University of Southern Queensland

Faculty of Engineering and Surveying

Chassis Design for SAE Racer

A dissertation submitted by:

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Abstract

This dissertation concerns the design and construction of a chassis for the Formula SAE-Aust race vehicle – to be entered by the Motorsport Team of the University of Southern Queensland.

The chassis chosen was the space frame – this was selected over the platform and unitary styles due to ease of manufacture, strength, reliability and cost. A platform chassis can be very strong, but at the penalty of excessive weight. The unitary chassis / body is very expensive to set up, and is generally used for large production runs or Formula 1 style vehicles. The space frame is simple to design and easy to fabricate – requiring only the skills and equipment found in a normal small engineering / welding workshop.

The choice of material from which to make the space frame was from plain low carbon steel, AISI-SAE 4130 ('chrome-moly') or aluminium. The aluminium, though light, suffered from potential fatigue problems, and required precise heat / aging treatment after welding. The SAE 4130, though strong, is very expensive and also required proper heat treatment after welding, lest the joints be brittle. The plain low carbon steel met the structural requirements, did not need any heat treatments, and had the very real benefits of a low price and ready availability. It was also very economical to purchase in ERW (electric resistance welded) form, though CDS (cold drawn seamless) or DOM (drawn over mandrel) would have been preferable – though, unfortunately, much more expensive.

The frame was designed using the USQ 2004 frame as a model for dimension, with a bit added to the cockpit for driver comfort and safety, and a 100 mm reduction to the wheelbase. The basic design targets were a 20% reduction in weight and a 40% increase in torsional rigidity. The weight target was met – 38 kg versus 49 kg – as was the torsional target – 485 N.m/° versus 214 N.m/° (yet to be physically verified). The finished space frame also possesses an elegant simplicity that is pleasing to the eye.

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Limitations of Use

Certification

I certify 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.

Anthony Michael O'NEILL
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Signature

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Date

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To my project supervisor, Mr Chris Snook, for encouraging me to undertake this topic and providing his valuable assistance and encouragement throughout the duration of the project.

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Chapter 1

1. Introduction

The introduction to this project is to be covered in the following manner:

i. Formula SAE-Aust Competition
ii. Project Details

1.1. Formula SAE – Aust Competition

The objective of the Formula SAE competition is to give engineering students from around the world the opportunity to participate in a team-based competition to design, fabricate and actually race (compete) a small formula type racing car. The rules of the competition are fairly open to encourage innovation and to help minimise costs. (Any form of motor sport that is highly restricted becomes very costly – e.g. Pro-Stock Drag Racing, NASCAR, Formula 1 etc - as each team is forced to highly develop every component – at great expense - to remain competitive). There exists, however, strict safety rules (everything about the car and the competition race course is focused primarily on safety – to the extent that a vehicle that complies with the letter of the rules but, in the opinion of the judges is not safe, will not be allowed to race) and a simple requirement that the engine is less than 0.610 litre swept capacity and must 'breathe' through a 20 mm diameter restrictor.

To add meaning to the competition, the assumption is made that the whole exercise is for the manufacture and evaluation of a prototype race vehicle to cater for the non-professional weekend racer. To this end, a business presentation must be given regarding the feasibility of manufacturing 4 such vehicles per day, and that the prototype vehicle should cost less than $US25,000. In simple terms, the vehicle must be effective and efficient, not only to race but to build and maintain. Expensive and exotic materials, specialised and difficult manufacturing processes and an end product that is difficult to repair (or dangerous if repaired incorrectly) or modify should be avoided. This philosophy shall be carried through this dissertation.

The judging categories are as follows:

Static Events	
Presentation	75
Engineering Design	150
Cost Analysis	100
Dynamic Events	
Acceleration	75
Skid-Pad	50
Autocross	150
Fuel Economy	50
Endurance	350
Total Points	**1,000**

Table 1: Judging Categories & Points

The 2005 Formula SAE Series consists of three separate competitions – the United States of America, the United Kingdom (GB) and Australia (for the Australasian countries). However, any team may compete in any competition.

1.2. Project Details

This Report covers the design and construction of the chassis for the Formula SAE Racer. The details for this are given below from the Project Specification:

1. Research SAE rules to determine safety and design requirements.
2. Review and critique designs used by other teams.
3. Determination of layout, suspension type and dimensions in consultation with Team.
4. Selection of materials to be used.
5. Determination of work processes (including quality control) for construction of frame.
6. Determination of imposed loads – suspension, engine, torsional etc.
7. Research and design a suitable mounting bracket for suspension, engine etc.
8. Testing of joint strength of selected material in configurations used in chassis.
9. Determination of optimal frame design (with regards to weight, deflection and torsional stiffness) by Finite Element Analysis.
10. Liaise with Team and Faculty Workshop in the construction of the frame.
11. Testing (and modification, if necessary) of frame to ensure compliance with design and safety objectives.

This entails research into the dynamic loads on a chassis, existing Formula SAE chassis designs, types of chassis, materials selection, construction methodology and physical testing of the completed chassis.

Chapter 2

2. The Chassis – what is it and what does it do?

In general, the chassis is the supporting frame of a structure whether it is an automobile or a television set. However, the dynamics of an automobile are somewhat more severe than a television set (unless, of course, the TV. set is being hurled from a hotel's tenth floor by some deranged 'pop star'.)

The purpose of the auto chassis is to link up the suspension mounting points, final drive, steering, engine / gearbox, fuel cell and occupants. The auto chassis requires rigidity for precise handling, light weight to minimise both construction and running costs and inertia, and toughness to survive the quite severe fatigue loads imposed by the driver, road surface and power plant (Fenton, 1980, p2).

The discussion on the basic types of chassis that can be used for the Formula SAE chassis will be carried out in the following order:

1. Platform
2. Space frame
3. Monocoque / unitary
4. Evolution of the sports chassis
5. Chassis strength

2.1. Platform Chassis

The original and oldest form of chassis – used for thousands of years – even before the invention of the wheel (a sled has a chassis). This is a clay model (probably a toy) from the Harrapa Civilisation (Indus Valley) from 4000 years ago.

Figure 1: Clay Model – 2000 BC **(Owen&Bowen,1967)**

The platform chassis did not change much over the following 3800 years – below is the Cugnot Steam Tractor, which was used for hauling heavy artillery during one of those indeterminable European wars that seem to have started when the Romans left and have continued until this day. However, this was also the beginning of the Industrial Revolution.

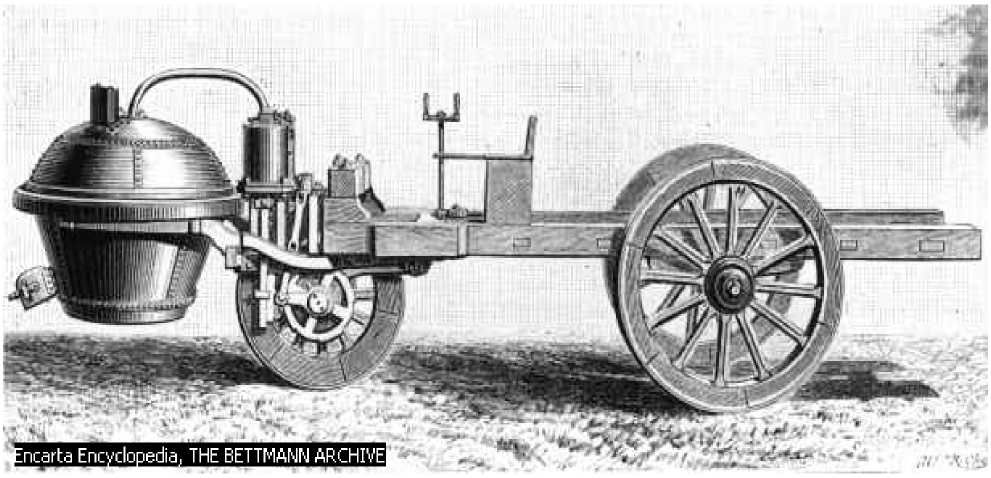


Figure 2: Cugnot Steam Tractor 1770 AD

This was a turning point in chassis design – it was the first known self-propelled road vehicle. The dynamics of this new development led to new and better things – like the horseless carriage just over 100 years later.

Figure 3: The Horseless Carriage 1890 AD

This development quickly led to the modern motorcar shown below:

Figure 4: The Modern Motorcar **(Owen&Bowen,1967)**

This was an important development for it led to the necessity of understanding the dynamics of the motor vehicle. What was suitable for a horse drawn cart was no longer suitable for a powered vehicle – and now those levels of power were becoming considerable, along with the demands of the motorist for safe and predictable handling, along with comfort and reliability. Bullock carts were no longer good enough, though Henry Ford continued to build cars with bullock cart rear suspension in Australia until the late 1980s.

The first type of chassis was the platform – shown below in *Fi*gure *5: The Platform Chassis.*

Figure 5: The Platform Chassis

This design suited the production methods of the early 20th century where a chassis and drive train were manufactured and then sent to a coachbuilder for the body to be attached to the top. (Still unable to leave the horse and cart mentality behind).

The platform chassis is simple to design and manufacture, but tends to be heavy if rigid. Also, with the platform chassis, the body is 'along for the ride' and contributes little to the overall rigidity of the vehicle. The platform chassis consists mainly of longitudinal beams – which need depth and mass for rigidity.

This design particularly suits trucks / trailers where an open platform is needed to carry loads of varying shape, size and mass. The manufacture of a platform chassis may be fully automated or by hand, depending on production requirements (e.g. mass produced trucks or specialised sports cars).

2.2. The Space Frame Chassis

The space frame uses a series of triangulated tubes to produce a structure – with each member in compression or tension. Historically, the Fokker Triplane of the Red Baron (Manfred von Richtofen) in 1917 made use of the space frame.(Bowen, 1980, p121).

The space frame can be simple or complex, as shown below:

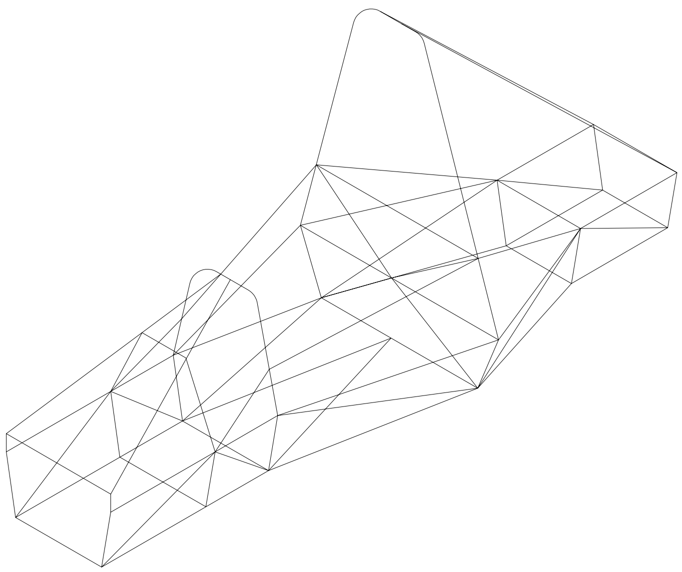


Figure 6: Simple Space Frame

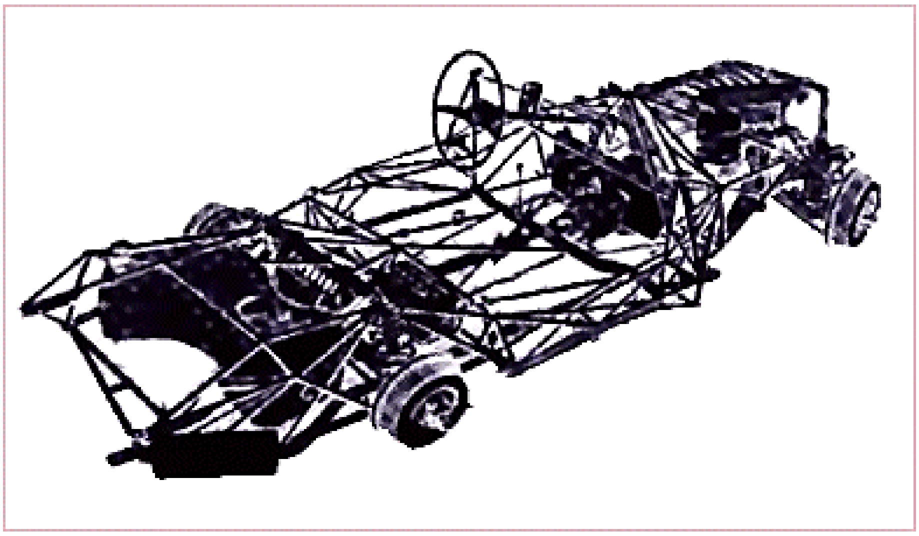


Figure 7: Complex Space Frame (Mercedes)

The difficulty of manufacture, maintenance and repair of the complex space frame shown in Figure 7 (a Mercedes sports car) has virtually seen the demise of such efforts in road going motor vehicles.

However, the potential simplicity of the space frame as shown in Figure 6 has ensured the continuing development of the space frame in the formula type race cars and also in specialist professional drag racing classes. Recent developments in Europe and the USA in hydroforming and automated procedures for construction have led to renewed interest in the spaceframe for mass produced vehicles due to the lack of expensive tooling and the ability to have a new vehicle designed and into production much more quickly (a marketing plus).

2.3. Unitary Construction

Monocoque can be defined as a type of vehicle construction in which the body is integral with the chassis. This means the chassis is less well defined (as in the platform chassis) and the body provides similar (or greater) strength as does the space frame.

In the monocoque, the body is not 'along for the ride' (as in the platform chassis) and contributes to the overall strength. This allows a large reduction in the mass of the platform parts of the monocoque. (The strength is directly related to the second moment of area – with a monocoque, because of the distance apart of the members, the actual areas of the material can be a lot less).

An excellent example was the 1960's Ford GT40 – noted for its exceptional rigidity, race winning ability and good looks.

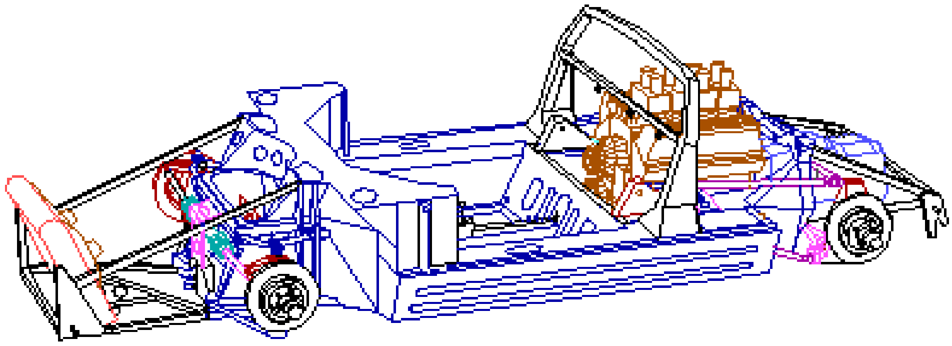


Figure 8: Ford GT40(Ford)

With proper design, the monocoque combines light weight, high strength and torsional rigidity. However, the economical construction of such vehicles is more suited to the long production runs of the modern motorcar – especially with automated (robotic) assembly lines.

2.4. Evolution of the Sports Chassis

As an interesting aside here, the evolution of the sports chassis followed a similar route as the normal road vehicle. (Fenton, 1980,p4)
The early race cars had platform chassis – and were big, heavy and slow. Then, after WW2, technology began to change with, initially, the addition of bracing tubes to the platform structure (thereby allowing a lighter platform with added rigidity). These bracing tubes became more numerous and the platform structure became less obvious – this evolution can be seen quite clearly below in *Figure 9: Evolution of the Sports Chassis:*

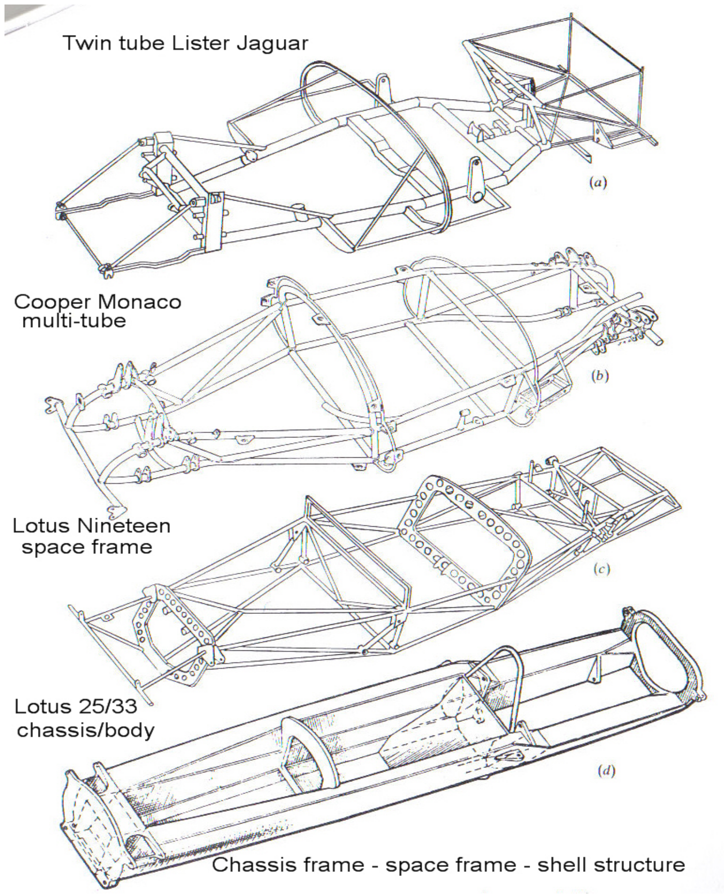


Figure 9: Evolution of the Sports Chassis **(Fenton,1980)**

This evolutionary process with the tubes continued until the space frame, then it was discovered that the unitary / monocoque type shell construction combined exceptional rigidity with light weight. One of the earliest, and best examples, was the twice Le Mans (24 hour endurance race) winning Ford GT40 of the early 1960's. Current racing vehicles using a unitary chassis make much use of expensive composites (carbon fibre etc).

2.5. Chassis Strength

The automotive chassis is affected by load transfers – longitudinal, lateral and diagonal. How little the chassis is actually affected by these loads is a measure of the chassis' worth.
Thompson, Rajic and Law in their *Design of a Winston Cup Chassis for Torsional Stiffness* state that increased torsional stiffness of a race car chassis improves vehicle handling by allowing the suspension components to control a larger percentage of a vehicle's kinematics – that is, the suspension can allowed to do its job properly (ed C Smith, 2004,p133). In their efforts, the torsional rigidity of the chassis was increased by 232% (baseline more than tripled) for a weight penalty of 5%. Of course, this is still subject to cost-benefit analysis – if the chassis was sufficiently rigid at the lower figure, then all that has happened is the vehicle has incurred a 5% weight penalty – which, in some classes of racing would be sufficient to make the vehicle uncompetitive. An example would be in US Pro Stock Drag Racing, where a 5% weight penalty would take the standing quarter mile elapsed time (e.t.) from 6.70 s to 6.81 s – in a typical 16 car field, the e.t.'s would range from 6.70 for the quickest to 6.75 for the slowest qualifier. Obviously, in this case, the time penalty for the 5% extra weight would make the car totally uncompetitive.

In order for a racing car, or any car, to handle properly it must be possible to actually tune the handling balance (Deakin et al, p107). This means that the front and rear axles can be tuned to give the same lateral acceleration ie a balanced chassis.
From this, an understeering car (which has insufficient traction at the front) may have this tendency lessened by reducing the load transfer at the front and increasing the load transfer at the rear. The catch is that this load transfer can only be accomplished if the chassis is stiff enough to transmit the torques involved.

2.5.1. Longitudinal Load Transfer:

This occurs under acceleration or braking and, as stated earlier, all forces can be regarded as acting through the vehicle's centre of gravity. This is shown below in Figure 10: Longitudinal Load Transfer.

