University of Southern Queensland

Faculty of Engineering & Surveying

FoES Formula SAE-A Space Frame Chassis Design

A dissertation submitted by

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in fulfilment of the requirements of

ENG4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: October, 2004

Abstract

The formula SAE-A (Society of Automotive Engineers - Australasia) competition is a purpose built competition for engineering students to apply their design and team working skills against each other in an engineering contest. Each team is responsible for the design of their vehicle and the smooth integration of the various components.

The design of a chassis for a formula SAE-A race car must contain all necessary components to support the car and the driver. It must also comply with the formula SAE-A 2004 rules. In order to produce a competitive vehicle with optimum chassis performance, many areas need to be studied and tested.

This project carried out all of the necessary background research required to sustain an accurate database of design criteria. This design criteria then allowed the design process and methodology to be derived and to allow for smooth construction of an efficient and effective spaceframe chassis.

Once construction of the chassis was completed, analyses were conducted to investigate the effects of working loads on the chassis. Finite element analysis was used to simulate the conditions of various load combinations. This analysis was verified by conducting similar physical tests on the chassis which ensured that the results were accurate. The results established that the deflections would be very minimal under working loads of the vehicle.

During the development and construction of the formula SAE-A racer, some areas for improvement were recognised and future recommendations were suggested.

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Limitations of Use

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Certification of Dissertation

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

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

Christopher Scott Baker

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Acknowledgments

My project supervisor Mr Chris Snook for providing this interesting and exciting topic and then providing his guidance, assistance and encouragement throughout the duration of the project.

The assistance and support of the University of Southern Queensland Mechanical Workshop staff. Namely Mr Chris Galligan and Mr Brian Aston.

A special thanks to Mr Chris Hannifan for volunteering his time, providing his experienced welding services and use of his personal workshop. Chris also donated the majority of material used for the construction of the chassis.

Christopher Scott Baker

University of Southern Queensland

October 2004

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Glossary of Terminology

Brake caliper	The part of the braking system that, when applied by the driver, clamps the brake disk/rotor to slow or stop the car.
Camber	The amount a tyre is tilted in or out from vertical. Described in degrees, either positive or negative.
Crossover bars	Chassis members that travel width ways across the chassis attaching one side to the other.
Master cylinder	Supplies hydraulic pressure to the brakes, and also has its own fluid reservoir to replenish the line if leaks occur.
Oversteer	When a car is at it's cornering limits, the rear tyres have a greater slip angle than the front, causing the back end to slide out wider than the front portion of the car.
Rails	Critical chassis members that travel the full length of the chassis, often parallel to each other.
Understeer	When a car is at it's cornering limits, the front tyres have a greater slip angle than the back tyres, causing the car to travel straighter or wider even though the driver is turning the steering wheel more.
Uprights	The upright attaches the wheel, brake disc, hub, brake caliper and steering arm to the car. The upright determines the king-pin inclination, and the final camber, caster, and toe settings of the wheel and tire.
Wishbones	Essentially the wishbones are connected to the chassis with rod-ends, allowing the wishbones to pivot up and down with the wheel's movement. They are triangulated to prevent the wheel from moving fore or aft of their designated position.

Chapter 1

Introduction

1.1 Project Introduction

Spaceframe chassis's have been in use since the start of the motor sport scene. A spaceframe consists of steel or aluminium tubular pipes placed in a triangulated format to support the loads from the vehicle caused by; suspension, engine, driver and aerodynamics.

There are two main types of chassis used in race cars, steel spaceframes and composite monocoque. Although spaceframes are the traditional style they are still very popular today in amateur motorsport. Their popularity maintains because of their simplicity, the only tools required to construct a spaceframe is a saw, measuring device and welder. The spaceframe still has advantages over a monocoque as it can easily be repaired and inspected for damage after a collision.

The chassis has to contain the various components required for the race car as well as being based around a drivers cockpit. The safety of the chassis is a major aspect in the design, and should be considered through all stages. The design also has to meet strict requirements and regulations set by the formula SAE-A organisers. Due to limited budgets and time constraints the design of the chassis will need to be geared towards simplicity and strength.

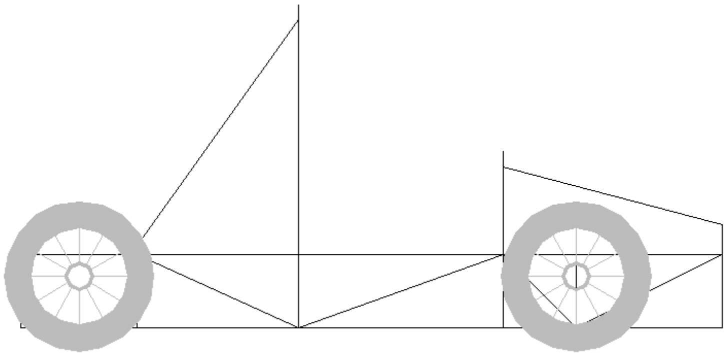

Figure 1.1: Basic Formula SAE-A Spaceframe Layout

1.2 Project objectives

The main objectives of this project were:

- Research background information relating to Formula SAE-A Rules and investigate similar formula SAE-A chassis designed by other institutions.
- Research effective spaceframe chassis characteristics and discover the effects of stress, torsion and deflection on a chassis with respect to vehicle handling and performance. In addition, the benefits and performance of different structure material also need to be taken into account.
- Design an effective and efficient spaceframe chassis that satisfies Formula SAE-A rules and regulations. The chassis design must be capable of being constructed from materials and resources available, while also considering other component requirements such as engine, drive train, suspension, etc.
- Conduct prototyping from preliminary designs and test appropriate factors.
- From testing, modify for improvements to the chassis.
- Fabricate chassis from specified material.
- Conduct non-destructive testing on the completed chassis for its response to loads.

The subsidiary objectives of the project were:

- Conduct tests on the fully fabricated chassis while in the form of a fully completed Formula SAE-A Racer for its impact on handling and performance.
- Recommend improvements or changes that could be implemented in a better formula SAE-A chassis.

A copy of the project specification is presented in Appendix A

1.3 Project Structure

This project is being undertaken with the cooperation of a USQ Formula SAE-A core design team consisting of eight members. Each team member is responsible for a different area. The eight key areas include:

- Team Manager
- Spaceframe Chassis
- Engine
- Suspension
- Drivetrain and Braking
- Steering
- Bodywork and Aerodynamics
- Cockpit Design and Vehicle Testing

This core design team is also supported by a formed USQ Motorsport club which assists in the construction stages and sourcing of sponsorship and components.

Chapter 2

Background

2.1 Spaceframe History

A space frame chassis uses a series of straight small diameter tubes to achieve strength and rigidity with minimal weight. The technique was formalisd during the Second World War, when they were used for the construction of fugalarge frames in combat aircraft. This design was first developed by Barnes Wallis who was an English aviation engineer. The advantages that the spaceframe offered to the the aircraft, was that it allowed the aircraft to obtain large amounts of damage to certian areas while still retaining enough strength to remain airborne. After the war in 1947, Dr Ferdinand Porsche used the concept to build his Cisitalia sports car. Soon after leading vehicle manufacturers such as Lotus and Maserati adopted the idea to produce race cars, these cars were nicknamed birdcage racing cars because of the multitude of tubes. Modern race cars are now constructed out of a single monocoque frame made from expensive fibre composite materials.

2.2 Current Frame

Currently, a spaceframe is defined by a series of load bearing members that are covered by panels that offer no load bearing support. Spaceframes however offer greater flexi-

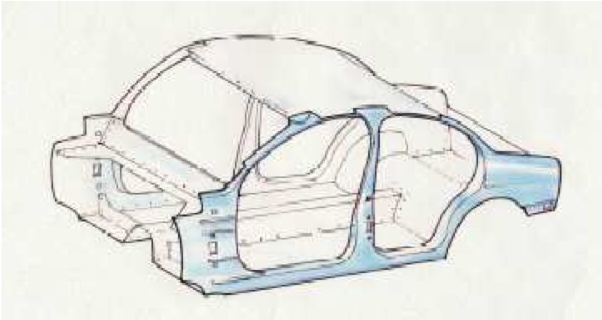

Figure 2.1: The welded Steel Monocoque Chassis. (pg39 (Reimpell 2001))

bility in terms of one off production, while also allowing a wide choice of materials such as steel, aluminium or composites. When multi and mass productions are required, spaceframes become very uneconomical compared to monocoque style frames.

Currently around 95 percent of world's automotive producers use the traditional welded steel monocoque frames as shown in figure 2.1. This form has provided an efficient and cost-effective means of mass production since the 1960s. Monocoque is defined as a structural skin, where outer panels (normally steel) are welded together early in production, contributing to the overall structural integrity of the vehicle.

2.3 Spaceframe Technology

Significant research is currently being undertaken in spaceframe technology to increase it's level of competitiveness against monocoque frames. With construction techniques expected to involve modern composite materials and advanced adhesives to form the chassis structure. Figure 2.2 shows a prototype of a space frame for a modern passenger vehicle. Once a solid spaceframe chassis is produced then the non-load bearing panels can be attached that are molded from a colour-cored thermoplastic. The advantage of this modern spaceframe construction and plastic panel technology is that the overall mass of the vehicle is reduced and the construction process has the potential of being more cost effective.

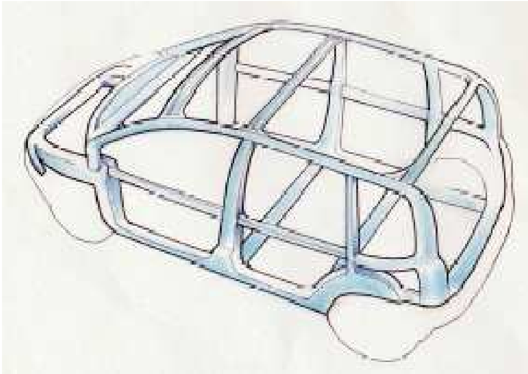

Figure 2.2: Modern Spaceframe, *Shown without external body panels*(pg40 (Reimpell 2001)).

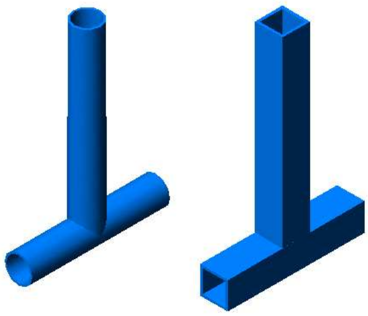

Figure 2.3: Section Joining

2.4 Fabrication Technoques

Traditionally, spaceframes were constructed from rectangular hollow section (RHS) tube as this was much easier to join and had flat surfaces to work from. RHS also allowed easy fabrication techniques, as all welded joints were flush. Modern spaceframes are now entirely fabricated from round tubular steel members to provide a torsionally ridged chassis frame. This process involves more complicated fabrication techniques as precision notching is required to achieve a strong structural join. These joining methods have been made much easier for hardened steels with the introduction of high quality tooling. The joining of two round tubes through notching also increases the amount of weld area increasing the strength, which can be seen in figure 2.3.

Modern welders and welding techniques have also improved the fabrications processes in construction of spaceframe chassis by allowing more complex welds to be achieved. Improved filler materials have also improved welding techniques and produce a stronger and cleaner weld. When cold drawn steels are used, tungsten Inert Gas (TIG) welding is preferred over Metal Inert Gas (MIG) welding. TIG welding produces smaller localised heat effected zone, preserving the steel harding properties.

Chapter 3

Literature Review

Spaceframes have been used in the construction of racing car chassis', since the introduction of car racing in the 1940's. Spaceframes are still commonly used today although they are loosing their competitiveness to fibre composite monocoque style chassis designs.

The performance capability of a vehicle on the road or race track can be related back to the chassis design. There has been much research conducted in the area of chassis design and how the chassis set up effects the vehicles response and performance.

3.1 Race Car Vehicle Dynamics

Racing is all about running every component to its limits and achieving maximum performance from the resources available. Professional racing teams spend enormous amounts of money on testing and research to achieve an edge over their competitors. Therefore race car vehicle dynamics has been heavily studied with all aspects and components of the race vehicle analysed. Every component of a racing vehicle is part of a complex system and the performance of many components often relies solely on the quality and performance of other components. For example, if a very high quality suspension system was attached to a soft flexible chassis, the majority of its performance would be wasted by the chassis flexing before the suspension spring can contract. This is why finding an optimum vehicle configuration is vital. (L. D. Metz 1998*a*)deemed that having the correct chassis set up and many component tuning options available, will allow for the system to achieve maximum potential.

It is a complex and difficult task to optimise a vehicle to perform at its full potential in different track conditions and events. This task can be further complicated if one vehicle has multiple drivers with each driver having a different preference for the vehicle set up. (L. D. Metz 1998*a*) also admits that it is unrealistic to produce a perfect optimum but by using modern design methods and some driver compromises, common faults can be overcome and performance satisfaction can be achieved.

(Stobart 2001) stated that the the first principal objective for an ideal chassis set up is to have cornering balance (neutral steer) under lateral load conditions to prevent over steer and under steer during cornering. The compromise between cornering performance and straight line speed is a difficult decision, which often has to be made by the team and decided upon early in the vehicle design. High chassis rigidity with vehicle corning ability can be achieved by having many triangular braces to stiffen the chassis. Forgoing the triangular braces will provide a lighter chassis with more high speed potential however will then result in increased body roll and deflection. If the chassis is over braced the increased weight will also increase the lateral loading of the chassis, again causing understeer and oversteer.

(Reimpell 2001) testified that the most common vehicle handling deficiencies, are often caused by poor and inadequate chassis designs. Excess body roll is the most common chassis deficiency caused by excessive deflection. During a turn when the lateral loads are high, generated deflection allows the vehicle to lean outwards of the turn causing the tyres to also lean and roll onto one side of the tyre track. This then reduces the contact surface between the tyre and the road. Under high lateral loads this contact surface will break and the vehicle will begin to drift laterally. When the vehicle goes into a drift it loses positive velocity and the set driving line, it is also extremely hard for the driver to control the vehicle and recover from the drift. Body roll can be reduced by increasing the rigidity of the chassis and therefore minimising the deflection. It can also be reduced by lowering the center of gravity in the chassis which will reduce the roll effect caused by lateral loads.

Because the chassis is a one piece rigid structure, it is unable to be adjusted for different track conditions. Therefore all component adjustments have to be made to suit the chassis. In professional racing teams, adjustable anti-roll bars can be used to provide some adjustment in the chassis for different conditions but require special setups which can only be applied to larger chassis.

(L. D. Metz 1998*b*) emphasised that the primary set up for a chassis is to be aware of the center of gravity(CG) of the vehicle. The best position for the CG is to be as low as possible to the ground while central along lateral and longitudinal axes. The CG determines the wheel loads which then effects wheel traction, breaking and cornering ability. The CG can be determined in chassis design using the setup location of each of the major components including engine, drivers seat, fuel and oil tanks. In many racing categories the vehicle must comply to a specified weight which allows teams to build the race car under weight and use ballast to meet the requirement. This ballast can then be positioned in the car to assist in tuning for varying track conditions.

3.2 Space Frames

Although Spaceframes have been extensively researched in the past, each style of vehicle if different and requires different characteristics, making the chassis requirements also differ for each type of vehicle. Spaceframe materials and fabrication techniques are generally universal across race vehicle categories. Spaceframe chassis are made from either Rectangular Hollow Section RHS steel, tubular steel or in some cases a combination of both. Tubular steel is found to be much more resistant to torsional loads because it has a constant axis for the moment of inertia, which is desirable in chassis performance.

(Reimpell 2001) stated that the common theory behind spaceframes is to create a chassis frame in a triangulated format to provide minimum deflection and maximum strength. If the frame is made from just a rectangular format it will be easily distorted under loads as shown in Figure 3.1. Triangulating the box by inserting a diagonal member, braces the frame, effectively reducing the amount of deflection. Increased

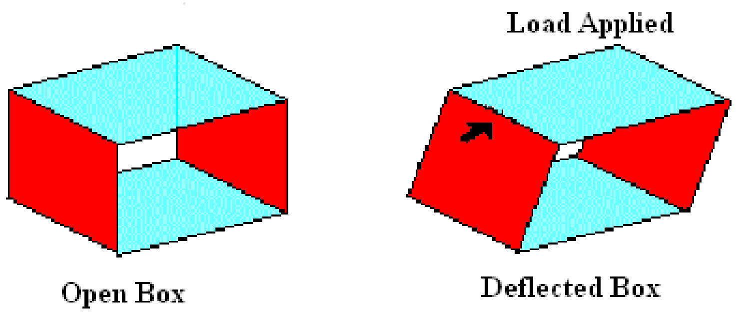

Figure 3.1: Box which is not Triangulated

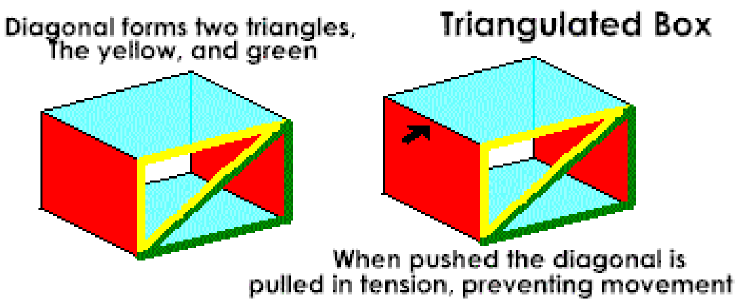

Figure 3.2: Triangulated Box

strength is gained when the section is loaded as shown in Figure 3.2. The diagonal member is stressed in tension and the end members are stressed in compression. If the force was applied in the opposite direction, the the diagonal member would be placed under compression and the ends will be placed in tension. As the diagonal member is longer and under higher loads it is more capable of buckling if compression loads are applied. For this reason it is important to know the load paths are and design so that the diagonals are under tension stresses.

