CHAPTER 1
INTRODUCTION

1.1 Background of Study

On December 2006, a team of engineering students from Universiti Teknologi Petronas (UTP) entered a competition organized by Society of Automotive Engineers (SAE) for the first time. On June 2009 , once again a team will be competing in the same event. This competition allows the students to design and manufacture a formula-style racing car and to compete against other universities in a number of events .As a member of 2007-2009 UTP-FSAE , I was responsible to develop the drive train system .To begin this project , investigation into the FSAE competition rules and regulations was required. The competition has strict guidelines on the safety features of the car and other limitations to restrict 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 drive train of this style of race car. This research will not only lead to design decision for next year car , but should also become a foundation for future students involved in developing a race car for UTP Formula SAE in future competitions. The task of designing and manufacturing requires a team of students each responsible for specific areas of the car .As a result, this project requires great teamwork and communication to be completed successfully , particularly in the time permitted. The team also responsible for fundraising and sponsorship efforts in order to gain enough finances to allow the purchase of all necessary components

1.2 Problem Statement

Drive train is defined as the mechanism design to deliver engine power and torque from the engine to the wheels .For this proposed project scope , the drive train design starts from the transmission output to the wheels .Current drive train components include driving sprocket, chain, driven sprocket , drive shaft that transfer the power to the wheels. The proposed project will look at alternatives designs to eliminate problems associated with chain drive system (reliability, power loss, ease of maintenance) and retain the same performance offered by chain drive system (high efficiency , lightweight).

1.3 Objectives and Scope of Study

The two main objectives of this study which is to design a new drive train system for a small race car to meet target performance by following the rules of competition and to perform design and analysis on the drive train system also proposing fabrication method for small scale production. The task involves are setting up the target specifications based on overall car performance requirements , design and analysis and iterations using computer aided tools , optimization of designed components to reduce mass. The final product will go through final modifications for manufacturing feasibility in limited production quantities.

CHAPTER 2
LITERATURE REVIEW AND THEORY

2.1 Background

To develop a drive train 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**

Table 1: Total Points Distribution

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 Drive Train

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

2.2.1 SAE constraints

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

· Four wheels not in a straight line
· Any transmission and drive train can be implemented
· All exposed high speed components must be guarded by scatter shields
· Only tires touching the ground

2.2.2 Transmission

A transmission or gearbox is used to vary the gearing ratios of a drive train during operation, to optimize a vehicles performance. An engine produces maximum power and torque at a specific rev range; therefore by utilizing 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 synchronizers. Synchronizers, also known as a synchromesh unit, act as a brake or clutch by making contact before the teeth to synchronize their speed through a frictional surface. This enables the teeth to be meshed easily and without noise or damage. Without synchronizers 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 drive train 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. Two components for power transfer system in the drive train are:

a) *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 tensioned 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.

b) *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 tensioned. 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 drive train 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 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 effects. Either the tire will break traction and slip or the tire will hold traction and act against the turning motion of the car creating under steer. 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 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.

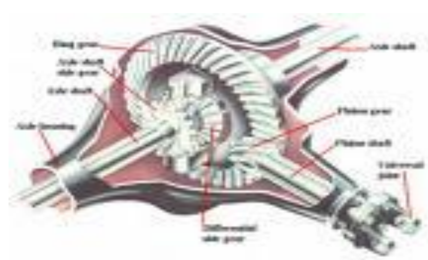


Figure 2 : Differential

2.2.6 Drive Shaft Joints

There are two commonly used joints throughout the drive train 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.

Figure 3: U-Joint

Figure 4 : CV joint

2.3 Wheels and Tyres

Tire selection is arguably the most important decision concerning the performance of a vehicle. The contact patches of the tires 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 tires. Providing your engine has sufficient power then the traction between the tires 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 tires. 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

· Tires are only to be cut or treaded by the manufacturer

· Dry tires may be slicks or treaded

· Rain tires must have a minimum tread depth of 2.4mm

2.3.2 Wheel and Tire Selection

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

Tires perform better when a larger area of rubber is in contact with the road. One way to increase this area is to use a tire 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 tires. This is why most forms of motorsport use a slick tire. Using a smooth flat tire with no tread means no void area and optimizes 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 tire maintaining the contact of rubber with the road. Without the treaded grooves the tire would ‘aquaplane’ over water lying on the road, meaning a film of water would remain between the tire 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 tire with some void area for dispersing water is required.

Tire Pressures are also extremely important when considering tire performance.Lower tire pressures will increase the contact area with the road but decreases the tire’s sidewall strength. Sidewall strength is extremely important for cornering to avoid the tire deforming. When a sidewall deforms, the tread shifts out from under the wheel and the tire begins to roll onto its side, not just losing grip but also the reaction time of steering inputs. At lower tire 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 tire 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 tire 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, minimizing deformation. Sidewall deformation robs traction, overheats tires and causes uneven wear. It is often argued that wider tires are better because they increase the area of rubber contacting the track surface. Whilst wider tires 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.

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. 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 5: Disc Brake

2.4.3 Brake Components 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
METHODOLOGY/PROJECT WORK

3.1 Drive train selection and development

The most significant part of this project was the selection and design of the drive train. To begin the process of selecting the best system for UTP-FSAE racing car , 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 drive train. The final decisions regarding the design and development of each drive train design has been summarized to show the reasoning behind the final selection for the drive train.

3.2 Centre shaft design and analysis

The design and analysis on the solid rear axle used in the 2006 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 analyze the shaft. The manufacturing process is also outlined as a record of how the design was created.

3.3 Differential Design

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 under steer or over steer and significantly improve cornering ability. This improvement in performance should secure more competition points, particularly in the skid-pad event.

3.4 Associated drive train components

This chapter focuses on the design of the remaining drive train 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.

3.5 Wheels, tire and braking system

The design considerations for the wheels, tires and braking system are all responsibilities covered in the scope of this project. The selection of the wheels and tires 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.