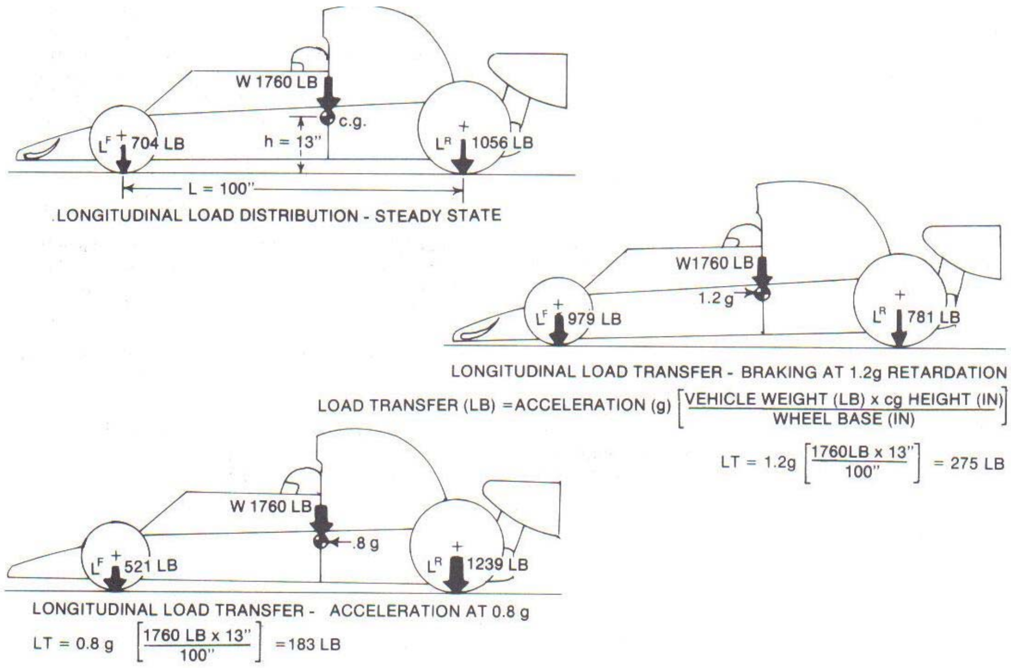


Figure 10: Longitudinal Load Transfer (C Smith,1984)

Longitudinal load transfer is given by the following formula:

LLT = (long accel x force down at axle x cg height) / wheelbase

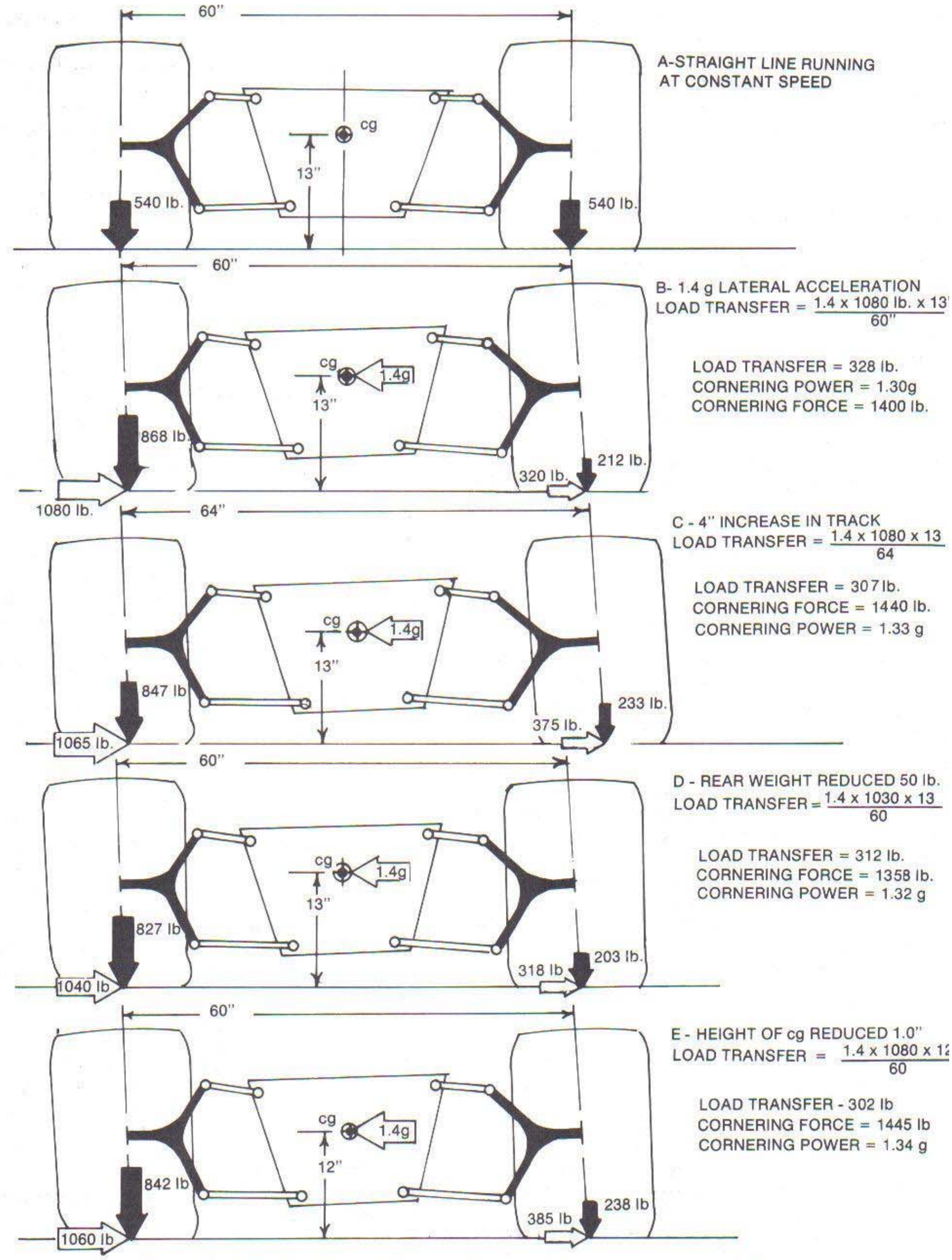
Eg accel 0.8g: LLT = (0.8 x 7848N x 0.3302m) / 2.540m = 816N

From the equation, it is obvious that longitudinal load transfer can be reduced by lengthening the wheelbase, lowering the centre of gravity, *adding lightness* (Chapman) or softening initial acceleration. (This assumes, naturally, that the chassis is strong enough to transmit these forces and not simply flex all out of shape).

Under braking, in particular, excessive load transfer can cause many problems – unloading the rear tyres (and reducing their braking ability) and loading up the front tyres (and uses up some of the suspension travel – allowing possible bottoming of parts of the vehicle).

2.5.2. Lateral Load Transfer:

This is caused by the centrifugal force generated by cornering, and exacerbated by braking (in the corners).



Simplified illustration of the relationship between track width gross weight, center of gravity height and lateral load transfer—and between lateral load transfer and cornering force

Figure 11: Lateral Load Transfer Considerations (C Smith,1984)

The basic lateral load transfer (LLT) equation is:

LLT = (lat accel x force down at axle x cg height) / track

Lateral load transfer can be reduced by lowering the centre of gravity or widening the track. The lateral load transfer can be generated in the following four ways:

1. The side forces generated by the tyres as they resist centrifugal force – these (instantaneous) forces are reacted on the sprung mass through the roll centres.
2. The physical compression of the outside springs from roll and by the deflection of the anti-roll bars (if fitted) this occurs over a finite time period.
3. By the jacking tendency of any independent suspension.
4. Lateral displacement of the c.g. due to roll – a minor effect.

The lateral forces act through the centre of gravity of the sprung mass and produce a moment around the roll centre. The roll couple will be resisted by the suspension springs and the anti-roll bars.
Chassis roll can cause unwanted changes on the wheel camber angles and, since these changes occur over a finite period of time, the result is instability and inconsistency in the vehicles handling behaviour.
Chassis roll may be reduced by stronger suspension springs, the use of anti-roll bars or by raising the roll centre relative to the centre of gravity. However, this last option has the undesirable effects of causing unfavourable wheel camber changes and high jacking thrusts. Neither of these is conducive to good handling.

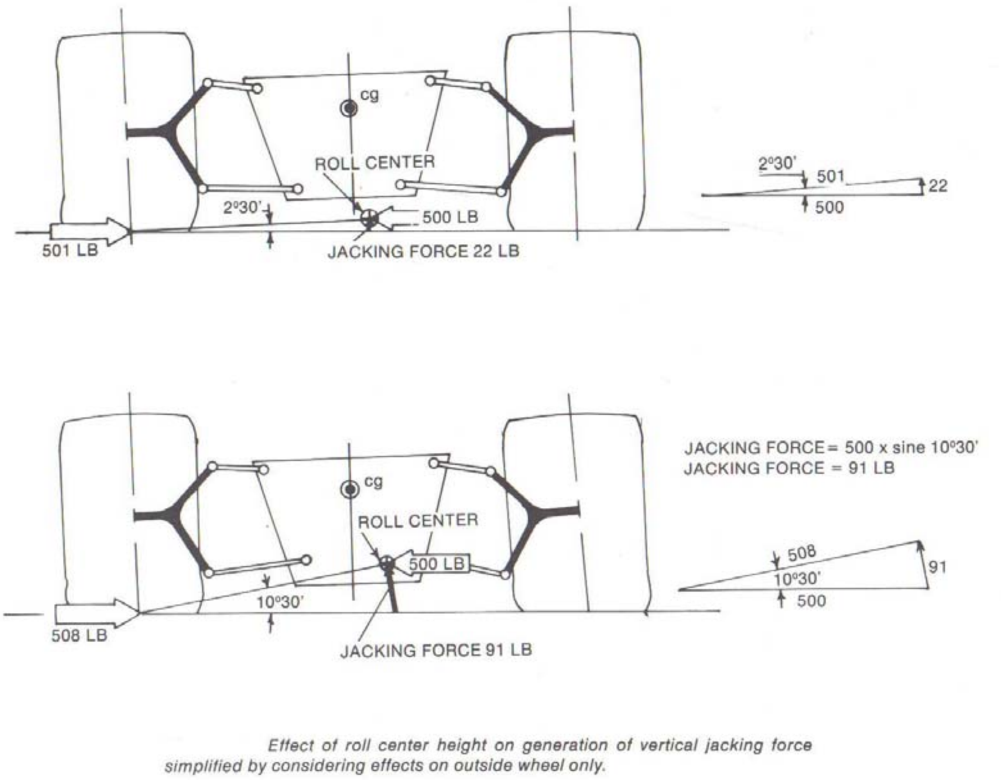


Figure 12: Vertical Jacking on Suspension (C Smith,1984)

The desirable situation is for the mass centroid axis and the roll axis to be parallel. When this occurs, the front and rear roll couples will be about equal and the vehicle will have linear front and rear roll generation and lateral load transfer – with the potential for predictable handling. Smith (1984) considers that having the front roll couple somewhat greater than the rear will cause some natural understeer and excess traction capacity at the rear for acceleration.

One of the important facets of this is to keep the centre of gravity as low as possible – the aim being to reduce the roll moment couple.

2.5.3. Diagonal Load Transfer:

This occurs when positive or negative acceleration is applied during lateral acceleration (cornering). With deceleration, the weight is transferred diagonally from the inside rear tyre to the outside front tyre – to the detriment of handling. Rear cornering power is lost by transferring load to the front, and front cornering power is lost by generating an understeer torque about the vehicle's c.g. and possibly also by overloading the front tyre or compressing the suspension spring to the point of unsuitable camber angle.

2.5.4. Required Chassis Torsional Rigidity and Strength

Deakin et al conclude that a Formula SAE racer, which has a total suspension roll stiffness of 500 – 1500 Nm/degree, requires a chassis stiffness between 300 and 1000 Nm/degree to enable the handling to be tuned (and noting that a flexible chassis will cause understeer).

This tends to follow USQ experience with the 2004 SAE car, which has a measured torsional rigidity of 214 Nm/degree – along with, amongst other traits, understeer. The 2004 USQ car appears to drive reasonably well, apart from the understeer and other minor construction matters, so this figure of 300 Nm/degree as a minimum appears to be founded in practice.

Fenton (1980,p7) gives a torsional stiffness for a normal family saloon as a minimum of 6500 Nm/degree and also gives the following formula for torsional stiffness of a chassis:

$$\mathbf{C = cd / D}$$

Where
- C = torsional stiffness in N/mm
- c = spring rate
- d = road wheel deflection
- D = torsional deflection of chassis

For a typical SAE racer, this equates to a torsional stiffness of 1000 N/mm, which for a track of width of 1200mm, becomes about 1090 Nm/degree, the upper end of Deakin et al's figure for an SAE racer.

Gaffney and Salinas, in their *Introduction to Formula SAE Suspension and Frame Design*, claimed a torsional rigidity of 2900 Nm/degree for the University of Missouri (Rolla) SAE racer, whereas the Laval University's 2004 SAE team claimed 2000Nm/degree for their car. These figures appear to be theoretical (and rather high) – their frames were not actually subjected to physical testing as was the 2004 USQ car.

Whilst there is a bit of conflict in the above figures – some seem rather higher than others – the fact remains that the 2004 USQ car, which was physically tested to 214 Nm/degree, had a reasonable level of handling. This is not to say that the 2004 USQ car had a chassis of sufficient rigidity – there is still an understeer problem and Deakin et al's figure of a minimum of 300 Nm/degree would appear to be a realistic minimum. It is intended to aim higher than this minimum. Longitudinal strength appears to be of secondary concern - if the chassis has adequate torsional rigidity, it will have quite sufficient longitudinal strength – and the factor that most affects handling is the efficient (or otherwise) transference of lateral loads.

Chapter 3

3. The SAE Chassis – A Particular Case

In the 'real world', every engineering project is subject to constraints. The SAE chassis is no different in this respect and, as such, is subject to the following constraints:

3.1. General Constraints

1. Low in cost
2. Easy to maintain
3. Reliable
4. Low production rate
5. Safe to repair

3.1.1. Low in Cost

The Formula SAE Competition sets a benchmark cost of $US25,000 as the maximum cost of the production of the vehicle. Any part that is manufactured from raw materials must be costed on the basis of the price per pound (or kilogram) in the Costing Table with the labour and machining activities likewise costed.

Engine / transmission is costed in accordance as to whether the engine is 'low performance' (2 valve industrial type engines – Briggs & Stratton), 'high performance' (2 valve motorcycle engines – early air cooled engines) or 'ultra-high performance'(3 or 4 valve motorcycle engines). In order to not disadvantage any team that does not have access to generous sponsorship from companies like Carpenter Steels or BHP, any part used, whether new, used or donated / pirated must be costed at its full new retail price (without any 'discounts') with any modifications to that part costed in accordance with set Costing Tables (see below).

COSTING TABLES

To assist in your process the following tables must be used in costing:

Common Materials and Cost Minimums Table

Mild steel, e.g. 1010, 1025	$0.30/pound
Alloy steel, e.g. 4130, Chrome Moly	$0.60/pound
Aluminum	$0.75/pound
Magnesium	$2.25/pound
Non-graphite composites	$88.18/kg ($40/pound)
Graphite-based composites	$220.50/kg ($100/pound)

Other materials such as plastics span such a vast range of uses and costs that a common price standard is impractical. Cost for composites and structural construction similar to fiberglass should be cost separately with a clear identification of the costs of all materials and processes. Obviously, process costs are in addition to the above material cost minimums.

OPERATIONS COST TABLE

Labor (all activity)	$35.00 / hr.
CNC Machine (time)	$70.00 / hr.
Welds	$0.14 / cm ($0.35/inch)
Saw or tubing cuts	$0.16 / cm ($0.40/inch)
Tube bends	$0.75 / bend
Non-metallic cutting	$0.08 / cm ($0.20 /inch)
Tube end preparation for welding	$0.75 / end
Drilled holes less than 1" diameter, any depth	$0.35 / hole
Drilled hole greater than 1" diameter	$0.35 / inch / hole
Reemed hole	$0.35 / hole
Tapping holes	$0.35 / hole
Sheet metal shearing	$0.20 / cut
Sheet metal punching	$0.20 / hole
Sheet metal bends	$0.05 / bend
Sheet metal stampings (process cost only)	$0.008 / sq. cm ($0.05 / sq. inch)
Sand castings (process cost only)	$6.61 / kg ($3.00 / pound)
Die castings (process cost only)	$8.82 / kg ($4.00 / pound)
Investment casting (process cost only)	$17.64 / kg ($8.00 / pound)
Plastic injection molding (process cost only)	$6.06 / kg ($2.75 / pound)

2005 Formula SAE® Rules

Table 2: Formula SAE Costing Tables

3.1.2. Easy to Maintain

This means precisely what it says – the frame type used must be easy to maintain. In this case, maintenance can mean cosmetically (keep it looking good – very important from a sales perspective, and pride of ownership) and protection from corrosion. In this case, the coating that looks good can also provide excellent corrosion resistance.

Ease of maintenance also means that, since this is a racing car, and racing cars occasionally fall over or run into hard objects, the frame must be easy to repair or modify. Since just about every weekend racer will have a welder of some sort (stick, MIG or oxy-acetylene) in his shed / workshop (or, even worse, a mate who's a welder), but won't have the knowledge or ability to perform specialist welding (TIG) or proper heat treatment, this should also be taken into account. In plain English, this means avoiding the 'chrome-moly' steels and favouring the plain carbon steels.

3.1.3. Reliability

Always keep the intended use and user in mind – a very important rule when designing virtually anything. In this case, it is for trouble free use by a non-technical weekend racer. What may be necessary for Formula 1 could prove to be quite unsuitable for this class of racing. With regards to the reliability of the frame, the following points are important:

- i: The frame will *not* be inspected after every race for cracks or any other fatigue related problems.
- ii. The frame will occasionally be *'adjusted'* with large hammers or the oxy torch.
- iii. The only defect that may be noticed will be a complete failure of a frame member.
- iv. The frame will be expected to last for the lifetime of the race car with little to no maintenance.

3.1.4. Low Production Rate

It is stipulated in the Formula SAE Rules that the production rate should be based on 4 vehicles per day for a limited production run – how ‘limited’ that run is, is not stated. However, it would be safe to assume that the run would not be for one week, and would, in all likelihood, extend for a period of at least a few months.

Though this is a limited production run, it must be remembered that this is a low cost weekend racer – not Formula 1. The entire Formula SAE racer would cost less than a set of Formula 1 brakes. This means that there is an economic constraint within the production rate constraint. It is not justifiable to set up to make a very limited production race vehicle using Formula 1 techniques. Simply put, hand built exotic carbon fibre and ‘unobtanium’ composite monocoque vehicles, whilst only low production rate, are probably out of contention. The fabrication methods will have to utilise readily available technology (read ‘low tech’) that requires only normally available skilled operators. Also, jigging and any other one-off tooling needed for the vehicle will have to be kept simple and to a minimum. The profit margins are small, so costs must be tightly controlled.

3.1.5. Safe to Repair

To a certain degree, this is covered by ease of maintenance. As mentioned earlier, the clientele are the low budget weekend racers who are after a vehicle that can be raced with a minimum of maintenance. They do not generally have access to specialised welding and heat treatment knowledge and facilities. However, this will not stop many of these people from attempting any frame repairs (or modifications) with whatever equipment they do have – generally a stick or MIG welder or with the oxy-acetylene. So, the best way around this is to use materials that can be safely welded at home with the above equipment.

3.2. Specific Constraints

These constraints which, whilst not applying directly to the chassis (apart from #6 – ‘specified crash protection’), must be considered in the design of the chassis.

1. Ground clearance – no touch ground
2. Wheels – minimum 203.2 mm (8”)
3. Suspension – fully operational with min 50mm travel
4. Steering – mechanical to at least 2 wheels
5. Brakes – must operate on all 4 wheels
6. Specified crash protection

3.2.1. Ground Clearance

The Formula SAE rules (3.2.1.) specify that no part of the vehicle shall touch the ground during the normal track events.
Since no minimum or maximum clearance is specified, it gives the designer freedom in this area to juggle roll centres, centre of gravity, suspension geometry, track width and wheelbase to achieve handling ‘utopia’.

3.2.2. Wheels

The Formula SAE rules (3.2.2.1.) specify a minimum wheel (rim) diameter of 203.2 mm (8” – USA imperial). There is no maximum wheel diameter stated. As above, this gives the designer freedom to play around with these variables.

3.2.3. Suspension

Rule 3.2.3. states that the vehicle must have an ‘operational’ suspension – as opposed to a set up that looks like a suspension, but is set up so firmly that it is basically ‘no suspension’, and handles like a go-kart. The rules also stipulate a minimum of 50.8 mm (2”) suspension travel – with the driver seated. Obviously, this constraint must be considered with the ground clearance and the wheel diameter.

3.2.4. Steering

Rule (3.2.4.) states that the steering must be to at least 2 wheels and must be connected mechanically – no electronic steering allowed. Also specified is the provision of mechanical stops to ensure the steering linkages do not lock and that the bits that go round and round do not hit the bits that do not. The chassis must have a suitable rigid part to which the steering rack may be attached.

3.2.5. Brakes

Rule (3.2.5.) covers the braking system and stipulates a dual circuit hydraulic system – electronically actuated brakes are not allowed. The braking system must be adequately protected.

3.2.6. Specified Crash Protection

The one area for which no compromise is allowed is safety. The rules stipulate certain minimum safety standards by specifying minimum steel sizes for the various parts of the frame. The main points will be outlined here, with a full copy of the rules in the Appendix.

The main points are shown below in Table 3: Specified Steel Sizes for Formula SAE Frame:

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) or 25.0 mm x 2.50 mm metric
Side Impact Protection, Front, Bulkhead, Roll Hoop Bracing & Safety Harness Attachment	25.4 mm (1.0 inch) x 1.65 mm (0.065 inch) or 25.0 mm x 1.75 mm metric

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

Table 3: Specified Steel Sizes for Formula SAE Frame

If alternative steel sizes are to be used, they must comply with the requirements of Rule (3.3.3.2.2) Steel Tubing Requirements shown in Table 4 below.

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

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.

Table 4: Alternative Steel Tubing

An important point to note here is that steel **must** be used for the main roll hoop and main roll hoop bracing – there is a total prohibition on the use of aluminium, titanium or composites for these components.

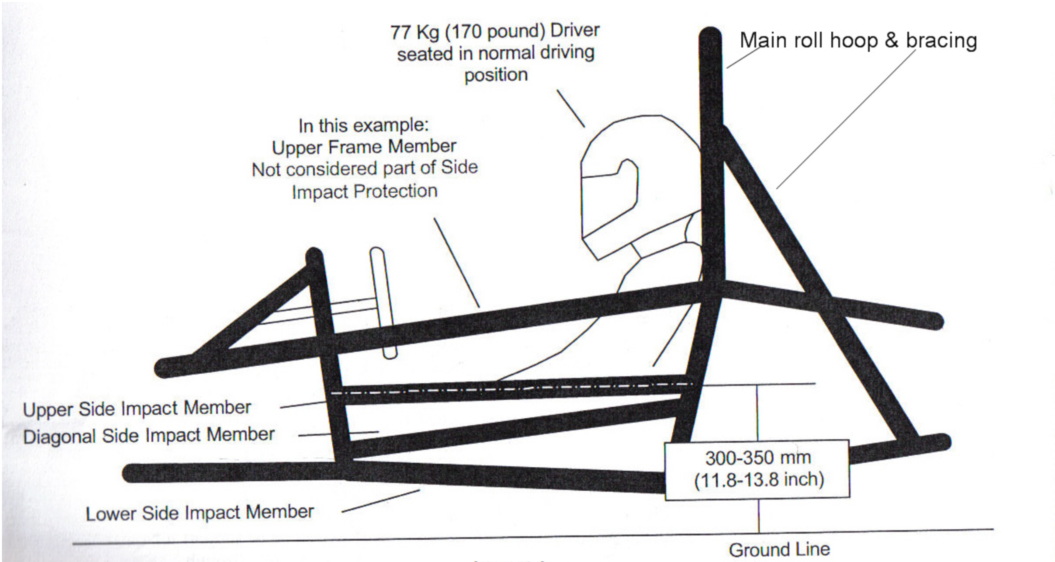


Figure 13: Side View of Formula SAE Frame

3.3. Selection of Chassis Type

This will be addressed by summarising the types of chassis available, along with their advantages and disadvantages.

1. Platform
2. Spaceframe
3. Monocoque

3.3.1. Platform

Good:

1. Easy to design
2. Inexpensive components
3. Easy to manufacture
4. Can be made with considerable longitudinal rigidity

Bad:

1. Heavy if rigid (beam construction)
2. Body along for ride

3.3.2. Spaceframe

Good:

1. Lightweight
2. High strength / rigidity
3. Design simplicity

Bad:

1. Labour intensive
2. Specialised welding / heat treatment may be necessary.
3. Suitable for short production runs only.

3.3.3. Monocoque

Good:

1. Lightweight
2. Very strong / rigid

Bad:

1. Generally expensive – tooling etc
2. Specialist skills / equipment
3. Suitable for mass production
4. Suitable for limited expensive runs (Formula 1)

3.3.4. Selection of Chassis Type

The selection of chassis type required consideration of the above general points, along with the specific requirements of the SAE Rules. These were:

1. Safety rules requiring steel hoops, braces etc.
2. Max production rate of 4 per day
3. Construction of 1 prototype

There were also the following pragmatic considerations:

1. Economics ('cheap').
2. Able to be manufactured in small workshop.
3. Able to be modified after construction.

And the winner is…….

The Space Frame

Why?

1. The safety regulations require considerable amount of steel tubing
2. Simplicity of design and manufacture
3. Light weight
4. Potential strength and torsional rigidity
5. Suitable for small production runs
6. Very suitable for the construction of a ‘one off’ prototype
7. Prototype can be easily modified as required
8. Prototype can be manufactured and modified very cheaply
9. Can be built in any small workshop.