3.3 Formula SAE-A

Many of the competing teams in the formula SAE-A competition list their race vehicle specifications, including their type of chassis and construction materials. They also display a large variety of pictures illustrating the construction process of the chassis and the methods used. Due to the competitive nature of the formula SAE-A event most teams are reluctant to publish detailed results and characteristics of their chassis. The Formula SAE-A competition organisers regularly publish a newsletter and general

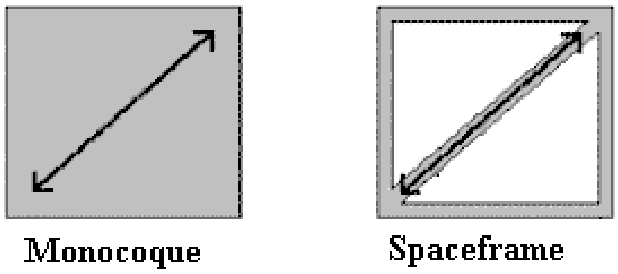

Figure 3.3: Achieving the Same Result (Oosthuizen 2004)

tips on getting started in the competition.

The competition rules relating to the compulsory impact members of the chassis have changed over the past years. Chassis that were built pre 2001 have different designs and setups that no longer comply to the SAE-A rules. Many of the more competitive teams with more experience and larger budgets opt to use a composite momocoque chassis because of its weight and performance properties. (Oosthuizen 2004) explains how monocoque chassis resist deflection and stresses similar to spaceframes however instead of having one diagonal, it has an entire panel to provide strength shown in figure 3.3.

3.4 Crashworthiness

(L. D. Metz 1998*c*), stated that from an engineering perspective, crashworthiness is the ability of the vehicle to prevent occupant injuries in the event of an accident. They also stated that crashworthiness is not the same as vehicle safety, and the two topics must be distinguished. The behavior of the structure such as a spaceframe under rapidly applied loads is commonly modelled using various analysis to provide a better understanding of the impacts experienced during a collision. Because the chassis contains the cockpit for the driver it is very important that the structural behavior of of the chassis under impact loads is known. The most common vehicle impacts occur at any angle on all vertical surfaces of the vehicle. (L. D. Metz 1998*c*) stated that the key to improving crashworthiness is to prevent 'second collision' where the occupant collides with the vehicle internals. The formula SAE-A rules enforce the use of five point racing harnesses and arm restraints to reduce occupant movements.

(Reimpell 2001) has conducted sudden impact tests with racing car chassis and concluded that the majority of serious injury is caused by sudden deceleration of the vehicle. This scenario is likely to occur when the vehicle collides with a solid stationary object, producing large amounts of energy which travel through the vehicle. Because racing chassis are designed for performance they are very rigid and therefore not very accepting to energy absorbtion. To absorb this energy separate energy absorbtion zones or crumble zones are attached to the bulk head of the chassis to assist in high energy collisions.

Chapter 4

Design Criteria

4.1 Determination of Physical Restraints

4.1.1 Introduction

The design of a chassis depends solely on the class of racing that the vehicle will be contesting. The chassis involved in this project is for a formula SAE-A class, where only one set of rules and requirements have to be met in order to contest in the event. These restrictions are often set by a governing body that runs the event and enforces that the rules are followed by the competing teams. There are also restraints placed on the class to ensure that safety measures are adhered to. In amateur motorsport classes including formula SAE-A there are also many restrictions to maintain a competitive competition and prevent teams with more financial resources from dominating the events.

4.1.2 Vehicle Requirements

Vehicle Design Objects

The design objectives of the class are set around a mock scenario where a design team is engaged to design and produce a prototype car for evaluation as a production item.

The intended sales market is a non-professional weekend autocross racer. Therefore, the car must have high performance in terms of its acceleration, braking, and handling qualities. It must also be low in cost, easy to maintain, and reliable. In addition, the cars marketability is enhanced by other factors such as aesthetics, comfort and use of common parts. The mock manufacturing firm is planning to produce 4 cars per day for a limited production run and the prototype vehicle should actually cost below $25,000. 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. (Section 1.2, 2004 Formula SAE-A Rules, Appendix D)

Body and Vehicle Configuration

The vehicle must be open-wheeled and open-cockpit design (a formula style body) with external wheels.The vehicle must also have a wheelbase of at least 1525 mm between centers. The vehicle must have four wheels that are not in a straight line. (Section 3.1.1 & 3.1.2, 2004 Formula SAE-A Rules, Appendix D)

Ground Clearance

Ground Clearance must be sufficient to prevent any portion of the car (other than tires) from touching the ground during track events. (Section 3.2.1, 2004 Formula SAE-A Rules, Appendix D). To accomdate this rule the team has elected to have a static vehicle ride height of 60mm without driver. The team has also elected to run 13inch wheels with an outside tyre diameter of 520mm. The chassis will have to have suspension mounting points capable of allowing this ride height.

4.1.3 Crash Protection

The driver must be protected from vehicle rollover and collisions. This requires two roll hoops that are braced, a front bulkhead with crush zone, and side protection members. Rollover accidents are often extreme and occur at high speeds when the forces acting on the vehicle are very large and cause substantial amounts of damage. The other serious accidents which can occur on race tracks is if a fast traveling vehicle collides with a stationery vehicle. This is why the bulkhead of the chassis requires bracing and crumble zones.

Main Hoop

This is the main rollover protection bar that is alongside or just behind the driver. This main hoop protects the drivers upper body in the event of a vehicle rollover. The main hoop must be constructed from a single piece of uncut tube that is attached to the base of the chassis. The main hoop must also be braced back to the main body of the chassis. The braces must also be at a horizonal angle of no less than 30 degrees. These rules ensure that the hoop is very strong and secure with no weak spots.

Front Hoop

The front hoop is the secondary rollover protection bar which is in front of the driver and above his/her legs near the steering wheel. This hoop protects the drivers arms and hands in a rollover. It also forms a safe rollover area with the main hoop that protects the driver's body in a rollover. The front hoop must also be constructed from a single piece of uncut tube the same as the main hoop and attached to the base of the chassis. The front hoop must also be braced forward onto the bulkhead of the chassis. (Section 3.3.4 2004 Formula SAE-A, Appendix D) specifies that if a rollover line was drawn between the front hoop and the main hoop, the top of the drivers helmet must be 50mm below this line as shown in figure 4.1.

Side Impact Protection

The driver must be protected from a side collision while seated in the normal driving position (2004 Formula SAE-A, Appendix D section 3.3.8.). This side impact protection is to protect the driver's body if another vehicle was to drive into the side of the chassis.

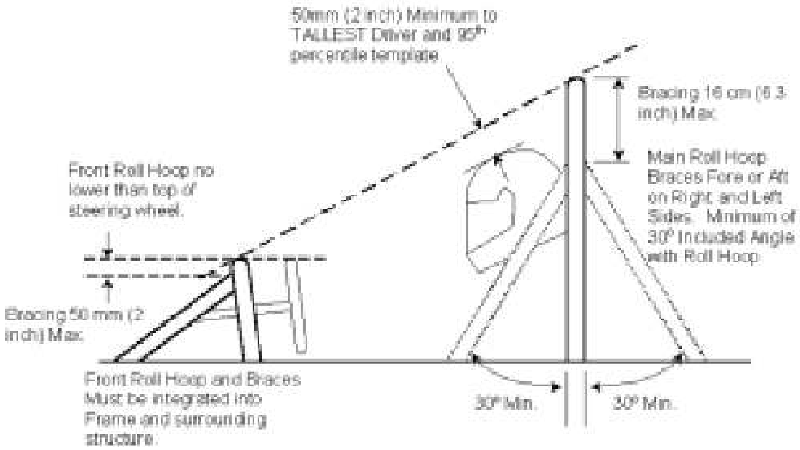

Figure 4.1: Chassis Restrictions (*source formula SAE-A rules pg 20*)

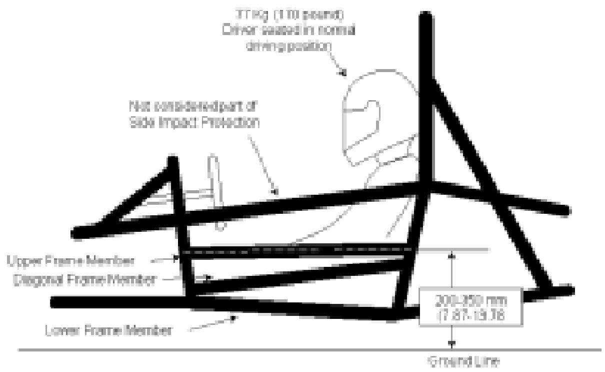

Figure 4.2: Side Impact Members (*source formula SAE-A rules pg 27*)

The side impact protection must include 3 frame members constructed from specified material.

- **Upper Member** must be between 200mm and 350mm from the ground and connect the main roll hoop with the front roll hoop.
- **Diagonal Member** must connect the upper and lower side impact members.
- **Lower Member** must connect the bottom of the main roll hoop with the bottom of the front hoop.

Crush Zone

The chassis must also have a crush zone forward of the major structure of the chassis (2004 Formula SAE-A, Appendix D section 3.3.1). The crush zone must be designed to absorb energy in the event of a head on collision. It must also be defined by two separated planes forward of the main chassis structure so in a head on collision they can crumple and decelerate the vehicle within an acceptable limit.

4.1.4 Component Restraints

When designing a chassis it is not only important that the vehicle is designed to the regulations but it must also be designed so that it can house the necessary components that are required in the vehicle. These components should include

- Engine
- Drive Train
- Suspension
- Human Factors

All of these components are other team member's projects and tight networking with them can determine what areas within the chassis need to be incorporated into the design.

Engine

Approximate engine dimensions for a 600cc motorbike engine are 500mm long, 550mm wide and 400mm high. The engine will also require custom mounting points on the chassis.

Drive Train

The chassis needs to accommodate the rear axle which is going to be approximately 300mm above the ground, the chassis also has to support bearing housing for the rear axle. The axle will have a 320mm drive sprocket which will need to be inline with the pinion sprocket on the engine. There also needs to be a clear line between the drive sprocket and the pinion sprocket for the chain to run. A 300mm diameter brake disk will also be attached to the rear drive shaft.

Suspension

The weight of the vehicle needs to be supported through the suspension. The wishbones for the front and rear suspension also need to be mounted to the chassis. The shock absorbers and springs need strong mounting points on the chassis which will produce large fluctuating loads.

Human Factors

One of the main purposes of the chassis is to provide a cockpit for the driver. The chassis must provide comfortable leg room so the driver can reach the peddles. It must also provide clear vision forward of the vehicle. The front plane of the front hoop will have to house controls and driving instruments. The steering wheel must also be within easy reach from the drivers seat which is under the main hoop.

4.2 Determination of Loads

4.2.1 Introduction

To design a chassis, assumptions need to be calculated as to the expected loads that could be experienced by the chassis. These loads should include the known static loads of the vehicle components such as driver and engine, while also including predicted dynamic loads which will occur through suspension and drive train components. Worst case loads should also be calculated and designed for to prevent the vehicle failing and injuring the driver. While the vehicle is stationery there are constant loads from the vehicle components and the self weight of the vehicle being transmitted through the suspension to the ground. Once the vehicle is in motion these components cause load paths that are much more complicated. When the vehicle is cornering, accelerating and braking these loads are then applied in different and varying directions. Radial forces are also produced throughout the chassis by rotating components.

4.2.2 Static Load paths

When the car is stationary the loads from the vehicle have to transfer from the various components through the spaceframe to the wheels and to the ground. When designing the chassis it is very important to be aware of these load paths so that the components are supported with minimal deflection. The main components that need to be analyzed are the engine and the driver because these two masses account for almost two thirds of the total mass of the vehicle, minor components account for the remaining weight.

4.2.3 Dynamic Load Paths

Dynamic vehicle loads are created from accelerating and braking, which are proved through Newton's law of F=ma. When the vehicle is braking large forces are produced by the brake calipers pressing on the disk brakes. When analyzing these accelerating and braking forces, most of the analysis will be on the driver and engine using Newton's second law (Giancoli 1991).

$$F = ma \tag{4.1}$$

where

$$\begin{aligned} F &= \textit{Applied Force} \\ m &= \textit{Mass of Component} \\ a &= \textit{Acceleration} \end{aligned}$$

From previous years results in the acceleration test, competitive vehicles have reached accelerations capable of 0 to 100km/hr in around 3 seconds. Assuming that the acceleration is constant the formula (Giancoli 1991):

$$a = \frac{v_f - v_i}{dt} \tag{4.2}$$

where

$$\begin{aligned} a &= \textit{Acceleration} \\ v_f &= \textit{Final velocity} \\ v_i &= \textit{Initial velocity} \\ dt &= \textit{time} \end{aligned}$$

When 100km/hour = 27.77m/s and the initial velocity is 0

$$\begin{aligned} a &= \tfrac{27.77-0}{3} \\ &= 9.25m/s^2 \end{aligned}$$

To allow for the acceleration not being constant, an acceleration value of 10m/s^2 will be used for the calculations. This generous force allows for a slight factor of safety.

Radial Loads

There are many radial loads applied to the chassis by the internal components when the vehicle experiences hard cornering. This is caused by the the components wanting to continue in a straight line while the chassis has changed paths. These forces prove difficult to calculate without physical testing and data logging. Research from other teams specifies that 'g' forces of up to 1.5 can be reached in their vehicles. This value can be used to estimate realistic forces that may be experienced under these conditions. To estimate the forces acting through the individual centres of gravity of each component, Newton's second law can once again be applied.

$$F = ma \quad (4.1)$$

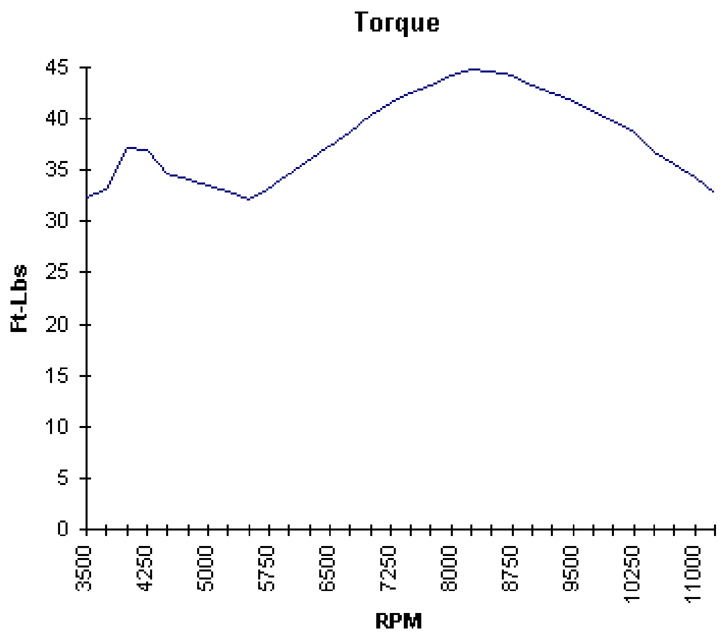

Figure 4.3: Torque vs RPM for YZF600 Engine *source www.superbikes.net*

where

$$\begin{array}{lcl} F & = & Applied\ Force \\ m & = & Mass\ of\ Component \\ a & = & Acceleration(Gravity \times 1.5) \\ Gravity & = & 10m/s^2 \end{array}$$

Torsional Loads

When the vehicle accelerates, the engine produces torque which gets amplified by the drivetrain and transmitted to the road through the tyres. This torque also has to be counteracted by the engine through the chassis. To design a chassis it is important to know what the maximum estimated torque will be. To estimate the torque a worst case assumption will be calculated. The maximum torque produced by the engine can be found on the engine power torque performance chart shown in figure 4.3. This graph shows that the engine produces a maximum torque of 45Ft.Lbs which is equivalent to 61N.m at 8500RPM. This maximum torque will be applied to the vehicle when it is leaving the start line and the engine will be in first gear, which then has a gear ratio of 2.85:1. The torque will also be amplified by being transferred through the clutch gear at a ratio of 1.7:1 and again through the chain drive to the rear axle which has

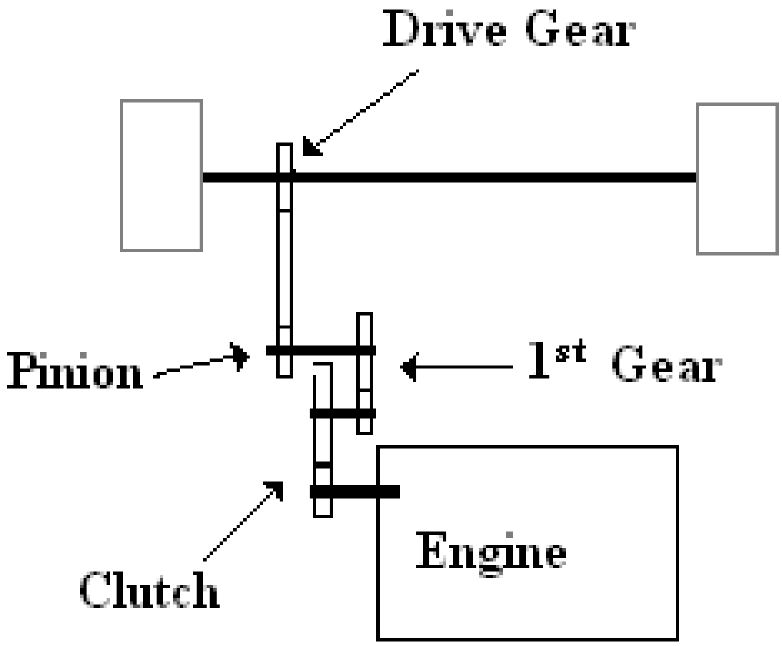

Figure 4.4: Drive Train Torque Ampliation

a ratio of 4.6:1 Once the torque has been transferred to the rear axle it can then be transmitted to the road through the wheels. Some of the torque will be lost through the clutch slipping and tyre traction.