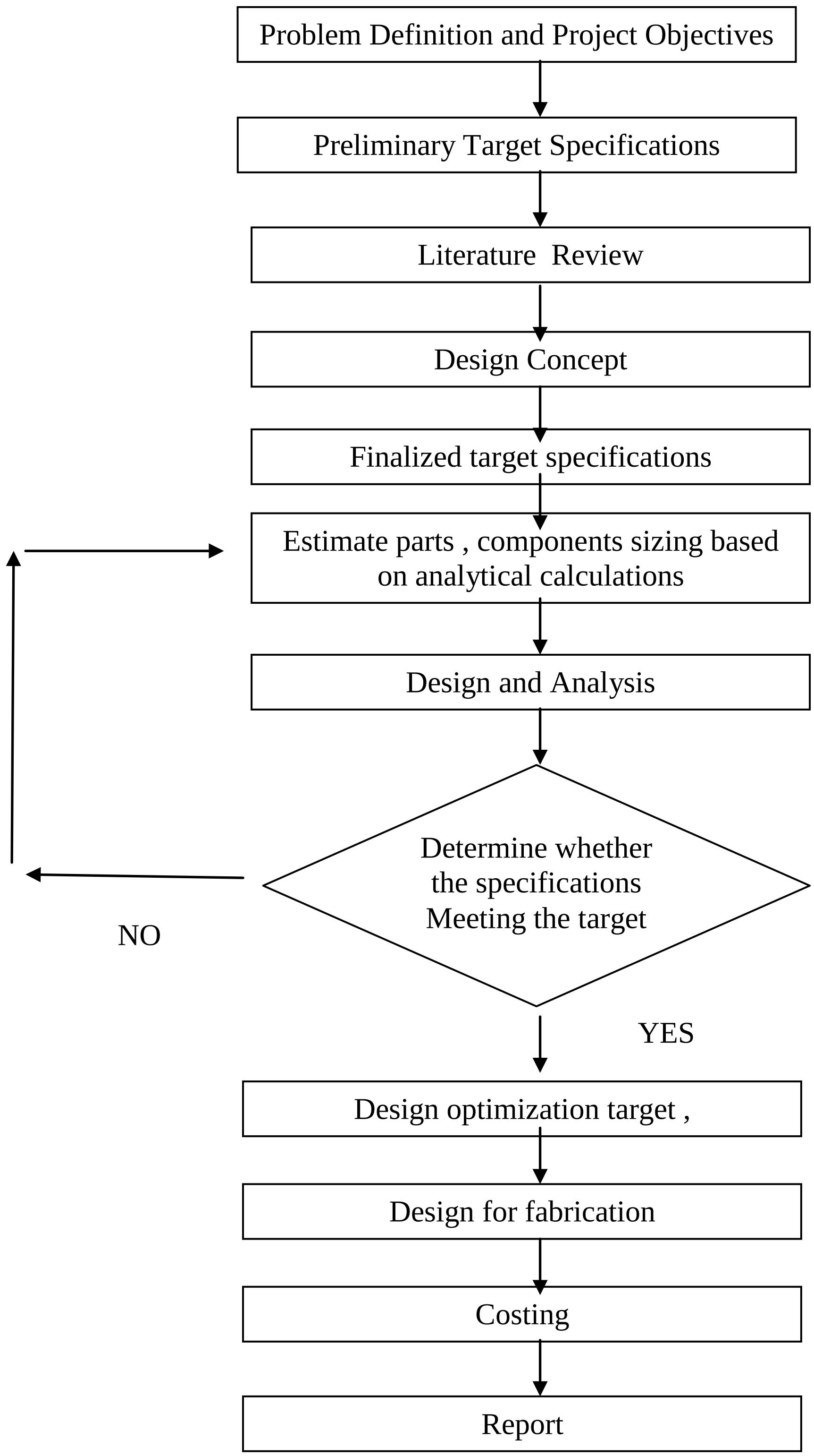


Figure 6 : Flow Chart

No	Activities	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
1	**Selection of Project Topic**																												
2	**Preliminary Work**																												
3	**Review on 2009 FSAE Regulations**																												
4	**Literature Review on Drive Train**																												
5	**Literature Review on Wheels and Tires**																												
6	**Literature Review on Braking System**																												
7	**Design Concept**																												
8	**Finalized target Specifications**																												
9	**Estimate parts**																												
10	**Analytical calculations**																												
11	**Design and Analysis**																												
12	**System Reliability Determination**																												
113	**Design optimization Target**																												
14	**Design for fabrication**																												
15	**Cost Estimation**																												

Figure 7 : Gantt Chart

CHAPTER 4
RESULT AND DISCUSSION

4.1 Drive train selection and development

4.1.1 Design criteria

Below are the criteria must be considered designing this year's drive train system car :

- The rear suspension also connects and supports some drive train components therefore the development of both these systems required close interaction and communication
- The cost of each design is another very important criteria that must be considered when selecting an appropriate design
- Maximizing weight reduction in every design

4.1.2 Design selection

Below are shown the advantages and disadvantages for all the five options of drive train system:

a) 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	· Ease of maintenance and replacement
· Cheap	· Simple design
· Light weight	. Long operating life
. Reliable	

However the disadvantages of this design are: -

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

b) 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

c) 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

d) The fourth design option was a chain drive system to a Torsen differential. The benefits to this design are: -

· Small, lightweight differential

· Increased performance characteristics

· No forced tire 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

e) The final design option considered was a chain drive system to a custom or modified limited slip differential :

The benefits to this design are: -

· Performance characteristics

· Design points

· No forced tire 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. Figure 8 shown the drawing of the Drive train system:

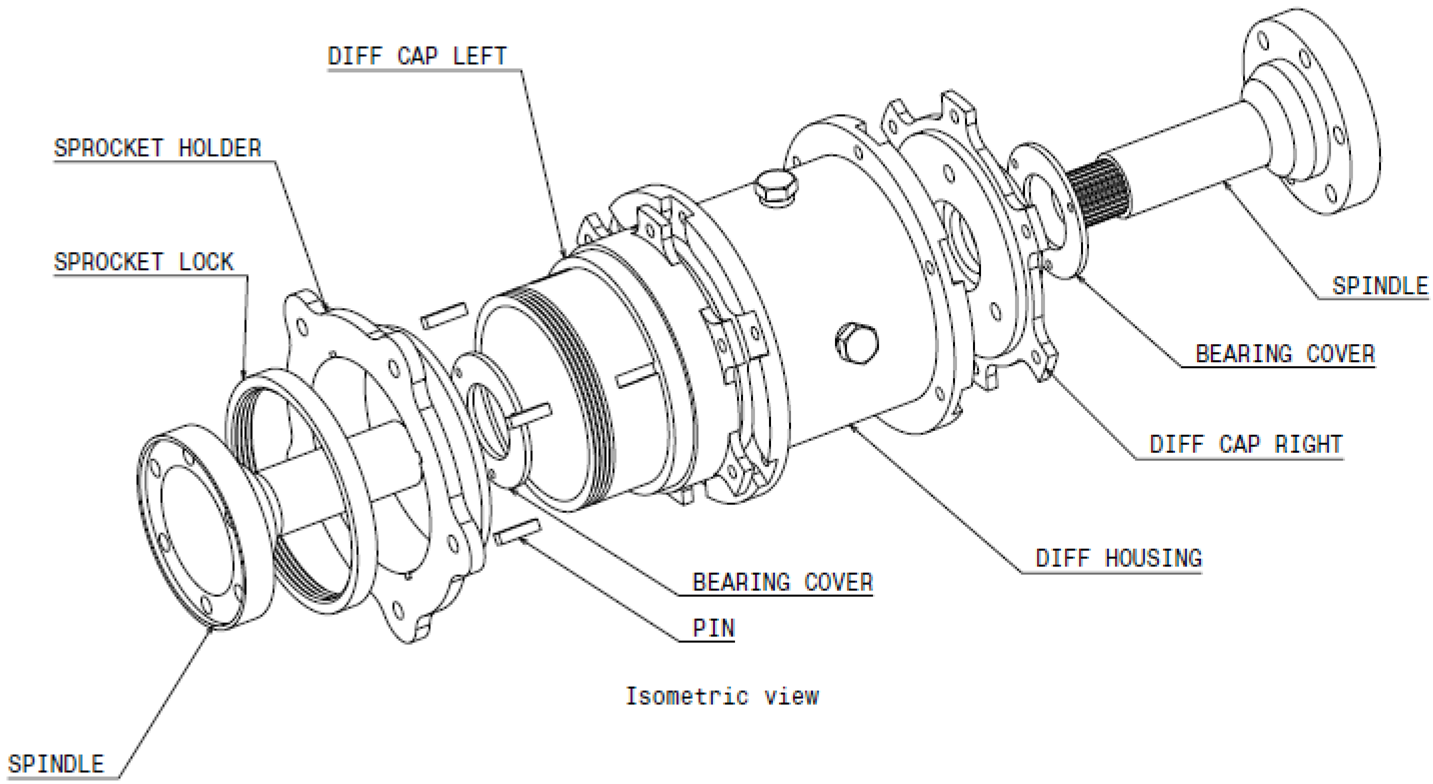


Figure 8 : 3 Dimension drawing of drive train

4.1.3 Drive train development

The development of the drive train 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 analyze 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 drive train design. The hub flanges from the Suzuki were also utilized, 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 finalized and manufactured. Details on the design and manufacture of the shaft are summarized 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 drive train and adds unnecessary component 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 caliper 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 drive train will be assembled and tested.

4.1.4 Final Gearing

The final gearing of the drive train 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 maximizing the torque and acceleration. The 2006 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 maximizing the cars potential performance. For these reasons it was decided to increase the final drive ratio of the drive train 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:
1st = 37/13 (2.846)
2nd = 37/19 (1.947)
3rd = 31/20 (1.550)
4th = 28/21 (1.333)
5th = 31/26 (1.192)
6th = 30/27 (1.111)

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.

4.1.5Axle loading

The loading of the axle must be determined to design the components of the drive train. 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 analyzing the tire friction possibly limiting the transfer of torque, and finally the predicted accelerations and cornering forces acting on the vehicle.

4.1 Centre shaft design and analysis

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 centre’s 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 finalize 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 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.