Having decided on the type of chassis to be used, the next step was to decide on the details. This is covered in Chapter 4.

Chapter 4

4. Materials for the SAE Chassis – Options and Selection

This Chapter considers the various materials that may be suitable for the construction of the frame – not only the physical properties, but also fabrication and economic considerations. The chapter is set out in the following manner:

1. Normal Operating Conditions
2. Required Properties of Fabrication Materials
3. Availability of Materials
4. Economic Considerations
5. Suitable Materials List
6. Fabrication Methods
7. Heat Treatment Requirements
8. Surface Treatments / Coatings
9. Selection of Materials for Chassis

4.1. Normal Operating Conditions

This section looks at the following:

1. What the SAE Frame Does
2. Operating Environment
3. Loads – Dynamic and Static.

4.1.1. What the SAE Frame Does

The frame is a structure that holds all the components (and the occupant) of the vehicle in the correct place. This includes the engine, drive train, suspension, fuel tank, steering etc under fairly arduous conditions.

One of these in particular – the suspension – requires that the frame has a high degree of torsional rigidity. This is to allow only the suspension to do the suspension's job – and not have the frame acting as a 'de facto' suspension.

On top of this, the frame has the task of protecting the occupant under any normally foreseeable event (rollover, collision etc).

4.1.2. Operating Environment

The operating environment of the frame is not particularly hostile, but it is fairly demanding – both for performance and longevity.

The frame will be exposed to the elements – rain, ambient temperatures in the range 0°C to 45°C, wind, all levels of humidity and sunshine (including heat and ultraviolet radiation).

The frame will also be exposed to its own mechanical environment – oils, solvents, petrol, ethanol and other petrol additives as well as heat caused by the engine (particularly the exhaust) and the braking system. Considering that the exhaust headers and the brake discs can actually glow red hot during severe operating conditions (for steel, this is in the range of roughly 560°C to 840°C) means that some thought must be given to firstly the placement of these 2 items and secondly, to their supporting structure.

In the first case, careful placement of header pipes and adequate clearance allowed for in the frame design will mean that little heat energy is transferred to the frame. Adequate airflow to the brake discs will help alleviate any problems in this area.

4.1.3. Loads

The loads on the chassis can be divided into the following:

1. Static Loads
2. Dynamic Loads

These loads will be dealt with in detail in Chapter 5.

4.1.3.1. Static Loads

The static loads are those that are due to the self-weight of the various components of the vehicle. These include:

1. Engine
2. Driver
3. Suspension
4. Frame
5. Ancillary components

It is important that the load paths from the various components of the vehicle are correctly determined and the frame designed accordingly. These static loads, when the vehicle is in motion, may be subjected to accelerations in the order of 4.5g under ‘normal’ operating conditions.

4.1.3.2. Dynamic Loads

These are the loads imposed on the frame during the normal course of vehicle operation eg cornering, braking, accelerating etc.
They include:

1. Accelerating
2. Braking
3. Cornering
4. Bumps / dips
5. Engine torque reactions
6. Drive train

Because of the potential magnitude of dynamic loads, it is important that these be considered carefully in the design process.
A rough estimation is forces of the order of 4.5g (bump) and 2.0g (normal cornering/accelerating) with a mass of 300kg – 13,000 N and 6,000 N respectively distributed in various directions through the frame and suspension to the ground. This rough estimate of magnitude would be sufficient for materials selection – though any surprises in force magnitude further into the design process may call for revisions in materials choice.
The over-riding consideration for the materials selection process is the fatigue loading to which the frame will be subjected.

4.2. Required Properties of Fabrication Materials

The operating environment subjects the frame to fluctuating loads – up to 13,000 N distributed unevenly through out the chassis, with a normal external operating environment (and localized temperatures in the vicinity of 250°C to 300°C).

The major properties of the materials are set out below in *Table 5: Properties Required for SAE Frame*.

This covers the mechanical, physical, chemical and dimensional properties of the materials. Many of these values are not quantified, because there is a reasonable degree of flexibility in the requirements.
Aluminium does not have the tensile strength of steel (about one third) but has a density of roughly one third that of steel – so for the same weight, aluminium would be on par with steel for strength. Fatigue, of course, is another matter.
Stiffness is another flexible requirement (pardon the pun) – extra struts / webs may compensate for a lack of stiffness in a material, if other factors are more favourable.
Excessive creep may only mean a shortened life for an otherwise excellent material – again, a compromise.
A higher density may be offset by much higher tensile and fatigue strengths e.g. steel vs aluminium.
Dimensional stability is, of course, of much importance with a frame – the suspension settings must not change with different ambient and other conditions.

Properties	
Mechanical	
Hardness	High degree of hardness
Fatigue	Very high fatigue resistance
Tensile	High tensile strength
Impact	High impact strength
Creep	Low creep characteristics
Wear	Very good wearing ability
Stiffness	High stiffness
Compression	High compressive strength
Shear	Very high shear strength
Physical	
Density	Low to medium density
Electrical	na
Magnetic	na
Conduction	High thermal conductivity - dissipation
Expansion	Low thermal expansion
Flammability	Very low
Melting Point	High - above 600°C
Chemical	
Environmental resistance	Resistant to solvents, oils, weather.
Composition	na
Bonding	na
Structure	na
Dimensional	
Flatness	Must maintain machined surfaces
Surface finish	Able to be easily machined
Stability	Must be stable at operating temperatures
Tolerances	To 0.5mm

Table 5: Properties Required for SAE Frame

4.3. Availability of Materials

The availability of potential materials is quite important – both for the construction of the prototype and the on-going production of 4 per day. These considerations are listed in *Table 6: Availability of Materials for SAE Frame* below.

Availability	
On Hand	Yes
Order from Warehouse	Yes
Minimum Order Requirements	No
Limited suppliers (proprietary)	No
Special Processing Required	
Casting	No
Forging	No
Extrusion	Yes
Moulded	No
Tooling Required	No

Table 6: Availability of Materials for SAE Frame

Because an over-riding consideration for the whole SAE Project is economy, ready availability of materials is quite important – both for on-going modifications during the development of the frame and for any repair work which may be necessary during the testing period. The use of proprietary materials (eg Vasco300 etc) should normally be avoided due to price and availability concerns in this country.

Keeping this in mind, for the purposes of this Project, materials available locally ‘off the shelf’ would be most desirable.

4.4. Economic Considerations

As mentioned previously, the design philosophy behind the Formula SAE Project is, to put it bluntly, "cheap". However, this is cheap in $$cost, not in performance or quality. The economic considerations are shown below in *Table 7: Economic Considerations for SAE Frame*.

Economics

Raw Material	Cheap
Quantity Required	
Millions	No
Thousands	No
Number / year	No
Few	Small regular quantities
Fabricability	
Formability	Good bending
Weldability	Very good weldability
Machinability	Not critical

Table 7: Economic Considerations for SAE Frame

From the above, it can be seen that commercially small regular quantities of a cheap material is required. This material should be easy to bend and to weld.

4.5. Suitable Materials List

With reference to the above physical parameters, the following materials would be feasible:

1. Aluminium
2. Low C steel
3. Alloy steels ('chrome-moly')

Other exotic materials such as titanium alloys, though eminently suitable for the chassis of a high-performance racing car, are ruled out on cost, availability and processing/fabricating difficulties.

The physical properties of a selection of metals, including aluminium and the above steels are shown below in *Table 8: Metals Properties*

Material	Density kg/m^3	Ultimate Strength: Tension, MPa	Ultimate Strength: Compression,[2] MPa	Ultimate Strength: Shear, MPa	Yield Strength[3]: Tension, MPa	Yield Strength[3]: Shear, MPa	Modulus of Elasticity, GPa	Modulus of Rigidity, GPa	Coefficient of Thermal Expansion, 10^{-6}/°C	Ductility, Percent Elongation in 50 mm
Steel										
Structural (ASTM-A36)	7860	400			250	145	200	77.2	11.7	21
High-strength-low-alloy										
ASTM-A709 Grade 345	7860	450			345		200	77.2	11.7	21
ASTM-A913 Grade 450	7860	550			450		200	77.2	11.7	17
ASTM-A992 Grade 345	7860	450			345		200	77.2	11.7	21
Quenched & tempered										
ASTM-A709 Grade 690	7860	760			690		200	77.2	11.7	18
Stainless, AISI 302										
Cold-rolled	7920	860			520		190	75	17.3	12
Annealed	7920	655			260	150	190	75	17.3	50
Reinforcing Steel										
Medium strength	7860	480			275		200	77	11.7	
High strength	7860	620			415		200	77	11.7	
Cast Iron										
Gray Cast Iron										
4.5% C, ASTM A-48	7200	170	655	240			69	28	12.1	0.5
Malleable Cast Iron										
2% C, 1% Si,										
ASTM A-47	7300	345	620	330	230		165	65	12.1	10
Aluminum										
Alloy 1100-H14										
(99% Al)	2710	110		70	95	55	70	26	23.6	9
Alloy 2014-T6	2800	455		275	400	230	75	27	23.0	13
Alloy-2024-T4	2800	470		280	325		73		23.2	19
Alloy-5456-H116	2630	315		185	230	130	72		23.9	16
Alloy 6061-T6	2710	260		165	240	140	70	26	23.6	17
Alloy 7075-T6	2800	570		330	500		72	28	23.6	11
Copper										
Oxygen-free copper										
(99.9% Cu)										
Annealed	8910	220		150	70		120	44	16.9	45
Hard-drawn	8910	390		200	265		120	44	16.9	4
Yellow-Brass										
(65% Cu, 35% Zn)										
Cold-rolled	8470	510		300	410	250	105	39	20.9	8
Annealed	8470	320		220	100	60	105	39	20.9	65
Red Brass										
(85% Cu, 15% Zn)										
Cold-rolled	8740	585		320	435		120	44	18.7	3
Annealed	8740	270		210	70		120	44	18.7	48
Tin bronze	8800	310			145		95		18.0	30
(88 Cu, 8Sn, 4Zn)										
Manganese bronze	8360	655			330		105		21.6	20
(63 Cu, 25 Zn, 6 Al, 3 Mn, 3 Fe)										
Aluminum bronze	8330	620	900		275		110	42	16.2	6
(81 Cu, 4 Ni, 4 Fe, 11 Al)										

(Table continued on page 740)

Table 8: Metals Properties (Beer, Johnston & DeWolf)

4.5.1. Aluminium

Aluminium's properties, as shown above, seem, superficially at least, to be ideal for the chassis of the SAE vehicle. It is light in weight, with yield strengths available in the range of 230 MPa to 500 MPa (with commercially 'pure' aluminium possessing a yield strength of a fairly useless 95 MPa).

The strongest normally available alloy – 7075 – requires the T6 heat treatment specification for its strength – and this is a big drawback for all aluminium alloys. After any welding, the aluminium must be heat treated / aged correctly to regain its rated strength – otherwise the strength (and the structure) is severely compromised.

The other drawback with aluminium is fatigue. *Figure 14: Stress – Loading Cycles Curves* shows aluminium (2024) and steel (1020HR). The steel has an *endurance limit* – a stress level for which an infinite number of load reversals may be endured. The aluminium (this particular alloy can be used for frames), on the other hand, does not possess an endurance limit, and will eventually fail – the number of load cycles being totally dependent on the magnitude of the load (barring, of course, stress risers and other metallurgical imperfections).

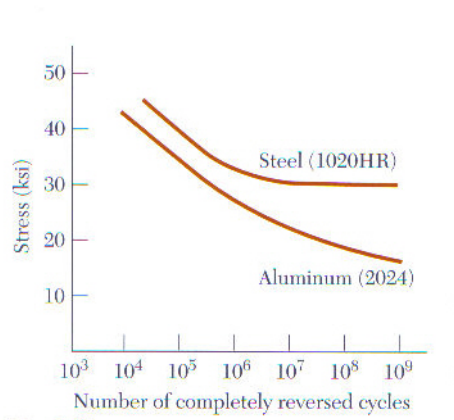


Figure 14: Stress – Loading Cycles Curves **(Beer,Johnston&DeWolf)**

The aluminium alloy (7075-T6) mentioned above is used in the fabrication of high performance connecting rods (con-rods) used extensively in drag racing. However, for this application, the rods are highly polished to eliminate any potential stress risers, and still have a limited life span – suitable for racing where an engine is rebuilt regularly, and the con rod is regarded as a 'consumable'. Such a rod is shown below in *Figure 15: Manley Aluminium Con Rod*:

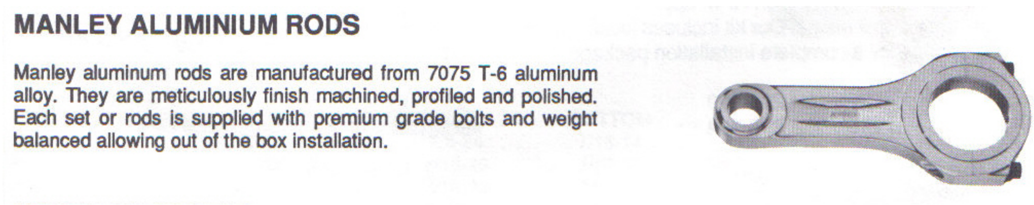


Figure 15: Manley Aluminium Con Rod (Lunati)

A frame constructed from aluminium (6061-T6 with a yield strength of 240 MPa would be suitable) would not be able to be polished to the same degree as the con rod – with the welds providing a particularly problematic area.

From the above, it can be seen that aluminium would be able to provide a light strong frame, but has the real disadvantage of requiring proper heat treatment and has a limited fatigue life.

4.5.2. Low Carbon Steel

Low carbon steesl show all the properties required for the chassis – with the exception of density. Yield strengths in the range of 250 MPa to 350 MPa are readily commercially available.

Somers (1993) considers weldability as being divided into two general classes:

1. Fabrication Weldability
2. Service Weldability

He goes on to state that *fabrication weldability* addresses the question:

> "can one join these materials by welding without introducing detrimental discontinuities?"

This is the area covering hydrogen-assisted cold cracks, hot cracks, reheat cracks, lamellar tearing and porosity.

He also states that *service weldability* concerns the question

> "Will the finished weldment have properties adequate to serve the intended function?"

This area deals with the effect of the welding thermal cycle on the heat- affected zone (HAZ), and as such, is dependent on both heat input and material thickness.

RB Smith (1993,p645), with regard to the above, considers that an ambient pre-heat and inter-pass temperature for low carbon steel (AISI-SAE 1017,1018,1019,1020,1021,1022,1023) for thicknesses less than 50mm with no requirement for post-weld heat treatment to be satisfactory. For the thicknesses to be used, this means no special heat treatment is necessary – a time and cost benefit.

As well, from *Figure 14: Stress – Loading Cycles Curves* above, it can be seen that steel has an endurance limit – the stress level at which the steel may be able to sustain an infinite number of load reversals.

In consideration, it can be seen that the low carbon steels are cheap, readily available, easily welded with no requirement for heat treatments in the tube thicknesses being considered. The only drawback with these steels is the density (weight).

4.5.3. Alloy Steels

In this section, the heat treatable low alloy (HTLA) steels are considered, in particular AISI-SAE 4130 steel (known as ‘chrome-moly’). HTLA steels show all the properties required for the chassis – with the exception of density. Yield strengths of 650 MPa are readily commercially available, though considerably more expensive than the low carbon steels.

Somers (1993) considers that these steels, though possessing good weldability, require proper heat treatment. This view is supported by C Smith (1984) – who considers that 4130 steel, <u>*properly heat-treated*</u>, is *virtually unbeatable* for applications such as a racing chassis. An important consideration here is that if any welding is done subsequently to the frame, proper heat treatment must follow, or the integrity of the frame will be compromised.

Again, from *Figure 14: Stress – Loading Cycles Curves* above, it can be seen that all steels have an endurance limit – the stress level at which the steel may be able to sustain an infinite number of load reversals. With an alloy steel such as 4130, this load would be considerably higher than for the low carbon steels.

In consideration, it can be seen that the HTLA steels are readily available, though expensive, are reasonably easily welded but have a very real necessity for the proper heat treatment processes. As well, there is the disadvantage of high density. In summary, an excellent but expensive choice.

4.6. Fabrication Methods

Each of the materials, aluminium, low C steel and alloy steel, is available in tubular sections in *electric resistance welded* (ERW) – steel only, *cold drawn seamless* (CDS) and *drawn over mandrel* (DOM), each being of higher quality and even higher price.
Each of the materials can be cut mechanically (cold saw, band saw, hole saw etc) and each can be welded using the appropriate method for that material – all methods being readily available to any workshop.

For this prototype space frame, there would only be one feasible fabrication method – hand built welded. The basic frame components – hoops, floor frame etc – would be marked out accurately on the floor and then members carefully cut and assembled using this ‘pattern’. The main hoops would be separately fully welded, then the other frame members would be ‘tacked’ into position until the full frame is cut and assembled. Once this is finished, final checks for dimensions would be done, then the frame fully welded. This has the advantages of allowing minor changes to be made during frame construction (if unforeseen problems arise) and requiring no tooling / jig costs.

Each of the materials is readily fusion welded, especially by the arc welding (AW) processes. Tungsten Inert Gas (TIG) welding would be the most appropriate. This will be discussed later in more detail.

The production of 4 frames per day would require the use of pre-cut components with jigs, and the frame would also be ‘hand welded’. Automated welding and assembly techniques would have no place in this process, as the low output would not offset the costs associated with current automation technology. Maybe sometime soon in the future…….try Dana Corporation and Audi.

4.7. Heat Treatment Requirements

Heat treatment will be discussed for each of the materials in the following order:

1. Low Carbon Steel
2. Alloy Steels
3. Aluminium

4.7.1. Low Carbon Steel

The low carbon steels generally do not respond to, or need, heat treatment (C Smith, 1984) – so no need – and this simplifies the fabrication process with these materials.

4.7.2. Alloy Steels

As mentioned in ***4.5.3. Alloy Steels*** above, the heat treatable low alloy (HTLA) steels only are considered, in particular AISI-SAE 4130 steel (popularly known as 'chrome-moly').

The major problem with these steels is in the heat affected zone (HAZ) with cracking in the coarse-grained region – to avoid this, the appropriate preheat and interpass temperature should be used. The post weld heat treatment (PWHT) of a chromium-molybdenum weldment is also referred to as a stress relief heat treatment. This is designed to reduce the residual stresses and to improve the fracture toughness of the HAZ and the weld metal. (Chen & Pollack, 1993)

With these steels, proper welding techniques and heat treatment processes are imperative, and must be performed.

4.7.3. Aluminium

As mentioned above, the aluminium alloy 6061-T6 would be suitable for the construction of a space frame. The ‘T6’ refers to the heat treatment / aging process required for this particular alloy. The T6 treatment is properly called “solution heat-treated and artificially aged” which means the alloy is heated to around 500°C for around 30 mins then quenched in water (at 80°C) – the alloy is then “artificially aged” where the alloy is heated to 200°C for 7 hours (which would bring the hardness to around 105 to 130 Brinell). (Oberg, Jones & Horton, 1980 p2242).

With the various aluminium alloys, it is imperative that proper welding techniques and solution heat treatment / artificially aging processes are used.

4.8. Surface Treatments / Coatings

Surface treatments / coatings will be discussed for each of the materials in the following order:

1. Low Carbon Steel
2. Alloy Steels
3. Aluminium

Metal products are almost always coated by one of the following:

1. Plating & related processes – hot dipping etc
2. Conversion coatings
3. Physical Vapour Deposition
4. Chemical Vapour Deposition
5. Organic Coatings
6. Ceramic coatings
7. Thermal / mechanical coating

Of these processes, for the SAE racer, only #1, #2, & #5 are relevant. Vapour depositions tend to be used for precision work (aluminium coatings on telescope mirrors, integrated circuits etc) and ceramic coatings would not be able to flex with the frame (and would crack) and thermal/mechanical coatings are expensive and tend to be used in aggressive environments and for wear/erosion protection. (Groover 2002)

Groover (2002) gives the main reasons for coating a metal as:

1. Corrosion protection
2. Enhance the appearance (marketing reasons)
3. Increase wear resistance
4. Decrease friction
5. Increase or decrease electrical conductivity
6. Preparation of surface for further processing
7. Rebuild worn or corroded surfaces

For the frame of the SAE vehicle, #1 and #2 from the above list are the only relevant ones and will be discussed further.

4.8.1. Low Carbon Steel

For low C steel, the major environmental problem is corrosion.
The marketing value in applying a cosmetic coating is also very important.

Of the methods outlined above, the only suitable ***plating*** method would be hot dipping (galvanising). However, for cosmetic reasons for a racing car, this is a poor choice. Zinc is also dangerous to weld without proper breathing protection and, if later modifications are made to the frame, the integrity of the coating is compromised (the various commercially available ‘cold gal’ paints cannot match the protection and appearance given by the hot dip zinc.)

The conversion coatings, such as phosphate and chromate, are more suitable as primers for subsequent painting, and the other conversion coating, anodising, is normally used for aluminium and magnesium.

The organic coatings include polymers and resins, either natural or synthetic, which can be applied as liquids or powders and then

subsequently dried or cured. These are more commonly known as 'paint' – acrylic lacquers, '2-pack' (epoxy and polyurethane paints) automotive enamels etc. These coatings are available with a wide range of properties and an even wider range of colours.
Powder coating uses a dry powder that is electrostatically fixed to the frame, then melted to allow subsequent re-solidification on the surface as a coating. These too are available in a wide range of colours.

The obvious choice for the frame would be a phosphate based primer with a suitable acrylic lacquer finish – both cosmetically acceptable and providing good corrosion resistance. Powder coating, though cosmetically superior, has the disadvantage of not being easily repaired in the *weekend warrior's* workshop.

4.8.2. Alloy Steels

The discussion above in *4.8.1. Low Carbon Steel* is totally relevant to the alloy steels in question here. Corrosion protection and cosmetics are both equally important, as is the ability to 'touch up' the coating after any repair work, both in the factory and in the workshop. A good primer with an acrylic lacquer would be very acceptable – and pick a colour that will easily show up any cracks that may develop in the frame (i.e. don't pick black!).

4.8.3. Aluminium

Though aluminium is a very reactive metal, the aluminium oxide that forms on the surface is also a very effective coating to protect against any further corrosion.

Conversion coating – anodising – is suitable for aluminium, but for this particular usage, would probably not be appropriate. Cosmetically, an anodised frame (*a nice Barbie Pink perhaps?*) would be out of place on this type of vehicle. Anodising a frame would be expensive, and difficult to 'touch up' or repair satisfactorily.

The various organic coatings (paints) tend not to be very successful, or popular, on aluminium racing car frames. Generally, such frames are left to run in the 'as bought' condition.

The only real coating choice for an aluminium SAE frame would be *no coating.* This, too, has the added advantage of allowing the frame to be effectively inspected for fatigue cracking – a very real problem for aluminium frames. And, if the owner has sufficient patience and skills or can afford to pay one of the commercial aluminium polishers, mill-finish aluminium can be polished to a very impressive lustre.