Engine Torque

$$\textit{Torque produced by engine} = 68N.m$$

$$\begin{aligned}\textit{Torque @ clutch} &= \textit{Engine torque} \times \textit{Clutch Ratio}\\ &= 68 \times 1.7\\ &= 116N.m\end{aligned}$$

$$\begin{aligned}\textit{Torque in } 1^{st} \textit{ Gear} &= \textit{Clutch Torque} \times 1^{st} \textit{ Gear Ratio}\\ &= 116 \times 2.85\\ &= 331N.m\end{aligned}$$

$$\begin{aligned}\textit{Torque @ Rear Wheels} &= 1^{st} \textit{ Gear Torque} \times \textit{Drive Ratio}\\ &= 331 \times 4.6\\ &= 1523N.m\end{aligned}$$

This load is produced at the back wheels but has to be counteracted by the engine which is held by the engine mounts. This torque is applied around the drive pinion on the engine.

Torsional Braking Loads

When the vehicle is braking large forces are produced by the brake calipers pressing on the disk brakes. These braking forces are the largest forces in the race car and produce a large moment due to the rotating nature of the brake disk. These loads are transmitted through the wishbones to the chassis on the front wheels and through the caliper mount on the rear wheels. Knowing the top speed of the vehicle and the time it takes while braking hard to come to rest, will provide sufficient data calculate the braking loads using the impulse-momentum theorem.

$$F = \frac{mv_f - mv_i}{dt} \tag{4.3}$$

where

$$\begin{aligned} F &= \textit{Applied Force} \\ m &= \textit{Mass of vehicle} \\ v &= \textit{final and inital velocity} \\ dt &= \textit{time} \end{aligned}$$

Assuming that the braking deceleration can be 100km/h to 0km/h in 3 sec:

$$\begin{aligned} F &= \frac{\text{250 x 0 - 250 x 27.7m/s}}{3} \\ &= 2308N \end{aligned}$$

This load has to be spread across the 3 brake disks, predominantly on the front 2. It is then estimated that each front disk would receive 1000N of the force. Assuming the front disks are 300mm outside diameter the force can be transferred to a moment of 150N.m to be shared by the 2 wishbones.

4.2.4 Defined Loads

These are approximations of loads that may be experienced by the formula SAE-A spaceframe chassis. It should be noted that all of these loads are calculated on assumptions however they are generally similar to real loads produced. Values are obtained neglecting some minor factors and using worst case scenario values to produce maximum loads in all cases, this is so the chassis is capable of withstanding all possible situations.

Static Loads

$$
\begin{aligned}
\text{Mass of engine} &= 60kg \\
\text{Mass of driver} &= 120kg \\
\text{Self mass of chassis} &= 50kg \\
\text{Estimate Total Mass of Vehicle} &= 250kg
\end{aligned}
$$

These loads will be applied in the direction of gravity through the engine mounts and through the seat. The suspension must hold the total mass of chassis as well as all components.

Acceleration and Braking Forces

Acceleration and braking forces are loads that are applied through the vehicle components under acceleration and braking. These forces travel through the component mounting points to the chassis. These forces need to be analyzed to ensure that extra forces are not applied to members that are already carrying large loads. Once again the engine and the driver will be analyzed.

- **Engine**

$$
\begin{aligned}
Acceleration\ Force\ on\ Engine &= Mass\ of\ Engine \times acceleration \\
&= 60 \times 10 \\
&= \underline{600N}
\end{aligned}
$$

When the vehicle is accelerating the force will be in the opposite direction of vehicle travel and will be transferred through the 6 engine mounts.

$$
\begin{aligned}
Deceleration\ Force\ on\ Engine &= Mass\ of\ Engine \times Deceleration \\
&= 60 \times 10 \\
&= \underline{600N}
\end{aligned}
$$

This force will be in the direction of vehicle travel and also have to be supported by the engine mounts.

- **Driver**

$$\begin{aligned} Acceleration\ Force\ on\ Driver &= Mass\ of\ Driver \times Acceleration \\ &= 120 \times 20 \\ &= \underline{1200N} \end{aligned}$$

This force of the driver will be widely spread through the seat and the seat mounts while some load will also be transferred through the race harness and steering wheel, the loads will be in the opposite direction of vehicle travel.

$$\begin{aligned} Deceleration\ Force\ on\ Driver &= Mass\ of\ Driver \times Deceleration \\ &= 120 \times 10) \\ &= \underline{1200N} \end{aligned}$$

This force will be in the direction of vehicle travel and the load will have to be fully supported by the driver's harness.

Cornering Loads

Due to centripetal acceleration there are forces directed towards the outside of the corner. This force is proportional to the velocity at which the vehicle travels around the corner. Centripetal acceleration is often measured in terms of gravity or G forces. An acceleration of 1.5 G's will be used to estimate the forces applied on the chassis

- **Engine**

$$\begin{aligned} Horizontal\ Force\ Produced &= Mass\ of\ Engine \times (Gravity \times 1.5) \\ &= 56 \times (9.81 \times 1.5) \\ &= \underline{825N} \end{aligned}$$

This force will be applied at 90 degrees to the direction of vehicle travel. The engine mounts will also have to transmit this load to the chassis.

- **Driver**

$$\begin{aligned} Horizontal\ Force\ Produced &= Mass\ of\ Driver \times (Gravity \times 1.5) \\ &= 120 \times (9.81 \times 1.5) \\ &= \underline{1770N} \end{aligned}$$

This force will be also applied at 90 degrees to the direction of vehicle travel and be transmitted to the chassis through the seat and racing harness.

4.3 Stresses Criteria

4.3.1 Introduction

Excessive stresses on the chassis can cause deflection, buckling, plastic deformation and eventually failure. This is why it is important to understand the principles of stresses and how they are formed and transferred through the chassis. The understanding of load paths through the chassis can also substantially influence the design of stress members.

4.3.2 Axial Stress

Axial stress occurs when loads are applied parallel to the direction of the material and can be in two forms; tension and compression. Axial stress is very common in spaceframes as they are made from a series of straight members, many of which are in the direction of the applied forces.

Tension Members

A tension member is a straight member subjected to two pulling forces applied at either end (Johnston 1992). When the load within the tension member coincides with the longitudinal centripetal axis of the member, the stress distributed through the member can be assumed to be uniform and defined by:

$$\sigma = \frac{P}{A} \tag{4.4}$$

where

$$\begin{aligned} \sigma &= \textit{Normal Stress} \\ P &= \textit{Load} \\ A &= \textit{Cross Sectional Area} \end{aligned}$$

When the normal stress of the tension member exceeds the yield strength of the material the member will experience plastic deformation which is permanent to the material. If the chassis experiences any plastic deformation, the frame will be considered ruined. The plastic deformation can leave the chassis permanently bent and twisted. When designing a chassis the working stresses should be well clear of the yield strength to avoid this deformation.

If the normal stress of a tension member exceeds the tensile strength of the material, failure of the member will occur. Usually the tensile strength of the material is extremely high and should even be well above the stresses reached in a collision.

Compression Members

A compression member is a straight member subject to two pushing forces applied at either end (Johnston 1992). The fundamental theories of buckling apply to compression members as the member will fail due to buckling long before the yield strength of the material is reached. This is why compression members are the main concern when axial loads are analyzed.

The length of the member is very critical when modeling buckling, because all members of the chassis are welded at both ends the effective length can be reduced to

$$L_e = 0.7L \tag{4.5}$$

This effective length can then be used in Euler's classical equation:

$$P_{cr} = \frac{\pi^2 EI}{L_e^2} \tag{4.6}$$

where

P_{cr}=*Critical Load*

E=*Modulus of Elasticity*

I=*Area Moment of Inertia*

L_e=*Effective Length*

This equation assumes that the member is perfectly straight and homogeneous. If the member is subject to a load below the P_{cr} load it may deflect slightly but the internal elastic moment will remain adequate to restore straightness to the member when the load is removed. When the P_{cr}load is exceeded the lateral displacement will produce an eccentric bending moment greater than the internal elastic restoring moment resulting in the member collapsing and no longer being able to carry load.

4.3.3 Deflection

Previous chapters have already explained the undesirable effects of deflection within the chassis. However, if the chassis was constructed so that no deflection would occur, it would require extensive amounts of material resulting in excess weight.

Deflection can be caused by many different stresses, such as axial forces in either tension or compression and even torsional stress caused by twist or rotation. The analysis of deflection can then become increasingly complicated with the introduction of biaxial stressing.

$$\delta = \frac{PL}{AE} \tag{4.7}$$

where

$$\begin{aligned}
\delta &= Deflection \\
P &= Load \\
L &= Length \\
A &= Cross\ Sectional\ Area \\
E &= Modulus\ of\ Elasticity
\end{aligned}$$

4.3.4 Bending

Bending stresses occur when a member is subject to a rotational moment load. This moment causes one side of the member to be in tension while the other is in compression. The bending stress can be calculated using:

$$\sigma_b = \frac{M_x y}{I_x} \tag{4.8}$$

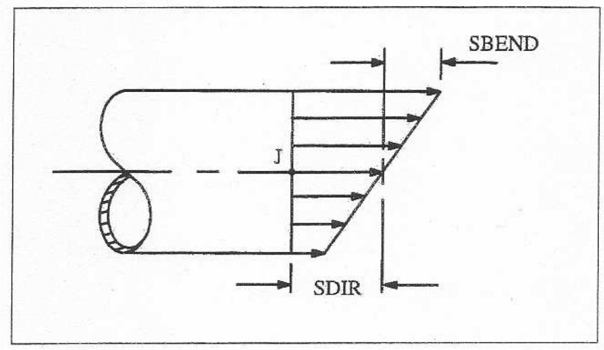

Figure 4.5: Bending Stress((MEC2402)Stress Analysis Study Book)

where

$$\sigma = \textit{Bending Stress}$$
$$M_x = \textit{Bending moment about the neutral axis}$$
$$y = \textit{Distance from the neutral axis}$$
$$I_x = \textit{moment of inertia of the cross section about the neutral axis}$$

As shown in figure 4.5, the maximum bending stress occurs at the outer surface. For bending situations, it is only important to have material at the outer most edge of the member, as this is where the maximum stresses occur. This is why hollow section tubes are excellent materials for resisting bending stresses. Bending stresses are common in chassis due to the large rotational moments caused by components such as the engine and drive train as well as other dynamic forces caused by vehicle travel.

4.3.5 Stress Analysis

Stresses can be measured and calculated using various techniques. The common methods used are to physically apply loads to the chassis and measure the deflections by sight or by attaching strain gauges. When the deflection is known the stress can be calculated. Stresses can also be calculated using simple formulas and hand calculations but this usually requires many simplifications to be made. When complex structures such as chassis are analyzed, the formulas become very large and complex, therefore computer programs are required to calculate the stresses involved.

When analyzing the formula SAE chassis both physical and numerical tests will be performed to calculate realistic stresses that might be experienced in the chassis under race conditions. Using both methods, comparisons can be made to verifying the accuracy of the results.

Numerical Testing

Because of the complexity of the spaceframe chassis, hand numerical calculations would prove extremely lengthy. Therefore the numerical tests will be completed using finite element analysis (FEA) software. This software allows complex numerical calculations to be performed in feasible time. Property settings required to conduct FEA can often be complicated to simulate the real conditions.

Physical Testing

Some physical tests were conducted on the chassis but were undertaken simply to verify the results of the FEA. Physical tests also ensure that there are no critical faults in the chassis.

Chapter 5

Material Selection

5.1 Introduction

Motorsport is a highly contested competition where teams seek to find any advantage to increase their vehicles performance. Different chassis materials can reduce the weight of the vehicle, improving the vehicle power to weight ratio. Material selection can also provide advantages by reducing member deflection, increasing chassis strength and can determine the amount of reinforcement required.

The formula SAE-A rules disallow the use of Titanium Alloy being used for chassis construction but permits all other viable materials. Feasible construction materials for a space frame would include:

- Plain carbon Steels
- Alloy Steels
- Aluminum
- Fibre composites

5.2 Material Specifications

To enforce a safe structure for the vehicle, the formula SAE-A rules specifies a baseline material size for key members. (2004 Formula SAE-A, Appendix D Section 3.3.3) states that the steel tube must be round, mild or alloy and contain a minimum of 0.1% carbon. The outside diameter must also be a minimum of 25.4mm for the hoops and have a wall thickness of 2.4mm. Different sections of the chassis are allowed to be different diameters but for fabrication simplicity the chassis will be constructed from the same material.

Alternative tubing geometry can be used besides the baseline, as specified in 2004 Formula SAE-A, Appendix D section 3.3.3.2. This rule allows larger diameter tubes to be used with a decreased wall thickness. Even with a larger diameter tube the minimum wall thickness is restricted to 2.1mm. There is no allowance for high performance steels and therefore all steels must be treated equally.

When using larger diameter tubes the preferred tube must have an equivalent, or greater, buckling modulus than the baseline material as specified in 2004 Formula SAE-A, Appendix D section 3.3.3. The equation for calculating buckling modulus is

$$\text{Buckling Modulus} = EI \tag{5.1}$$

where

$$E\text{=Modulus of Elasticity}$$

$$I\text{=Area Moment of Inertia}$$

Because all steels have to be treated equally the modulus of elasticity is going to be the same. Therefore

$$I_{baseline} = I_{new} \tag{5.2}$$

where I for tube is

$$I = \frac{\pi}{64}(d_o^4 - d_i^4)$$

and

$$d_o\text{=outside diameter}$$

$$d_i\text{=inside diameter}$$

Table 5.1: Percentage Carbon Ranges for Plain Carbon Steel (Marshek 1999)

Low Carbon Steel	Up to 0.05% carbon
Mild Steel	Between 0.05% and 0.3% carbon
Medium Carbon Steel	Between 0.25% and 0.6% carbon
High Carbon Steel	Between 0.55% and 1.1% carbon

While complying to the rules, 31.75mm diameter tube can be used with a thickness of 2.1mm and still have a slightly larger buckling modulus than the baseline size of 25.4mm.

5.3 Material Selection

5.3.1 Steel

Steel is a highly versatile alloy of iron and carbon. Other alloying elements such as Silicon, Manganese, Sulphur, Molybdenum, Phosphorus, Nickel and Chromium, can be added to improve its material properties. Steel can be divided into two main groups; Plain Carbon or Non-Alloy Steel and Alloy Steel. Many different forms of steel are available depending on its individual makeup of elements.

Plain Carbon Steel

Plain carbon steels contain carbon as the principal alloying element with only small amounts of other elements added. The strength of plain carbon steel increases with the percentage of carbon as shown in figure5.1. While an increase in carbon improves the strength of the material, it decreases its ductility making it more susceptible to brittle fracture. In the Plain Carbon Steel group there are three main types which are graded depending on their percentage of carbon content.

- **Low carbon steel** is the most widely used steel as it is also the cheapest. Low carbon steel is easy to form and cast. It is commonly used for applications where high strength is not required.

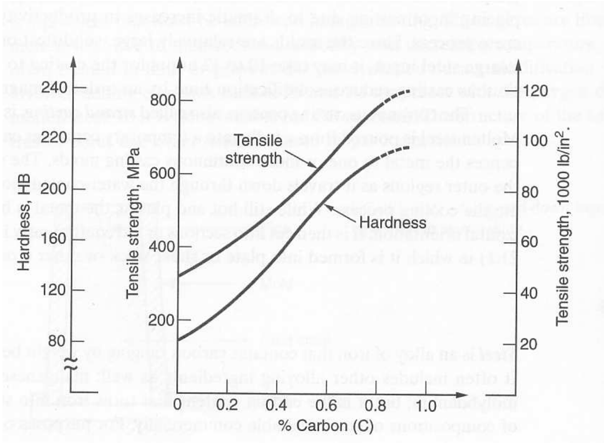

Figure 5.1: Tensile Strength and Hardness of Plain Carbon Steels. (D.R.Askeland).

- **Medium carbon steel** is between low and high carbon steels, and has high strength while still having some ductility. It still provides moderate strength while still maintaining affordability.
- **High carbon steel** is specifically for high strength applications where stiffness and hardness are needed. High carbon steels also have a high resistance to wear.

Heat treatment and tempering processes carried out on these steels can improve their hardness and/or toughness properties, depending on the methods used.

Alloy Steels

Alloy steels are iron-carbon steels that contain significant additional alloying elements. Alloy steels have superior mechanical properties to plain carbon steels. Common alloying elements that are added include Chromium, Manganese, Molybdenum, Nickel and Vanadium. The percentage of alloying elements added can influence mechanical properties to increase strength, hardness, hot hardness, wear resistance, fatigue resistance and toughness.

- **Stainless Steel** is the generic name for a number of different high alloy steels used primarily for their resistance to corrosion. The one key element they all share is that they must a minimum of 12% chromium. Although other elements, particularly Nickel and Molybdenum are added to improve corrosion resistance.

The main advantage to using a steel which is corrosion resistant is that it will have an extremely long life and strength is not lost to rust. The disadvantage with stainless is that it is very expensive.

- **Chrome Molybdenum SAE4130** is a high alloy steel which contains Silicon, Chromium and Molybdenum. These alloying elements give the steel superior strength compared to other common steels. The alloying elements also provide a protective barrier within the steel to increase the corrosive resistance. Another advantage of chrome molybdenum steel is that it's weldability is very good. The disadvantages of chrome molybdenum steel is that it is brittle therefore can become fatigued when exposed to fluctuating loads. Chrome Molybdenum is also very expensive and hard to find a supplier.