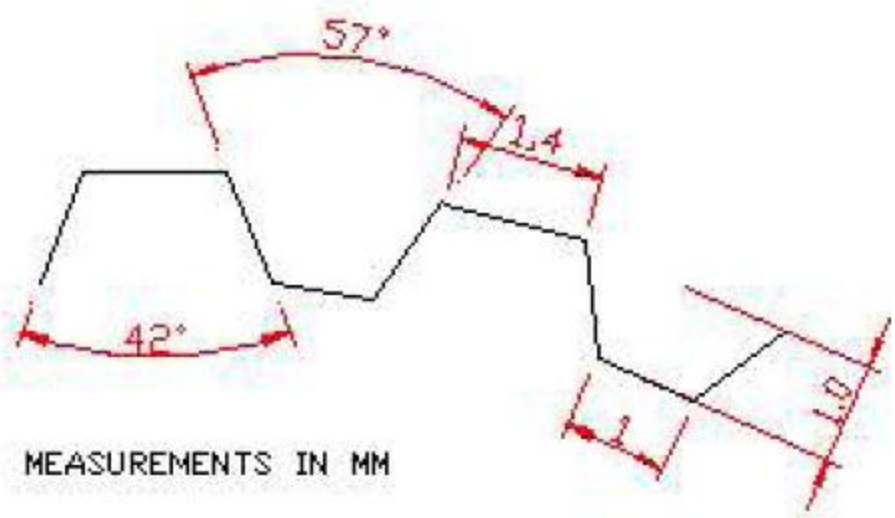


Figure 9 : Spline Profile

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

Table 2 : Spline Dimensions

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-

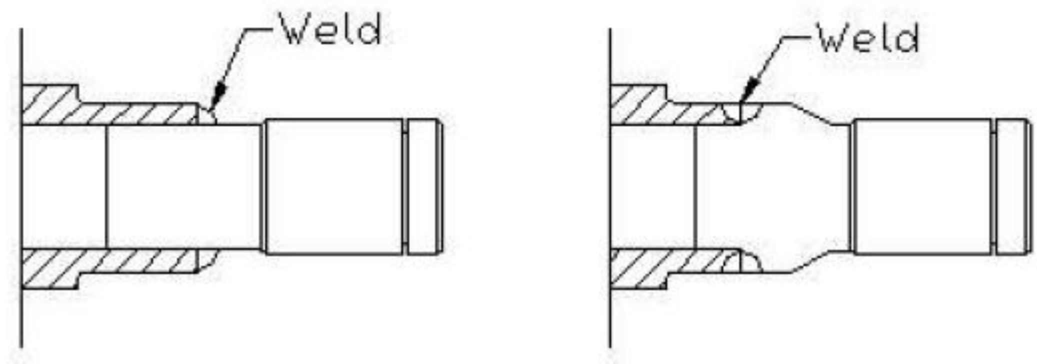
40mm O.D. However outside diameters are not of particular importance, as the shaft will be machined during manufacture.

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 analyzing 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 UTP technician, 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



Fiure 10 : 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.

4.2.6 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 tires. 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:-

$$F = T / r_p$$

$$F = 664.5/0.1517$$

$$F = 4381.37N$$

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 or hold it in place. The initial chain tension is neglected as well as any centrifugal effects of the chain.

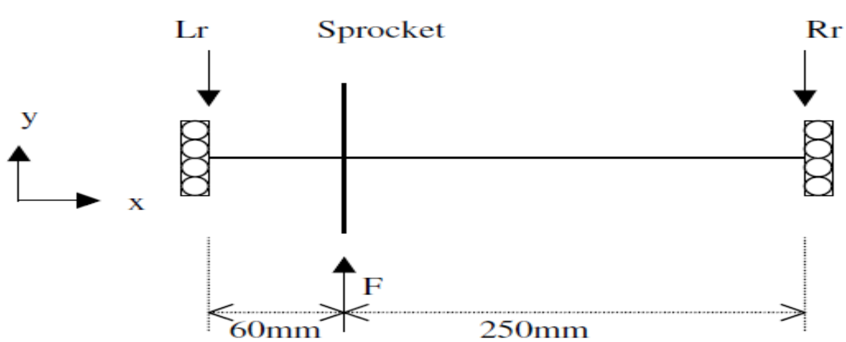


Figure 11: Free Body Diagram of Centre Shaft

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

$$\sum M_{L}=0$$
$$0=(4381.37 \text{ x } 0.06) - (Rr \text{ x } 0.31)$$
$$Rr=848N$$

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

$$\sum F_{Y}=0$$
$$F= R_r + L_r$$
$$L_r= 4381.37-848$$
$$=3533.37N$$

The shear force, bending moment and torque diagrams are illustrated below in Figure 11

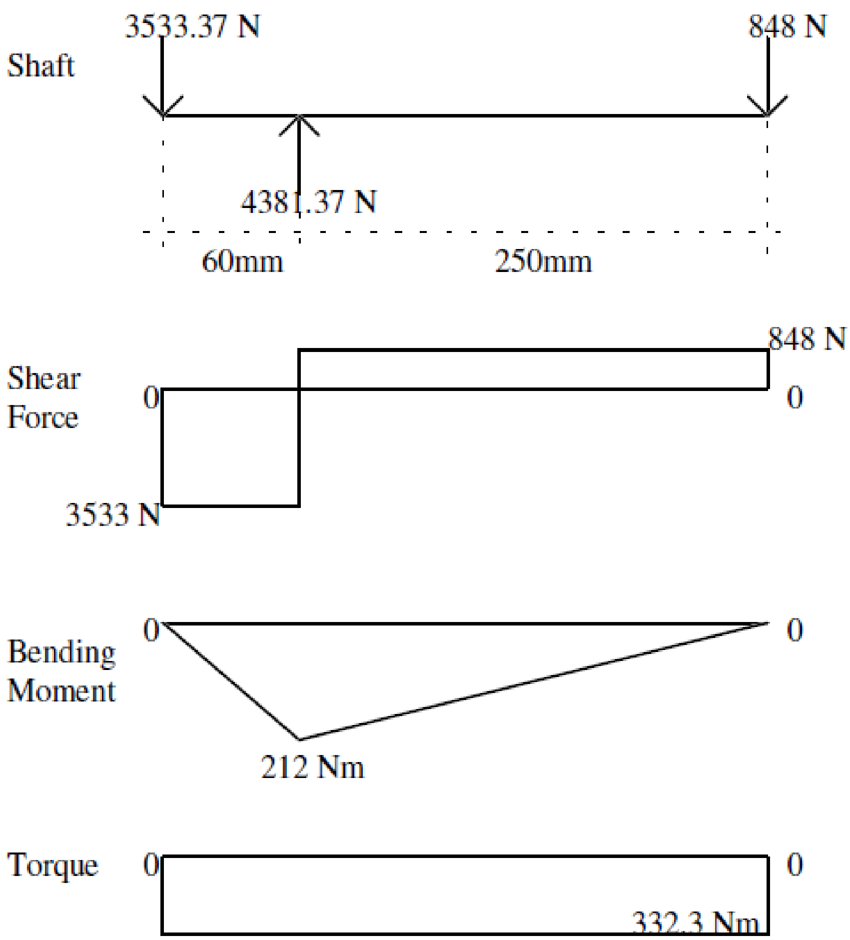


Figure 12 : Force Diagram

4.2.7 Stress Analysis

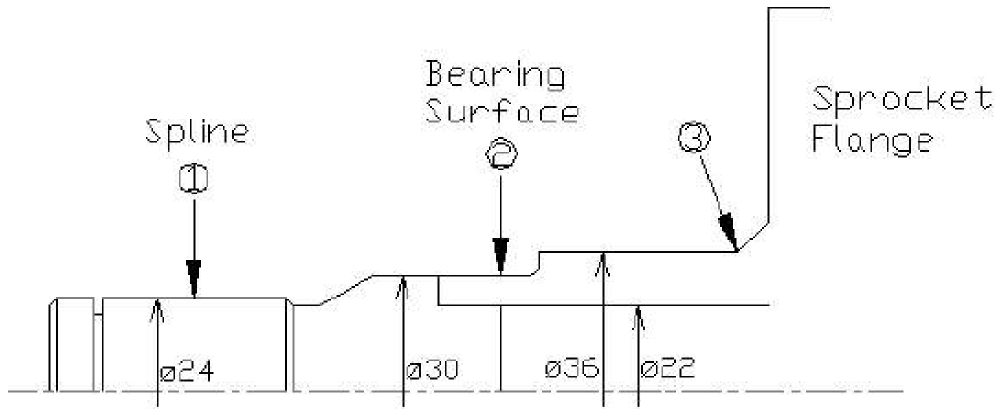


Figure 13 : Critical Location for Analysis

The first point of concern is the possible shearing of the involute spline. 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):-

$\tau = Tr/J$

where: t = 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:

$$J = (\pi d^4)/32$$

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

$$\tau = (332.3 \times 0.011) / (\pi \times 0.022^4)/32$$

$$= 158.94 \text{Mpa}$$

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):

Where σY= Yield strength (MPa)

τ_{max} = Maximum shear stress (MPa)

Therefore the equivalent tensile strength under normal loading conditions is:

$$\sigma = 158.94/0.58$$

$$= 274 \text{Mpa}$$

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

τ= (664.5 x 0.011) /[(π x 0.022^4)/32]

= 317.83Mpa

The equivalent tensile strength for extreme loading conditions is:

σ=317.83/0.58

=548Mpa

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).

$$\tau = VQ/lb \qquad \sigma = Mc/I$$

Where:

V = Shearing force (N)

Q = First moment of area (m^3)

I = Moment of inertia (m4)

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:

$Q \quad = Ay$

$= [1/2 * p(r\,o^{2} - r\,i^{2})][4(r\,o - r\,i)/3\,p]$

$=2/3(r_0^3 - r_i^3)$

$=2/3(0.015^3 - 0.011^3)$

$=1.36 \times 10^{-6} m^3$

$I = (\pi/4)(r_0^4 - r_i^4)$

$= (\pi/4)(0.015^4 - 0.011^4)$

$=0.28 \times 10^{-6} m^4$

Therefore the direct shear stress is:

$\tau_d =(3533.37 \times 1.36 \times 10^{-6})/(0.28 \times 10^{-6} \times 0.03)$

$=0.572$Mpa

Torsional shear stress is then found.