4.9. Selection of Materials for Chassis

The original materials choice list gave the following:

1. Aluminium
2. Low Carbon Steel
3. Alloy steel

4.9.1. Aluminium

The aluminium was light, but not as strong as the steels, and has problems with fatigue, along with fussy heat treatments / aging and difficulty in welding. Subsequent ‘Owner modifications’ would most certainly not have any necessary heat treat / aging done and, as a consequence, would be severely structurally compromised.
Aluminium tends to be more expensive than low carbon steel, but less expensive than the alloy steel in question here.
Aluminium does not need any protective coatings, but looks a little ‘spartan’ without a cosmetic coat.

4.9.2. Low Carbon Steel

The low carbon steel was heavy but strong, with a potential ‘infinite’ fatigue life, easy to weld and needing no special heat treatments. It is very agreeable to ‘Owner modifications’.
Low carbon steel is inexpensive and readily available ‘off the shelf’ in a large range of sizes – though some of the less popular sizes are only available on ‘special order’.
Though prone to rust, low carbon steel is easily painted, and with the correct coating choice, is easily and successfully repaired.

4.9.3. Alloy Steel

The alloy steel (AISI-4130) was heavy, very strong with again, a potential ‘infinite’ fatigue life. However, it is more difficult to weld and must be properly heat-treated. Subsequent ‘Owner modifications’ would most certainly not have any necessary heat treatment done and, as a consequence, would also be severely structurally compromised. The alloy steel (4130) is expensive and not readily available ‘off the shelf’ - it must be generally specially ordered on a *job lot* basis from specialist suppliers in the capital cities. (Though the country of manufacture should be chosen carefully).

Though prone to rust, alloy steel is easily painted, and with the correct coating choice, is easily and successfully repaired.

4.9.4. Final Materials Choice

From the above, it was decided to use steel. The choice was to use one of the following:

SAE 1020 DOM 350 (low C steel) or
SAE 4130 CDS 650 (‘chrome – moly’)

Of these two, the final choice was:

1020 DOM 350

Because:

It is considerably cheaper
It iseasy to weld and form
Repairs/modifications can safely be done by Owner
Can be easily painted in wide range of colours

Chapter 5

5. The SAE Chassis – Design & Construction Methodology

This chapter looks at the design of the chassis, including the design criteria used. It also looks at the selection of the work processes to be used in the construction of the frame, along with relevant quality control methodology – both for the prototype frame and for the subsequent SAE production rate of 4 frames per day.

This chapter is set out in the following order:

1. Design
2. Work Processes
3. Quality Control Methodology

5.1. Design

The design process was approached in the following order:

1. Design Criteria
2. Design Process

5.1.1. Design Criteria

The design criteria were approached in two areas:

1. Dimensions
2. Applied Loads

5.1.1.1. Dimensions

It was decided at the beginning of the design process that there would be minimal changes between the 2004 car and the 2005 car. This was so decided as the 2004 car was reasonably successful (except for a few minor design faults) and the philosophy for 2005 was to be ***evolution*** of the car and ***devolution*** of responsibilities.

The aim was simple – less weight and more power. Each of these has the simple effect of improving the power to weight ratio – and hence accelerative performance. Less weight also benefits handling and braking – less weight means less inertia which, in turn, means better cornering and braking.

With this in mind, it was decided to keep the dimensions fairly similar – with a slight reduction in wheelbase, the rationale being that a shorter wheelbase gives better cornering at the expense of high-speed stability. The course is not designed for high speed, so this is no loss at all. Even the 'drag strip' component of the competition is only 75 m long – allowing terminal speeds in the vicinity of 100 kph for the faster cars. The same car, over a proper quarter mile (400 m) drag strip, would reach a terminal speed of around 165 kph.

From the above, it was deduced that the following dimensions would apply:

Wheelbase	**1650 mm**
Front Track	**1285 mm**
Rear Track	**1165 mm**

This is shown below in *Figure 16: Dimensions of 2005 SAE Car*

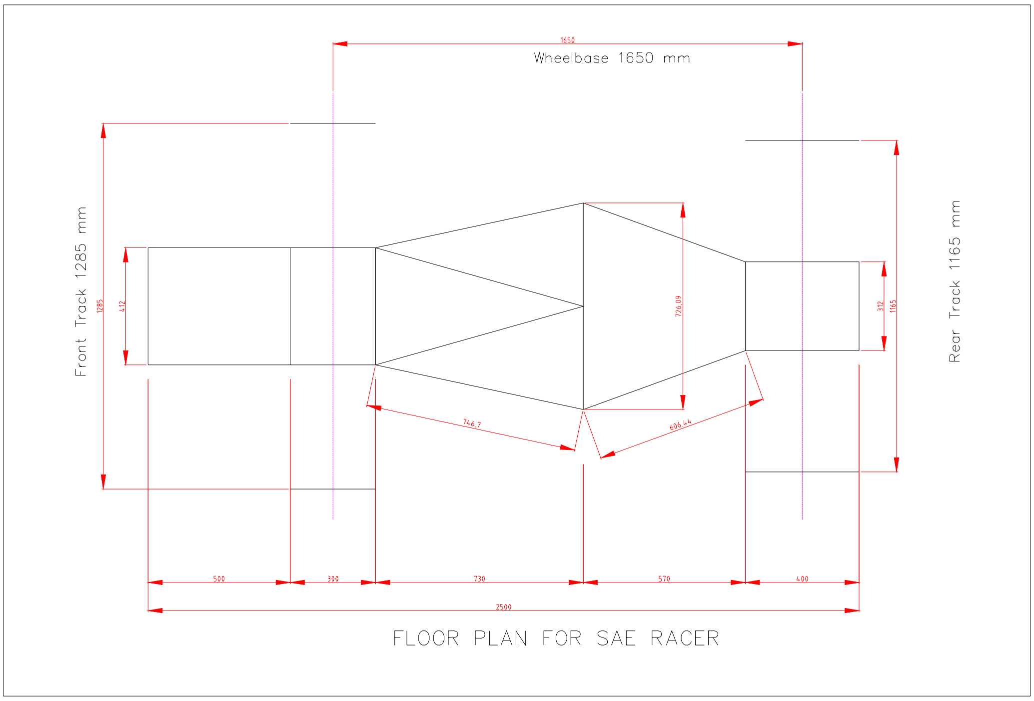


Figure 16: Dimensions of 2005 SAE Car

It was also decided to give the car a bit more space in the width of the driver's compartment – for where the larger driver's knees normally reside. This is shown below in *Figure 17: Mid-Rail Dimensions of 2005 SAE Car:*

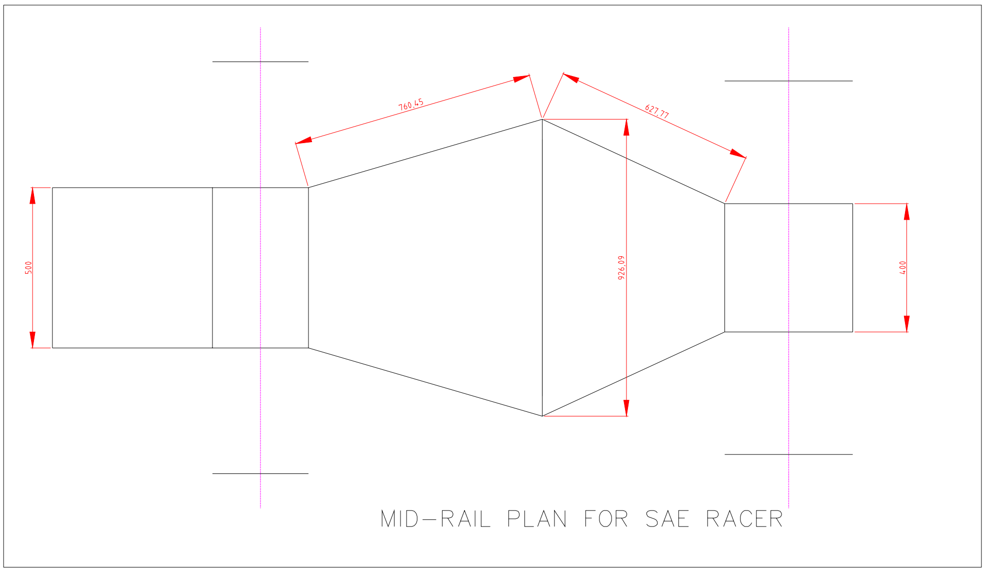


Figure 17: Mid-Rail Dimensions of 2005 SAE Car

This extra width is shown below in *Figure 18: Main Hoop*

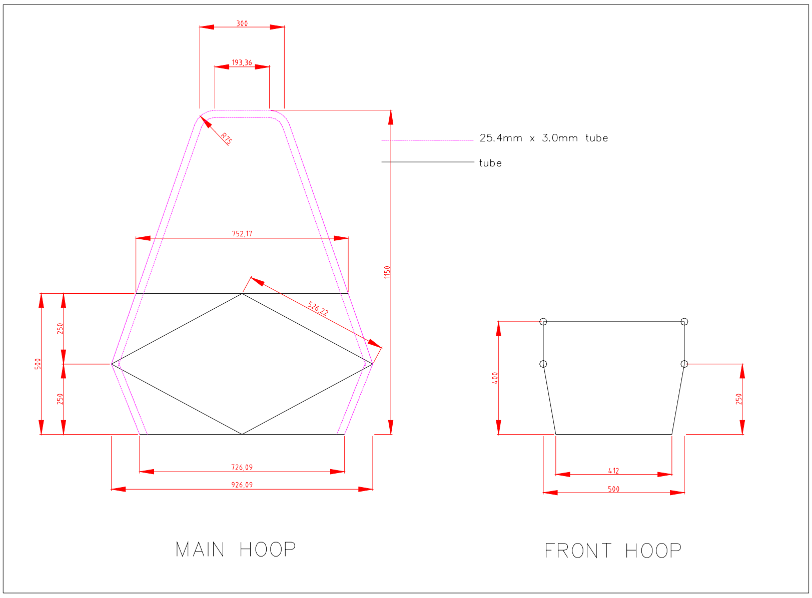


Figure 18: Main Hoop

Once the basic dimensions had been decided, it was then necessary to set a weight target, in accordance with the evolutionary philosophy espoused earlier. The 2004 chassis had weighed in at around 50 kg – a fairly excessive figure. This being the case, a target of a 20% weight reduction, whilst not only *not* compromising strength but actually enhancing it, was set. So, the aim was – a considerably stronger frame with much less weight.

The weight target: **40 kg (max).**

Simply put, this meant that, since 20% less steel was going to be used to achieve a greater strength, a more intelligent usage of the steel was required.

5.1.1.2. Applied Loads

The chassis of the SAE Racer has to withstand certain loads and transmit others – all the time maintaining its structural integrity. Sounds simple and, if done properly, is simple. A good chassis should have a simplicity and elegance that should appeal to the eye.

The types of loads applied to the SAE chassis (and, for that matter, any chassis) are as follows:

1. Static Loads
2. Dynamic Loads

1. Static Loads

These are the loads carried by the chassis with the vehicle sitting on the tarmac, fuelled up and ready to go – with a driver, fully outfitted, strapped into position. The static loads distributed by the chassis are shown below in *Figure 19: Static Loads on 2005 SAE Chassis*

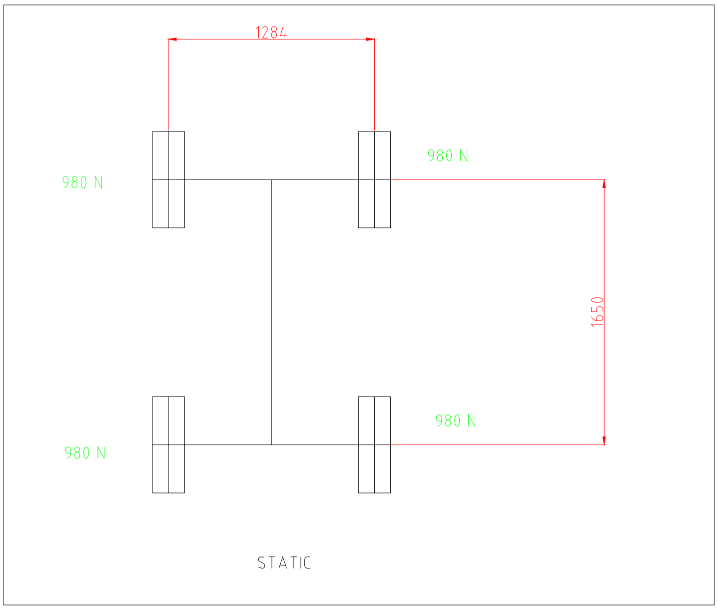


Figure 19: Static Loads on 2005 SAE Chassis

This calculation is based on the rather conservative (read 'heavy') total weight estimate of 400 kg (fuelled with driver).
This would be made up of the following:

1.	Engine / gearbox	75 kg
2.	Driver	120 kg
3.	Chassis	40 kg
4.	Peripherals	135 kg
5.	Bodywork	30 kg
	Total	**400 kg**

These static weights, though important, are just that – static. It would be most unlikely, even for a Ford, for the engine to just fall onto the ground (though it did happen to the front end on some early 1960s Falcons due to premature ball joint failure).
What *is* important is the dynamic loads that these static weights put on the chassis when the car is doing what it was intended to do – race.

2. Dynamic Loads

As mentioned above, these are the loads that are generated when the vehicle is moving - this incudes accelerating, braking, cornering and hitting the odd bump, gutter or pothole.

Fenton (1980, p14) gives the following figures (which include a factor of safety of 1.5):

+ / - 4.5 g **vertical (hitting a bump)**
+ / - 1.5 g **fore and aft (braking & accelerating)**
+ / - 1.5 g **cornering LH or RH**

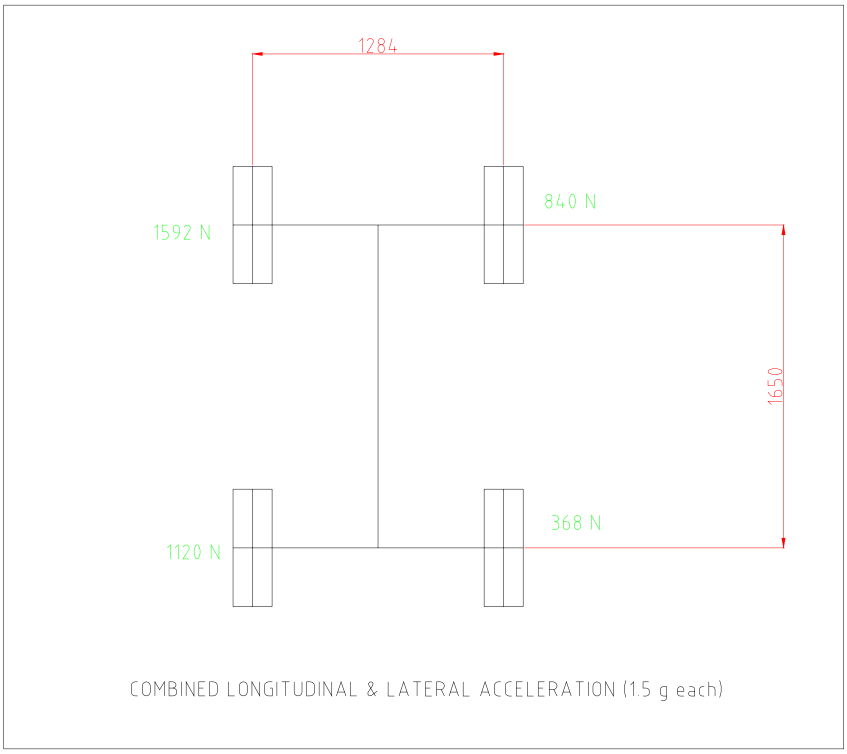


Figure 20: Dynamic Load Distribution

These figures need to be qualified somewhat for an SAE vehicle. The figures from the 2004 USQ entry for the standing 75 m are in the order of 5.7 seconds. From *Table 9: Drag Strip Performance* below, some interesting information emerges. The time of 5.47 s shown equates to the time of 5.7 s when 0.2 s is added for delay in the initial start.

Weight	360	kg							
Horsepower	37	hp		speed	speed				
Distance (m)	n	k	time	mph	m/s	dt	dv	a = dv/dt	a (g)
0	0	0	0	0	0	0	0	0	0
5	80.467	0.034	0.90	18.97	8.48	0.90	8.48	9.43	0.96
10	40.234	0.136	1.43	23.91	10.69	0.53	2.20	4.17	0.43
15	26.822	0.306	1.87	27.36	12.23	0.44	1.55	3.49	0.36
20	20.117	0.544	2.27	30.12	13.46	0.40	1.23	3.11	0.32
25	16.093	0.849	2.63	32.44	14.50	0.36	1.04	2.86	0.29
30	13.411	1.223	2.97	34.48	15.41	0.34	0.91	2.67	0.27
35	11.495	1.665	3.29	36.30	16.23	0.32	0.81	2.53	0.26
40	10.058	2.175	3.60	37.95	16.96	0.31	0.74	2.41	0.25
45	8.941	2.752	3.89	39.47	17.64	0.29	0.68	2.31	0.24
50	8.047	3.398	4.17	40.88	18.27	0.28	0.63	2.23	0.23
55	7.315	4.111	4.45	42.20	18.86	0.27	0.59	2.15	0.22
60	6.706	4.893	4.71	43.44	19.42	0.27	0.56	2.09	0.21
65	6.190	5.742	4.97	44.61	19.94	0.26	0.53	2.03	0.21
70	5.748	6.659	5.22	45.73	20.44	0.25	0.50	1.98	0.20
75	5.364	7.645	5.47	46.79	20.92	0.25	0.48	1.93	0.20

Table 9: Drag Strip Performance

Some interesting facts emerge from this Table – the first is that to shift 360 kg down the strip in that time requires 37 hp. Various dynamometers can be calibrated in imaginative ways to give widely varying results – but the drag strip gives a rather unbiased (and, at times, unflattering) estimate of power based on weight shifted (or work done) and allows a real comparison of different engine outputs. All that is needed is the elapsed time and the total vehicle weight. An important point to note here is that these calculations are based on correct gearing for the vehicle and the distance travelled. With the USQ 2004 SAE vehicle, to do the time of 5.7 seconds required 37 horsepower – the engine may have produced more, but the gearing and driving style did not use any more than 37 hp. (A good example would be, even if the gearing was correct, when the engine made maximum power at 8500 rpm but the driver took the engine to 7500 rpm).

The other interesting information to come from the table is the actual acceleration of the car. Some writers assume particularly high and constant acceleration rates for the SAE racer (and, presumably, for any vehicle).

A perusal of the table will show a high initial acceleration, which quickly reduces as the car traverses the drag strip. Appendix 1 & 2 give some more graphs and tables for lighter weights and higher engine outputs.

Figure 21: Speed vs Distance Travelled below shows how speed initially increases quite quickly (from zero) but then the rate of increase (*acceleration*) tapers off.

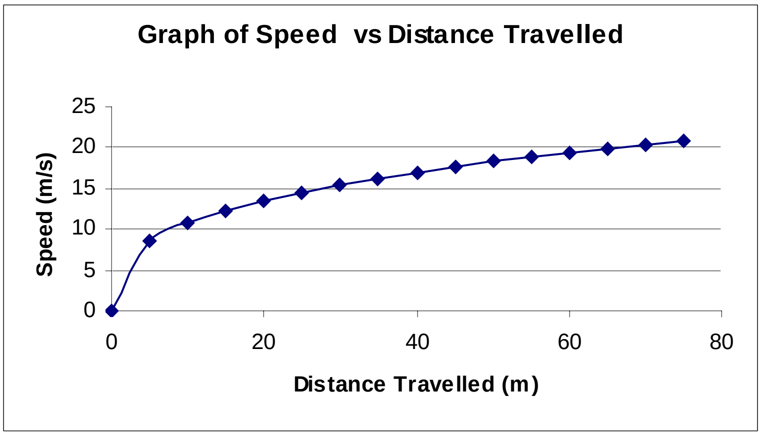


Figure 21: Speed vs Distance Travelled

Figure 22: Acceleration vs Distance Travelled shows an initial high rate of acceleration, which reduces fairly quickly to much less than the assumed 1 g. These realistic amounts of acceleration can be used in determining longitudinal and lateral load transfer during cornering scenarios (useful for calculating suspension geometry changes). It is worth noting here that the calculations are done every 5 metres and the accuracy is based on that distance.

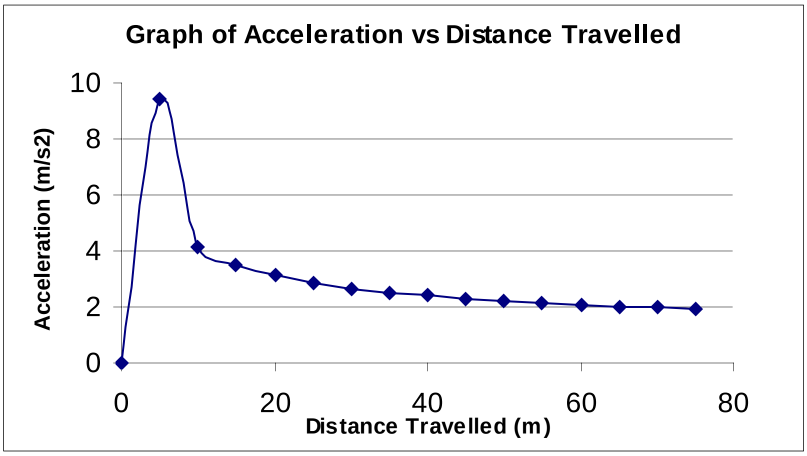


Figure 22: Acceleration vs Distance Travelled

Figure 23: Acceleration (g) vs Distance Travelled below show the acceleration again, but this time in ‘g’ (same curve, different scale).

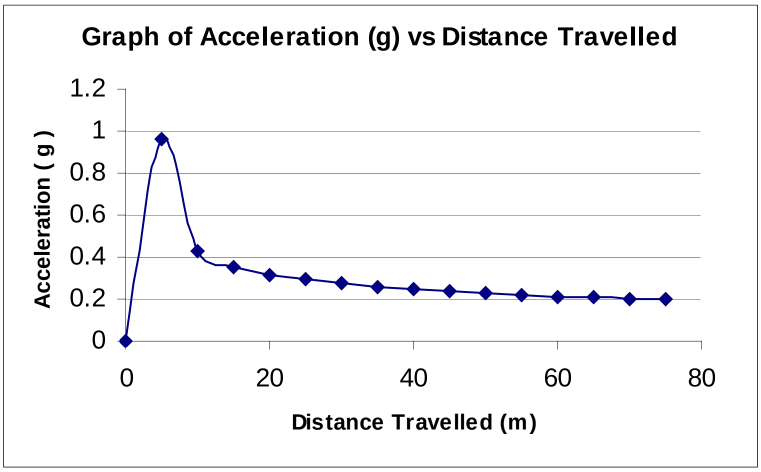


Figure 23: Acceleration (g) vs Distance Travelled

Table 10: Drag Strip Time Sheet below shows the level of performance of the USQ 2004 SAE car. A quarter mile time of 16.81 is not ‘earth-shattering’ - being on par with most passenger cars available today. The quicker Ford / Commodore V8s run 13.5 to 14.0 seconds, with the author’s modified street vehicles normally running 10.3 to 10.8 seconds (which equates to 0-160 kph in 6 seconds).

