014	1916	1835
	5796	4583	3855	3370	3024	2764	2562	2400	2268	2157
	7282	5697	4747	4113	3660	3321	3057	2845	2673	2529
	3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
N4	548	647	707	746	774	796	812	825	836	845
	53	276	409	499	562	610	647	677	701	721
	-640	-244	-6	152	265	350	416	469	512	548
	-1531	-912	-541	-294	-117	16	119	202	269	325
	-2621	-1729	-1195	-838	-584	-393	-244	-125	-28	53
	-3908	-2695	-1967	-1482	-1135	-875	-673	-511	-379	-269
	-5393	-3809	-2858	-2225	-1772	-1432	-1168	-957	-784	-640

When $T > \mu N$; Tyre will slip (Shaded areas show wheel slip)

Solid Rear Axle

Road Tyre		**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/	**uN3**	938	869	827	800	780	765	753	744	737	730
30 km/hr		1285	1129	1035	973	928	895	869	848	831	817
40 km/hr		1770	1493	1326	1215	1136	1077	1031	994	963	938
50 km/hr		2394	1961	1701	1527	1404	1311	1239	1181	1134	1094
60 km/hr		3156	2532	2158	1909	1730	1597	1493	1410	1341	1285
70 km/hr		4057	3208	2699	2359	2117	1935	1793	1680	1587	1510
80 km/hr		5097	3988	3323	2879	2562	2325	2140	1992	1871	1770
		3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/	**uN4**	384	453	495	522	542	557	569	578	585	592
30 km/hr		37	193	287	349	394	427	453	474	491	505
40 km/hr		-448	-171	-4	106	186	245	291	328	358	384
50 km/hr		-1072	-639	-379	-206	-82	11	83	141	188	228
60 km/hr		-1834	-1211	-836	-587	-409	-275	-171	-88	-20	37
70 km/hr		-2736	-1886	-1377	-1037	-795	-613	-471	-358	-265	-188
80 km/hr		-3775	-2666	-2001	-1557	-1240	-1003	-818	-670	-549	-448

Solid Rear Axle

Slick		**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/	**uN3**	1608	1490	1418	1371	1337	1311	1291	1276	1263	1252
30 km/hr		2202	1935	1775	1668	1591	1534	1490	1454	1425	1400
40 km/hr		3034	2559	2274	2084	1948	1846	1767	1703	1652	1608
50 km/hr		4104	3361	2915	2618	2406	2247	2123	2024	1943	1876
60 km/hr		5411	4341	3700	3272	2966	2737	2559	2416	2300	2202
70 km/hr		6956	5500	4627	4044	3628	3316	3074	2880	2721	2589
80 km/hr		8738	6837	5696	4936	4392	3985	3668	3415	3207	3034
		3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/	**uN4**	658	777	848	895	929	955	975	990	1003	1014
30 km/hr		64	331	491	598	675	732	777	812	841	866
40 km/hr		-768	-293	-8	182	318	420	499	563	615	658
50 km/hr		-1838	-1095	-649	-352	-140	19	143	242	323	390
60 km/hr		-3145	-2075	-1434	-1006	-700	-471	-293	-150	-34	64
70 km/hr		-4689	-3234	-2360	-1778	-1362	-1050	-808	-614	-455	-323
80 km/hr		-6472	-4571	-3430	-2669	-2126	-1719	-1402	-1148	-941	-768

Torsen (Bias ratio 3:1)