5.3.2 Aluminum

Aluminum is a nonferrous metal with very high corrosion resistance and is very light compared to steels. Aluminum cannot match the strength of steel but its strength-to-weight ratio can make it competitive in certain stress applications. Aluminum can also be alloyed and heat treated to improve it mechanical properties, which then makes it much more competitive with steels however the cost increases dramatically.

Aluminum alloys are also available but are very specialist materials. These alloys are extremely strong and light, compared to all other materials. They are also very expensive and not readily available in tube form. The primary use for aluminum alloys are for military, aircraft and space applications.

5.4 Tube Production

Hollow steel sections can be produced through many different methods. These methods can also influence the mechanical properties of the material.

5.4.1 Cold Drawn Seamless

Cold drawn seamless (C.D.S) tube is produced by the piercing method. A heated billet moves through pressure rolls as it is driven over a stationary mandrel to produce a hot finished seamless tube. This hollow section is then cold drawn through a die to precision finished dimensions. Cold drawn seamless tubing was once the highest performing mechanical tubes on the market but are now closely matched by C.D.W (see section 5.4.3).

The manufacturing process produces excellent tolerances, mechanical properties, and reduced surface defects. The hardness and strength properties are also increased by the amount of cold reduction (**?**).

The material properties of C.D.S fulfill chassis requirements, however the material costs do not justify its use compared to other suitable materials.

5.4.2 Electric Resistance Welded

Electric resistance welded (E.R.W) tube is the cheapest and most common type of steel tube available. It is produced from steel strip then cold formed and then electric resistance welded to complete its shape. The welding process involves slightly overlapping the strip and producing a thin continuous weld along the overlap. The welding process does not involve any filler material being added. The temperature is produced by applying electrical current through the overlap fusing the two layers together. This process is made continuous by using wheel electrodes as shown in figure 5.2.

5.4.3 Cold Drawn Electric Resistance Welded

Cold drawn electric resistance welded (C.D.W.) tube is produced from steel strip and electric resistance welded similar to E.R.W. However C.D.W is cold drawn to finished dimensions. Because of its high product flexibility, C.D.W. is the most versatile and widely sought mechanical tubing grade. A variety of thermal treatments can be applied to alter the mechanical properties and machinability. Modern E.R.W. processes

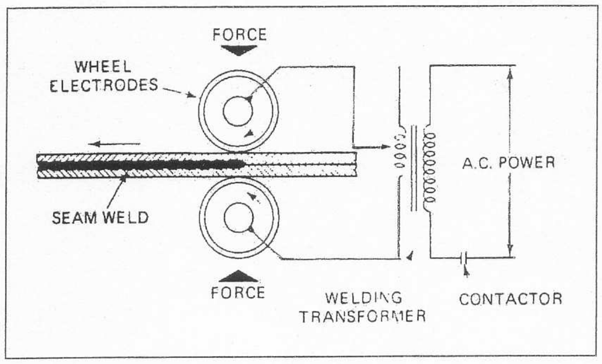

Figure 5.2: Process of Seam Welding((MEC2202)Manufacturing Process Study Book)

guarantee the weld to be as strong, or stronger, than the parent tube body.

C.D.W. is used for a large variety of machine parts where closer tolerances and higher mechanical properties are needed. The high mechanical properties make it an excellent material for a chassis in addition to its affordability.

5.5 Comparisons

To decide on the most appropriate material for the spaceframe chassis, all of the materials needed to be compared so their advantages and disadvantages can be assessed. Alloy steels and aluminum alloys are probably the ideal materials as their properties are superior to others. However extremely high costs associated with these materials makes them unviable for use in spaceframe construction. The use of plain carbon steels is much more affordable while still having sufficient strength.

Pure aluminum is also a possible material and is reasonably affordable and very light but it is the weakest and will require extra reenforcement to produce a rigid chassis. This extra material increases the weight reducing the materials weight advantage. Aluminum is very hard to work with as it requires very skilled welding and is a overall softer metal.

When comparing possible steel tube dimensions the 25.4mm x 2.4mm tube and 31.75mm x 2.1mm tube where the only tube dimensions considered. The 31.75mm x 2.1mm tube has a marginally higher material content but a larger bending and bucking modulus compared to the 25.4mm tube.

Table 5.2: Material Property Comparision of Possible Tubes.

Tube Types	Yield Strength	Tensile Strength	Mass kg/m	Cost $/m
31.75mm x 2.1mm Welded Medium Carbon steel	150Mpa	210Mpa	1.35	5
31.75mm x 2.1mm C.D.W. Medium Carbon steel	250Mpa	350Mpa	1.40	10
31.75mm x 3.175mm Alunimum	75Mpa	110Mpa	0.617	7.5

The different tube formation methods can increase or decrease the material properties. Cold working increases strength while having a weld seam that could produce a brittle area. For these reasons cold drawn seamless is the superior forming process but is too expensive. Cold drawn electric resistance welded tube has very similar properties to cold drawn seamless and is much more affordable and practical. The mechanical and economic properties for the proposed materials are shown in table 5.2

5.6 Conclusion

The material decided upon for the Formula SAE spaceframe chassis was a 31.75mm diameter 2.1mm thick C.R.W. medium carbon steel. This steel was chosen as it was readily available and provided superior strength compared to other affordable materials. This material has been specifically designed for vehicle spaceframes and roll cages while having great weldability and being easy to work with. A Technical Data Sheet for this chosen material is attached in Appendix E.

Chapter 6

Design Process and Methodology

6.1 The Design Process

The engineering design process is the decision making process which integrates the basic science, mathematics and engineering principles required in a project. The design process begins with an identified need, in this case it is the need for the formula SAE racer to have a chassis. There are many design steps which were taken prior to this project commencing, these included:

- Conseptulization
- Feasibility assessment
- Decision to proceed

The design process of this project incorporates the following steps:

- Development of Work
- Preliminary Design
- Prototyping and Redesign
- Detailed Design

- Construction Planning
- Construction

Completing all of these design steps chronologically will satisfy the aim of the project efficiently. By following this design process, the available resources can be optimally converted into a functional formula SAE chassis.

6.1.1 The Development of Work

The development of work includes the organization and the breakdown structure of the design process. The organization phase is a very board area and is continued throughout the entire design process. The main design organization for the chassis is to coordinate with all the other formula SAE projects and ensure that amalgamation of the components is possible and smooth. The organization aspect also includes the management decisions and developing a order of construction.

The breakdown structure allows easier management of the project on the condition that the relationship between the separate areas is closely maintained. The breakdown of the chassis comprises of the frontend which is the forward structure of the chassis including the main hoop and the rearend which is rearward of the main hoop. This breakdown was conducted for ease of design and manufacture. The frontend was considered priory as both steering and suspension design were dependent on these dimensions.

6.1.2 Preliminary Design

The preliminary design process is the evaluation leading up to the selection of the best overall design. The preliminary design also includes the overall system configuration, basic schematics and layout. The first step was to ensure that the control parameters are met, these include:

- The Formula SAE-A Rules
- Constraints Set by Other Components

- Finance Funding
- Project Timeline

The first designs were sketched using Autocad a two dimensional computer aided drafting software program. During this design stage the entire chassis was designed even through the engine type and rear suspension format were still undecided. The volume area required for the drivers cockpit required human factors data, which was obtained from another project.

During the preliminary design several possible chassis styles were considered, which all met the required control parameters. A decision then had to be made on one of the designs as the preliminary design. This design would then progress through to the next stage of the design process.

6.1.3 Prototyping and Redesign

Prototyping is a fundamental part of design as it allows models of the design to be tested before the final design is committed. Prototyping is used to emphasize the value of the design by constructing an inexpensive model which is then an aid to help grasp the relative size and the interrelationship of the design. Prototyping also helps resolve problems associated with component interface that may not be obvious in the preliminary design.

For the cockpit of the chassis it is very hard to design for human factors allowing comfort for the driver. This involves analyzing the ergonomics of the compartments where the body goes. Anthropometric data was first used to approximate the size of the cockpit area for the preliminary design. A real size prototype of the cockpit was then constructed out of light timber. This allowed the drivers to physically sit in the cockpit while components such as the drivers seat, steering and front suspension could also be fitted. Figure 6.1 shows the prototype frame with a driver, testing the parameters of the human body with the cockpit.

Once the prototype had been tested, modifications and adjustments could be included

Figure 6.1: Prototype Chassis

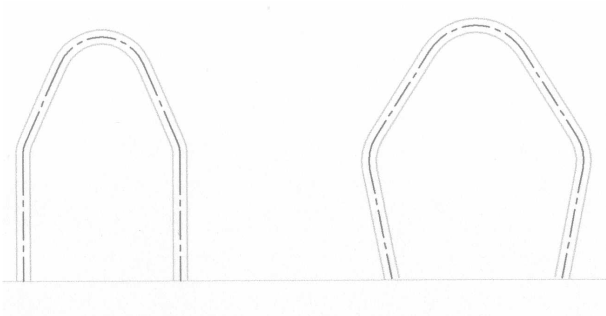

Figure 6.2: Changes Made to Front Hoop

into the design. The addition of these improvements are known as redesign. Without the prototype many of these design changes would not been picked up and could have led to costly problems further into the design process. These design changes included the widening of the front hoop in the middle section as shown in figure 6.2 this will provide more driver knee room and comfort. The main hoop was also widened to allow for a bigger seat and provide more shoulder protection. The widening of the main hoop as shown in figure 6.3 allowed for the seat to then be moved further back under the hoop. By making this change then allowed for the height of the hoop to be lowered while still protecting the drivers head in a roll over accident. The overall length of the cockpit was also lengthened to provide more room for pedal boxes and master cylinders. This was required due to the design of the pedal boxes being unknown at the time and conservative action was taken.

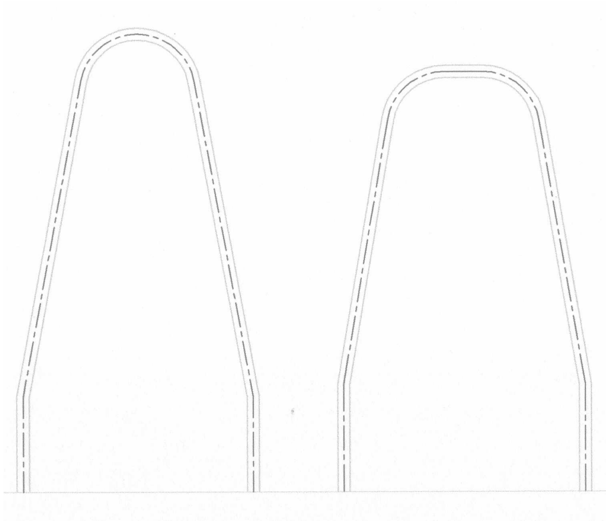

Figure 6.3: Changes Made to Main Hoop

6.1.4 Detailed Design

The purpose of the detailed design phase is to develop a system of drawings and specifications that completely describes the final design. It is at this stage of the design process where every part of the chassis is specified in detail. It is also during this stage were the component requirements are incorporated into the design.

The detailed design also includes specifications relating to:

- Operating parameters
- Maintenance requirements
- Material requirements
- Reliability
- Product design life

The detailed design should be completed with detailed drawings to allow for member manufacture as well as assembly drawings to aid in the fabrication of the chassis.

The detailed design for the chassis mainly involved converting the 2 dimensional preliminary design into a 3 dimensional tubular solid model. From the solid model all of

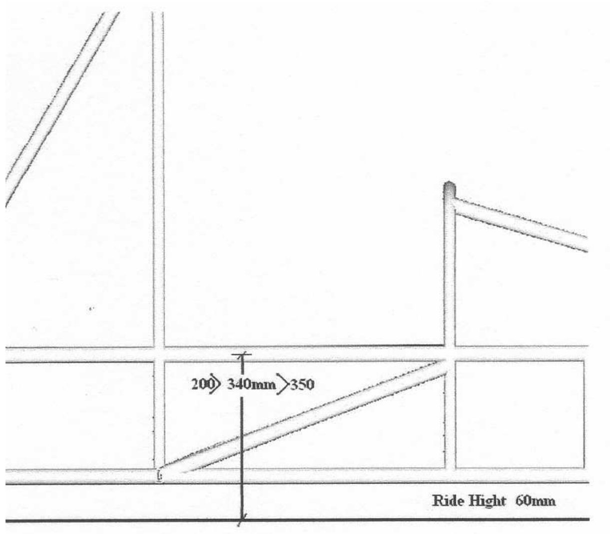

Figure 6.4: Upper Member Ruling

the complex joinery could be modeled illustrating how each member had to be notched. Once the solid model was approved detailed drawings were then produced for manufacture.

There were very few product specifications allocated in the detailed design as they were set prior to the design process or as part of the formula SAE rules.

There is very little maintenance required on the chassis except for observing the welds for cracks. If a crack is found in a weld then it needs to be ground out and rewelded. The material requirements have also been specified in the material selection process. The reliability of the chassis has been rated very high. The design of the chassis requires high reliability due to the severe nature of failure if one did occur. The chassis design life is closely related to the reliability. Once the reliability of the chassis starts to decline the life of the chassis is over. For the majority of racing car chassis, the chassis life is ended due to collision before it's design life has been reached.

Part of the detailed design is to insure that the formula SAE rules and regulations are met. The rules and regulations were taken in account in the preliminary design stage but because many changes were made since, it is important to have a second check. Figure 6.4 shows the solid model of the final design being checked to ensure that the upper member of the side impact protection in between the correct heights.

6.1.5 Construction Planning

The construction planning process is initiated as a review to identify what equipment, machines and tooling will be required to perform the construction operations for the product. It will often include the sequence of procedures and the accuracy required, along with the estimated production time.

The construction planning stage is often a process that is skipped and then causes many problems further in the construction stage. Therefore by having a well planed construction process will improve the efficiency and decrease the change of delays. The construction process for the space frame firstly involved sourcing the required material from a supplier.

The process also required locating a suitable workshop for construction. The workshop resources that were required included a welding bay and TIG welder, tube notching equipment, pipe benders and a spray booth. For construction of the spaceframe chassis two main workshops were used. A private workshop at Willowbank was used for measuring, cutting and notching of the pipe. This workshop also had the specialized notching equipment that was required and was not available anywhere else. The university mechanical workshop was used for the welding as it had a very well equipped welding bay and was much more assessable for the team. The workshop also had very large painting booth which was required to properly and safely paint the chassis.

Neither of the workshops had tube benders that could successfully bend the selected pipe for the front and main hoops, many inquiries had to then be made to find a suitable bender. Eventaually a workshop in Ipswich was found with a mandrel bender that could bend the tube. Planning was then arranged to courier the pipe and detailed drawing to Ipswich and return.

For the early stages of the of the construction, trips to willowbank had to be organized and additional helpers sourced to achieve maximum output during these work sessions.

6.1.6 Construction

The construction process for the chassis was very time consuming and required around 40 man hours from start to finish. The fabrication steps required firstly measuring the lengths of the tube and cutting them at the required length, an extra 2 centimeters had to be added to the end of each cut to allow for the notching.

The notching involved a special custom jig that was attached to the toolpost of a metal lathe. The pipe was held in the jig and a hole saw was turned in the chuck. This method allowed high precision notching at a variety of angles required. When the bent front and main hoop arrived they had to also be cut to length and notched.

Due to the engine and rear suspension setup not being finalised at this stage, the frontend of the chassis was constructed first. This also allowed for front suspension and steering component designs to be finalised on completion. It also allowed the construction of the frame to commence without being delayed by other components.

To assist with the welding of the frame all the members were lightly sanded to remove any surface corrosion or burs. They were then rinsed in solvent to remove any oil or coolent left on the pipes.

The fabrication of the frame required the members to be firmly clamped in position. Each member was then checked to ensure that it was level and square with adjoining members. Many minor adjustments could then be made my lightly tapping the member, before then being tack welded in place.

When the rear suspension setup and engine were decided, the rear chassis design could be finalised. Fortunately the existing rearend design of the chassis could accommodate these components and no redesign was required. This also allowed the construction to commence immediately on the rearend. Figure 6.5 shows the front end of the chassis with the rearend setup being checked.

When the entire chassis was tack welded together the members could then be fully welded. By originally tack welding the chassis first meant that if any changes needed to be made, then it was a simple and easy task of breaking the tack and retacking the

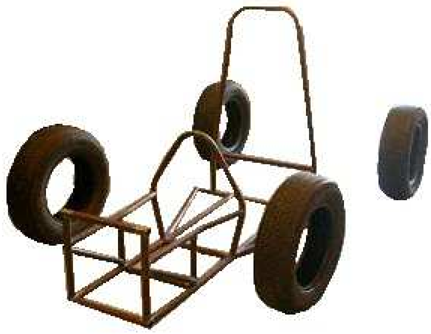

Figure 6.5: Frontend of Chassis

member elsewhere. By welding the entire chassis at once also allowed the welder to assess all the weld areas and then adjust his welding method to reduce any residual stresses and warping in the chassis.

When the chassis was completed it was taken to the painting booth and given a light coat of metal primer, this prevented any surface corrosion from forming. When the mounting brackets had to be attached to the chassis the primer was scraped off in that area, to provide a clean welding surface for the mount. Prior to race day the chassis will be lightly sanded and reapplied with the primer before a final paint coat is applied.

Chapter 7

Assessment of Chassis

7.1 Introduction

Assessing the chassis properties is a critical testing phase to ensure that the chassis will perform under the applied loads with out failure. The position of the center of gravity, the amount of member deflection and the torsional stiffness are all significant parameters that will influence the overall driving performance of the vehicle. These parameters will all be analysed using various procedures to ensure that the chassis will perform to it full potential. The material stresses reached could possible be another testing area but under standard driving conditions these stresses are minimal and will not be tested.