$\tau_t =Tr/J$

$=(664.5 \times 0.03)/[(\pi \times (0.03)^4/32]$

$=250.7$Mpa

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

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

$= 0.57+ 250.68$

$=251.25$Mpa

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

$$\sigma = 251.25/0.58$$
$$= 433\text{Mpa}$$

Now the combined bending and torsional stresses acting on the shaft must be considered. The bending diagram in Figure 11 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.

$$\sigma_b = Mc/I$$
$$= (212 \text{ x } 0.018) / \ [\pi/4(0.018^4 - 0.011^4)]$$
$$\sigma_b = 53.79\text{Mpa}$$

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

$$\tau_t = Tr/J$$
$$= (664.5 \text{ x } 0.018)/[\ \pi(\ 0.036^4\ 0.022^4)/32]$$
$$= \ 84.29\text{Mpa}$$

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).

$$\sigma_1 ,\ \sigma_2 = [(\sigma_x + \sigma_y)/2] \underline{+} \ [\tau_{xy}{}^2 + [(\ \sigma_x - \sigma_y)/2]^2]^{1/2}$$

$$^{=}[(53.79)/2] \underline{+} \ [84.29^2 + [(\ 53.79)/2]^2]^{1/2}$$
$$= -62.06\text{Mpa}$$

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

$$\sigma_y^{2} = \sigma_1^{2} - \sigma_1 \sigma_2 + \sigma_2^{2}$$
$$= 115.39^{2} – 115.39 \times (-62.06) + (-62.06)^{2}$$
$$\sigma_y^{2} = 155.97\text{Mpa}$$

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.

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

Table 3: Factor of safety summary

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.

CHAPTER 5
CONCLUSION AND RECOMMENDATIONS

5.1 Project Summary

The aim of this project was to design and develop a drive train system, including wheels, tires 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 2006 vehicle, and optimizing the performance to provide a competitive race car. The project aims have been achieved with the design and development of a drive train system, including further investigation into the implementation of a differential and brake system for future years. The selection of wheels, tires 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 lowcost. 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.. A set of 13 inch diameter wheels was selected for use to allow the front disc brakes to fit inside the rim.

5.2 Recommendations

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 tires are identical. After this the tires can be fitted and hopefully assembled straight onto the car. The disc brake assembly should not require any modifications from the arrangement in the 2006 car. However the mounting of the rear brake caliper requires consideration to allow it to slide back with the brake rotor when the centre shaft is adjusted to tension the chain.

For UTP-FSAE 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 drive train any further requires the implementation of a differential. I recommend that the University Torsion 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.

5.3 Conclusion

The aim of this project has been achieved so far, with the design and selection of the drive train, 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. These Project were completed though the success of the drive train development won't be clear until the car is completed and tested.

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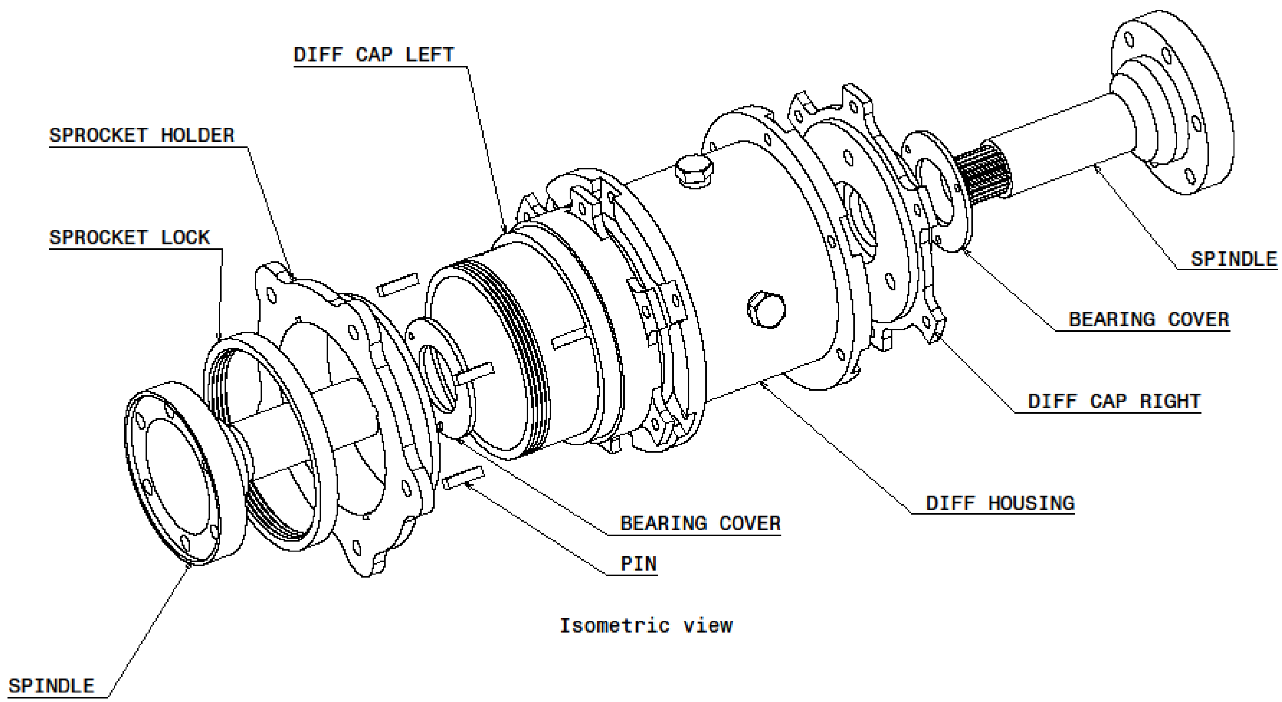
APPENDICES

APPENDIX I : Drive Train Technical Drawing

APPENDIX II : FSAE 2009 rules

APPENDIX I

Drive Train Technical Drawing



Isometric view

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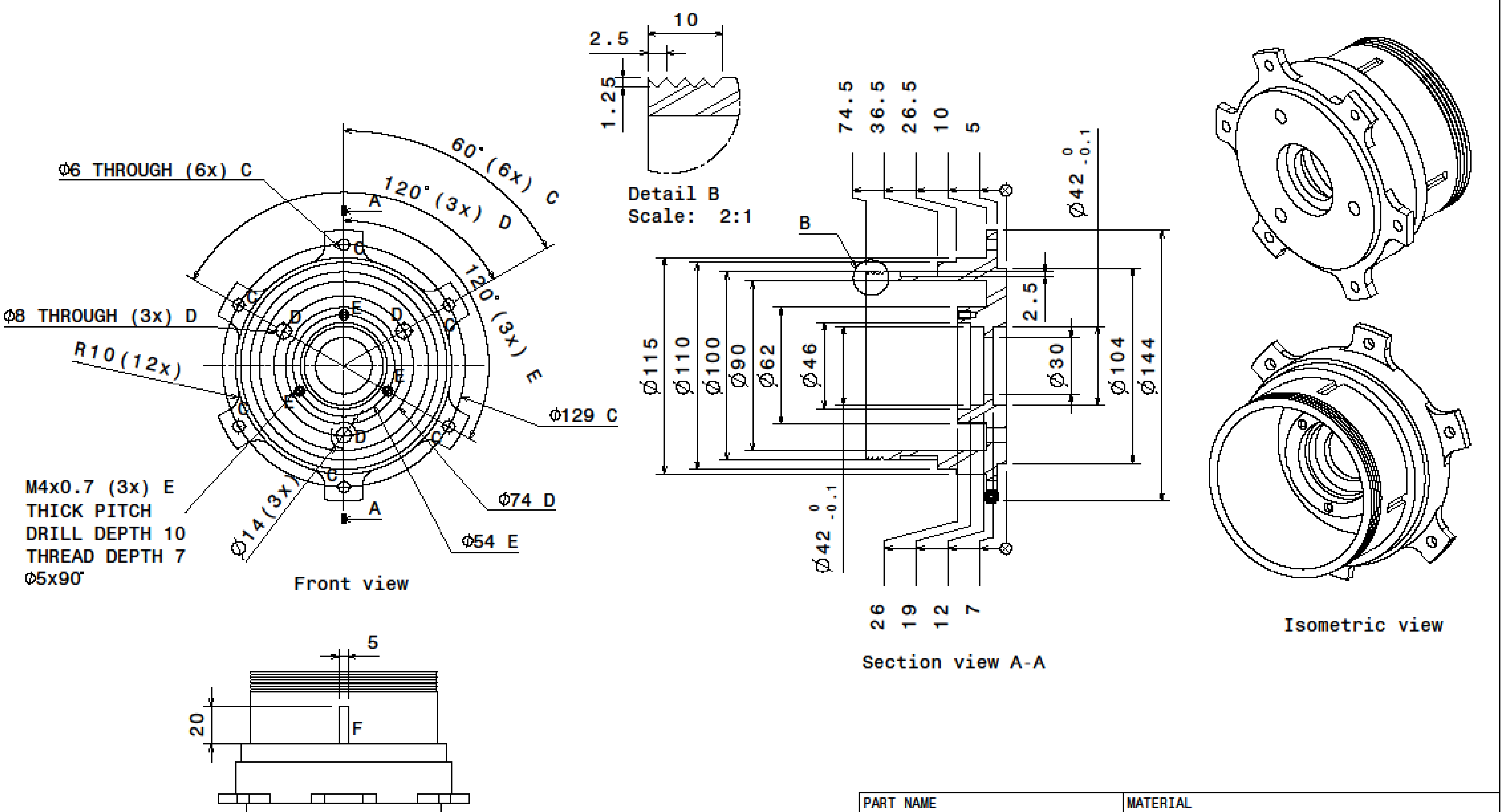
CHECKED -	DATE -	UiTM FORMULA SAE		
APPROVED -	DATE -			