The Formula SAE cars feel quick because of their size and proximity to the ground.

Dragstrip 60'	2.341
330'	6.82
660'	10.64
1/2 track speed	64.52
1000'	13.98
E.T. (1320')	16.81
m.p.h.	81.7

Table 10: Drag Strip Time Sheet

From all the above, it may be deduced that Fenton's values are reasonable – even though normal driving acceleration goes nowhere near 1 g, the initial acceleration may well exceed that figure considerably (if only for a very short period of time). Braking decelerations of the order of 1 g are not uncommon in passenger cars (even 30 years ago, a Toyota Corolla could achieve a 1 g stop) and should be easily achieved (one would hope) in a Formula SAE vehicle – old jungle saying – *the fastest vehicle is the one with the best brakes*.

Another important factor with acceleration is the loss of traction due to the lifting effect on the right rear wheel. Milliken & Milliken (2002, p478) give, as a comparison (a passenger vehicle), figures of 0.56 g for an open differential and 0.625 g for a locked differential (no differential)

The formula given is:

$$A_{Xmax} = \frac{\mu \frac{a}{\ell}}{\left(1 - \mu \frac{h}{\ell}\right)}$$

where:

μ	=	traction co-efficient
a	=	distance from front axle to CG.
h	=	distance from ground to CG.
l	=	wheelbase

For a Formula SAE car, similar to the 2005 USQ vehicle, the following calculations would be representative:

μ	=	1.2 for racing slick
a	=	0.825 m
h	=	0.265 m.
l	=	1.65 m.

These numbers give a maximum accelerative force of:

$$\mathbf{A_{Xmax} = 0.74\ g}$$

From this, and the realistic acceleration curves given earlier, it would be fairly safe to assume the following:

1. The vehicle would have traction problems in a full power drag type start.
2. The vehicle would not have straight line traction problems once its speed exceeded 10 m/s (35 kph)

Of course, it must be realised, that the level of driving skill is critical to the amount of traction a vehicle exhibits. Poor and undisciplined drivers tend to produce large amounts of *power oversteer* and can also produce huge amounts of *understeer* where none existed for the good driver.

Torsional Rigidity.

The main consideration for the chassis with regard to acceptable handling is the torsional rigidity. The efficient transfer of loads by the suspension – and the consistency of that suspension's performance – is dependent, to a very large degree, on the torsional rigidity of the chassis. A flexible chassis acts as a 'de facto' suspension – and how do you tune and adjust a 'rubbery' chassis? English cars of the forties and fifties were notorious for their flexible chassis – prompting, no doubt, Colin Chapman's (of Lotus fame) oft quoted statement

" Any suspension will work if you don't let it".

Would he have been referring to making an English car handle acceptably by stiffening up the suspension to the point that the only working suspension was the flexibility of the chassis?

However, in the Formula SAE racer, it is intended to have a chassis rigid enough to allow the suspension to function correctly – and to be tuned (that is, changes can be made to the suspension to produce the desired changes in handling – reliably and with repeatability).

Deakin et al conclude that a Formula SAE racer, which has a total suspension roll stiffness of 500 – 1500 Nm/degree, requires chassis stiffness to be between 300 and 1000 Nm/degree to enable the handling to be tuned (and noting that a flexible chassis will cause understeer).

This tends to follow USQ experience with the 2004 SAE car, which has a measured torsional rigidity of 214 Nm/degree – along with, amongst other traits, understeer. The 2004 USQ car appears to drive reasonably well, apart from the understeer and other minor construction matters, so this figure of 300 Nm/degree as a minimum appears to be founded in practice.

Fenton (1980,p7) gives a torsional stiffness for a normal family saloon as a minimum of 6500 Nm/degree and also gives the following formula for torsional stiffness of a chassis:

$$\mathbf{C = cd / D}$$

Where C = torsional stiffness in N/mm
c = spring rate
d = road wheel deflection
D = torsional deflection of chassis

For a typical SAE racer, this equates to a torsional stiffness of 1000 N/mm, which for a track of width of 1200mm, becomes about 1090 Nm/degree, the upper end of Deakin et al’s figure for an SAE racer.

Gaffney and Salinas, in their *Introduction to Formula SAE Suspension and Frame Design*, claimed a torsional rigidity of 2900 Nm/degree for the University of Missouri (Rolla) SAE racer, whereas the Laval University’s 2004 SAE team claimed 2000Nm/degree for their car. These figures appear to be theoretical (and rather high) – their frames were not actually subjected to physical testing as was the 2004 USQ car.

Whilst there is a bit of conflict in the above figures – some seem rather higher than others – the fact remains that the 2004 USQ car, which was physically tested to 214 Nm/degree, had a reasonable level of handling. This is not to say that the 2004 USQ car has a chassis of sufficient rigidity – there is still an understeer problem and Deakin et al’s figure of a minimum of 300 Nm/degree would appear to be a realistic minimum. It is intended to aim higher than this minimum.

Longitudinal Rigidity.

Longitudinal strength appears to be of secondary concern - if the chassis has adequate torsional rigidity, it will have quite sufficient longitudinal strength – and the factor that most affects handling is the efficient (or otherwise) transference of lateral loads. Small longitudinal deflections, in themselves, have no effect on the lateral load transfers which strongly affect handling through the changes in suspension geometry.

Summary:

From the above, it can be seen that the primary design criteria are:

1. Wheelbase 1650 mm
2. Front Track 1285 mm
3. Rear Track 1265 mm
4. Weight - 40 kg (max)
5. Torsional Rigidity – 300 N.m / ° (minimum)

Other design considerations, for example, engine mounting brackets, will be done for each case, taking into account the masses involved and the accelerative and other forces acting on that component.

5.1.2. Design Process

The design process followed a fairly logical sequence (as it should). This process is set out below:

1. Sketch
2. Autocad
3. Finite Element Analysis
4. Commonsense
5. Redo the above until acceptable.

5.1.2.1. Sketch

This was the imaginative part. The load paths and subsequent frame triangulation had to be determined at this stage. This was done the old fashioned way – on paper with a pencil and a large eraser.

The first series of sketches resulted in the following drawing:

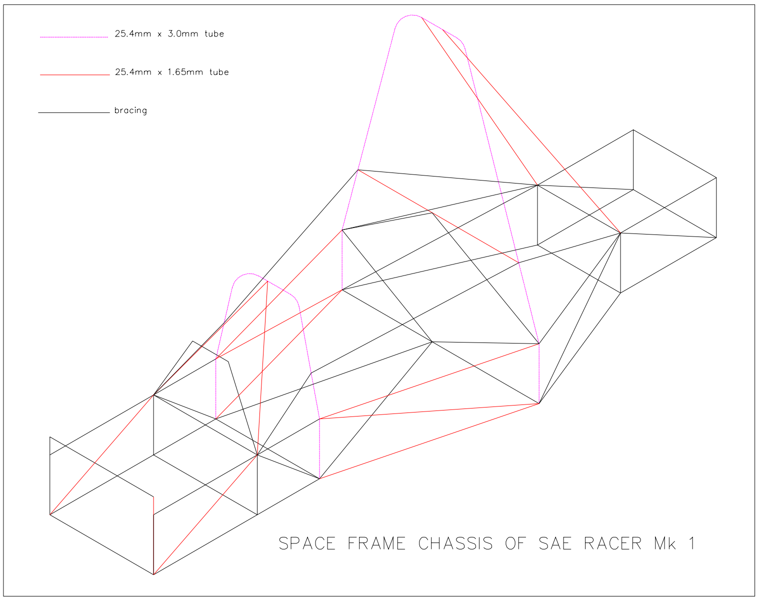


Figure 24: Space Frame – Mark 1

This was followed by the following series of drawings:

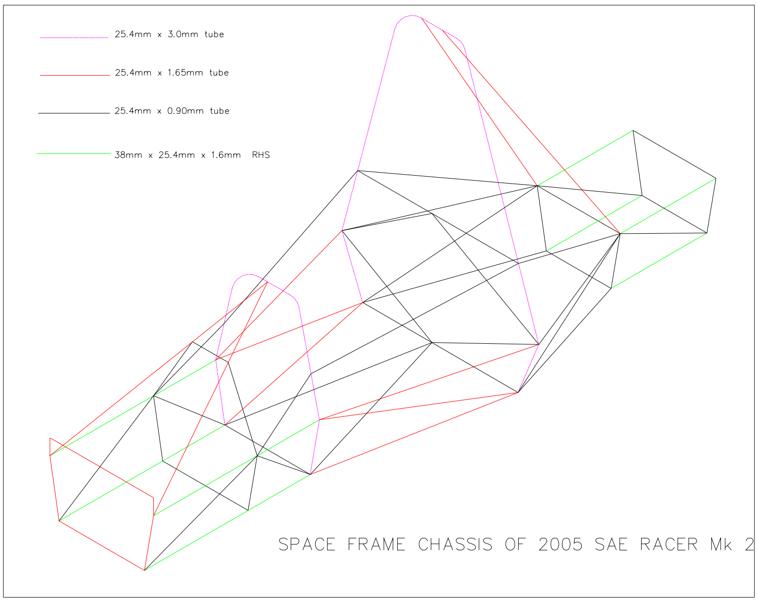


Figure 25: Space Frame – Mark 2

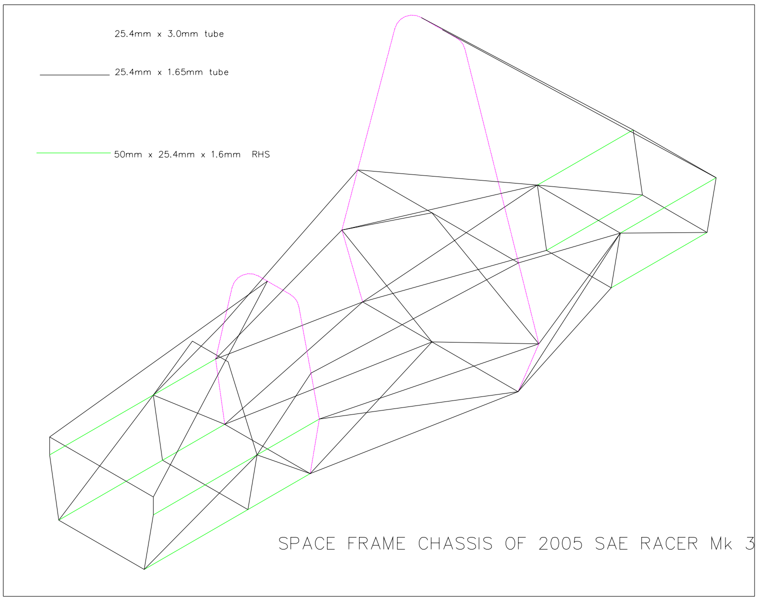


Figure 26: Space Frame – Mark 3

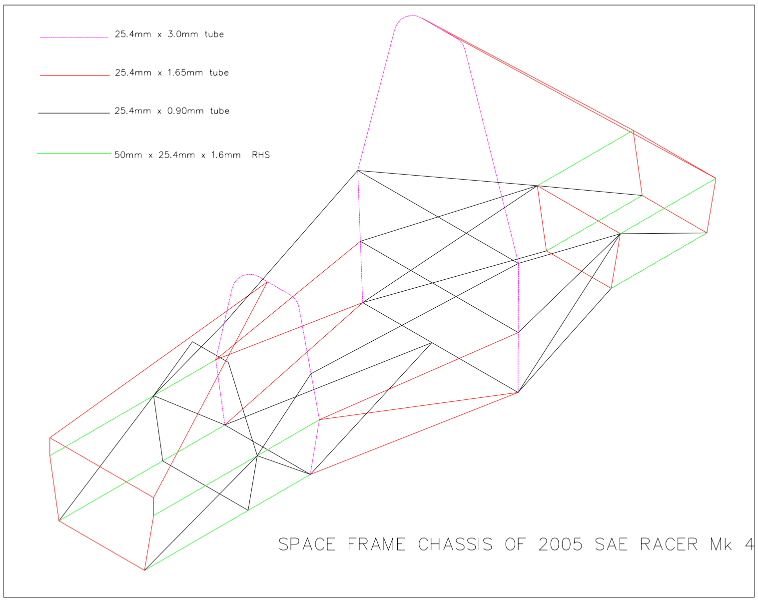


Figure 27: Space Frame – Mark 4

The sketching culminated in the layout shown below.

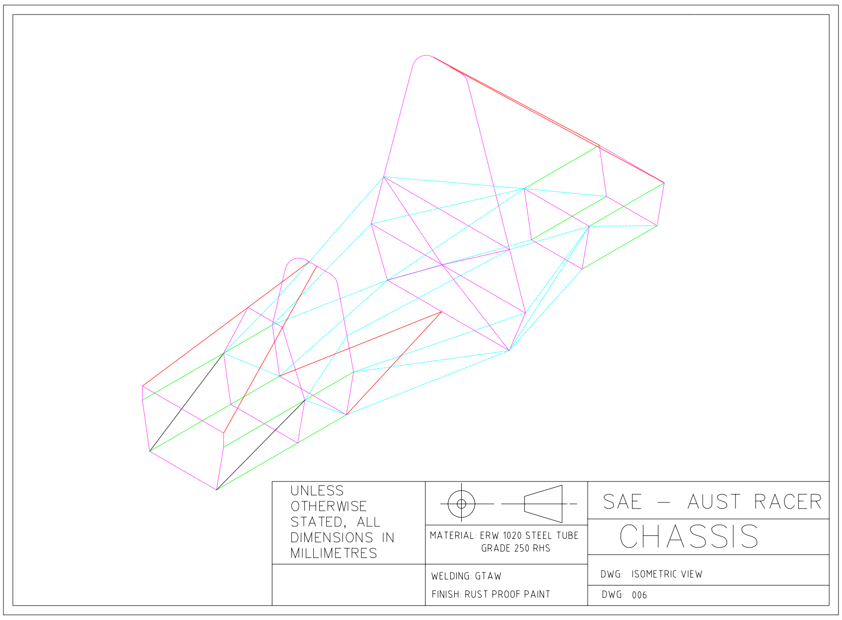


Figure 28: Final Chassis Layout.

This series of drawings actually included all of the above processes – the iterative process of refinement.

5.1.2.2. Autocad

This was the process of actually turning the sketch into a proper drawing – with Autocad being the most convenient tool with which to do this. Because the frame is a 3-dimensional construct, and Autocad works in 2-dimensions, this was an opportune time to draw plans that would be useful in the workshop when manufacturing time came.

It was decided to draw the following views:

1. Floor Rail Frame
2. Mid-Rail Frame
3. Hoops
4. Anthropometrical Data
5. Full Frame – plan & elevations

These drawings are shown below – for the sake of brevity, only the final versions of these are shown, and that these included the other iterative steps described earlier (FEA etc).

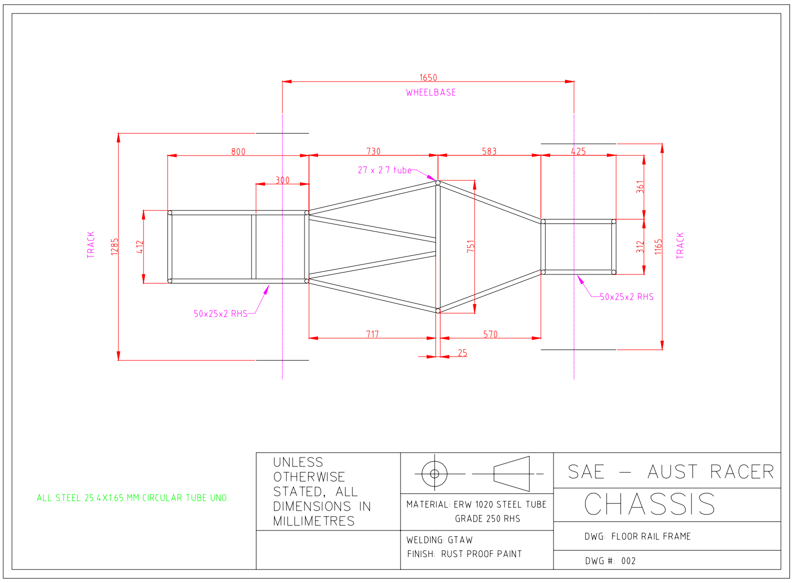


Figure 29: Final Floor Rail Frame

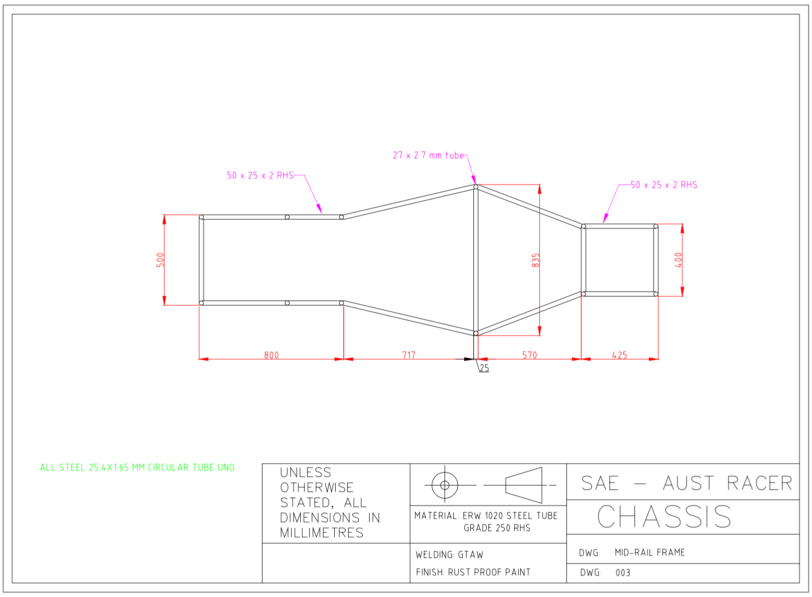


Figure 30: Final Mid-Rail Frame Plan

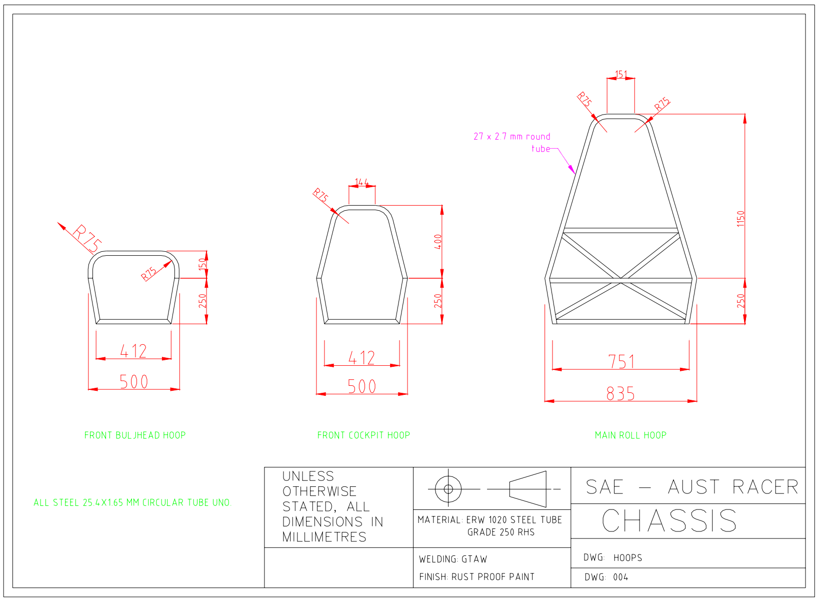


Figure 31: Final Hoops Elevation

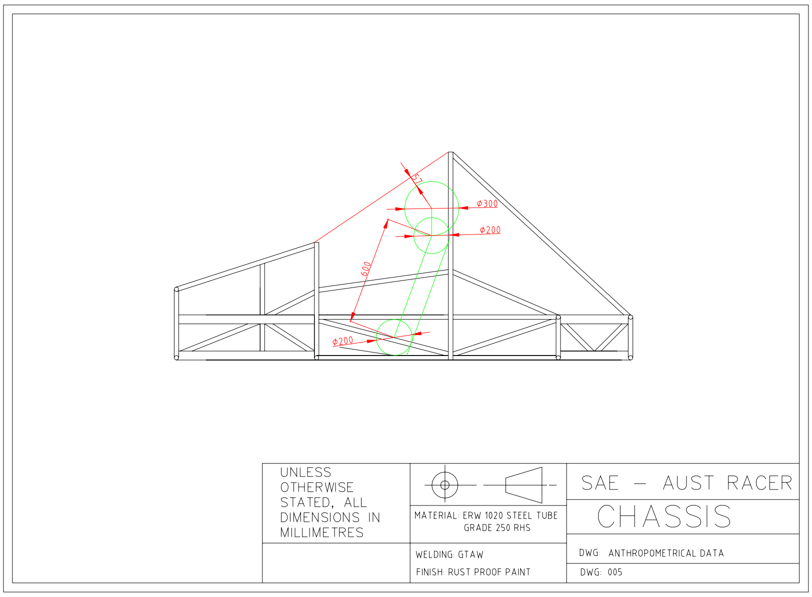


Figure 32: Anthropometrical Data

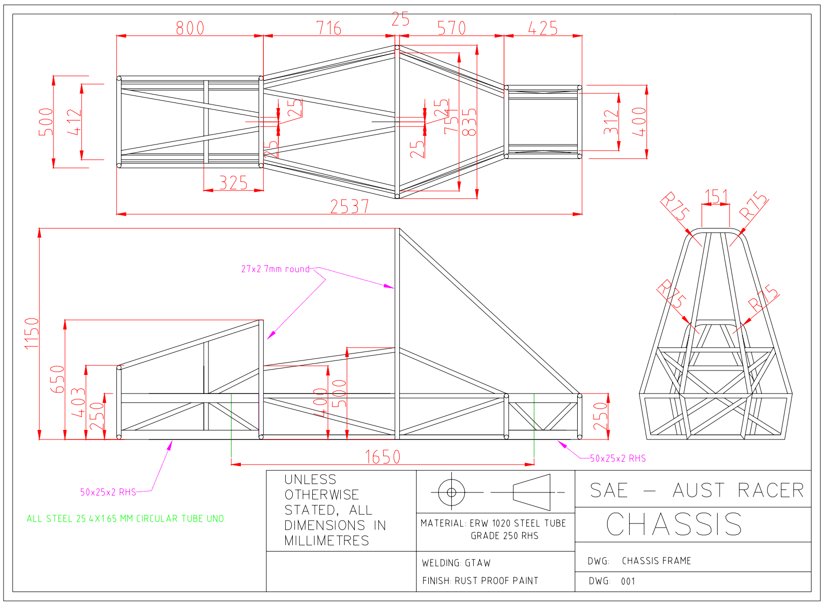


Figure 33: Final Frame – Plan & Elevations

5.1.2.3. Finite Element Analysis

The Finite Element Analysis (FEA) part of the process proved to be educational and informative – with the answers being of the commonsense variety that left the thought “Why didn’t I do it that way in the first place?”