The material properties used in the analysis of the chassis are very critical. They can cause severe calculation errors which could then lead to incorrect results showing that the chassis is much stronger or stiffer that it actually is. The material properties for the CDW tube used in the construction of the chassis are listed in table 7.1.

The material properties were taken from tables and calculated using various formulae, sources can be sighted in Appendix E

Table 7.1: Isotropic Material Properties for CDW Steel.

Young's Modulus	200GPa
Density	$7800kg/m^3$
Poisson's Ratio	0.3
I_{xx}	$11711mm^4$
Cross sectional Area	$95.38 \times 10^{-6}m^2$

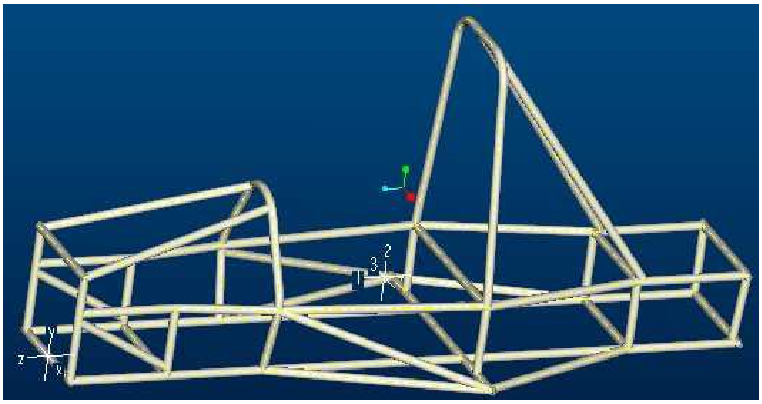

Figure 7.1: Center of Gravity of Chassis

7.2 Center of Gravity

The center of gravity (CG) is a simple representation of the objects position of weight. It is a single central point at which forces can act on the body as a whole.

(Giancoli 1991) defines the center of gravity as an imaginary point that cannot be seen or touched. However, no matter how small or large the system is, it can be picked up at this point and will remain in balanced equilibrium.

There are many methods of estimating the CG but none are 100% accurate. The CG is used in assisting with determining how the chassis will perform under race conditions.

The CG for the formula SAE chassis was determined using ProEngineer. A solid model of the chassis was drawn up and the center of gravity was found using a model analysis function. The function requires the density of the material and then calculates the position. The CG was found to be 225mm in the Y axis and 1191mm in the Z axis from the coordinate origin X,Y,Z. The CG can also be shown in figure 7.1 as the 1,2,3 origin.

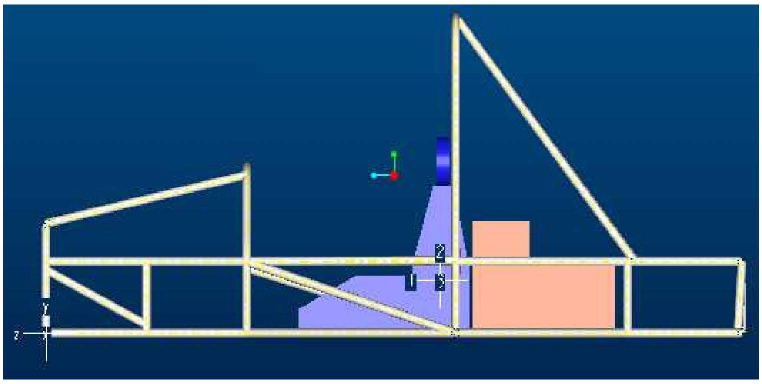

Figure 7.2: Center of Gravity of the Major Components

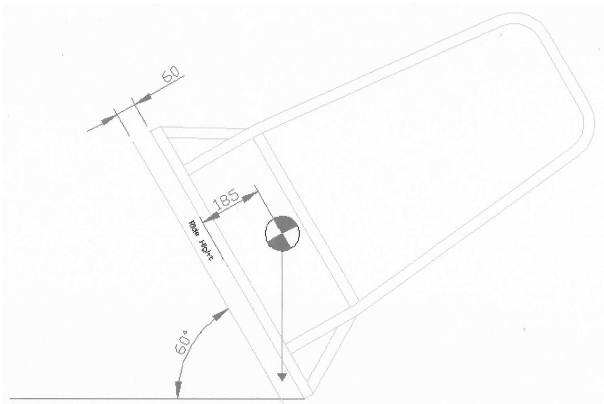

Figure 7.3: Vehicle on Tilt Test at 60 Degrees

To find the center of gravity of the vehicle the other major components had to also be simulated. The motor and the driver had to be constructed as solid models and then the density was adjusted so that the engine had a mass of 58kg and the mass of the person was 100kg. The engine and driver models were then assembled in the solid model with the chassis to determine the overall CG as shown in figure 7.2.

The CG of the major components is 1377mm in the Z direction and 185mm up in the Y direction and is also shown as the 1,2,3 origin in figure 7.2. Now that the vehicle CG has been estimated the wheel load percentages can also be calculated. This will assist in the suspension design and vehicle set up. Some performance characteristics can also be predicted depending on the CG of the vehicle. For example, less body roll will be experienced if the CG is low. The estimated CG point can also be used to ensure that the vehicle passes the static tilt test. Provided that the force of gravity which is a vertical line through the CG, is lower than the bottom tyre contact surface, then the vehicle will not tip over. Using the CG calculated in ProEngineer, the formula SAE

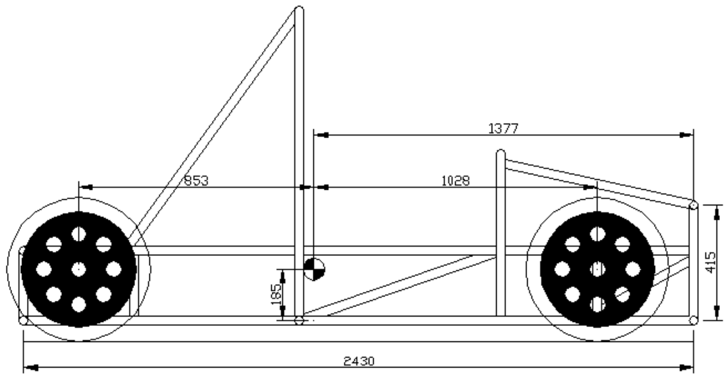

Figure 7.4: Distance from the CG to the Wheels

vehicle should pass the tilt test as shown in figure 7.3. The figure shows the force of gravity line inside the wheel. Figure 7.3 also shows that the ride height also needs to be taken into account in the test, therefore the real CG hight of the vehicle is 245mm.

The position of the CG also shows how the wheel loads are going to be distributed from the front to the rear. Ideally, for premium performance, the CG should be in the center of the vehicle half way between the wheel centers. This allows the weight to be evenly distributed across all four wheels. Figure 7.4 shows that the CG of the formula SAE vehicle causes the load to slightly favor the rear wheels. The percentages are still very close to 50% which is the ideal. This CG position can also be slightly improved my adding other components further to the front of the vehicle.

7.3 Deflection Analysis

Chassis deflection is one of the major design faults and is the largest contributor to vehicle failure in the formula SAE competition (P. Clark Formula SAE-A Technical Support). These chassis deflections are caused by having component mounts in the middle of an unsupported beam. When the component produces a force the load transfers through the mount causing a large bending moment in the member. These loads are usually formed under hard cornering and braking conditions when performance in critical.

The formula SAE-A chassis was analysed for deflection using ANSYS finite element

Table 7.2: FEA Model Report

Analysis Type	Structural
Package Used	ANSYS 5.5
Elements Used	3d Elastic Beam (Beam4)
Number of Nodes	292
Number of Elements	318

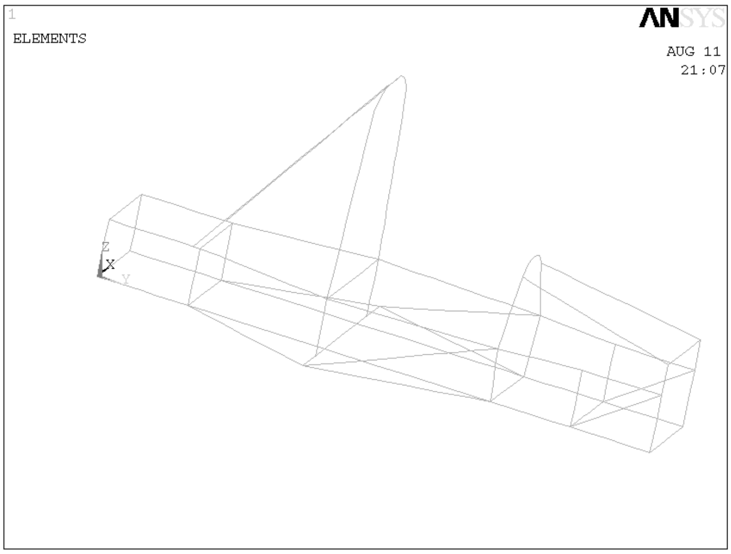

Figure 7.5: 3 Dimensional Model in ANSYS

analysis (FEA) software. The chassis was drawn up in 3 dimensions as a series of lines as shown in figure 7.5. Each line was then meshed into beam elements. These elements are ideal for measuring deflections as they are only a single element that can be given the material properties of the tubular members used. Each line was meshed into 6 beam elements to give sufficient accuracy. A plot of the nodes is shown in figure 7.5. This plot allows the user to view the size of the elements. Table 7.2 shows some of the ANSYS model properties used to simulate the FEA results, this short report can also be used to compare any further FEA analysis that may be conducted.

When modeling with FEA software it is difficult to match the element properties with the real material properties. There are usually many errors experienced in the first couple of trials due to incorrect assumptions and units. For this reason, a simple

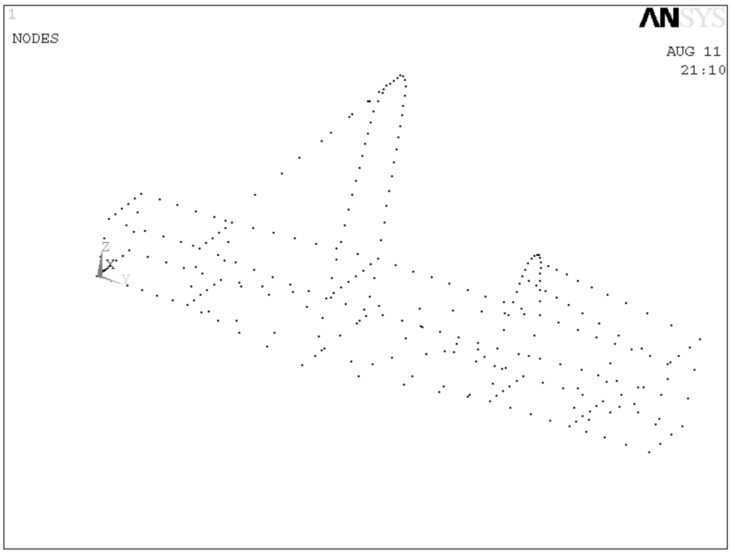

Figure 7.6: Nodes of the Meshed Chassis

analysis was conducted in FEA which could be easily verified with the physical frame. The test involved constraining the rear of the chassis and applying a simple downward load to the front bottom member of the bulkhead as can be seen in figure 7.7. The maximum deflection from ANSYS was recorded to be 16mm for a load of 850N. A similar test was then simulated on the real chassis to measure if similar results could be achieved. The rear of the chassis was clamped to a solid press bench. The press, along with 'G' clamps were used to constrain the rear of the chassis to the bench. A static mass was then applied to same front member of the chassis to copy the FEA test. A laser level and a rule were used to measure the deflection in the chassis. The deflection from the physical test was measured to be 14mm which was 2mm less that the ANSYS results. The 2mm difference was considered to be very close and was even expected due to the welded joints slightly increasing the stiffness. From this comparison analysis test, it was assumed that the software result was very accurate and further, more complicated tests could be conducted.

All of the predicted load applications from the design criteria were all tested in ANSYS. The tests simulated the forces generated by the masses of the engine and the driver under acceleration, deceleration, and corning applications. The conditions were simulated

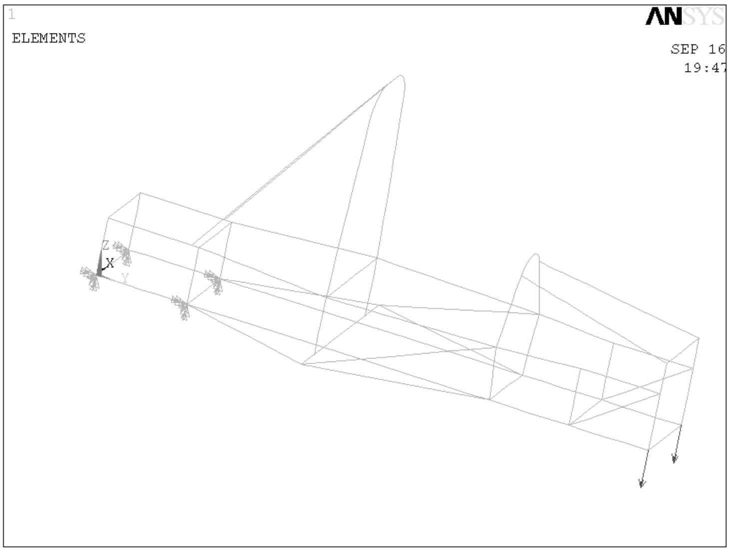

Figure 7.7: ANSYS Comparison with Physical Chassis

Figure 7.8: Chassis in Testing Rig

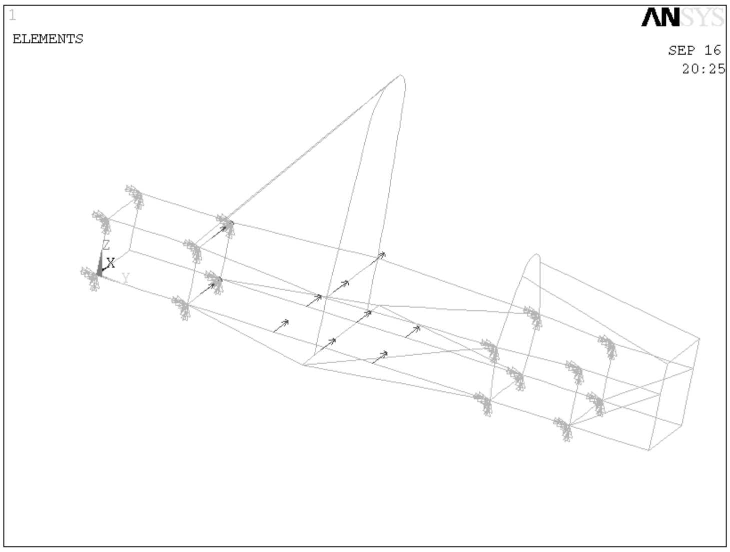

Figure 7.9: Analysis of Cornering Loads

by constraining the chassis at the wishbone attachments and applying the forces at the component mounting points on the chassis. Figure 7.9 shows one of the model chassis simulations. This example also shows where the chassis was constrained and where the forces were applied. The simplicity of the models allowed for these three tests to be solved very quickly in the FEA program. The ANSYS results showed that the deflections caused by these simulated loading conditions were very minimal. The maximum deflection recorded was for the decelerating condition where the program outputted a deflection of 0.0787mm. A plot for these results can be seen in figure 7.10,the figure shows deflections that have been severely amplified so the deflected areas can be seen, it is not an estimate of what the deflections will look like.

These FEA results show that deflection caused by the components in acceleration, deceleration and corning application will be very minimal and will not effect the performance of the chassis.

The braking forces caused by the caliper were then modelled in the FEA software. For the condition to accurately simulated, the uprights and the wishbones had to also be modelled and meshed in the program. As calculated in the design criteria, only the

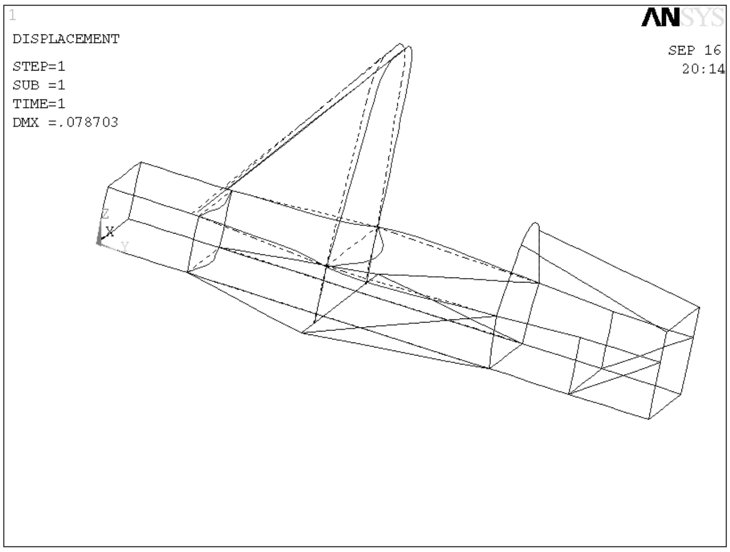

Figure 7.10: Deflection Analysis for Decelerating Loads

front brakes will be modelled as they supply most of the braking force. The load was applied as a moment through a keypoint in the middle of the upright. The rear of the chassis was constrained to simulate the weight that would be holding the chassis down. Figure 7.11 shows the conditions applied in the modelling of the braking simulation.

The results of the braking forces recorded a maximum deflection of 0.103mm. According to (Howard" 2000) the forces caused by the brake caliper are the largest working forces produced in a track racing vehicle. These forces caused minimal deflection as the wishbones spread the moment, reducing its intensity. The fact that the wishbones are also attached to very rigid points on the chassis helps to minimise any deflection created. These results confirm that the deflection caused by the braking forces will not effect the chassis performance and that the mounting points for the wishbones have been correctly chosen.