Bottom view

NOTES:
1. UNLESS OTHERWISE STATED:-
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   - TOLERANCE ON LINEAR DIMENSION ±0.25
2. THERE ARE FOUR (4) FEATURES F AND 90° APART
3. REMOVE ALL BURSS AND SHARP EDGES

PART NAME: SF02-DT-CAP LEFT, DIFF HOUSING		MATERIAL: ALUMINIUM 7075		
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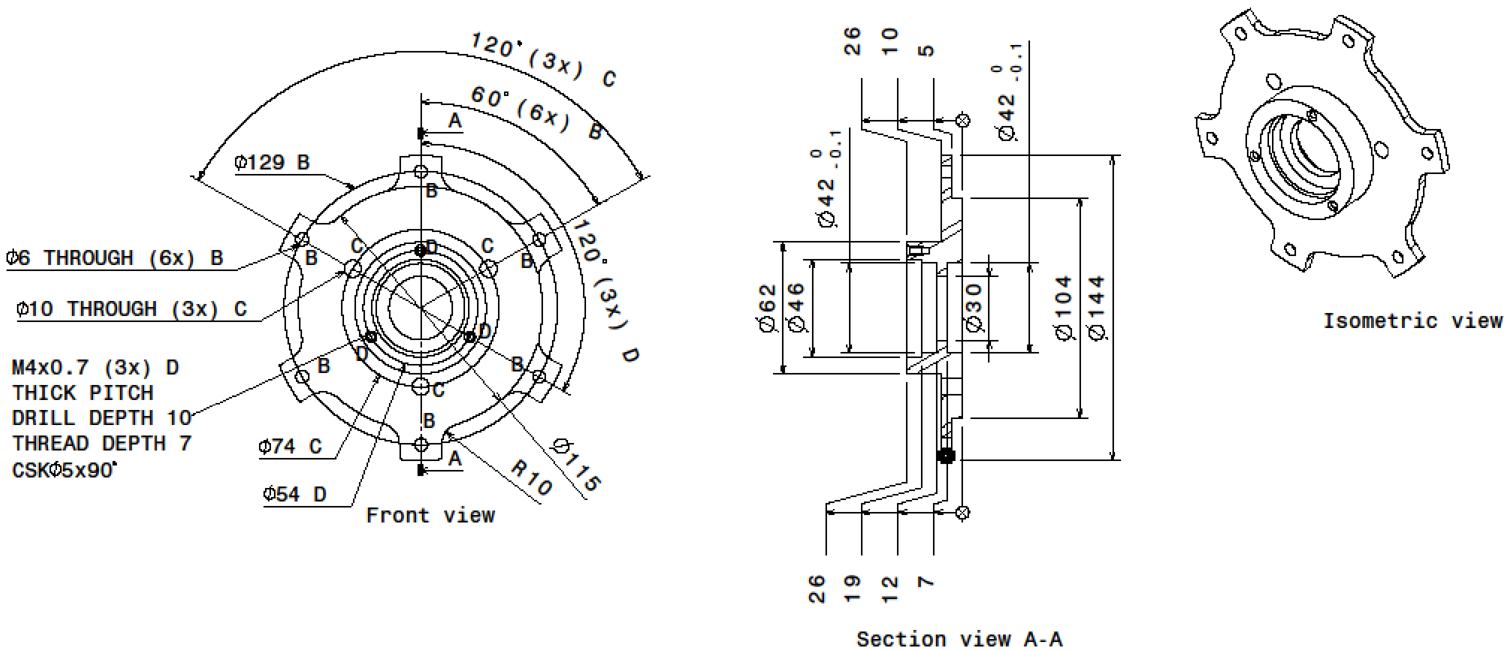

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   - TOLERANCE ON LINEAR DIMENSION ±0.25
2. REMOVE ALL BURSS AND SHARP EDGES

PART NAME SF02-DT-CAP RIGHT, DIFF HOUSING		MATERIAL ALUMINIUM 7075		
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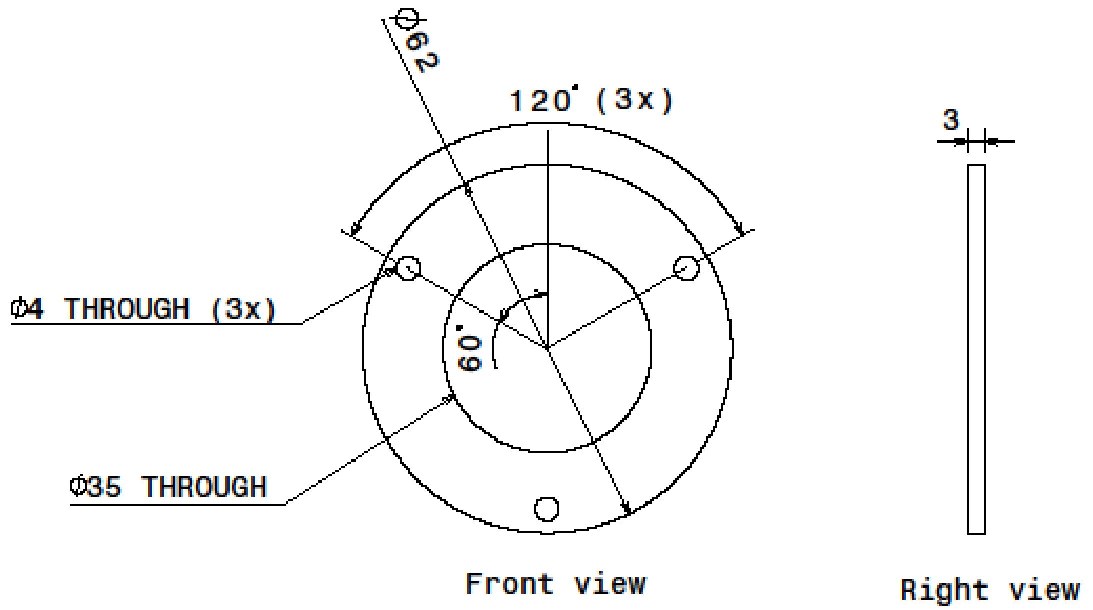
APPROVED -	DATE -			