Road Tyre	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/uNo	938	869	827	800	780	765	753	744	737	730
30 km/hr	1285	1129	1035	973	928	895	869	848	831	817
40 km/hr	1770	1493	1326	1215	1136	1077	1031	994	963	938
50 km/hr	2394	1961	1701	1527	1404	1311	1239	1181	1134	1094
60 km/hr	3156	2532	2158	1909	1730	1597	1493	1410	1341	1285
70 km/hr	4057	3208	2699	2359	2117	1935	1793	1680	1587	1510
80 km/hr	5097	3988	3323	2879	2562	2325	2140	1992	1871	1770
	3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/uNi	384	453	495	522	542	557	569	578	585	592
30 km/hr	37	193	287	349	394	427	453	474	491	505
40 km/hr	-448	-171	-4	106	186	245	291	328	358	384
50 km/hr	-1072	-639	-379	-206	-82	11	83	141	188	228
60 km/hr	-1834	-1211	-836	-587	-409	-275	-171	-88	-20	37
70 km/hr	-2736	-1886	-1377	-1037	-795	-613	-471	-358	-265	-188
80 km/hr	-3775	-2666	-2001	-1557	-1240	-1003	-818	-670	-549	-448

Torsen (Bias ratio 3:1)

Slick	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/uNo	1608	1490	1418	1371	1337	1311	1291	1276	1263	1252
30 km/hr	2202	1935	1775	1668	1591	1534	1490	1454	1425	1400
40 km/hr	3034	2559	2274	2084	1948	1846	1767	1703	1652	1608
50 km/hr	4104	3361	2915	2618	2406	2247	2123	2024	1943	1876
60 km/hr	5411	4341	3700	3272	2966	2737	2559	2416	2300	2202
70 km/hr	6956	5500	4627	4044	3628	3316	3074	2880	2721	2589
80 km/hr	8738	6837	5696	4936	4392	3985	3668	3415	3207	3034
	3	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
20 km/uNi	658	777	848	895	929	955	975	990	1003	1014
30 km/hr	64	331	491	598	675	732	777	812	841	866
40 km/hr	-768	-293	-8	182	318	420	499	563	615	658
50 km/hr	-1838	-1095	-649	-352	-140	19	143	242	323	390
60 km/hr	-3145	-2075	-1434	-1006	-700	-471	-293	-150	-34	64
70 km/hr	-4689	-3234	-2360	-1778	-1362	-1050	-808	-614	-455	-323
80 km/hr	-6472	-4571	-3430	-2669	-2126	-1719	-1402	-1148	-941	-768

D.2 Gear Tables

Maximum Performance from Vehicle

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	4	59.23	943.19
2nd	1.947	346.3	4	86.58	645.26
3rd	1.55	435	4	108.75	513.69
4th	1.333	505.7	4	126.43	441.77
5th	1.192	565.5	4	141.38	395.04
6th	1.111	606.9	4	151.73	368.20

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	4.5	52.64	1061.09
2nd	1.947	346.3	4.5	76.96	725.91
3rd	1.55	435	4.5	96.67	577.90
4th	1.333	505.7	4.5	112.38	496.99
5th	1.192	565.5	4.5	125.67	444.42
6th	1.111	606.9	4.5	134.87	414.22

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	5	47.38	1178.99
2nd	1.947	346.3	5	69.26	806.57
3rd	1.55	435	5	87.00	642.11
4th	1.333	505.7	5	101.14	552.21
5th	1.192	565.5	5	113.10	493.80
6th	1.111	606.9	5	121.38	460.25