The co-ordinates of each junction in the frame were determined, then these were used to build up a 3-dimensional object in ANSYS7. Since the torsional rigidity was of primary concern, it was decided to test the frame by restraining the front suspension mounting points and applying a moment (4 x 1000N) to the rear suspension mounting points, as shown below in *Figure 34: SAE Frame showing Torsional Loads.*

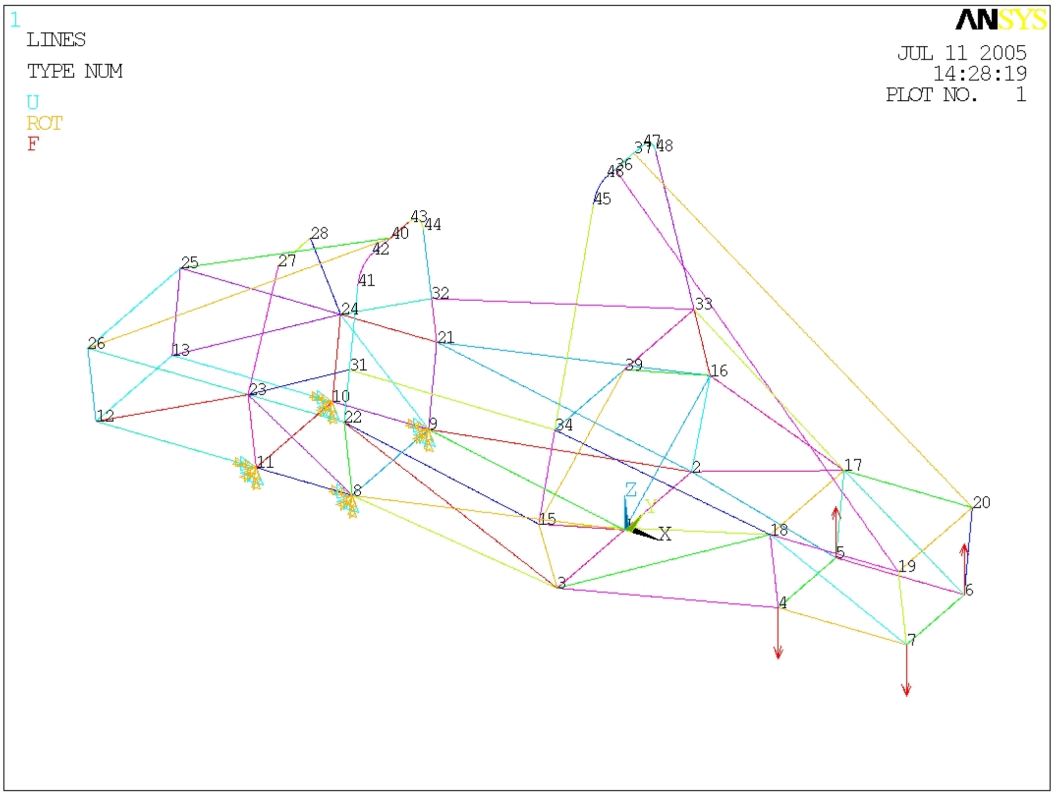


Figure 34: SAE Frame showing Torsional Loads

The resultant deflections are shown below in *Figure 35: First Design with Torsional Rigidity Test.*

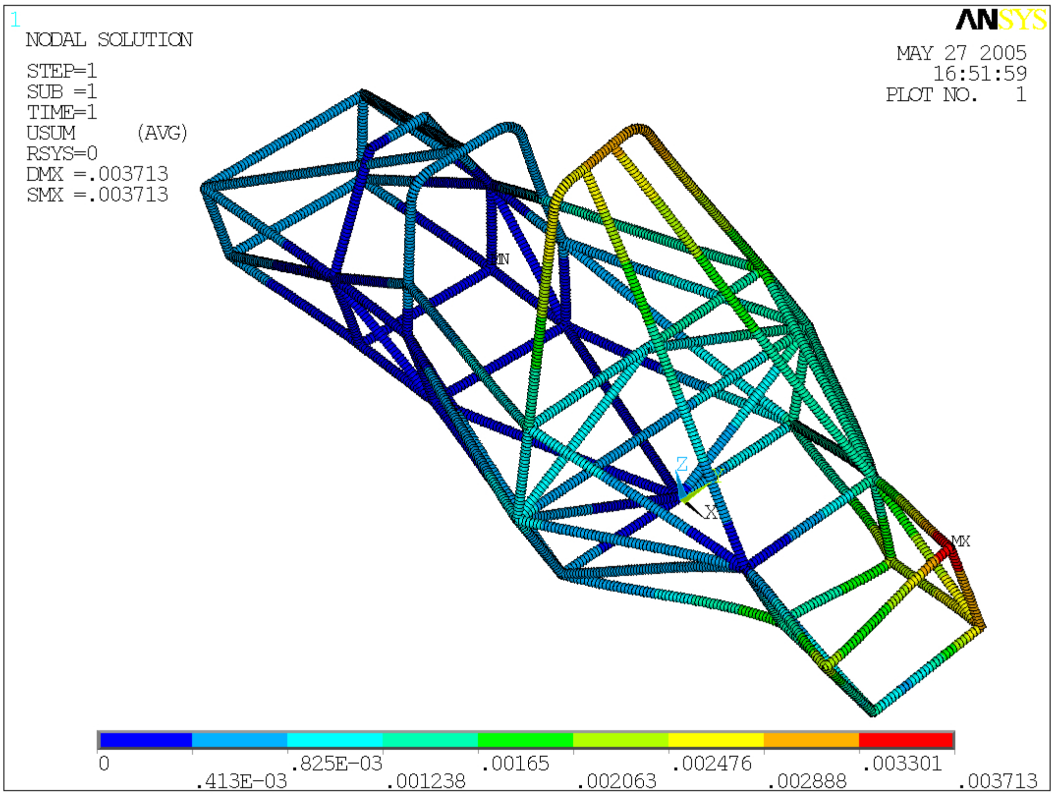


Figure 35: First Design with Torsional Rigidity Test

An inspection of *Figure 35: First Design with Torsional Rigidity Test* above shows that, even though the deflection in the diagram is exaggerated, the movement is concentrated in the rear suspension section. The obvious modification is to move the main hoop brace bars to the rear of the frame from their original position in front of the rear suspension section. The results of this are shown below in *Figure 36: Modified SAE Frame with Torsional Rigidity Test*

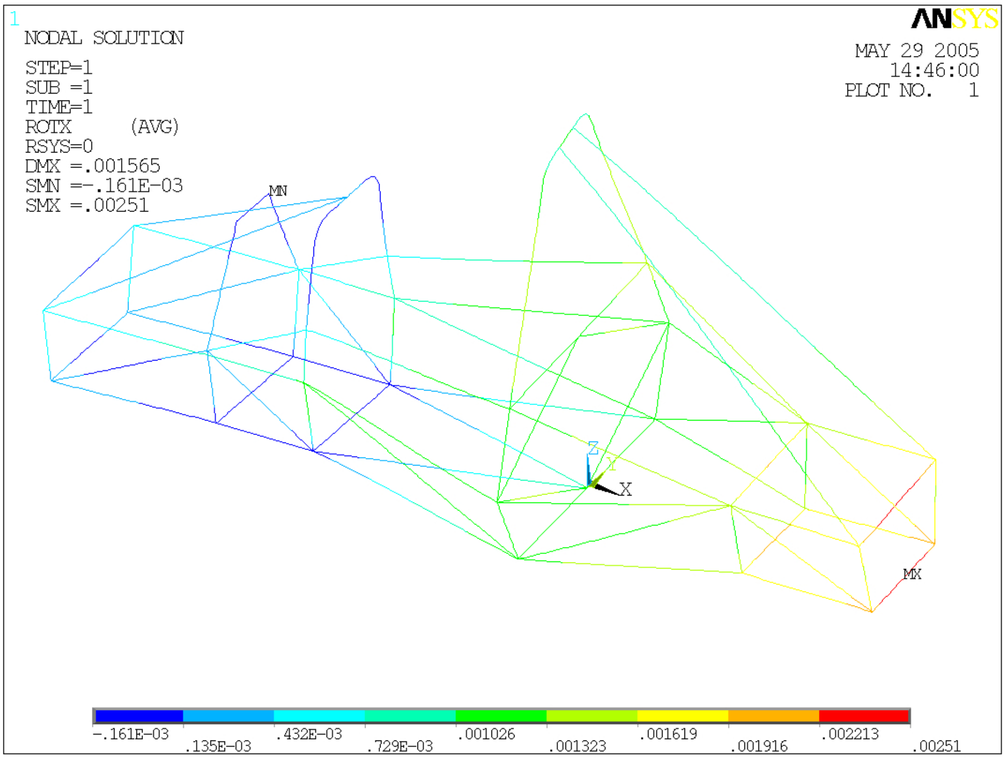


Figure 36: Modified SAE Frame with Torsional Rigidity Test

The chassis for *Jettison 1* (USQ 2005 car) was tested for longitudinal deflection by clamping the rear to a bench and a load of 850 N applied to the front of the bulkhead where a deflection of 14 mm was recorded. A test of deflection from where the suspension is mounted would be more relevant for this purpose.

To get a measure of the longitudinal strength of the frame, the rear was restrained and 4000N applied to the front of the frame. The results of this are shown below in *Figure 37: Longitudinal Strength of Frame*

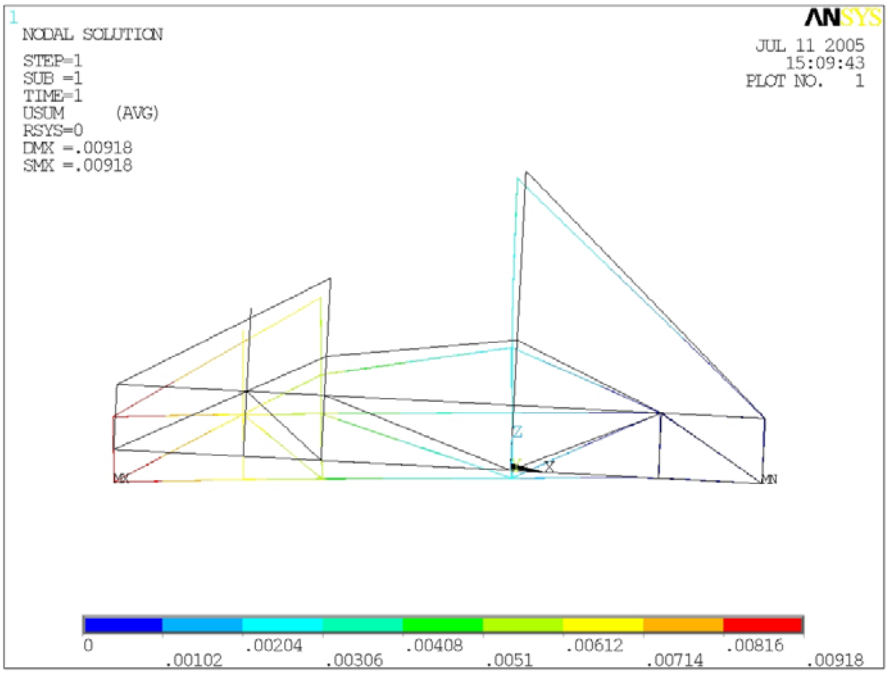


Figure 37: Longitudinal Strength of Frame

Finite Element Analysis Results

Torsional rigidity

The moment applied of 4000N at a distance of 156 mm from the longitudinal central axis equates to 624 N.m.

The amount of deflection in the 156 mm was 3.5 mm and this equates to a total deflection of 1.285°

Hence:

Torsional rigidity = 624 N.m / 1.285° = 485 N.m / degree

Longitudinal rigidity

The load applied was 4000N at the front of the frame.
The deflection indicated by the ANSYS7 analysis was 9.2mm.

Hence:

Longitudinal rigidity = 4000N / 9.2mm =435 N/mm

(Jettison equated to 53 N/mm)

The only other design feature that was somewhat different to the 2004 Car was the 10° rake to the sides. This is shown below in *Figure 38: 10° Rake to Frame:*

Figure 38: 10° Rake to Frame

This was done for the express purpose of making suspension geometry somewhat easier to design. Whilst equal length and parallel suspension arms keep the wheel camber constant in cornering, they play havoc with the roll centres and track widths. Since the roll centre is the point about which the centre of gravity (CG) tends to rotate (generating a moment) it is critical that this point has no large or sudden changes. This moment is what determines the down force on the tyres, and a sudden increase on one tyre would mean a sudden decrease on the opposite wheel – with a resultant unexpected loss of traction. Changes to the track widths can only occur when a tyre slides

– that is, breaks traction. This is also unadvisable during hard cornering.

It is more advisable to use unequal and non-parallel suspension arms which, whilst not controlling wheel camber to the same degree, can keep the track widths constant and reduce the migration of the roll centres during cornering.

To make this design easier, it was decided to rake the sides of the frame – and also to make the longitudinal mounting frame members of 50 x 25 x 2.0 rectangular hollow section (RHS), both for the added strength and for the convenience of having a flat surface to which to fix brackets.

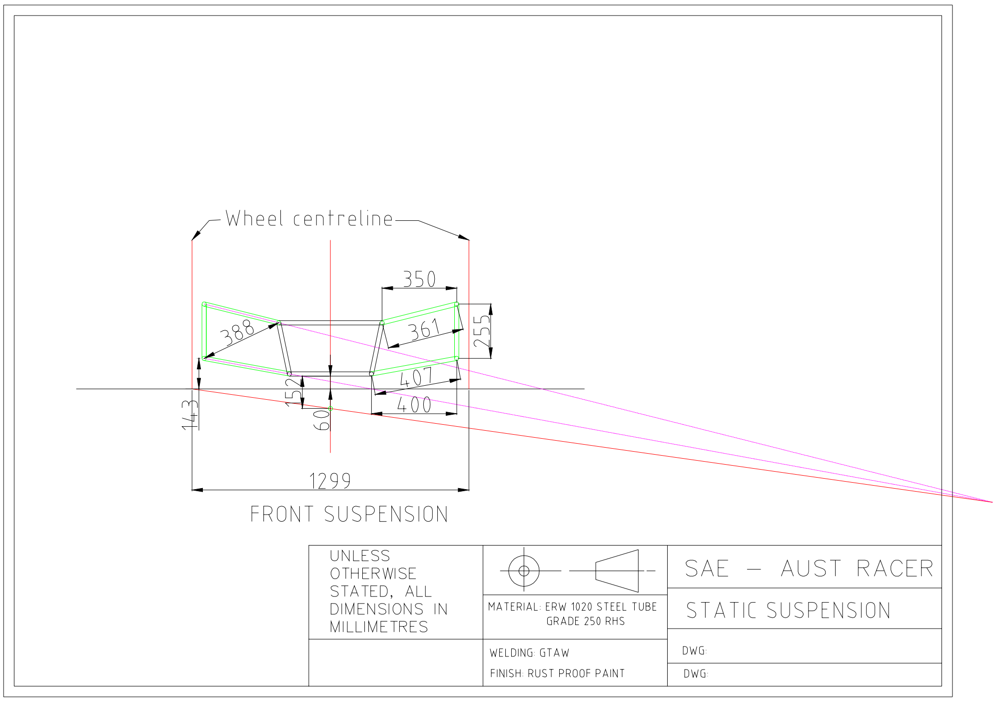


Figure 39: Suggested Static Suspension Geometry

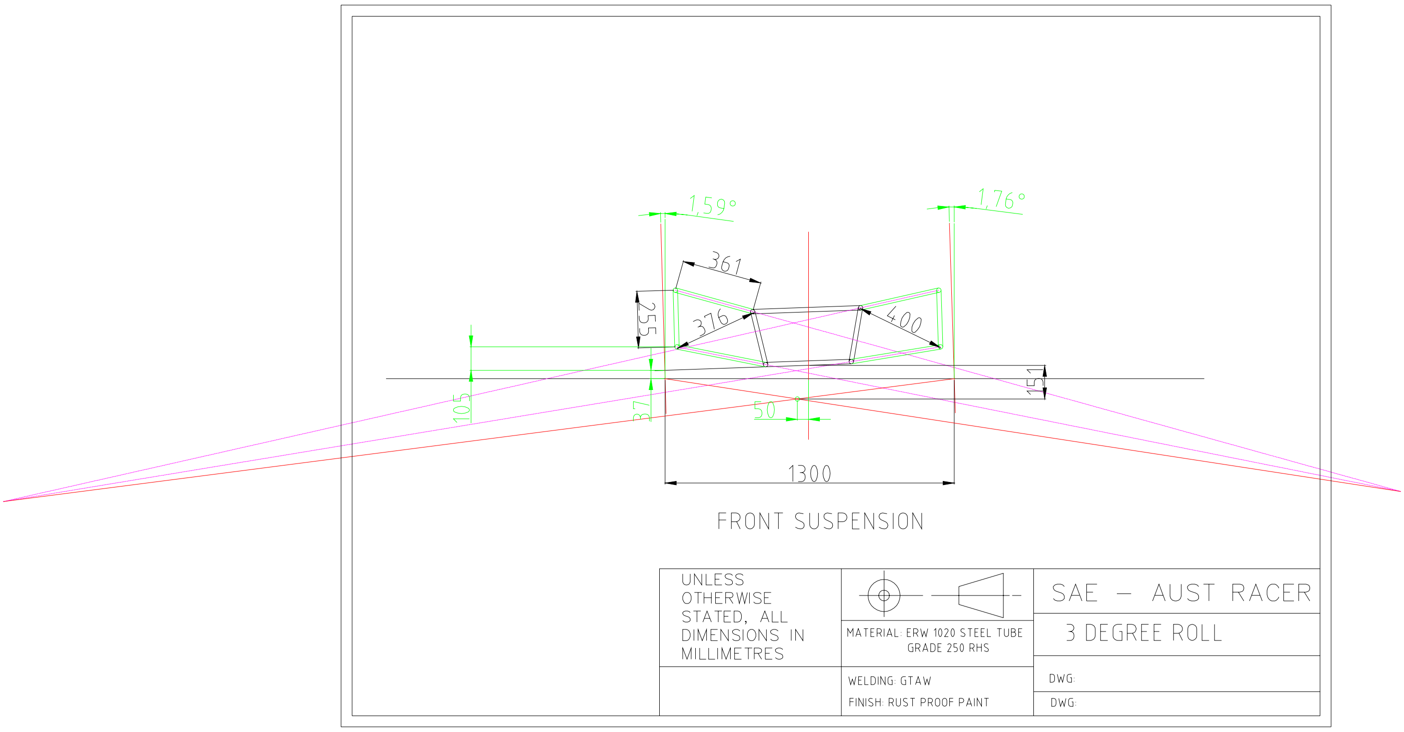


Figure 40: Suspension Geometry with 3° Roll

From the above 2 diagrams, it can be seen that, even during 3° roll (which is pretty severe cornering), the track width changes only by 1 mm and the roll centre's vertical position only changes by 1 mm with a 50 mm horizontal migration. These changes would have little detrimental effect on the handling of the vehicle. Please note that this is only a suggested suspension geometry and is shown only for the purpose of showing the advantages of the unequal length, non-parallel wishbone suspension geometry, which is easier to design and fit with raked sides.

5.2. Work Processes

The work processes to be used for the manufacture of the frame were as follows:

1. Set Out
2. Steel Cutting
3. Tube Bending
4. Welding Processes
5. Use of Jigs

5.2.1. Set Out

Since this was the prototype, it had been decided that the frame was to be 'hand built welded' – this would mean little or no jigging, and cutting and fitting members following a logical construction process. The first step would be to mark out a full size floor plan on the workshop floor. The following drawing was used for this purpose.