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PART NAME SF02-DT-COVER, BEARING		MATERIAL ALUMINIUM 7075		
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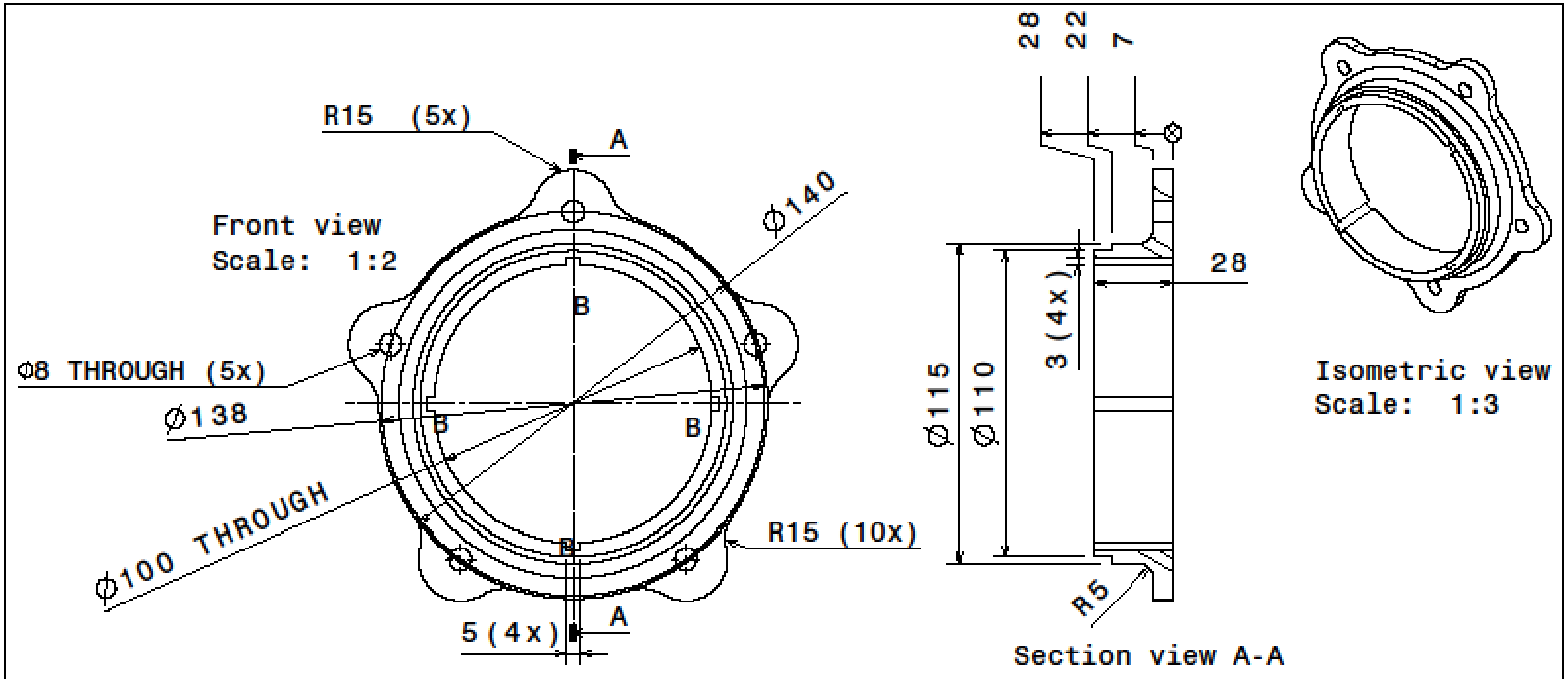

NOTES:-
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2. FEATURES B ARE 90° APART
3. REMOVE ALL BURSS AND SHARP EDGES

PART NAME SF02-DT-HOLDER, SPROCKET		MATERIAL ALUMINIUM 7075		
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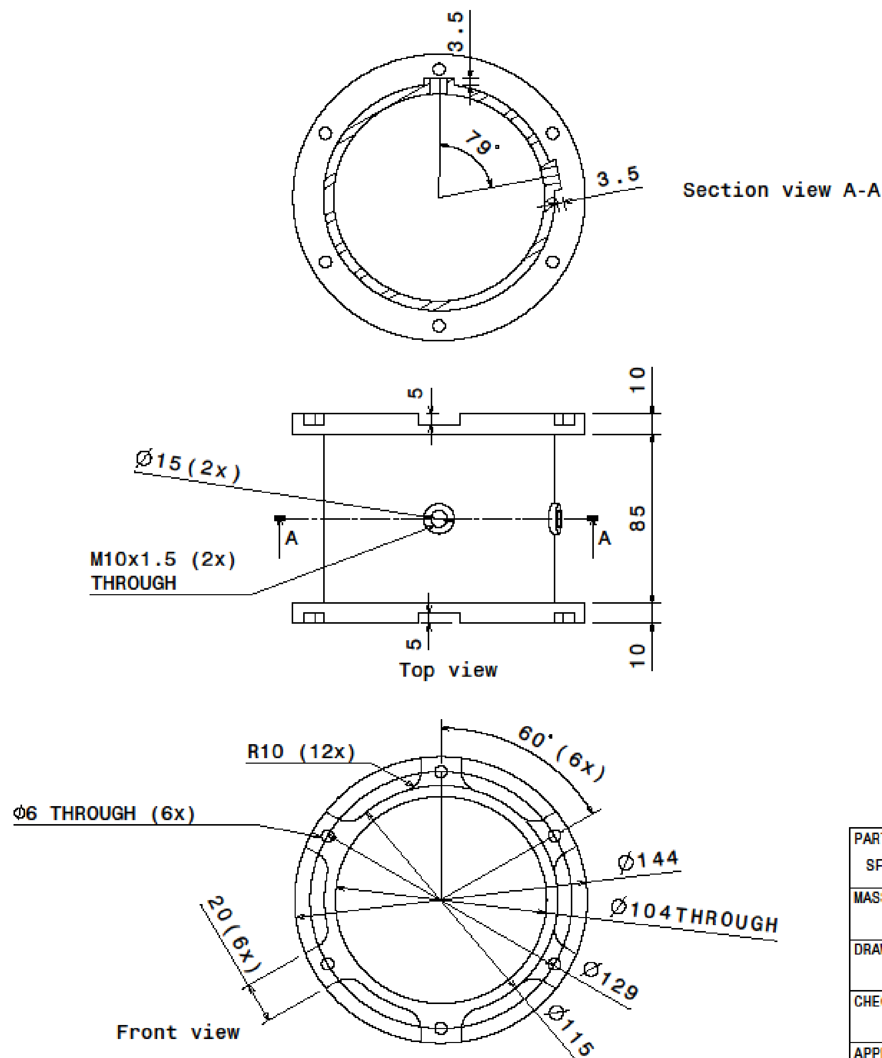
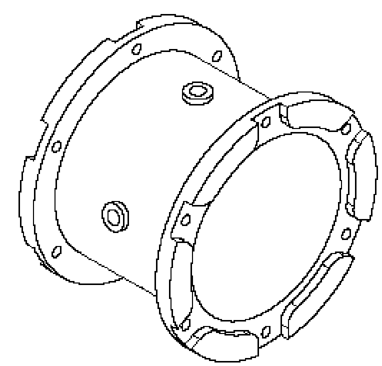
CHECKED -	DATE -			
APPROVED -	DATE -			





Isometric view

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2. FRONT VIEW AND REAR VIEW FEATURES ARE SAME
3. REMOVE ALL BURSS AND SHARP EDGES

PART NAME SF02-DT-HOUSING, DIFFERENTIAL		MATERIAL ALUMINIUM 7075		
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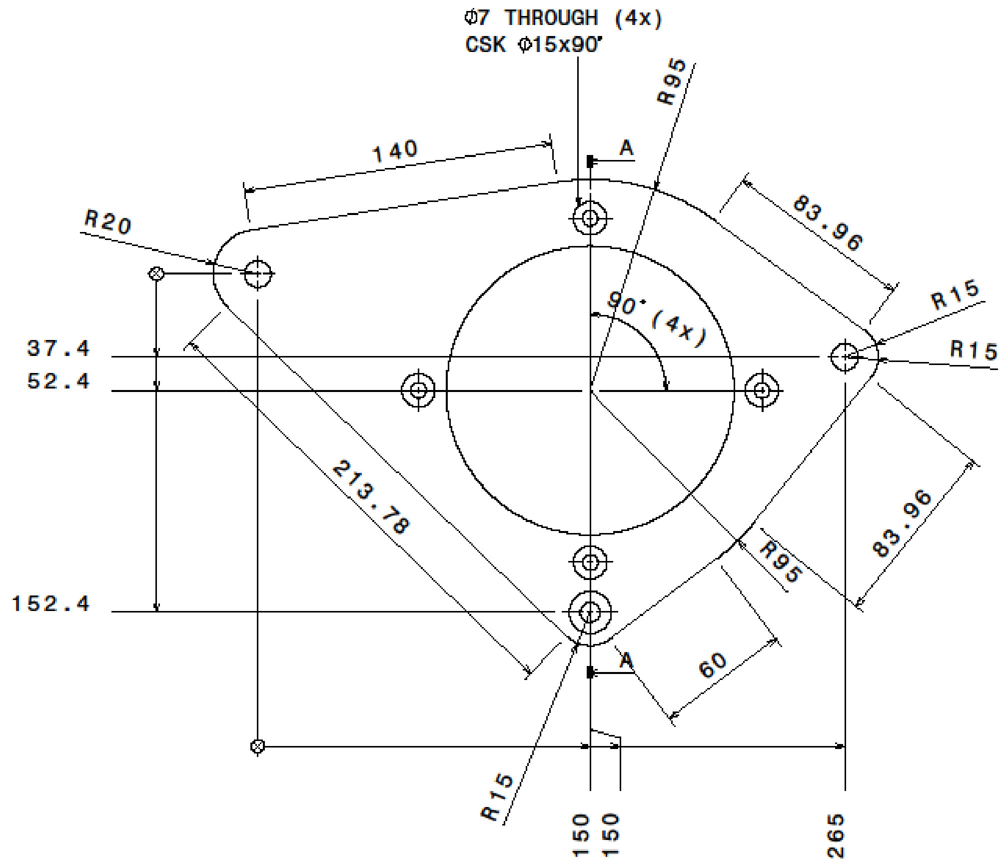
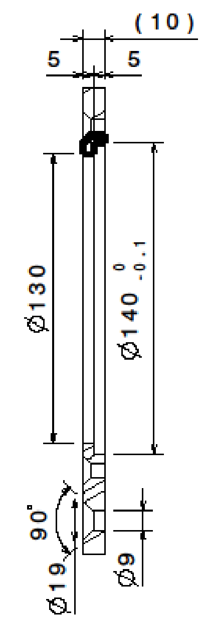