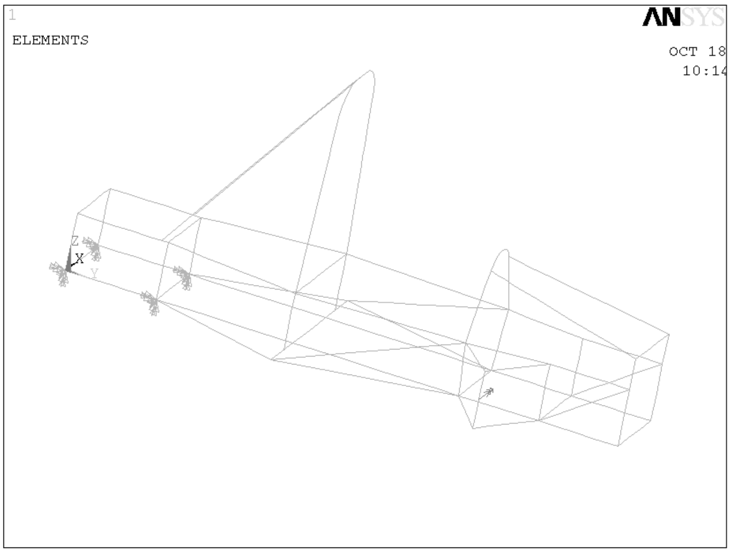

Figure 7.11: Load Setup for Braking

7.4 Torsional Stiffness

The torsional stiffness of a chassis is very important and is often measured to compare different chassis. The torsional stiffness is the rate which the chassis resists twist and is measured in Newton meters per degree of twist. The torsional stiffness for the chassis was required for the FSAE-A Design Specification Sheet which is in appendix C this to be submitted to the event organisers. It was decided to calculate the torsional stiffness manually by conducting a physical test. The test was conducted by constraining the rear of the chassis on the press similar to the physical deflection test. A steel bar was attached inside one of the horizonal bulkhead members, to create a leaver arm of 500mm from the center axis of the chassis. A weight of 80kg was then attached to the lever arm to generate a moment. The vertical movement of the lever arm was measured to be 15mm. From this measurement the torsional stiffness could then be calculated. Figure 7.12 shows a 2 dimensional representation of the load applied and the results recorded.

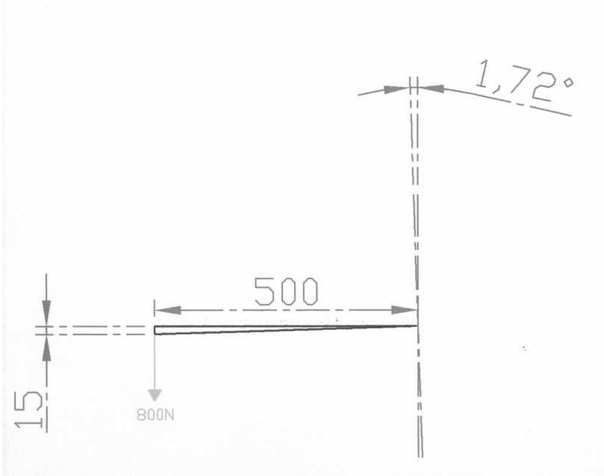

Figure 7.12: Torsional Stiffness Measurements

Formula used to calculate the torsional stiffness was:

$$Torsional\ Stiffness = \frac{Moment}{Degrees\ of\ Twist} \tag{7.1}$$

where

$$\begin{aligned} Moment &= 80\text{kg x } 10ms^2 \text{ x } 0.5\text{m} \\ &= 400Nm \\ Degrees &= 1.72 \end{aligned}$$

$$\begin{aligned} Torsional\ Stiffness &= \frac{400}{1.72} \\ &= 233Nm/deg \end{aligned}$$

This torsional stiffness measurement appears to show that the chassis is very stiff which will allow for superior performance. The absence of any data of what other formula SAE-A teams have recorded makes comparison prove difficult.

7.5 Conclusion

The assessment of results is a very important section of any design project. It allows the SAE-A team to predict the performance characteristics of the chassis without driving the vehicle. Because the chassis supports the driver's cockpit, it is important to ensure that the chassis will support the working loads of the vehicle without failure.

The results gained in this section show that the chassis will experience very minimal deflections under race conditions. This is also going to allow the chassis to perform at an optimum level in conjunction with the rest of the vehicle setup.

The results could be interpreted to show that the chassis is oversized, and weight could have been saved by the use of smaller members. It also however needs to be recognised that the chassis was designed to be flexible for various mounting points which could not be finalised until after the initial chassis deign stages.

Chapter 8

Recommendations

8.1 Introduction

When designing and building anything for the first time, it is almost impossible to predict all of the possible design criteria required to create the perfect design. As this was the first formula SAE chassis designed at the university a conservative approach was taken to prevent failure and achieve flexibility.

During the construction stages of the vehicle when components were being attached to the chassis, many small areas were found where design improvements could be made. These areas were unknown at the design stages of the chassis but flexibility allowed completion to occur without any major rework required.

8.2 Design Improvements

The main design fault found, was the chassis compatibility with the wishbones and the front upright setup. When the wishbones were attached to the bottom rail of the chassis they were on a slight upward angle. It was then detected that this angle caused the rod ends to fail at the full travel position. This problem was overcome by placing additional braces to the chassis. At the front they were placed between

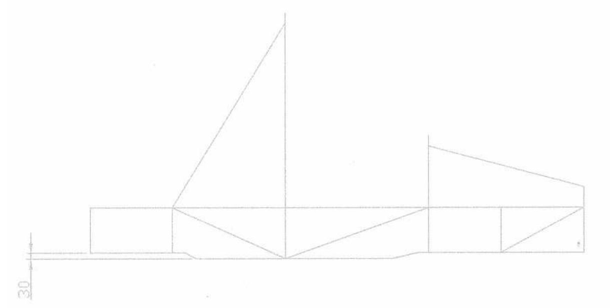

Figure 8.1: Chassis with Raised Front and Rear Rails

the front roll hoop and a support bar, and at the rear they were placed between the two vertical members. Ideally, the most functional design would have incorporated having the bottom rails of the chassis slightly raised at the front and rear as shown in figure 8.1. The middle portion of the chassis should still remain lower to obtain the low center of gravity. This approach would slightly complicate the chassis but simplify the mounting points. Another small design improvement that was recognized with the suspension setup, was to have the bottom rails narrower than the top rails. This would slightly offset the suspension pivot axis. The offset would produce a small amount of positive camber angle when the suspension is compressed and negative camber when in rebound. Overall, this would improve the suspension characteristics and improve the cornering and handling performance of the vehicle.

Recommendations for a future chassis could also include the positioning of the seat mounting points. To securely mount the seat, two horizontal bars were required. One has to run between the two bottom rails which supports the front of the seat and provides an attachment point for the submarine strap on the racing harness. The other bar which supports the back of the seat runs between the main hoop and supplies the mounting points for both shoulder straps on the racing harness. These bars had to be placed very close to already existing members on the chassis. If the type and placement of the seat was known prior to the original design of the chassis, then the structural members could have been repositioned to also support the seat and this double up would not have occurred.

The final area that could be improved is the length of the cockpit. When the pedals are adjusted for taller drivers, the room left in the front of the cockpit is slightly cramped with minimal room for master cylinders and brake bias bars. This problem was not realized in the prototyping stage as the seat type and position was changed. If the length of the cockpit was lengthened by another 100mm it would greatly assist in the design of the pedals and improve the driver's comfort.

8.3 Optimizing Chassis Design

When competing in racing events it is important to optimize all possible areas to obtain an advantage over the other teams. The main area where the chassis can be optimized is the weight. Reducing the weight of the chassis will increase the overall performance of the vehicle. The most obvious way to reduce the weight of the chassis is to use smaller and lighter tubes for all members that do not carry critical loads. Many of the experienced and competitive teams use up to 4 or 5 different sized tubes throughout the frame depending on the load condition. To achieve this design the designer has to be aware of all the critical mounting points. These hard points have to be set early in the design process so the chassis can accommodate these load paths produced.

8.3.1 Alternative Materials

The performance characteristics and the weight of the chassis can be improved by using materials with greater mechanical properties. By using a higher performance grade material, less would then be required to obtain the same chassis strength and deflection properties. If the teams' budget allowed, future chassis could be constructed out of chrome molybdenum or a high grade of Aluminum. The additional cost of using these high performance materials can range from 70% more for aluminum and around 300% more for chrome molybdenum.

As seen in the cost report in appendix F the mounts integral to the frame can add up to almost 20kg, which is almost another 50% of the chassis weight. Many of these mounts only carry light loads and could easily be constructed from light aluminum. Because aluminum cannot be welded to steel, attaching the brackets would prove complicated. However if the chassis was also constructed out of aluminum the brackets could be easily welded on. Another advantage of constructing the chassis out of aluminium is that no corrosion will occur. If aluminium was used it would be recommended then not to paint over the aluminium. Leaving the welds visible would allow for easy inspection of fatigue cracks. The main disadvantage with aluminium construction is that the complicated welding techniques required are currently not available to the university.

8.4 Conclusion

There are only a few recommendations of improvement that could possibly be made to the chassis if a new formula SAE-A chassis was going to be designed and constructed. Combining the use of a higher grade material and different sized members could significantly improve the chassis performance, it could be approximated that a weight saving of between 10 and 20kg could be easily achieved. There are possibly many more design changes that could be recommended to the chassis to improve the performance. Without testing the vehicles performance as a complete system makes determining further recommendations difficult. Many other improvements could be found by networking with other formula SAE-A teams to experience what design methods work well, and what methods could result in failure on race day.

Chapter 9

Conclusion

Like any vehicle, the chassis is the backbone of the system. Every critical component relies on the chassis either directly or indirectly. The driver of the vehicle also relies on the chassis for protection in the event of an accident.

This dissertation covers the procedures that were used to successfully design and construct a functional formula SAE-A chassis.

9.1 Summary of Project

The following objectives have been addressed:

Background Chapter 2 presented the history and evolution of spaceframe technology. It also covered the relevant construction procedures required in creating a spaceframe.

Literature Review Chapter 3 is a summary of recent literature used to help gain knowledge and techniques required to design an efficient and effective spaceframe chassis.

Design Criteria Chapter 4 discusses all of the areas that in some form influence the design of the chassis.

Figure 9.1: Completed Formula SAE-A Chassis

Material Selection Chapter 5 covered all of the possible materials that could be used in the construction of the chassis and the advantages and disadvantages of each.

Design Process and Methodology Chapter 6 followed the design process planning that was used for the chassis and the construction procedures taken.

Assessment of Chassis Chapter 7 discussed the analysis procedure conducted on the chassis and the results achieved.

Recommendations Chapter 8 covers all of the improvements that would be recommended for future formula SAE-A chassis.

9.2 Achievement of Project Objectives

All of the project objectives were met with the completion of a efficient and effective spaceframe chassis for a formula SAE-A racer as illustrated in figure 9.1. Throughout the process of this project many difficulties were encountered and overcome to reach a successful completion. Unfortunately due to time constraints the subsidiary objectives were unable to be completed although some recommendations for improvement were suggested for a future design of a formula SAE-A chassis.

9.3 Further Work

The only further work that would be required for this project would be to fully complete the subsidiary project objectives. This involves the analysis of the chassis as a component of the completed formula SAE-A Racer. Reporting on its impact on handling and performance of the vehicle would be very useful.

Further work that could additionally be conducted on the chassis is in the area of stress analysis. Study into the stresses experienced under working conditions and in the event of a collision could also be highly useful in the understanding of chassis characteristics.

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Appendix A

Project Specification

This appendix contains a copy of the project specification that was drawn up as part of the requirements of the project work, for the University of Southern Queensland. It details the objectives of the project.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

For: **Christopher Baker**

Topic: FoES Formula SAE-A Space frame Chassis Design

Supervisor: Mr Chris Snook

Project Aim: To design a chassis of a formula SAE-A Race car. The chassis must be designed to be able to contain the necessary components and driver. It must also comply with the formula SAE-A 2004 rules.

PROGRAMME: **Issue A, 16th March 2004**

1. Research background information relating to Formula SAE-A Rules and similar formula SAE-A chassis designed by other institutions.
2. Research effective space frame chassis characteristics and design methods involving the effects of stress, torsion and deflection on a chassis with respect to vehicle handling and performance. Also taking into account the benefits and performance of different materials.
3. Design an effective and efficient space frame chassis that is to Formula SAE-A regulations and capable of being constructed from materials and resources available, while also taking into perspective other team members requirements such as engine, drive train, suspension, etc.
4. Construct a solid model prototype from the design and test appropriate factors like strength, rigidity and deflection of the chassis using finite element analysis.
5. From prototype testing, modify improvements to the chassis and generate required detailed and assembly drawings.
6. Apply to the USQ Mechanical workshop for the construction of the chassis.
7. Conduct non-destructive testing on the completed chassis for its response to torsion and deflection

   If time Permits

8. Conduct tests on the fully completed chassis while in the form of a fully completed Formula SAE-A Racer for its impact on handling and performance.
9. Recommend improvements or changes that could be implemented in a better formula SAE-A chassis.

Appendix B

ProEngineer Model Analysis

This appendix contains the ProEngineer model mass properties for the chassis(CHASSIS) model and the chassis with the model engine and driver(CHASSISFULL).

CHASSIS
VOLUME = 5.6610526e+06 MM∧3
SURFACE AREA =5.3479661e+06 MM∧3
DENSITY = 7.7000000e-09 TONNE / MM∧3
MASS=4.3590105e-02 TONNE

CENTER OF GRAVITY with respect to CHASSIS coordinate frame: X Y Z 4.5203358e-02 2.2538069e+02 -1.1914589e+03 MM

INERTIA with respect to CHASSIS coordinate frame: (TONNE * MM∧2)

INERTIA TENSOR: Ixx Ixy Ixz 8.8691563e+04 -3.8569153e+00 -1.1367099e+00 Iyx Iyy Iyz -3.8569153e+00 8.5608218e+04 1.1916488e+04 Izx Izy Izz -1.1367099e+00 1.1916488e+04 7.4263869e+03

INERTIA at CENTER OF GRAVITY with respect to CHASSIS coordinate frame: (TONNE * MM∧2)

INERTIA TENSOR: Ixx Ixy Ixz 2.4597946e+04 -3.4128208e+00 -3.4843833e+00 Iyx Iyy Iyz -3.4128208e+00 2.3728824e+04 2.1115691e+02 Izx Izy Izz -3.4843833e+00 2.1115691e+02 5.2121640e+03

PRINCIPAL MOMENTS OF INERTIA: (TONNE * MM∧2) I1 I2 I3 5.2097557e+03 2.3731218e+04 2.4597961e+04

ROTATION MATRIX from CHASSIS orientation to PRINCIPAL AXES: 0.00018 0.00398 -0.99999 -0.01140 0.99993 0.00398 0.99993 0.01140 0.00022

ROTATION ANGLES from CHASSIS orientation to PRINCIPAL AXES (degrees): angles about x y z -86.792 -89.772 -87.446

RADII OF GYRATION with respect to PRINCIPAL AXES: R1 R2 R3 3.4571221e+02 7.3784655e+02 7.5120002e+02 MM

CHASSISFULL
VOLUME = 1.1337635e+08 MM^3
SURFACE AREA = 7.3487433e+06 MM^2
AVERAGE DENSITY =1.7814771e-09 TONNE / MM^3
MASS = 2.0197737e-01 TONNE

CENTER OF GRAVITY with respect to CHASSISFULL coordinate frame: X Y Z 1.8833934e+01 1.8546921e+02 -1.3767803e+03 MM

INERTIA with respect to CHASSISFULL coordinate frame: (TONNE * MM^2)

INERTIA TENSOR: Ixx Ixy Ixz 4.2989987e+05 -7.0518094e+02 5.2448303e+03 Iyx Iyy Iyz -7.0518094e+02 4.2098637e+05 5.1671781e+04 Izx Izy Izz 5.2448303e+03 5.1671781e+04 1.6502543e+04

INERTIA at CENTER OF GRAVITY with respect to CHASSISFULL coordinate frame: (TONNE * MM^2)

INERTIA TENSOR: Ixx Ixy Ixz 4.0099140e+04 3.4921712e-01 7.5188802e+00 Iyx Iyy Iyz 3.4921712e-01 3.8061785e+04 9.6789359e+01 Izx Izy Izz 7.5188802e+00 9.6789359e+01 9.4831138e+03

PRINCIPAL MOMENTS OF INERTIA: (TONNE * MM^2) I1 I2 I3 9.4827842e+03 3.8062113e+04 4.0099141e+04

ROTATION MATRIX from CHASSISFULL orientation to PRINCIPAL AXES: -0.00025 -0.00018 -1.00000 -0.00339 0.99999 -0.00018 0.99999 0.00339 -0.00025

ROTATION ANGLES from CHASSISFULL orientation to PRINCIPAL AXES (degrees): angles about x y z 0.000 -89.982 -0.194

RADII OF GYRATION with respect to PRINCIPAL AXES: R1 R2 R3 2.1667888e+02 4.3410530e+02 4.4557025e+02 MM

MASS PROPERTIES OF COMPONENTS OF THE ASSEMBLY (in assembly units and the CHASSISFULL coordinate frame)

DENSITY MASS C.G.: X Y Z

CHASSIS MATERIAL: UNKNOWN 7.70000e-09 4.35901e-02 1.77617e+01 2.26012e+02 -1.19209e+03 ENGINE MATERIAL: UNKNOWN 9.12660e-10 5.79995e-02 1.91466e+01 1.55151e+02 -1.70818e+03 PERSON MATERIAL: UNKNOWN 2.27300e-09 1.00388e-01 1.91189e+01 1.85381e+02 -1.26551e+03

Appendix C

Formula SAE-A Design Specification Sheet

This appendix contains a copy of the design specification sheet submitted to the formula SAE-A event organisers and design judges.