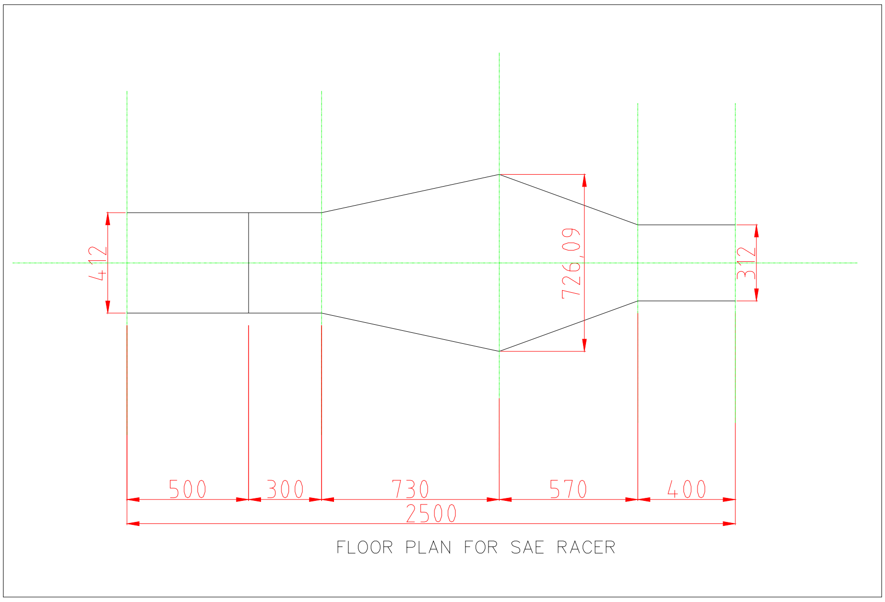


Figure 41: Floor Set Out Plan

The next step was to draw the various hoops needed on the floor – these hoops were to be used to build the frame around. Particularly important as the rules stipulated that the main hoop must run, uncut, from the bottom of the frame on one side to the bottom of the frame on the other side. These are shown in the drawings below:

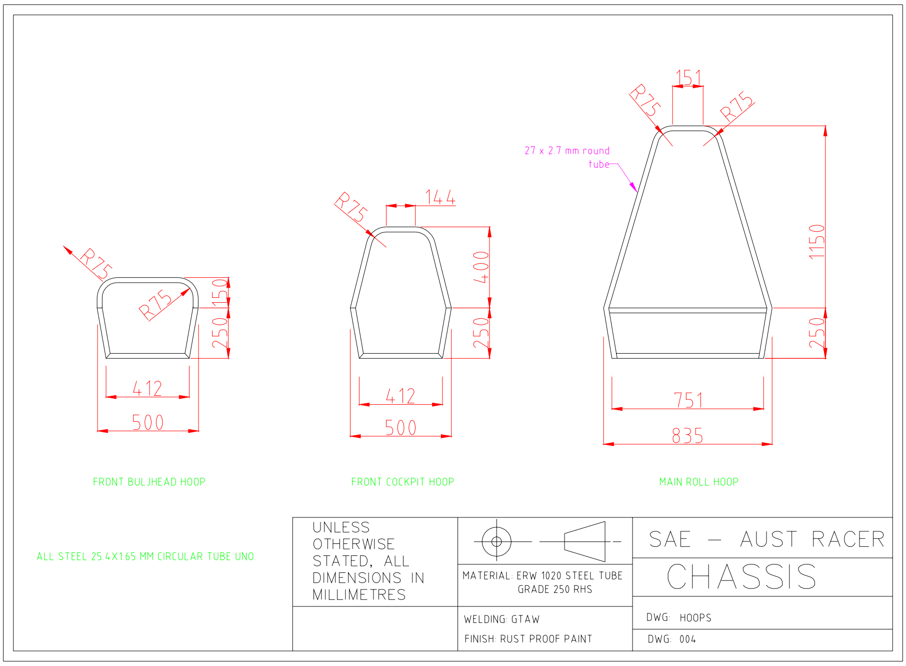


Figure 42: Hoop Construction Drawing

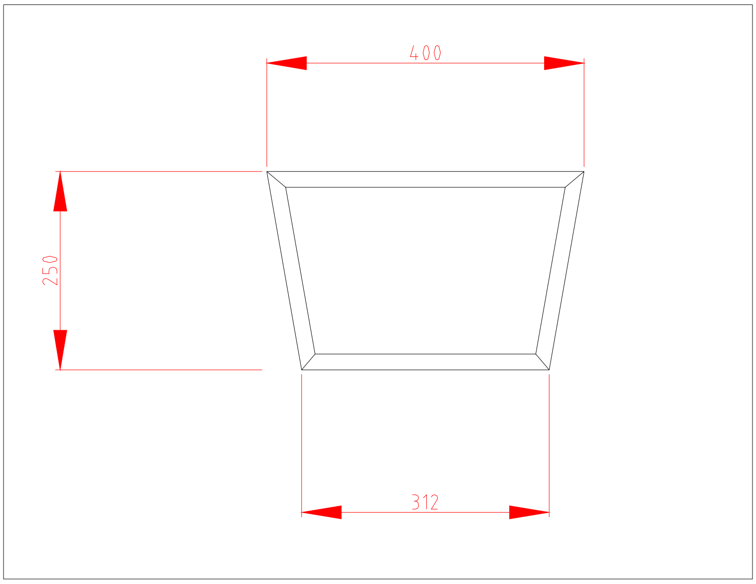


Figure 43: Rear Hoops – 2

5.2.2. Steel Cutting

There were 2 types of cut needed for the construction of this frame – curved and straight. The straight cuts were to be simply done by a friction cutter (drop saw). Any heat effects from this process would be negligible compared to the subsequent effects of welding.

The curved – generally circular at various angles – needed something better than hand cutting each to shape. To make this process a lot simpler and quicker, it was decided to manufacture a *pipe notcher* that used standard, and readily commercially available, hole saw blades. This was successfully done by Bronson Hansen, a final year mechanical engineering student.

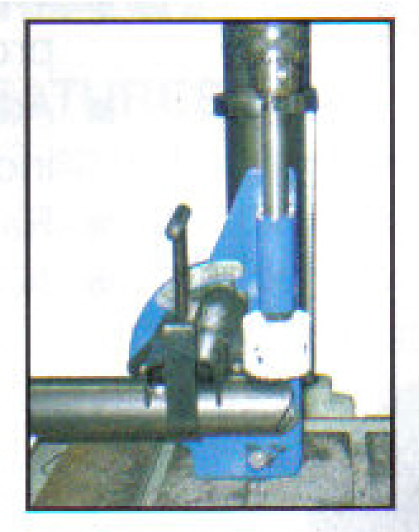


Figure 44: Typical Pipe Notcher

The pipe notcher that was made can also handle RHS and SHS as well as round tube, along with the capacity to cut the holes at any angle up to around 50°. Using this device, a typical hole took around 30 seconds to cut – this was, of course, followed up by 10 to 15 seconds of cleaning the cut on a linisher.

The individual lengths of pipe were to be marked on the floor and then cut to size – hand fitting each piece to its proper place.

5.2.3. Tube Bending.

In the Formula SAE Rules, the following requirement applies to tube bending:

> *3.3.4.1.(B) The minimum radius of any bend, measured at the tube centreline, must be at least three times the tube outside diameter.* ***Bends must be smooth and continuous with no evidence of crimping or wall failure****.*

This initially caused some problems, with the first set of hoops displaying severe crimping. However, the second set was bent at Toowoomba Specialised Welding using a Bramley Pipe Bender, as shown below.

Figure 45: Bramley Pipe Bender

The hoops were successfully bent using this pipe bender and showed no signs of crimping or any other form of distress. (As an aside, this bender was also used to bend the runners of the inlet manifold designed by Melinda Plank for the 2005 Car). It was initially suggested that mandrel bending would be the only process that would prove satisfactory, but this was not the case.

5.2.4. Welding Processes

Groover (2002) states that welding is divided into 2 major categories – *fusion welding* and *solid state welding*. Since solid state welding requires pressure or heat and pressure, the process is not suitable for notched tubing in a 3-dimensional frame.

Fusion welding is accomplished by the melting of the two parts to be joined – in most normal methods, a filler material is also added. Groover also lists the following fusion welding methods:

1. Arc welding – consumable and non-consumable electrodes
2. Resistance welding
3. Oxyfuel gas welding
4. Others – electron & laser beam, electroslag & thermite

Of these, category 1 methods are the most commonly used for this type of fabrication. Categories 2 and 4 are for specialised applications – seam welding of tubing etc. Category 3 can be used for this type of fabrication, but tends to add a bit too much heat to the tubes.

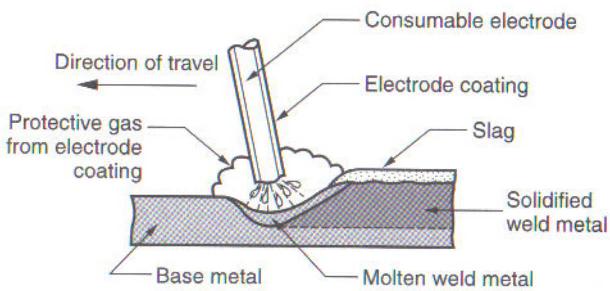


Figure 46: SMAW or stick welding (Groover)

Arc welding commonly consists of the consumable methods of shielded metal arc welding (SMAW), commonly known as 'stick welding' – shown above - and gas metal arc welding (GMAW) or MIG welding and the non-consumable electrode methods of gas tungsten arc welding (GTAW), normally called TIG Welding (Tungsten Inert Gas), and plasma arc welding (PAW).

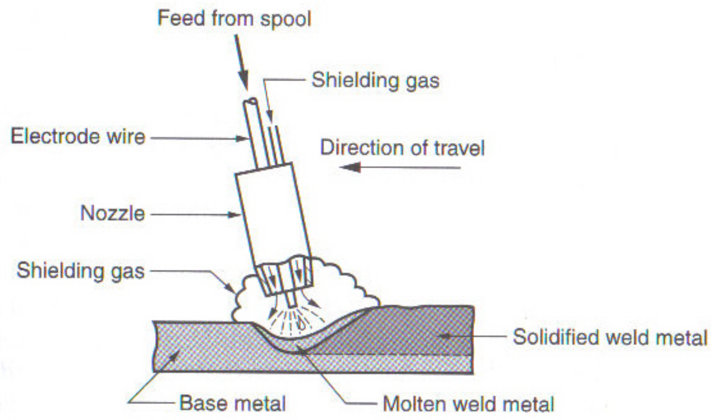


Figure 47: GMAW or MIG welding (Groover)

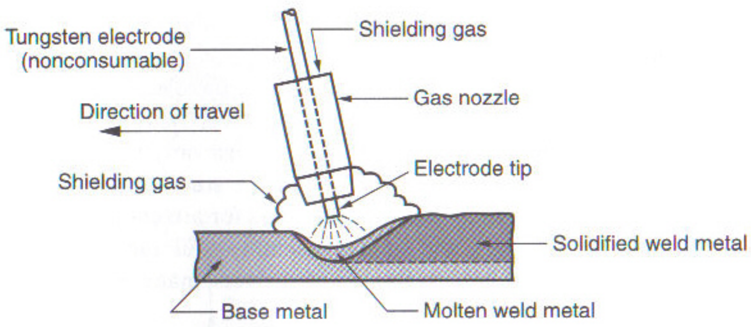


Figure 48: GTAW or TIG Welding (Groover)

So the choice came down to one of the 3 methods shown above. In reality, not actually possessing a stick welder made the choice between MIG and TIG welding. Stick welding tends to be a little messy and requires the manual removal of the protective slag.

MIG and TIG welding both provide a good quality weld that is free from slag, as both use protective gasses. TIG, however, has the added advantage of being able to be done with or without a filler material – depending on the job. The TIG method produces a higher quality spatter free weld, and is very suitable for welding the various steel alloys and aluminium. It is, however, more expensive than MIG welding.

Given the above considerations, it was decided to ‘tack’ the frame together with a MIG welder and finish the welds with a TIG welder.

5.2.5. Use of Jigs

For the prototype, it was decided to mark out a pattern on the floor for the various components of the frame, then hand cut and fit each of the members.

This method worked quite well for the prototype allowing, as it did, any minor modifications that may have been expedient to make with minimal fuss. After all, this is what a prototype is for – to uncover any problems that may not be obvious during the design phase. Hence, no jigs were used during the manufacture of the prototype.

However, for the proposed manufacture of 4 frames per day (which is what would have to happen if 4 *cars* were made per day), various jigs would be necessary.

The jigs needed will be discussed later.

5.3. Quality Control Methodology

Evans and Lindsay (1989) refer to quality engineering as the process of designing quality into a product and predicting potential quality problems before production. This came about because of the realisation that the traditional method of inspecting for quality only removed defects *after the fact* (if at all) – and that did nothing to reduce the (often very high) cost of such defects. This realisation led to, amongst other things, the formulation of various statistical quality control methods such as process control charts and acceptance sampling (Grant & Leavenworth, 1980). Design and manufacturing must be co-ordinated to produce an item that can be manufactured with consistent acceptable quality and minimal waste of both materials and labour. Another important factor these days is *product liability* – the product must be made to specifications (providing those specifications are correct – which is also part of the quality process).

For this to happen, the following are important:

1. The basic design should be simple and easy to make
2. Worksheets should be simple and **no** ambiguity.
3. Worksheets should give ‘ownership’ to the relevant worker.
4. Worksheets should include ‘signed off’ quality control checks.

The prototype is an important part of this process – its construction can show up problems not foreseen, as can the subsequent physical testing of the prototype, regardless of what the computer models show.

Chapter 6

6. The SAE Chassis – Manufacture

This chapter deals with the actual manufacturing of the SAE frame, which was done at the workshop of Toowoomba Specialised Welding, with the help of fellow mechanical engineering student, Bronson Hansen and Toowoomba Specialised Welding's Michael Garner (Proprietor) and his 'off-sider' Craig Rodgers.

This section deals with the following:

1. Worksheets
2. Quantity take-off
3. Manufacturing Process
4. Problems Encountered (& Solutions).
5. The SAE Frame

It should be kept in mind that the Formula SAE Competition calls for a production capability of 4 frames per day and, even though this prototype was hand built, a small workshop such as Toowoomba Specialised Welding could handle such a production rate without any expensive tooling and with current equipment.

6.1. Worksheets

The worksheets used for the prototype were taken directly from the plans drawn for the frame – it made sense to draw *useful* plans.
A typical, and most useful, one is shown below – this was used to mark the basic frame out on the floor of the workshop.

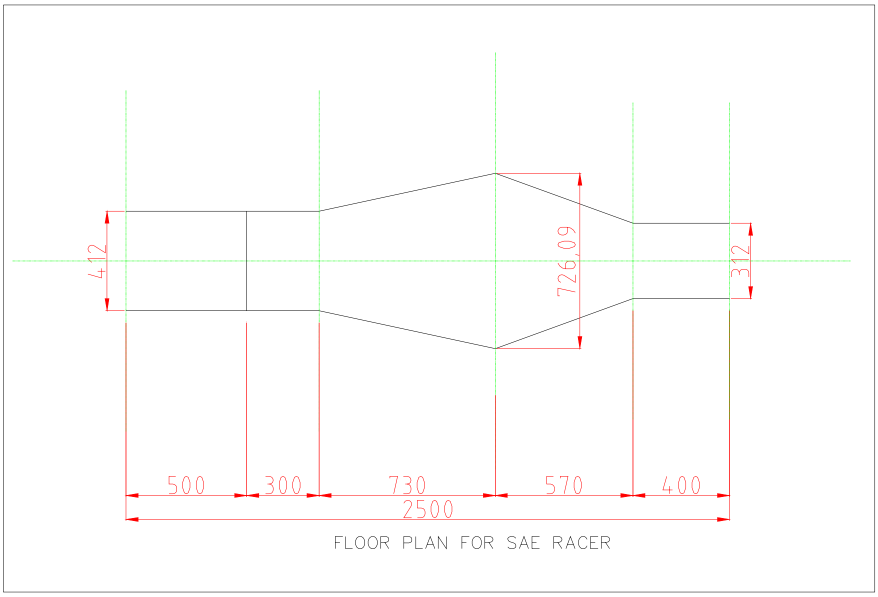


Figure 49: Floor Plan for Frame

As mentioned in the previous chapter, the hoops were also marked out on the floor. This process had to be done carefully, with much double-checking – this part *had* to be correct. This process was done by 2 people – each checking / confirming the other's interpretations of the plans.

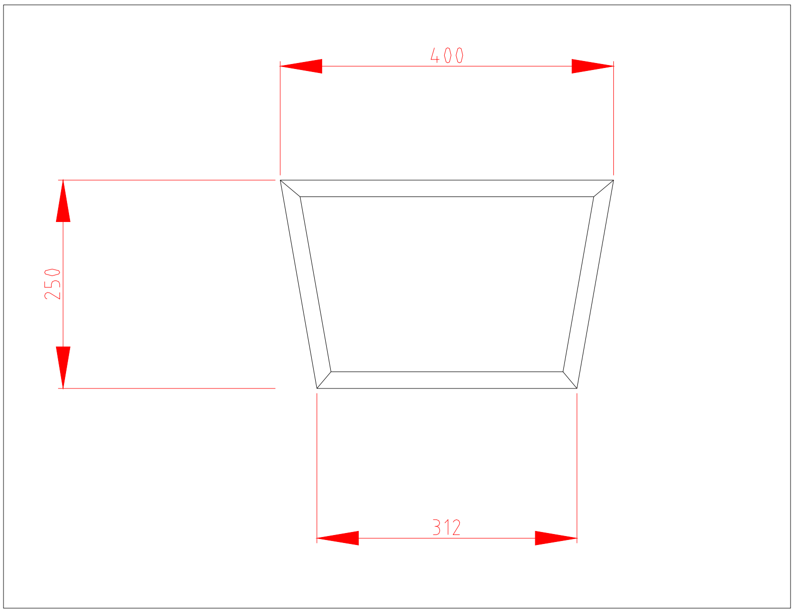


Figure 50: Rear Hoops

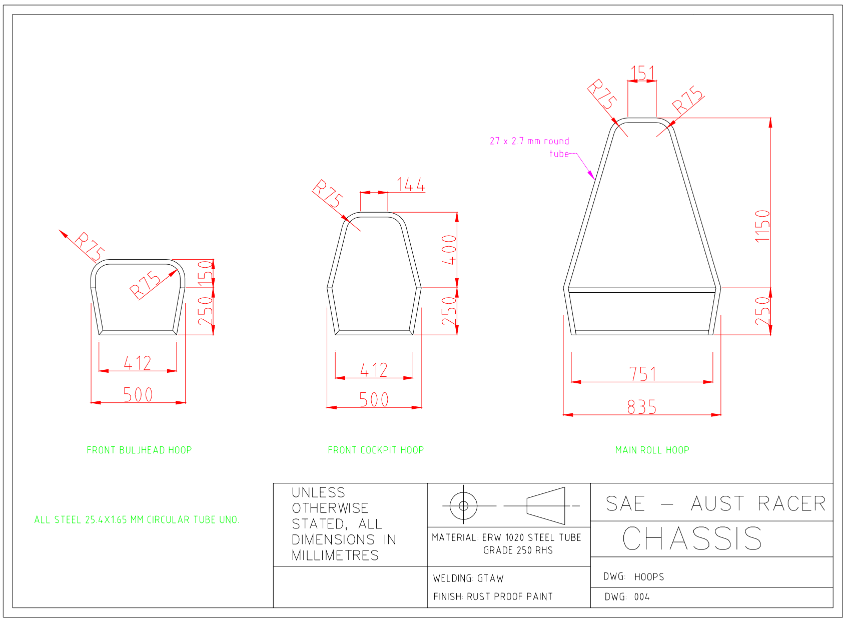


Figure 51: Hoops for Frame

The above were used as worksheets for the construction of the frame. The hoops were cut, bent and then welded together. These were then held in the vertical position whilst the horizontal components of the frame were measured, cut and then tacked into position. This was done with reference to the worksheet below:

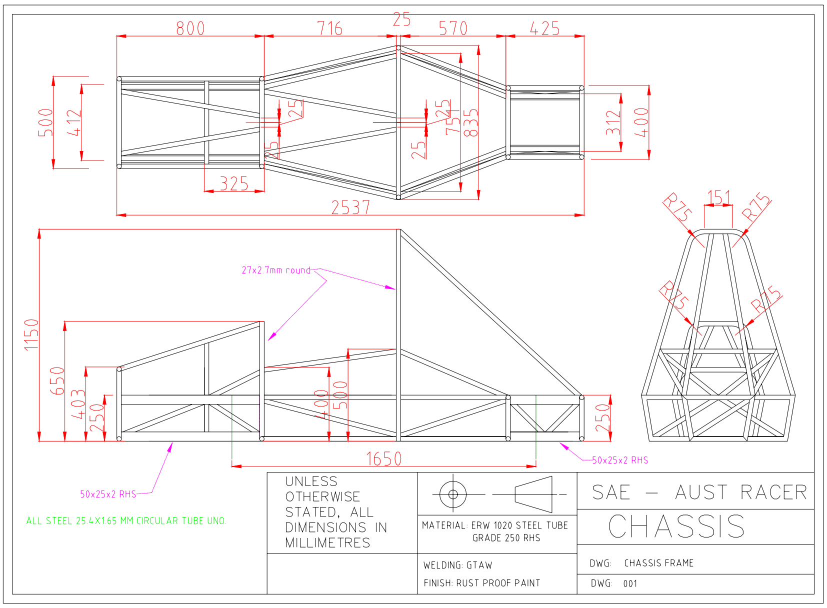


Figure 52: Full Frame

Quality control, or management, started with the design of the frame – it had to be built by the designer, so simplicity was the key. There was no part of the frame that was super critical in dimensions or location – and this made for manageable fabrication tolerances (for a prototype). The worksheets gave only those dimensions that were critical to the fabrication process, and in a manner that made construction easy. Hoops were kept vertical, and all ‘in-fill’ members were kept straight. The other factor, which helped with the quality management, was the selection of only 2 sizes of tubular steel – the larger (black) size being used for the hoops (for mandated safety reasons) and the smaller unpainted one being used for the rest of the tubular frame. The front and rear suspension mounting longitudinal members were of 50x25x2.0 RHS – these were easy to spot. This sizing policy made the use of the correct size member very easy and also determinable with a casual inspection – no need for careful measurements or metallurgical testing.

6.2. Quantity Take-off

The quantity take-off was simplified by the use of only 2 tube sizes and 1 RHS size. The Autocad program facilitated the measuring of individual member lengths. These individual lengths were then tallied with special regard to the commercially available lengths – 6.1 m in these sizes, and an order raised through the USQ Workshop.

6.3. Manufacturing Process

The manufacturing process was described above in the Worksheet Section – it simply consisted of, for the prototype, of marking out the basic floor frame, then using the hoops to ‘lay the keel’, and then ‘going for it’.
For a production rate of 4 per day, a different method would have to be used. The following would be necessary:

1. All members would be pre-cut and stored in special bays. If the steel is paid for monthly, then a month’s production of frames (80) could be cut at a time. Special test members would be necessary to check lengths and angles at appropriate times.

2. Jigs would be made for the hoops.

3. A jig would be made for the floor frame

4. Go – No Go gauges would be needed for each section of the frame.

5. One worker would specialise in tacking / assembling the frames, signing off each frame.

6. One worker would specialise in TIG finish-welding the frames, signing off each frame.

7. When the frame is finished, the worker would use appropriate Go – No Go gauges to ensure the frame was within specifications, and signed off accordingly.

The fine details of the above would be determined, if and when, the on-going order for 4 frames per day was received.

6.4. Problems Encountered (& Solutions)

The problems encountered in the manufacturing of the frame were minimal due to the planning and thought that went into the design. (I *had* to say that, didn't I?) The only consequential problem encountered was the initial bending of the hoops by the USQ Workshop. There appeared to have been some sort of communication problem – a 6.1 m length of tube was sourced of which 1.5 m was in excess of requirements. This 1.5 m length was supposed to be bent first to check the quality of the bending before the rest of the tube was bent. However, all the hoops were bent, with obvious crimping. It was initially thought that this may be acceptable, but a subsequent reading of the Formula SAE Rules clarified the situation – not usable. Unfortunately, this caused a delay in the manufacturing of the frame of the order of 6 or so weeks. The *window of opportunity* in Toowoomba Specialised Welding's Workshop had only been 'open' for that particular weekend.

6.5. The SAE Frame

It would be appropriate here to show some photographs of the finished frame. It would also be relevant to show here a photograph of the USQ 2004 frame for comparative purposes (with apologies to the 2004 USQ Team for the unflattering photograph – unfortunately the only one available).