Front view



Section view A-A

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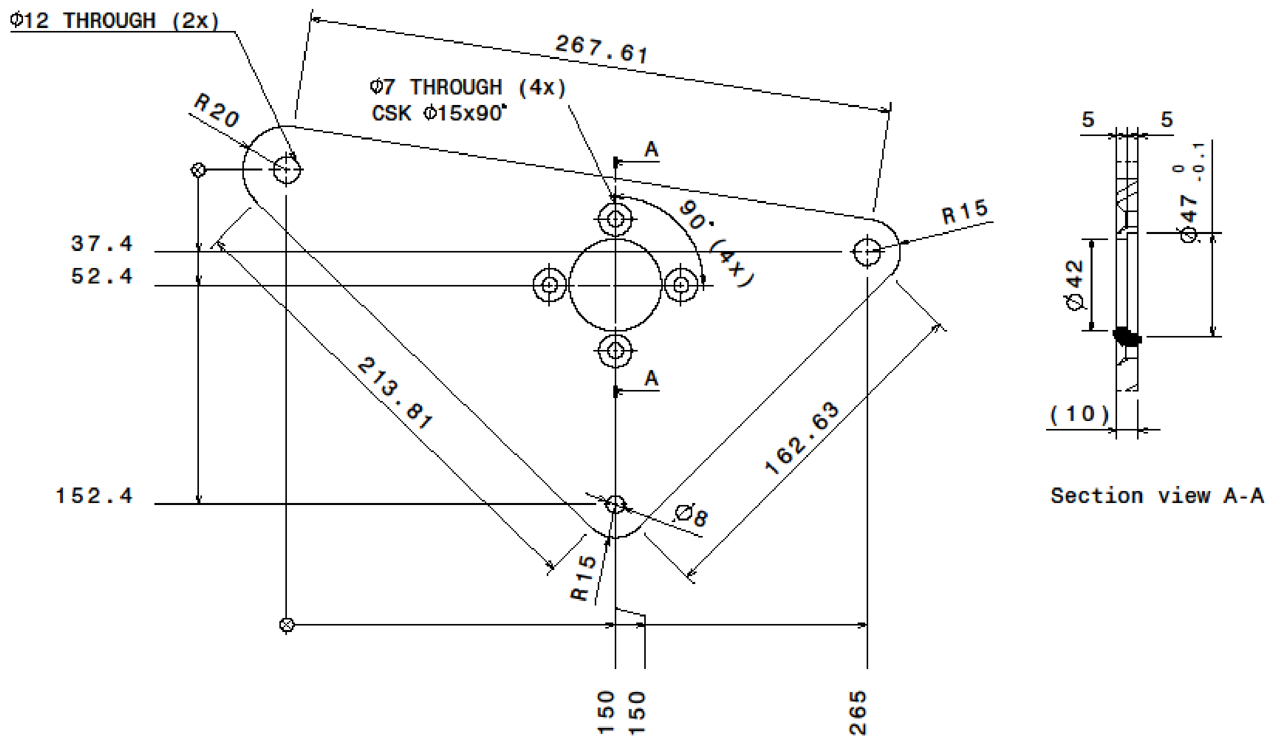

Front view

Section view A-A

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CHECKED -	DATE -			
APPROVED -	DATE -			

NOTES:

1. UNLESS OTHERWISE STATED:-
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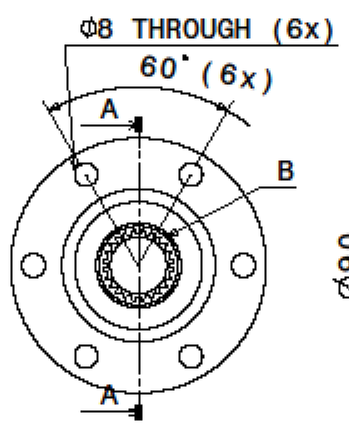
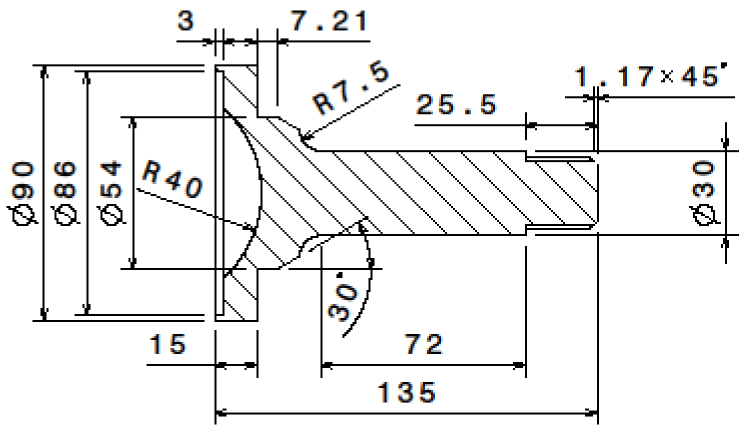
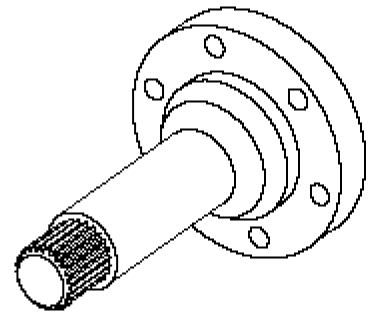