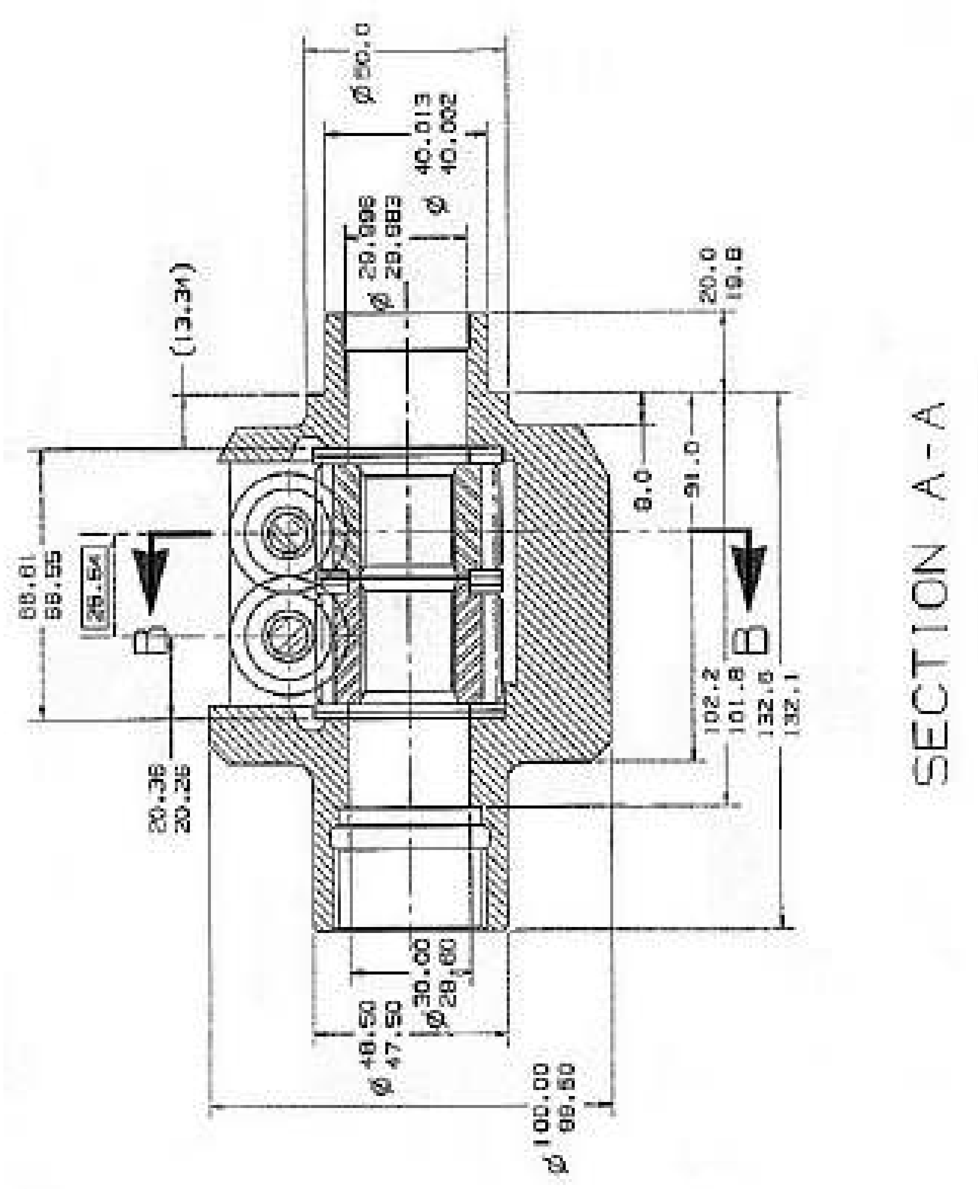
Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	5.5	43.07	1296.89
2nd	1.947	346.3	5.5	62.96	887.23
3rd	1.55	435	5.5	79.09	706.32
4th	1.333	505.7	5.5	91.95	607.43
5th	1.192	565.5	5.5	102.82	543.18
6th	1.111	606.9	5.5	110.35	506.27

Gear	Gear ratio	Front Sprocket (rad/s)	Reduction Ratio	Speed (km/hr)	Max Torque
1st	2.846	236.9	6	39.48	1414.79
2nd	1.947	346.3	6	57.72	967.88
3rd	1.55	435	6	72.50	770.53
4th	1.333	505.7	6	84.28	662.65
5th	1.192	565.5	6	94.25	592.56
6th	1.111	606.9	6	101.15	552.30