FSAE-A Design Spec Sheet 2004

Competitors: Please replace the sample specification values in the table below with those appropriate for your vehicle and submit this to with your design report. This information will be reviewed by the design judges and may be referred to during the event.
--Please do not modify format of this sheet. Common formatting will help keep the judges happy!
--The sample values are fictional and may not represent appropriate design specs.
--Submitted data will NOT be made public or shared with other teams.

Car No	13
University	University of Southern Queensland

Dimensions	Front	Rear
Overall Length, Width, Height	3200mm, 1495mm, 1180mm	
Wheelbase	1800mm	
Track	1300mm	1200mm
Weight with 68 kg driver (280kg total - to be confirmed)	126kg (To be confirmed)	154kg (To be confirmed)

Suspension Parameters	Front	Rear
Suspension Type	Unequal length double wishbone	Equal length double wishbone
Tyre Size and Compound Type	185/60R13 Falken	185/60R13 Falken
Wheels	13 x 6 inch. 110mm offset.	13 x 6 inch 110mm offset.
Design ride height (chassis to ground)	60mm	60mm
Center of Gravity Design Height	240 mm located 1400 mm from front of chassis (To be confirmed)	
Suspension design travel	26mm Jounce / 26mm Rebound (To be confirmed)	26mm Jounce / 26mm Rebound (To be confirmed)
Wheel rate		
Roll rate		
Sprung mass natural frequency (in vertical direction)		
Jounce Damping		
Rebound Damping		
Motion ratio		
Camber coefficient in bump		
Camber coefficient in roll		
Static Toe and adjustment method	Adjust by steering rack tie rods.	Adjust by tie rods.
Static camber and adjustment method	Static -1 degree. Rod end adjustment on wishbones.	Static -1 degree. Rod end adjustment on wishbones.
Front Caster and adjustment method	Static 5 degrees. Rod end adjustment on wishbones.	
Front Kingpin Axis	5 degrees non-adjustable.	
Kingpin offset and trail	36.25 mm offset. 0mm trail.	0mm offset. 0mm trail.
Static Akerman and adjustment method	100% Ackerman	
Anti dive / Anti Squat	0%	0%
Roll center position static	To be determined	To be determined
Roll center position at 1g lateral acc	To be determined	To be determined
Steering System location	Rear steer below and parallel to upper wishbone.	

Brake System / Hub & Axle	Front	Rear
Rotors	Modified rear FZR600 motorcycle 245mm	Singel modified front FZR600 motorcycle 298mm
Master Cylinder	Mechanical bias bar. Twin 3/4 inch cylinders with integral reservoir.	
Calipers	Front 4 piston FZR600 motorcycle	Rear 2 piston FZR600 motorcycle
Hub Bearings	SKF LM67048/Q Angular Contact roller bearing	AR Telstar front wheel bearing
Upright Assembly	Student built mild steel with integral brake calliper mounts	Student built mild steel
Axle type, size, and material	Solid 1020 mild steel 31.75 mm diameter	Solid 1020 mild steel 38mm diameter. 25mm at CV joint spline

FSAE-A Design Spec Sheet 2004

Ergonomics	
Driver Size Adjustments	Fixed seating position, adjustable steering wheel height and adjustable pedal position
Seat (materials, padding)	Polyethylene base with customised padding to suit each driver
Driver Visibility (angle of side view, mirrors?)	200deg side visibility, mirrors placed on side of cockpit at front
Shift Actuator (type, location)	Left hand gear lever within close proximity to steering wheel on LHS
Clutch Actuator (type, location)	Steering column mounted lever on RHS upper quadrant
Instrumentation	Tachometer, water temperature guage, neutral light and oil pressure warning light
Frame	
Frame Construction	Steel tube spaceframe
Material	31.75mm x 2.1mm cold drawn steel tube. 250 Mpa Yield Strength
Joining method and material	TIG welded
Targets (Torsional Stiffness or other)	
Torsional stiffness and validation method	To be determined (Validated through non destructive testing and Finite Element Analysis.
Bare frame weight with brackets and paint	48 kg (To be confirmed)
Crush zone material	2mm Folded aluminium sheet with foam filler
Crush zone length	150 mm
Crush zone energy capacity	To be determined
Powertrain	
Manufacture and Model	1993 Yamaha FZR600
Displacement	599cc
Fuel Type	98 RON
Induction	Naturally Aspirated
Max Power design RPM	11 000 rpm
Max Torque design RPM	8 500 rpm
Min RPM for 80% max torque	6 700 rpm
Effective Intake Runner Length	75 mm
Effective Exhaust runner length	520 mm
Exhaust header design	2 into 1 by 2 (Twin exhaust system)
Fuel System (manf'r)	Modified Mikuni carburettor with student built log type intake manifold
Fuel System Sensors	N/A
Injector location	N/A
Intake Plenum volume	900 cc estimated (currently adjustable)
Compression ratio	12:01
Fuel Pressure	0.3 bar
Ignition Timing	Standard Ignition (non-adjustable)
Coolant System and Radiator location	Radiator and Electric fan on thermoswitch. Mounted in side pod on LH of main roll hoop at driver shoulder height.
Fuel Tank Location, Type	Student built aluminium tank mounted within chassis structure below drive axle.
Muffler	Student built aluminium muffler with removable baffle. Length 260mm. Diameter 100mm.
Drivetrain	
Drive Type	Chain and sprocket (530 chain) through solid rear axle
Differential Type	N/A
Final Drive Ratio	4.00 standard. (15 tooth front 60 tooth rear) Adjustable with front sprocket changes to 4.28 and 4.61.
Vehicle Speed @ max power (design) rpm	
1st	59 (51) Brackets show 4.61 final drive ratio. All units km/h
2nd	86 (75)
3rd	108 (94)
4th	126 (109)
5th	141 (121)
6th	151 (131)
Half shaft size and material	4340 Steel (Estimated) 25mm diameter. Manufactured from Ford Telstar CV shafts.
Joint type	Inner plunging type CV joint. Outer fixed type CV joint. Both AR/AS Ford Telstar

Appendix D

Formula SAE-A 2004 Rules

This appendix contains the necessary chapters of the rules that relate to any chassis restrictions.

3.1 GENERAL DESIGN REQUIREMENTS

3.1.1 Body and Styling

The vehicle must be open-wheeled and open-cockpit (a formula style body). To protect the driver, there shall be no openings through the bodywork into the driver compartment from the front of the vehicle back to the roll bar main hoop or firewall other than that required for the cockpit opening. Minimal openings around the front suspension components are allowed.

3.1.2 Wheelbase and Vehicle Configuration

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

3.1.3 Vehicle Track

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

3.2 CHASSIS RULES

3.2.1 Ground Clearance

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

3.2.2 Wheels and Tires

The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter. 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 shall not be changed after the static judging has begun. Tire warmers are not allowed. No traction enhancers may be applied to the tires after the static judging has begun.

3.2.3 Suspension

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

3.2.4 Steering

The steering system must affect at least two wheels. The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four-bar linkage at one of the pivots). The stops may be placed on the uprights or on the rack and must prevent the tires from contacting suspension, body, or frame members during the track events. Allowable steering free play will be limited to 7 degrees total measured at the steering wheel.

The steering wheel must be mechanically connected to the front wheels, i.e. "steer-by-wire" is prohibited.

3.2.5 Brake Systems

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

3.2.5.1 Brake Over Travel Switch

A brake pedal over-travel switch must be installed on all cars. This switch shall be installed so that in the event of brake system failure such that the brake pedal over travels, a switch must 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 shall be mounted between wheel centerline and driver's shoulder level vertically and approximately on vehicle centerline laterally.

3.2.6 Jacking Points

A jacking point, which is capable of supporting the car's weight and of engaging the organizers' "quick jacks", must be provided at the rear of the car.

The jacking point is required to be:

(A) Oriented horizontally and perpendicular to the centerline of the car

(B) Made from round, 25.4 mm (1.0 inch) O.D. aluminum or steel tube

(C) A minimum of 300 mm (11.8 inches) long

(D) Exposed around the lower 180 degrees of its circumference over a minimum length of 280 mm (11 in)

The height of the tube is required to be such that:

(A) There is a minimum of 75 mm (3 in) clearance from the bottom of the tube to the ground measured at tech inspection,

(B) With the bottom of the tube 200 mm (7.9 in) above ground, the wheels do not touch the ground when they are in full rebound.

3.3 CRASH PROTECTION

The driver must be protected from car rollover and collisions. This requires two roll hoops that are braced, a front bulkhead with crush zone, and side protection.

3.3.1 Definitions

These definitions apply throughout Section 3.3.

A. Main Hoop--The main rollover protection (roll bar) alongside or just behind the driver.

B. Front Hoop--Rollover protection (roll bar) in front of the driver above his/her legs near the steering wheel.

C. Frame Member--A minimum representative piece of tubing as defined by Section 3.3.3, Minimum Material Requirements.

D. Major Structure of Chassis--That portion of the chassis that is within the envelope of frame members or structure that meet the requirements of 3.3.3. The upper portion of the main hoop and its bracing are not included in defining this envelope.

E. Crush Zone--A deformable area forward of the major structure/ bulkhead of the chassis designed to absorb energy.

3.3.2 Safety Structure Equivalency

Designs that use alternative materials or tubing sizes to those specified in Section 3.3.3.1 - Baseline Steel Material, and which protect the driver to an equal or greater extent than required by Section 3.3.3.1, will be allowed, provided they have been judged as such by a technical review. Approval will be based upon the engineering judgment and experience of the technical judge.

The technical review is initiated by completing the Safety Structure Equivalency Form using the format given in Appendix A-1. The form must be submitted no later than the date given in the "Action Deadlines" located in the Appendix.

3.3.3 Minimum Material Requirements

3.3.3.1 Baseline Steel Material

The safety structure of the car, which comprises of the main roll hoop, the front roll hoop, the side impact structure, the roll hoop bracing and the front bulkhead, shall be constructed of:

Either:

Round, mild or alloy, steel tubing (minimum 0.1% carbon) of the minimum dimensions specified in the following table,

Or:

Approved alternatives per Section 3.3.3.2

ITEM or APPLICATION	OUTSIDE DIAMETER x WALL THICKNESS
Main & Front Hoops	25.4 mm (1.0 inch) x 2.4 mm (0.095 inch)
Side Impact Protection, Front Bulkhead, Roll Hoop Bracing & Safety Harness Attachment	25.4 mm (1.0 inch) x 1.65 mm (0.065 inch)

Note: The use of alloy steel does not allow the wall thickness to be thinner than that used for mild steel.

3.3.3.2 Alternative Tubing and Material

3.3.3.2.1 General

Alternative tubing geometry and/or materials may be used. However, if a team chooses to use alternative tubing and/or materials:

(A) The material must have equivalent (or greater) Buckling Modulus EI (where, E = modulus of Elasticity, and I = area moment of inertia about the weakest axis)

(B) Tubing cannot be of thinner wall thickness than listed in 3.3.3.2.2 or 3.3.3.2.3.

Note: To maintain EI with a thinner wall thickness than specified in 3.3.3.1, the outside diameter MUST be increased.

(C) A Safety Structure Equivalency Form must be submitted per Section 3.3.2. The teams must submit calculations for the material they have chosen, demonstrating equivalence to the minimum requirements found in Section 3.3.3.1 for yield and ultimate strengths in bending, buckling and tension, for buckling modulus and for energy dissipation.

The main roll hoop and main roll hoop bracing must be made from steel, i.e. the use of aluminum or titanium tubing or composites are prohibited for these components.

3.3.3.2.2 Steel Tubing Requirements

Minimum Wall Thickness Allowed:

MATERIAL & APPLICATION	MINIMUM WALL THICKNESS
Steel Tubing for Front and Main Roll Hoops	2.1 mm (0.083 inch)
Steel Tubing for Roll Hoop Bracing, Front Bulkhead & Safety Harness Attachment	1.65mm (0.065 inch)
Steel Tubing for Side Impact Protection	1.25 mm (0.049 inch)

Note: All steel is treated equally - there is no allowance for alloy steel tubing, e.g. SAE 4130, to have a thinner wall thickness than that used with mild steel.

3.3.3.2.3 Aluminum Tubing Requirements

Minimum Wall Thickness:

MATERIAL & APPLICATION	MINIMUM WALL THICKNESS
Aluminum Tubing	3.175 mm (0.125 inch)

The equivalent yield strength shall be considered in the "as-welded" condition, (Reference: WELDING ALUMINUM (latest Edition) by the Aluminum

Association, or THE WELDING HANDBOOK, Vol . 4, 7th Ed., by The American Welding Society), unless the team demonstrates and shows proof that the frame has been properly solution heat treated and artificially aged.

Should aluminum tubing be solution heat-treated and age hardened to increase its strength after welding, the team must supply sufficient documentation as to how the process was performed. This includes, but is not limited to, the heat-treating facility used, the process applied, and the fixturing used.

3.3.3.2.4 Composite Materials

If any composite or other material is used, the team must present documentation of material type, e.g. purchase receipt, shipping document or letter of donation, and of the material properties. Details of the composite lay-up technique as well as the structural material used (cloth type, weight, resin type, number of layers, core material, and skin material if metal) shall also be submitted. The team must submit calculations demonstrating equivalence of their composite structure to one of similar geometry made to the minimum requirements found in Section 3.3.3.1. Equivalency calculations shall be submitted for energy dissipation, yield and ultimate strengths in bending, buckling, and tension. Submit the completed Safety Structure Equivalency Form per Section 3.3.2.

No composite materials are allowed for the main hoop or the front hoop.

3.3.4 Roll Hoops

The driver's head and hands must be protected from contact with the ground in any rollover attitude.

This requires a main hoop (roll bar) near the driver and a front hoop. Refer to Figure 1 on the next page.

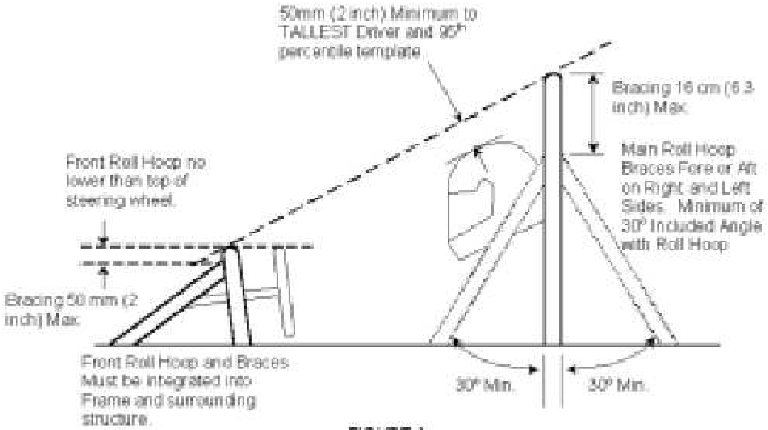

FIGURE 1

3.3.4.1 Main and Front Hoops – General Requirements

(A) When seated normally and restrained by the seat belt/shoulder harness, a straight line drawn from the top of the main hoop to the top of the front hoop must clear by 50.8 mm (2 inches) both the tallest driver's helmet and the helmet of a 95th percentile male (anthropometrical data).

A two dimensional template will be used to represent the 95th percentile male and ensure compliance.

-A circle of diameter 200 mm (7.87 inch) shall represent the hips and buttocks.

-A circle of diameter 200 mm (7.87 inch) shall represent the shoulder/cervical region.

-A circle of diameter 300 mm (11.81 inch) shall represent the head (with helmet).

-A straight line measuring 600 mm (23.62 inch) shall connect the centers of the two 200 mm circles.

-A straight line measuring 150 mm (5.9 inch) shall connect centers of the upper 200 mm circle and the 300 mm head circle.

With the seat adjusted to the rearmost position, the bottom 200mm circle will be placed in the seat, and the middle 200mm circle, representing the shoulders, will be positioned on the seat back. The upper 300 mm circle will be positioned up to 25.4 mm

(1 inch) away from the head restraint (i.e. where the driver's helmet would normally be located while driving).

(B) Both the main hoop and front hoop must each be formed from closed section metal tubing. No composite materials are allowed for the main hoop or the front hoop.

(C) Both the main hoop and front hoop must extend to the bottom of the chassis. Each hoop shall extend from the lowest frame member on one side of the car, up and over and down to the lowest frame member on the other side.

(D) The minimum radius of any bend, measured at the tube centerline, must not be less than three times the tube diameter. The bends shall be smooth and continuous with no evidence of crimping or wall failure.

(E) Proper gussets and tube triangulation must be used to ensure that the main and front hoops are securely attached to the primary structure.

(F) A 4.5 mm (0.18 inch) inspection hole must be drilled in a non-critical location of both the main hoop and the front hoop to allow verification of wall thickness.

3.3.4.2 Main Hoop

(A) The main hoop must be constructed of steel per Section 3.3.3.1 or 3.3.3.2.

(B) The use of aluminum alloys, titanium alloys or composite materials is prohibited for the main hoop.

(C) The main hoop must be formed from a single piece of uncut, continuous, closed section metal tubing that extends from the lowest frame member on one side of the car, up and over and down to the lowest frame member on the other side.

(D) In the side view of the vehicle, the portion of the Main Roll Hoop that lies above its attachment point to the Major Structure of the Chassis shall be within 10 degrees of the vertical.

(E) In the front view, the vertical members of the main hoop must not be less than 380 mm (15 inches) apart (inside dimension) at their attachment to the chassis.