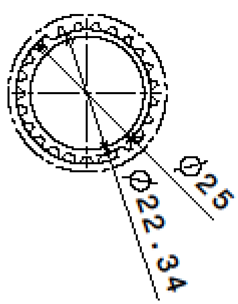
Front view



Section view A-A

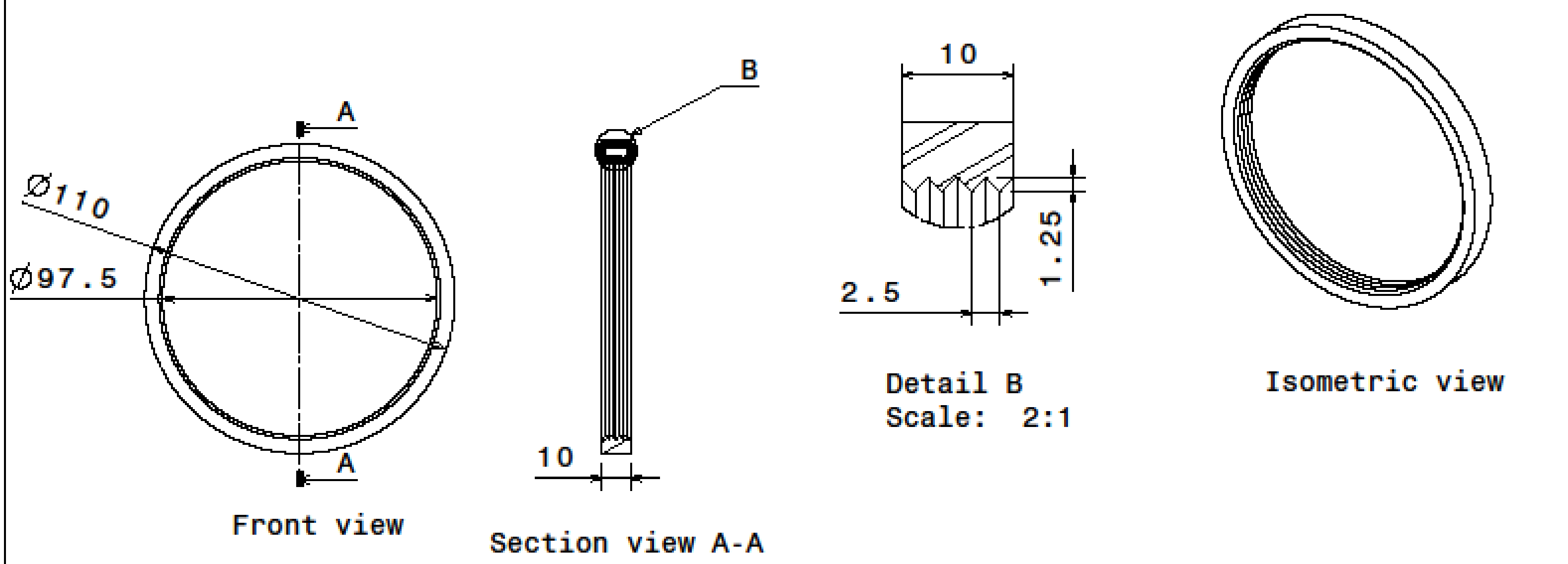


Isometric view
Scale: 1:2



Detail B
Scale: 1:1

PART NAME SF02-DT-SPINDLE		MATERIAL STEEL 4340		
MASS -		DRG. NO. -		
DRAWN -	DATE -	PROJECTION	SCALE 1:2	SHEET -
CHECKED -	DATE -	IIP FORMULA SAE		
APPROVED -	DATE -			



NOTES:

1. UNLESS OTHERWISE STATED:-
   - ALL DIMENSIONED TO BE MACHINED 3.2
   - TOLERANCE ON LINEAR DIMENSION ±0.25
2. REMOVE ALL BURSS AND SHARP EDGES

PART NAME SF02-DT-SPROCKET LOCK		MATERIAL ALUMINIUM 7075		
MASS -		DRG. NO. -		
DRAWN -	DATE -	PROJECTION	SCALE 1:2	SHEET -
CHECKED -	DATE -			
APPROVED -	DATE -			

APPENDIX II

FSAE 2009 rules

ARTICLE 8: POWERTRAIN

8.1 Engine Limitation

8.1.1 The engine(s) used to power the car must be four-stroke piston engine(s) with a displacement not exceeding 610 cc per cycle.

8.1.2 The engine can be modified within the restrictions of the rules.

8.1.3 If more than one engine is used, the total displacement can not exceed 610 cc and the airfor all engines must pass through a single air intake restrictor (see 8.6, “Intake SystemRestrictor.”)

8.1.4 Hybrid powertrains utilizing on-board energy storage are not allowed.

8.2 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.

8.3 Starter

Each car must be equipped with an on-board starter, and be able to start without any outside assistance at any time during the competition.

8.4 Air Intake System

8.4.1 Air Intake System Location

All parts of the engine air and fuel control systems (including the throttle or carburetor, and the complete air intake system, including the air cleaner and any air boxes) must lie within the surface defined by the top of the roll bar and the outside edge of the four tires. (See Figure 13).

8.4.2 Any portion of the air intake system that is less than 350 mm (13.8 inches) above the ground must be shielded from side or rear impact collisions by structure built to Rule3.24,3.25, or 3.26 as applicable.

8.5 Throttle and Throttle Actuation

8.5.1 Carburetor/Throttle Body

The car must be equipped with a carburetor or throttle body. The carburetor or throttle body may be of any size or design.

8.5.2 Throttle Actuation

The throttle must be actuated mechanically, i.e. via a cable or a rod system. The use of electronic throttle control (ETC) or “drive-by-wire” is prohibited.

8.5.3 The throttle cable or rod must have smooth operation, and must not have the possibility of binding or sticking.

8.5.4 The throttle actuation system must use at least two (2) return springs located at the throttle body, so that the failure of any component of the throttle system will not prevent the throttle returning to the closed position.

Note: Throttle Position Sensors (TPS) are NOT acceptable as return springs.

8.5.5 Throttle cables must be at least 50.8 mm (2 inches) from any exhaust system component and out of the exhaust stream.

8.5.6 A positive pedal stop must be incorporated on the throttle pedal to prevent over stressing the throttle cable or actuation system.

8.6 Intake System Restrictor

8.6.1 In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor.

8.6.2 Any device that has the ability to throttle the engine downstream of the restrictor is prohibited.

8.6.3 The maximum restrictor diameters are:

- Gasoline fueled cars - 20.0 mm (0.7874 inch)

– E-85 fueled cars – 19.0 mm (0.7480 inch)

8.6.4 The restrictor must be located to facilitate measurement during the inspection process.

8.6.5 The circular restricting cross section may NOT be movable or flexible in any way, e.g. the restrictor may not be part of the movable portion of a barrel throttle body.

8.6.6 If more than one engine is used, the intake air for all engines must pass through the one restrictor.

8.7 Turbochargers & Superchargers

8.7.1 Turbochargers or superchargers are allowed if the competition team designs the application. Engines that have been designed for and originally come equipped with a turbocharger are not allowed to compete with the turbo installed.

8.7.2 The restrictor must be placed upstream of the compressor but after the carburetor or throttle valve. Thus, the only sequence allowed is throttle, restrictor, compressor, engine.

8.7.3 The intake air may be cooled with an intercooler (a charge air cooler). Only ambient air maybe used to remove heat from the intercooler system. Air-to-air and water-to air intercoolers are permitted. The coolant of a water-to-air intercooler system must comply with Rule 8.10.

8.8 Fuel Lines

8.8.1 Plastic fuel lines between the fuel tank and the engine (supply and return) are prohibited.

8.8.2 If rubber fuel line or hose is used, the components over which the hose is clamped must have annular bulb or barbed fittings to retain the hose. Also, clamps specifically designed for fuel lines must be used. These clamps have three (3) important features, (i) a full 360 degree (360°) wrap, (ii) a nut and bolt system for tightening, and (iii) rolled edges to prevent the clamp cutting into the hose. Worm-gear type hose clamps are not approved for use on any fuel line.

8.8.3 Fuel lines must be securely attached to the vehicle and/or engine.

8.8.4 All fuel lines must be shielded from possible rotating equipment failure or collision damage.

8.9 Fuel Injection System Requirements

The following requirements apply to fuel injection systems.

8.9.1 Fuel Lines – Flexible fuel lines must be either (i) metal braided hose with either crimped-onor reusable, threaded fittings, or (ii) reinforced rubber hose with some form of abrasion resistant protection with fuel line clamps per 8.8.2. Note: Hose clamps over metal braided hose will not be accepted.

8.9.2 Fuel Rail – The fuel rail must be securely attached to the engine cylinder block, cylinder head, or intake manifold with brackets and mechanical fasteners. This precludes the use of hose clamps, plastic ties, or safety wire.

8.9.3 Intake Manifold – The intake manifold must be securely attached to the engine block or cylinder head with brackets and mechanical fasteners. This precludes the use of hose clamps, plastic ties, or safety wires. The use of rubber bushings or hose is acceptable for creating and sealing air passages, but is not considered a structural attachment.

8.10 Coolant Fluid Limitations

Water-cooled engines must only use plain water, or water with cooling system rust and corrosion inhibitor at no more than 0.015 liters per liter of plain water. Glycol-based antifreeze or water pump lubricants of any kind are strictly prohibited.

8.11 System Sealing

8.11.1 The engine and transmission must be sealed to prevent leakage.

8.11.2 Separate catch cans must be employed to retain fluids from any vents for the coolant system or the crankcase or engine lubrication system. Each catch-can must have a minimum volume of ten (10) percent of the fluid being contained or 0.9 liter (one U.S. quart) whichever is greater.

8.11.3 Catch cans must be capable of containing boiling water without deformation, and be located rearwards of the firewall below driver's shoulder level. They must have a vent with a minimum diameter of 3 mm (1/8 inch) with the vent pointing away from the driver.

8.11.4 Any crankcase or engine lubrication vent lines routed to the intake system must be connected upstream of the intake system restrictor.

8.12 Transmission and Drive

Any transmission and drivetrain may be used.

8.13 Drive Train Shields and Guards

8.13.1 Exposed high-speed equipment, such as torque converters, clutches, belt drives and clutch drives, must be fitted with scatter shields in case of failure.

8.13.2 Scatter shields for chains or belts must not be made of perforated material.

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

8.13.4 Belt drive - Scatter shields for 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).

8.13.5 Attachment Fasteners - All fasteners attaching scatter shields and guards must be a minimum 6mm grade M8.8 (1/4 inch SAE grade 5).

8.13.6 Attached shields and guards must be mounted so that they remain laterally aligned with the chain or belt under all conditions.

8.13.7 Finger Guards – Finger guards may be made of lighter material.