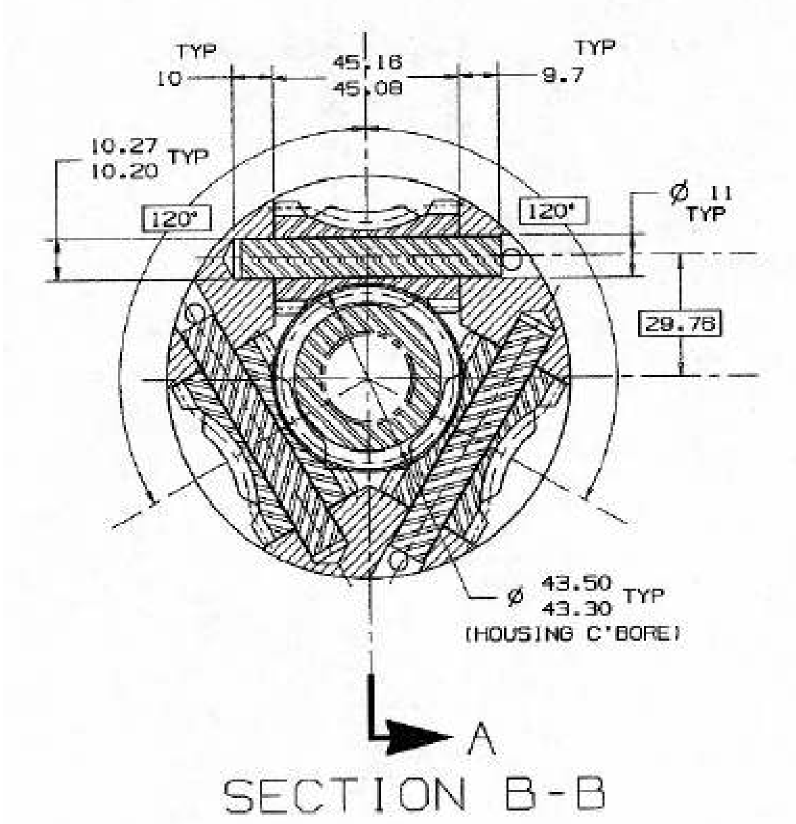
Appendix E

Detailed Drawings & Solid Model

E.1 Torsen Differential Drawings



Torsen Differential

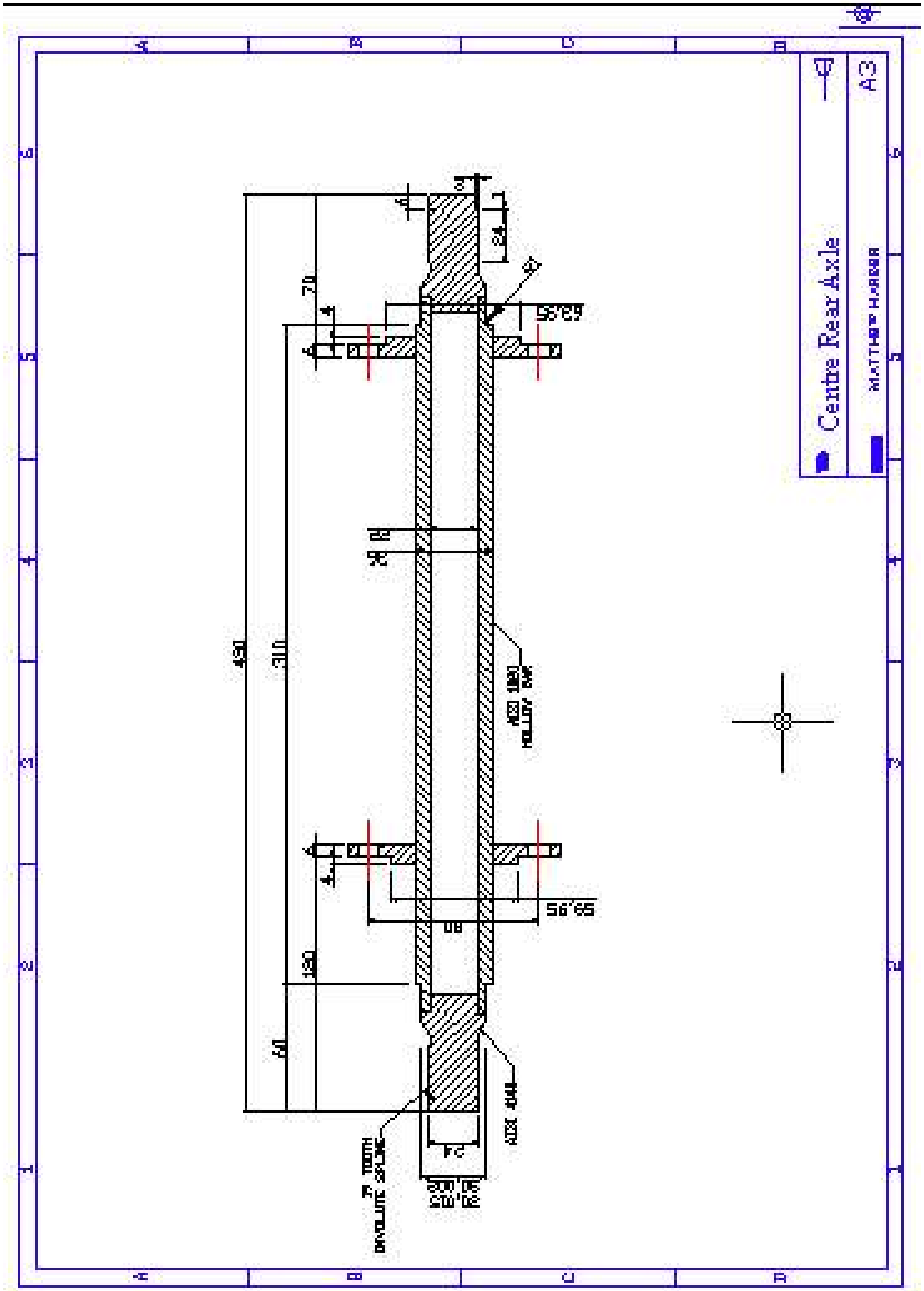


Torsen Differential Detailed Drawing

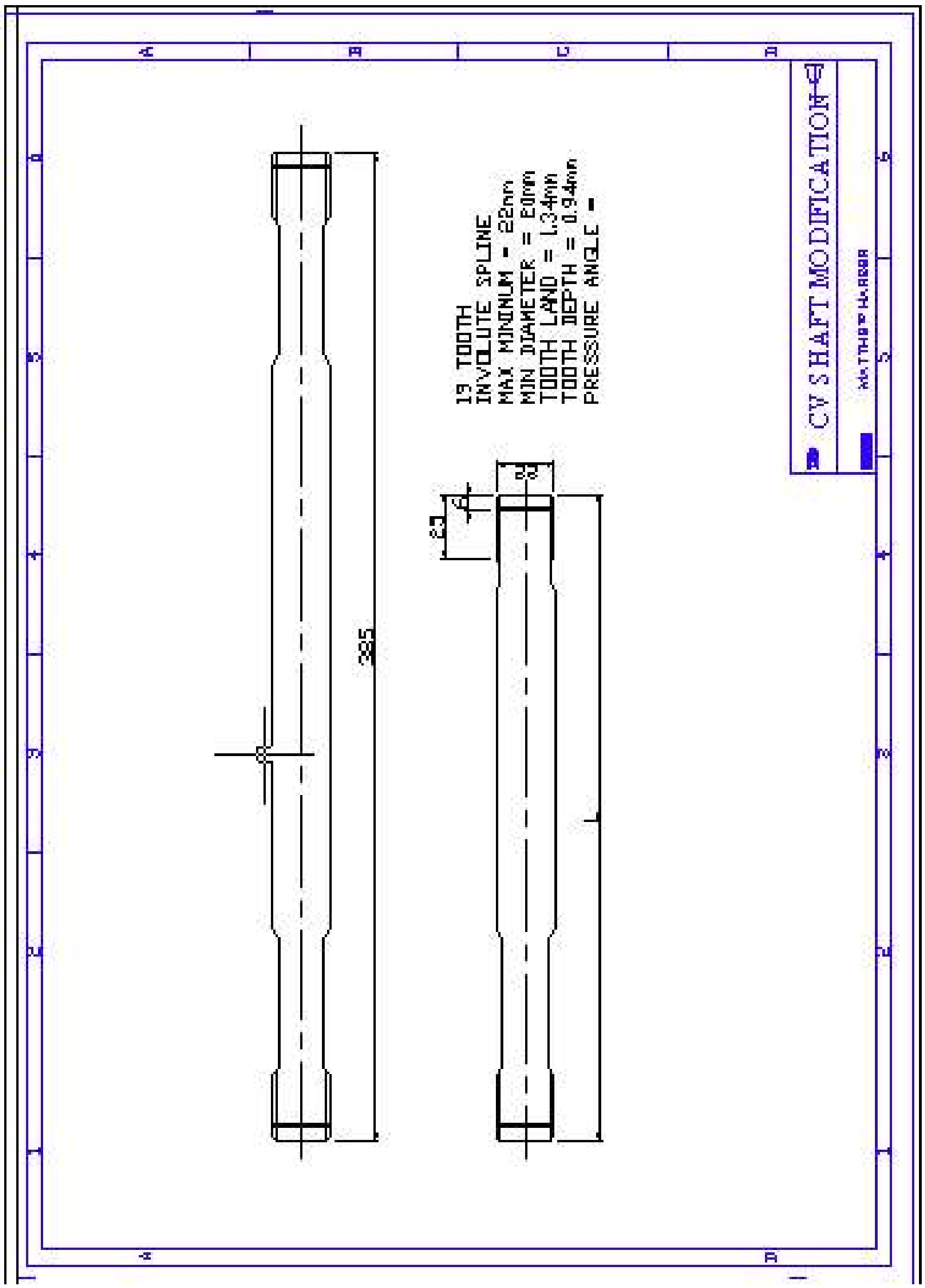