(**F**) On all monocoque chassis, the main hoop must be continuous and extend down to the bottom of the chassis. Mechanical fasteners must be used to ensure positive attachment of the Roll Hoop to the monocoque. All bolts (or solid rivets) used must be 8 mm (5/16 inch) minimum diameter. The number of bolts used and their placement is for the team to determine, but proof must be submitted to show equivalency to a welded tubular chassis that meets Section 3.3.3. Mounting plates welded to the roll hoop shall not be less than 2.0 mm (0.080 inch) thick steel (or the equivalent in aluminum). Backup plates of equal thickness must be used on the opposing side of the composite structure to prevent crushing the core. All teams are required to submit a Safety Structure Equivalency report per Section 3.3.2, showing the integrity of their proposed design.

3.3.4.3 Front Hoop

-The front hoop must be constructed of material per Section 3.3.3.1 or 3.3.3.2.

-The front hoop must be formed from closed section metal tubing. No composite materials are allowed for the front hoop.

-The front hoop must be no lower than the top of the steering wheel in any angular position.

-The front hoop must extend to the bottom of the chassis. It must extend from the lowest frame member on one side of the car, up and over and down to the lowest frame member on the other side. With proper gusseting, it is permissible to make it from more than one piece of tubing.

3.3.5 Roll Hoop Bracing

3.3.5.1 Main Hoop Bracing

-The main hoop bracing must be constructed of steel, with dimensions per Section 3.3.3.1 or 3.3.3.2.

-The main hoop must be braced in the fore or aft direction on the left and right sides.

-In side view, the main hoop and the main hoop bracing cannot be on the same side of the vertical line through the top of the hoop, i.e. if the main hoop leans forward, the bracing must be forward of the main hoop, and if the main hoop leans rearward, the bracing must be rearward of the main hoop.

-Braces must be attached as near as possible to the top of the hoop but must not be more than 160 mm (6.3 inches) below the top and at an included angle of at least 30 degrees.

-The braces must be straight, i.e. without any bends.

3.3.5.2 Front Hoop Bracing

-The front hoop bracing must be constructed of material per Section 3.3.3.1 or 3.3.3.2.

-The front hoop must have two braces extending forward to protect the driver's legs.

-These braces shall be attached as near as possible to the top of the hoop, but must not be more than 50.8 mm (2 in.) below the top of the hoop.

-The front hoop bracing should extend to the structure in front of the driver's feet; but in any case it must be integrated into the chassis to provide substantial support for the front hoop. When monocoque construction is used as bracing for the front hoop, it must be approved on an individual basis. Submit the "Safety Structure Equivalency Form".

3.3.5.3 Other Bracing Requirements

-Braces attached to monocoque chassis must be welded to plates not less than 2.0 mm (0.080 inch) thick and backed up on the inner side by plates of equal thickness using solid rivets or bolts 8 mm (5/16 inch) minimum bolt diameter through the nonferrous material.

-Roll hoop bracing may be removable. Any non-permanent joint shall be of the double-lug design as shown in figures 2 and 3. Each lug shall be at least 4.5 mm (0.177 in) thick steel, measure 25 mm (1.0 in) minimum perpendicular to the axis of the bracing, and be as short as practical along the axis of the bracing. All joints must include a capping arrangement (figure 2) and/or a doubler (figure 3), fabricated of at least 1.65 mm (.065 inch) steel. If a doubler is used, it must extend at least 120 degrees around the frame member. The pin or bolt shall be 10 mm Grade 9.8 or 3/8in Grade 8 minimum. The attachment holes in the lugs and in the

3.3.6.4 Non-Crushable Objects

All non-crushable objects (e.g. batteries, master cylinders) must be rearward of the bulkhead. No non-crushable objects are allowed in the crush zone.

3.3.7 Frontal Impact Protection – Others

People shall not be endangered by contact with sharp edges on the forward facing bodywork or other protruding components. All forward facing edges on the bodywork that could impact people, e.g. the nose shall have forward facing radii of at least 38 mm (1.5 inches). This minimum radius shall extend to at least 45 degrees relative to the forward direction, along the top, sides and bottom of all affected edges.

3.3.8 Side Impact Protection

The driver must be protected from a side collision while seated in the normal driving position. Side impact must meet the requirements listed below. The material requirements are given in 3.3.3.

3.3.8.1 Tube Frames

A minimum of three (3) tubular members must be used for Side Impact Protection. These side impact members must be located on each side of the driver while seated in the normal driving position. See Figure 4. The three (3) frame members defined below must meet the requirements given in 3.3.3.

Upper Member

A member must connect the main roll hoop and the front roll hoop at a height between 200 and 350 mm (7.87 and 13.78 inches) above the ground with a 77kg (170 pound) driver seated in the normal driving position. The upper frame rail can be used as the upper side impact member **if** it meets the height, diameter and thickness requirements of the latter.

Diagonal Member

At least one (1) diagonal member per side must connect the upper and lower side impact members forward of the main roll hoop and rearward of the front roll hoop.

Lower Member

A member must connect the bottom of the main roll hoop and the bottom of the front roll hoop. This lower side impact member is normally the lower frame rail/frame member.

Alternative geometry to the minimum requirements given above must be approved prior to competition. Teams must submit a Safety Structure Equivalency Form per Section 3.3.2.

3.3.8.2 Composite Monocoque

The section properties of the sides of the vehicle must reflect impact considerations. Non-structural bodies or skins alone are not adequate to meet the side impact rule. Teams building composite monocoque bodies must submit the Safety Structure Equivalency Form per Section 3.3.2. Submitted information should include: material type(s), cloth weights, resin type, fiber orientation, number or layers, core material, and lay-up technique.

3.3.8.3 Metal Monocoque

These structures must meet the same requirements as tube frames and composite monocoque. Teams building metal monocoque bodies must submit the Safety Structure Equivalency Form per Section 3.3.2

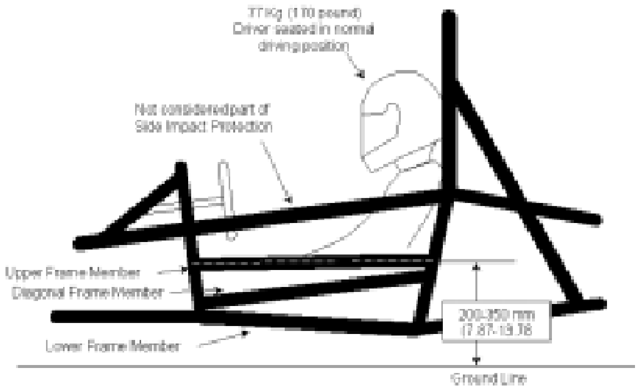

Figure 4

Appendix E

Material Properties For CDW

This appendix displays the material properties for the chosen steel to be used for the construction of the chassis. This appendix contains a copy of the technical data sheet from the supplier. The appendix also contains the isotropic material properties used in the ANSYS analysis and the references and calculations used to obtain these properties.

INDIA

TECHNICAL DATA SHEET

"TRU-WEL CDW 350"

STEEL TUBES FOR MOTOR SPORTS APPLICATIONS

PRODUCT GROUP: Cold Drawn Electric Resistance Welded Steel Tubes (CDW).

SPECIFICATION: TPI-ED/ROPS/350

CONDITION: As Drawn.

MECHANICAL PROPERTIES

Yield Strength 250 MPA
Tensile Strength 350 MPA
Elongation 25%

CHEMICAL PROPERTIES

Carbon 0.17 - 0.22%
Manganese 0.30 - 0.60%
Phosphorous 0.05% max.
Sulphur 0.05% max.
Silicon 0.45% max.

TOLERANCES

On Outside Diameter:
0 - 25 mm +/- 0.10 mm
25 - 50 mm +/- 0.15 mm
50 - 75 mm +/- 0.20mm
75 - 100mm +/- 0.25 mm

On Wall
0 - 2 mm +/- 0.10 mm
2.10 - 4 mm +/- 0.15mm
4.10 - 5.5 mm +/- 0.20 mm

OTHER DETAILS

All tubes are 100% Eddy Current Tested according to AS2084/1987. Destructive Testing consists of Tensile, Crushing and Flattening tests. Lengths up to 7000 mm +10 mm / - zero. Straightness 1 / 1000 mm.

BR BURRA RESOURCES PTY LTD

PO Box 8104, Camberwell North, Victoria 3124 Australia
Tel: 61 3 9836 7536 Fax: 61 3 9888 5511 Email: burra.tubes@bigpond.com

Isotropic Material Properties

Young's Modulus of Elasticity (Ex) 200GPa (Johnston 1992)

Density (ρ) $7800kg/m^3$ Table 10 (Johnston 1992)

Poission's Ratio (PRXY) 0.3 (Askenazi 1999)

$$\text{Cross Sectional Area (CSA)} = \frac{\pi}{4}(d_o^2 - d_i^2) \tag{E.1}$$

where

$$\begin{aligned} d_o &= \text{outside diameter(32mm)} \\ d_i &= \text{inside diameter(28mm)} \end{aligned}$$

Therefore

$$\begin{aligned} CSA &= \tfrac{\pi}{4}(32^2 - 28^2) \\ &= 188mm^2 \end{aligned}$$

$$\text{Second Moment of Inertia } (I_{yy}I_{zz}) = \frac{\pi}{64}(d_o^4 - d_i^4) \tag{E.2}$$

where

$$\begin{aligned} d_o &= \text{outside diameter(32mm)} \\ d_i &= \text{inside diameter(28mm)} \end{aligned}$$

Therefore

$$\begin{aligned} I_{yy}I_{zz} &= \tfrac{\pi}{64}(32^4 - 28^4) \\ &= 21299.99mm^4 \end{aligned}$$

Appendix F

Cost Report

As a minor part of this team project, a cost report had to be submitted to the formula SAE-A judges. This appendix contains the cost report sections that are related to this individual project. The cost report also includes the construction procedure that would be used to construct the chassis in a mass production setup.

Cost Report

Spaceframe *(Frame / Frame Tubes / Welding / Tubes Cuts/Bends)*

Construction Procedure for Spaceframe

The space frame was started by firstly measuring out the lengths of tube using a tape measure; an extra 2 cm was added to each member to allow for notching. The beams that required bending where then bent using a mandrel bender. The tubes were then cut using a cold saw. Once the tubes were cut they were then notched using a lathe and custom tube notching jig. The jig allowed easy notching for different angles. Once all the tubes where notched they where briefly cleaned in solvent to remove any coolant which improves the weldability. The chassis members where then clamped in place and tack welded. When all the members where in place, the frame was clamped down to a large level surface and the welds were completed. This prevented minimal member movement caused by the welding process.

Sub	Qty	Description	Length(m)	Weight(kg)	$/unit	Cost
A		Material		49.5	0.66	32.67
B	11	Bends			0.80	8.8
C	50	Cuts	1.5		16.00	24
D	82	Notching				35.87295
E		Assembly				69.6
F	74	welds	10.56		14.00	147.84
					Total	318.783

Sub	Qty	Amount	Unit	Manning	Description	total time(min)	$/unit	Cost
D	82	0,75	min	1	Lathe	61.5	0.58	35.67
E			min	2	assembly	60	0.58	69.6

The amount of steel required for all the members was calculated using the spread sheet in Table1. The members where all measured off the solid model. The bends where just simply counted by hand as there are only 11. The number of cuts was calculated just by using the number of members. The length of the cuts was worked out to be 30mm for each cut as this was the diameter of the tube. The notching process was calculated as labour time on the lathe, it was worked out the average time to notch a tube was about 45seconds. The assemble time was about an hour with 2 people, this also includes the solvent wipe. The length of the welds was found by measuring the length of the common welds using a length of fishing line and a rule (process shown in figures). The welds were then counted, because the chassis is symmetrical one side was counted and the answer was doubled at the end. The counting of the welds can be seen in table 2.

Mounts Integral to Frame

The integral mounts include the engine mounts, the wishbone suspension mounts, drive train mounts and floor plan mounts.

The suspension mounts are constructed from 6mm plate steel that was marked out from a stencil and then cut out using a plasma cuter. The holes where then drilled out using the pedestal drill. Once they where all cut they could be clamped to the chassis and welded.

The engine mounts where all made from different types of material. The mounts where all measured out, cut and drilled before they where welded together. Once they where made they where clamped to the chassis in the correct spot and welded on.

The floor pan mounts were made from strip steel that is 12mm wide and 3mm thick. The strips were cut using a metal shearer. The holes were then drilled in 4 goes drilling 4 mounts at once.

The drive shaft mounts were out of 50mm wide sheet that was 5mm thick; it then had two holes drilled in each mount. The mounts were then clamped on tho the chassis and welded on.

Suspension mounts

Sub	Qty	Description	Length(m)	Volume(m^2)	Weight(kg)	$/unit	Cost
A		Material			17.16	0.66	11.3256
B		cutting					37.12
C	16	Drilling				0.35	5.6
D		assembly					9.28
E	40	welds	4			16.00	64
						Total	127.3256

$2.2 \times 10^{-3}\ m^3 \times 7800 kg/m^3 = 17.16 kg$

Sub	Qty	Amount	Unit	Manning	Description	$/unit	Cost
B		16	4 min	1	cutting	0.58	37.12
D		16	1 min	1	assembly	0.58	9.28

Engine mounts

Sub	Qty	Description	Length(m)	Volume(m^2)	Weight(kg)	$/unit	Cost
A		Material			0.8	0.66	0.528
B		cutting					11.6
C	6	drilling				0.35	2.1
D		assembly					3.48
E	10	welds	1.4			16.00	22.4
						Total	40.108

$2.2 \times 10^{-3}\ m^3 \times 7800 kg/m^3 = 17.16 kg$

Sub	Qty	Amount	Unit	Manning	Description	$/unit	Cost
B		10	2 min	1	cutting	0.58	11.6
D		6	1 min	1	assembly	0.58	3.48

Floor Pan mounts

Sub	Qty	Description	Length(m)	Volume(m^2)	Weight(kg)	$/unit	Cost
A		Material			0.31	0.66	0.2046
B		cutting					1.856
C	4	drilling				0.35	1.4
D		assembly					1.856
E	32	welds	0.64			16.00	10.24
						Total	15.5566

$(0.07 \times 0.012 \times 0.003) \times 16 = 4.032 \times 10^{-5}$
$4.032 \times 10^{-5} \times 7800 = 0.31 kg$

Sub	Qty	Amount	Unit	Manning	Description	$/unit	Cost
B		16	0.2 min	1	cutting	0.58	1.856
D		16	0.2 min	1	assembly	0.58	1.856

Drive Shaft mounts

Sub	Qty	Description	Length(m)	Volume(m^2)	Weight(kg)	$/unit	Cost
A		Material			0.468	0.66	0.30888
B		cutting					2.32
C	4	drilling				0.35	1.4
D		assembly					1.16
E	2	welds	0.2			16.00	3.2
						Total	8.38888

$(0.1 \times 0.05 \times 0.006) \times 2 = 6 \times 10^{-5}$
$6 \times 10^{-5} \times 7800 = 0.468\text{kg}$

Sub	Qty	Amount	Unit	Manning	Description	$/unit	Cost
B		2	2 min	1	cutting	0.58	2.32
D		2	1 min	1	assembly	0.58	1.16

Nose Cone

Construction Procedure for nose cone.

The nose cone was started by etching the shapes on to the aluminium sheet. The shapes where then cut out of the sheet by using a gelatine. When the shapes where cut the areas that where going to be bent where annealed using an oxy-torch. When the shapes cooled the bends could be made using a sheet metal bender. Once all the sides where constructed the nosecone could be held together and the holes were drilled and ribited together. When the nosecone was made the inside was then filled with 'Selleys No More Gaps'

Sub	Qty	Description	Volume(m^2)	Weight(kg)	$/unit	Cost
A	3	Aluminum Steet 2mm		1.4	1.65	2.3142
B	20	Sheet metal Shearing			0.20	4
C		Annealing				4.64
D	10	Sheet metal bending			0.05	0.5
E		Assembly				1.16
F	18	Drilled holes			0.35	6.3
G	18	Aluminum Ribits				35.873
G		Foam fill		0.2		2
					Total	56.787

$0.26\text{m}^2 \times 0.002 = 520\text{x}10^{-6}\text{m}^3$
$520\text{x}10^{-6}\text{m}^3 \times 7800 = 1.4\text{kg}$

Sub	Qty	Amount	Unit	Manning	Description	$/unit	Cost
C		8	1 min	1	Oxy-torch	0.58	4.64
E			2 min	1	assembly	0.58	1.16

Table 1

bulkhead	top	415	
	bottom	415	
	right	415	
	left	415	
bottom rails	front right	1430	
	front left	1430	
	rear right	1000	
	rear left	1000	
top rails	front right	750	
	front left	750	
	rear right	1100	
	rear left	1100	
Side impact Protection	upper member right	735	
	upper member left	735	
	Diagonal member right	770	
	Diagonal member left	770	
	Lower member right	770	
	lower member left	770	
Upper side impact bar	left	850	
	right	850	
Uprights	front right	250	
	front left	250	
	rear right	250	
	rear left	250	
Upper Uprights	right	230	
	left	230	
Steering Bar	across	350	
Jacking Bar	across	300	
	spacer1	50	
	spacer2	50	
back plane	top	415	
	bottom	415	
	right	250	
	left	250	
cross overs	front bottom	415	
	rear bottom	415	
	rear top	415	
Front diagonal braces	right	430	
	left	430	
rear pod	right	650	
	left	650	
Main crossover		900	
Main hoop crossover		600	
Front hoop		1400	
Main hoop		2540	
main hoop braces	right	1070	
	left	1070	
front hoop brace	right	730	
	left	730	
total length of tube		32455	mm

		32.455	m
volume		0.006349	m^3
mass		49.51891	kg
	tube OD =	31.75	mm
	thickness =	2.1	
	cross section	195.6117	mm^2
		0.000196	m^2
	density	7800	kg/m^3

Table 2

welds/ cm	number
12	48
15	10
17	2
20	10
24	4
total	74
Total Length (cm)	1056

Figures