INTERNAL STRAIGHT SIDED SPLINE DATA

NUMBER OF TEETH	28
MODULE	0.79375
SPACE WIDTH ANGLE	68°
PITCH DIAMETER	(22.2250)
SPACE WIDTH EFFECTIVE MIN.	1.294
SPACE WIDTH ACTUAL	1.331-1.370
DIMENSION OVER 2 MEASURING PINS	19.696-19.753
PIN/BALL DIAMETER	1.60
MAJOR DIAMETER	23.300-23.400
FILLET RADIUS	MAX. 0.30
MINOR DIAMETER	21.600-21.700

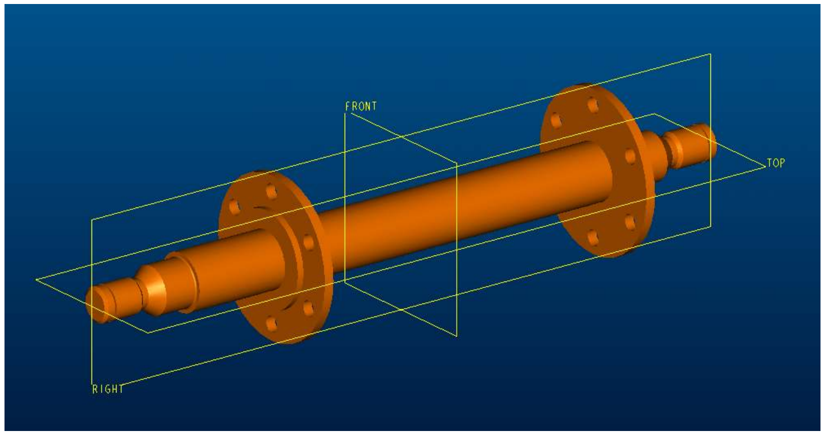
E.2 Centre Shaft Drawing



E.3 CV Shaft Modification



E.4 Solid Model



Centre Shaft in Pro Engineer