Figure 53: Frame from USQ 2004 Car

Figure 54: 2005 Frame

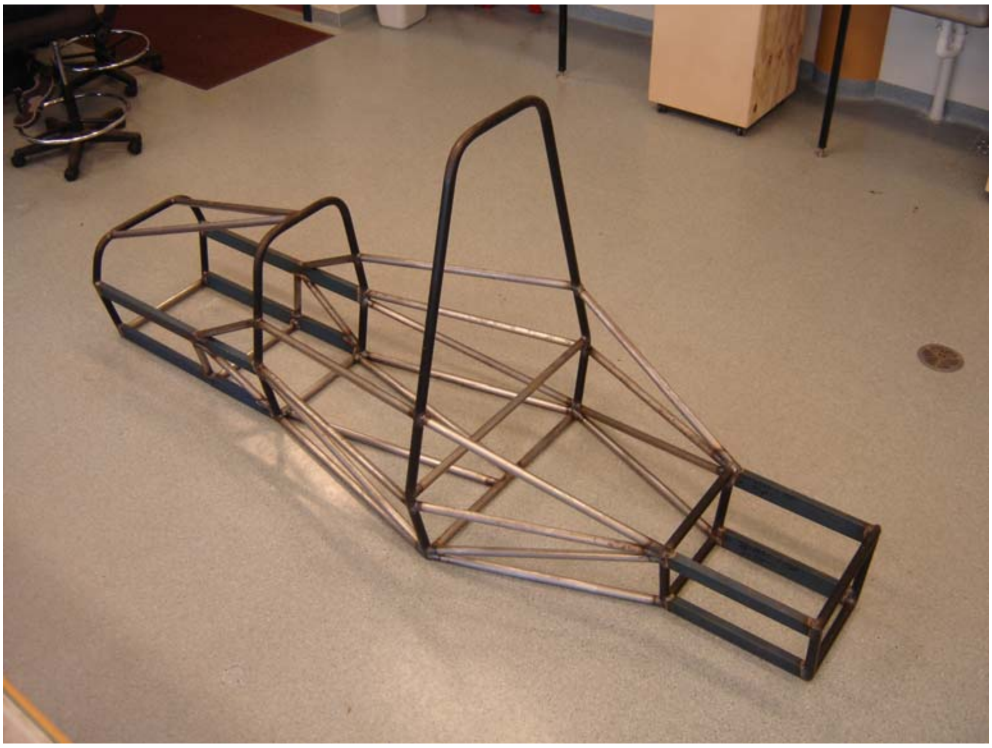


Figure 55: 2005 Frame

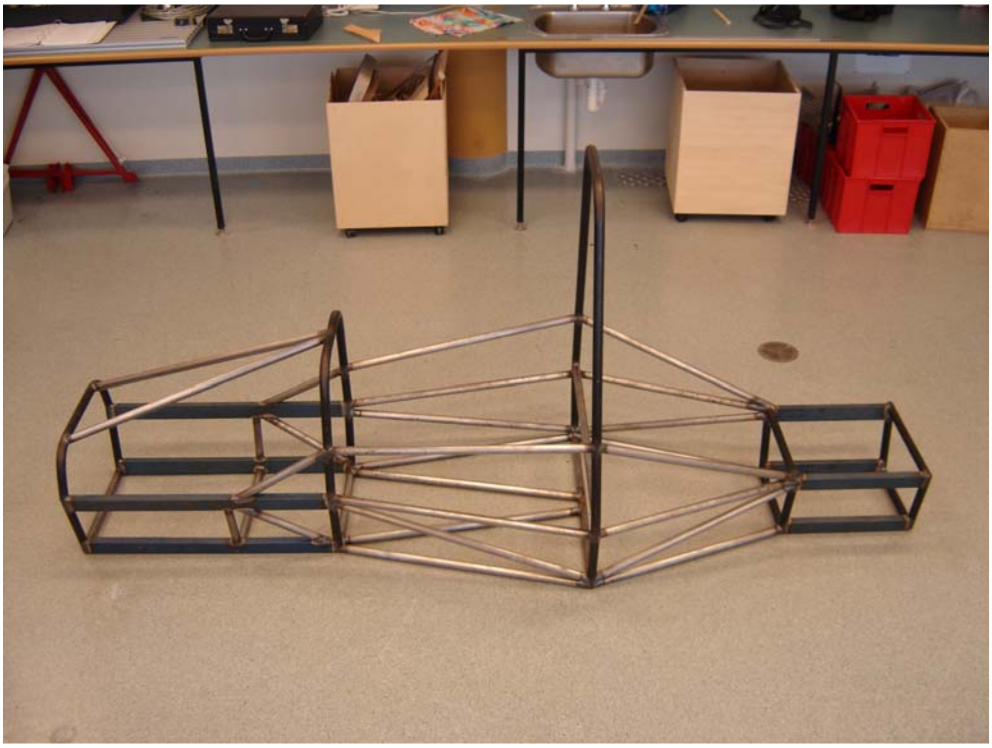


Figure 56: 2005 Frame

Chapter 7

7. The SAE Chassis – Testing & Appraisal

This chapter deals with the actualities of the 2005 Frame (shown below).

Figure 57: 2005 Frame

When time permits, the frame is to be subjected to physical testing in order to validate, or otherwise, the FEA results and to point out any areas where the frame may need strengthening.

This is to be done through the use of a specifically built test rig.

1. Test Rig
2. Testing Procedure

7.1. Test Rig

A fairly simple apparatus and method is proposed for testing the torsional rigidity of this frame – and other future frames. It is best if the actual displacement at small intervals along the frame can be determined. This will show which areas of the frame have the highest deflection ("weakest").

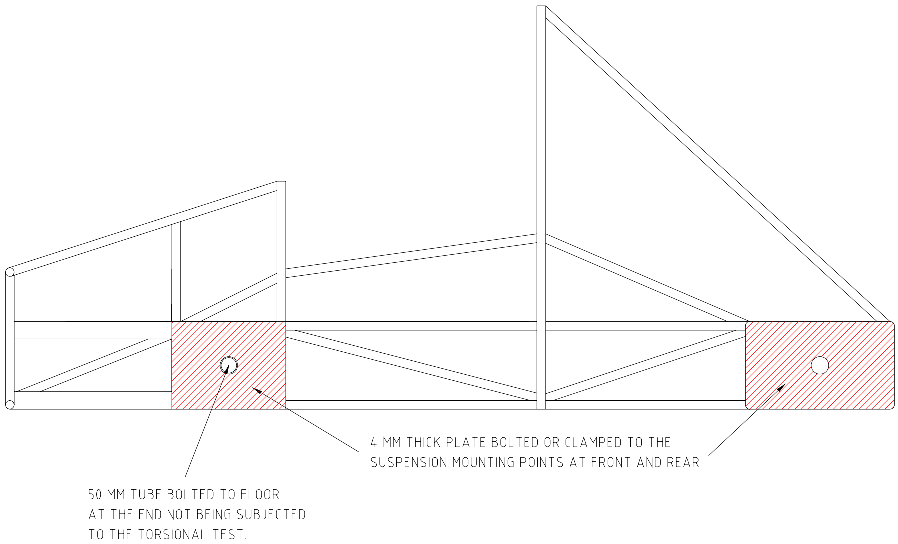


Figure 58: Frame with Test Plates in Position

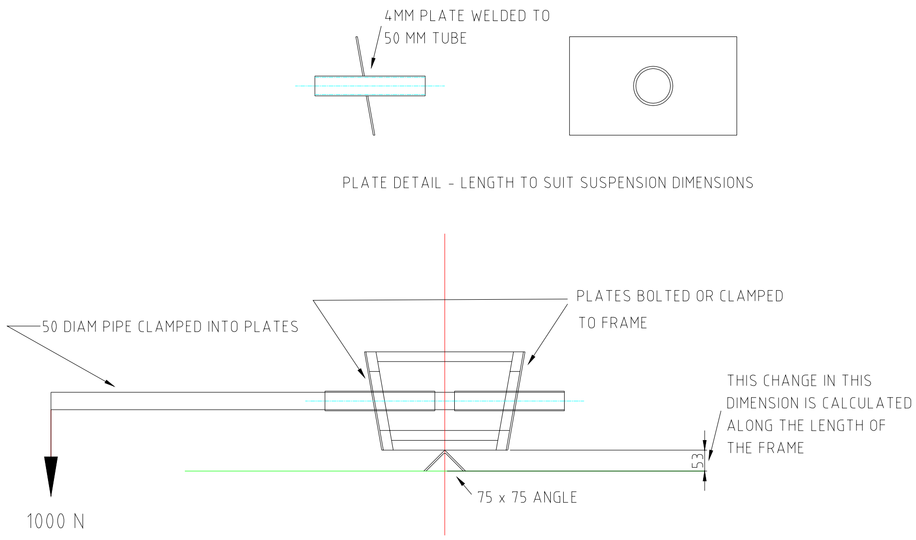


Figure 59: Frame with Moment Applied

The test rig consists of 2 plates for each end of the frame – at the suspension mounting points. The plates have a length of tube, of sufficient size to allow a 50 mm x 3 mm circular tube to pass through with a fairly close fit, welded to them. A length (around 1.2 M) of 50 mm tube is passed through the plate tubes into the frame.

7.2. Testing Procedure

The testing procedure is, obviously, set out in point form:

1. The frame is set up with the plates bolted or clamped to the front and rear suspension mounting positions. In the first instance, the moment is to be applied to the rear of the frame. Accordingly, the front of the frame is to be fixed to the test bed. This is done by passing a length of 50 mm tube through both the front plates and fixing each end of the tube to the test bed. Properly done, this procedure will effectively restrain the front suspension mounts.

2. Next, a length (around 1.2 M) of 50 mm tube is passed through the plate tubes into the rear of the frame. This tube is to be used to apply the load for testing purposes.

3. Prior to any load being applied, the bottom rail of the frame is marked every 200 mm with a felt pen. After this is done, a careful measurement (recorded) is made of the distance from the surface of the test bed to the bottom of the frame at each of the marked (200 mm apart) positions on both sides of the frame.

4. To this is applied a load to generate at moment, e.g. 100 kg weight applied at a distance of 1.0M from the centre of the frame. Of course, this weight must be loaded carefully to avoid any impact loadings which may cause irreparable damage to the frame.

5. Once the load is applied, go and have a cup of coffee / tea. This will give the frame a chance to stabilise and anything weak will become self-evident (i.e. break). It will also put the tester in a fresh state of mind for doing the measurements accurately.

6. Carefully measure (and record) the new distances between the bottom of the frame rail and the test bed on both sides. Calculate the difference in measurements.

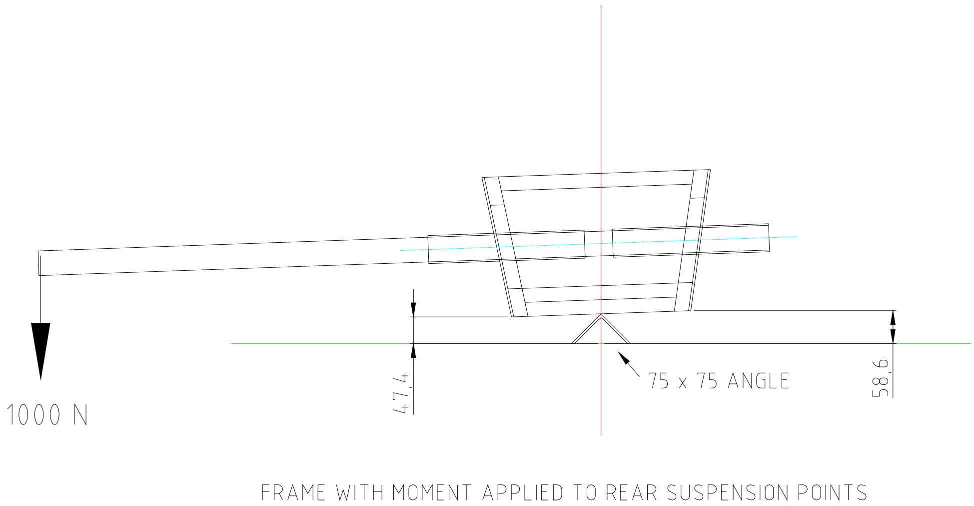


Figure 60: Frame with Load Applied (Exaggerated)

7. The actual overall torsional rigidity of the frame can be determined by the measurement taken at the points directly underneath the load carrying 50 mm tube. An exaggerated example is shown above. Typical realistic figures could be as below:

Applied Moment:	1000N.m
Deflection:	3.0 mm
Centre to edge (radius)	156.0 mm
Rotation:	$\mathrm{Sin}^{-1}(3.0/156.0)$ = 1.102°
Torsional Rigidity:	1000 N.m / 1.102° = **910 N.m / °**

Longitudinal testing

This set up can be used for longitudinal testing, but any frame that displays acceptable torsional rigidity will have sufficient longitudinal stiffness because the triangulation necessary to achieve torsional strength will also give sufficient longitudinal strength.

However, if a measure of longitudinal strength is required, the same test rig can be used. Two transverse lengths of 75 mm x 75 mm angle must be used – one each side of the end that is to be “fixed”. The fixed end is then bolted securely to the test bed. The load is applied to both ends of the 50 mm tube at the other end and deflections again measured. The longitudinal strength is the load per mm deflection. This set up is shown below.

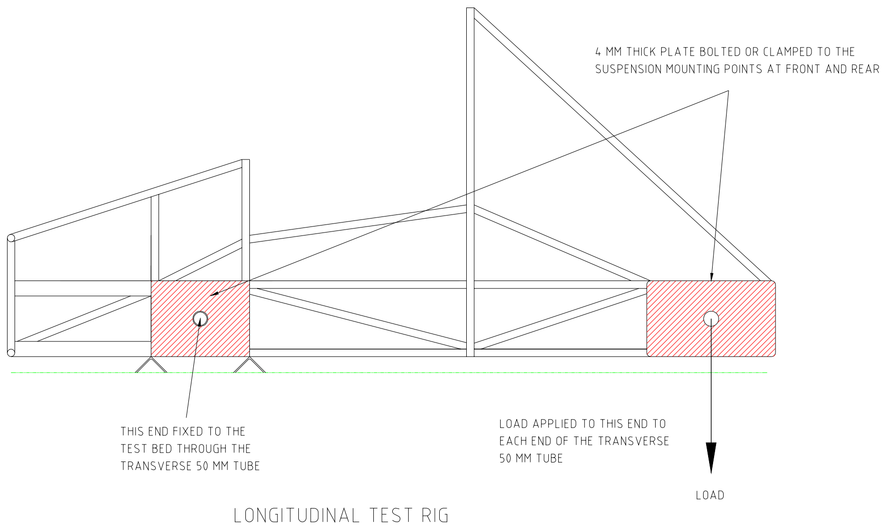


Figure 61: Longitudinal Test Rig

As mentioned earlier, the results of the torsional testing may be directly entered into a graph showing the cumulative frame deflection along the frame. This would give very valuable information on the rigidity of the frame along its length. Knowing where the weakest points are allows worthwhile modifications to be made – prior to the addition of all the other components of the racing car.

7.3. Quality and Appearance

Critical to the success of any product, regardless of the actual worth of the product, is the aesthetic value. For a product to sell, it must ‘look the part’.

For the frame, this means:

1. It must obviously look like what the clientele expects for a space frame.
2. It must possess an elegance of design.
3. It must demonstrate quality in manufacture.
4. It must perform adequately.
5. It must be reasonably priced.

Below is a ProEngineer version of the frame:

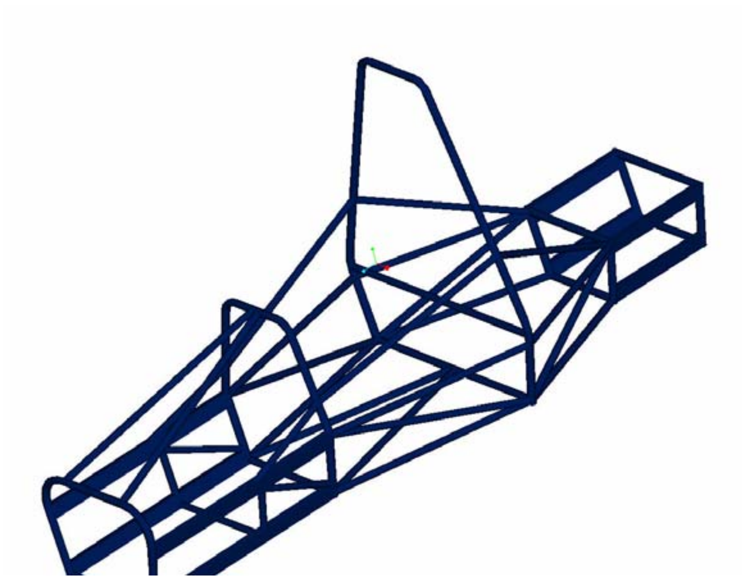


Figure 62: ProEngineer Version of Frame (N.Arvind.Doss)

These considerations shall now be addressed.

1. **It must obviously look like what the clientele expects for a space frame.**

This one is fairly easily addressed: the frame does actually look like a space frame. It has triangulation in all sections, and has fairly well defined load paths integrated into the design.

2. **It must possess an elegance of design.**

The design is not cluttered – a thinking examination of the frame's members and their purposes will quickly show the necessity of each member. The frame is unencumbered with complexity and (in the eye of this beholder, at least) possesses a beauty born of simplicity.

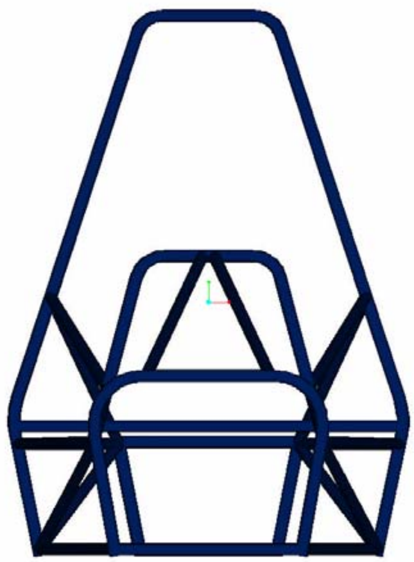


Figure 63: A Study in Frontal Elegance

3. **It must demonstrate quality in manufacture.**

The frame was TIG welded by a welding tradesman – all joints pass a visual inspection with regards to fit and weld quality. All members on each side demonstrate the symmetrical nature of the frame along the longitudinal axis. There are no *delicate adjustments by Thor*.

4. It must perform adequately.

This is a measure of the car's actual road behaviour and cannot be really determined until the car is finished and road tested. However, the torsional rigidity figures from the FEA indicate that the frame would perform more than adequately. The frame from the 2004 USQ SAE racer tested at 214 N.m and is probably best described as a tubular platform chassis rather than a space frame, due to its almost total lack of triangulation. This frame still worked to a reasonable degree on the road. The 2005 frame is also 39 kg as opposed to 49 kg for the 2004 frame.

5. It must be reasonably priced.

Since the frame is fabricated from low carbon steel and is simple in design, it would have to be cheaper than the exotic alternatives. If this frame fulfils the requirements satisfactorily, then any money spent in excess of what this frame costs is wasted.

Materials				
Supplier's Name & Ph#		**Quantity (m)**	**$/unit**	**Cost**
	50x25x2 RHS	5.20	10.00	$52.00
	27x2.7 CHS	8.19	4.00	$32.76
	25.4x1.65 CHS	27.41	3.00	$82.23
	40x40x2 flat	1.20		
Operations Costs		**Quantity**	**$/unit**	
Supplier's Name & Ph#	**Cuts (m)**	4.60	16.00	$73.60
Toowoomba Specialised Welding	**# Drilled Holes**	84.00	0.35	$29.40
Ph 0422 576 460	**# bends**	8.00	0.75	$6.00
	# end preps	138.00	0.75	$103.50
	Welds (m)	14.15	14.00	$198.10
Operations Labour				
Supplier's Name & Ph#		**Hours**	**$/hr**	
Toowoomba Specialised Welding	**Setout**	2.00	35.00	$70.00
Ph 0422 576 460	**Assembly**	10.00	35.00	$350.00
		Total Cost		**$997.59**

Table 11: Costing Data for Frame

The pricing given above is, of course, cost price.

Summary:

This is a simple and elegant frame that should perform more than adequately. It is relatively light and strong, and, being fabricated from low carbon steel, is cheap to produce and, if the need arises, can be easily repaired or modified in any reasonable home workshop. It is the ideal frame for the ‘sportsman’ racer for whom this type of vehicle is intended.

Chapter 8

8. Auxiliary Mounting Brackets

This chapter looks at the design of the auxiliary mounting brackets to be used on the SAE frame:

1. Suspension Brackets
2. Engine Mounts

8.1. Suspension Bracket

To design a suspension bracket, the following must be done:

1. Determine loads
2. Design bracket

8.1.1. Loads

As mentioned earlier in this worthy tome, the static loads are not as critical as the dynamic loads, which are the ones that break and bend things.

To determine these loads, the static weight (with driver) shall be taken as 400 kg, the equivalent being 4000 N (for convenience).

Vertical Bump Force:

The first force to be calculated is the vertical force acting at the wheel due to hitting a bump – this is + / - 4.5 g. Assuming 50 / 50 weight distribution, then there would be 1000 N force downward at each wheel. Taking the acceleration of the wheel to be 4.5g gives:

Since: $F = F_{static} \times \text{acceleration}$

Then: $F = 1000\ N \times (4.5) = 4500\ N$

However, this force is distributed over 4 brackets, so:

Force on each Bracket: $F = 4500\ N / 4 =$ **1125 N$_{(vertical)}$**

Braking Force:

Under this category are actually braking and acceleration, but since braking can be done at a much higher rate of (negative) acceleration over a longer period of time than can the opposite, then calculations for braking only should be satisfactory.

Fenton (1984) gives a design rate for deceleration of –1.5g. The total force needed for a 1.5 g stop is:

Since: $\text{Force} = m \times a$

$\text{Force} = 400\ kg \times 9.81\ m/s^2 \times 1.5$

Then: $\text{Force} = 6000\ N_{\text{(longitudinally)}}$

However, as before, this force is distributed over 4 brackets, but not evenly this time. The front brakes, due to longitudinal load transfer and the resultant increased traction on the front tyres, in a properly set up vehicle will provide approximately two thirds of the stopping power. Hence, the force will be divided as follows:

Force on Front Wheels:	Total Force x 0.667
Force on Front Wheels:	6000 N x 0.6667
Force on Front Wheels:	4000 N
Force on each Front Wheel	2000 N

Force on Each Bracket = 2000N / 4 = 500 N (longitudinally)

Braking Moment:

The other important component of braking forces is the moment that is applied to the suspension during braking. This can be calculated as follows:

Brake Rotor Diameter	0.150 m
Force Applied	500 N
Moment Generated	500 N x 0.150m

Moment Generated = 75 N.m

Cornering Force:

Fenton (1980, p14) gives + / - 1.5 g for cornering forces.

Since:	Force = m x a
	Force = 400 kg x 9.81 m/s^2 x 1.5
Then:	Force = 6000 N $_{(laterally)}$

Force on Front Wheels:	Total Force x 0.5
Force on Front Wheels:	6000 N x 0.5
Force on Front Wheels:	3000 N
Force on each Front Wheel	1500 N

Force on Each Bracket = 1500N / 4 = 375 N$_{(transverse)}$.

Summary of Forces on Bracket

For simplicity, and to allow for the 'worst case scenario', it will be assumed that each bracket on the car will be subject to the same forces.

Bump	**1125 N$_{(vertical)}$**
Braking Force	**500 N $_{(longitudinally)}$**
Braking Moment	**75 N.m**
Cornering	**375 N$_{(transverse)}$**

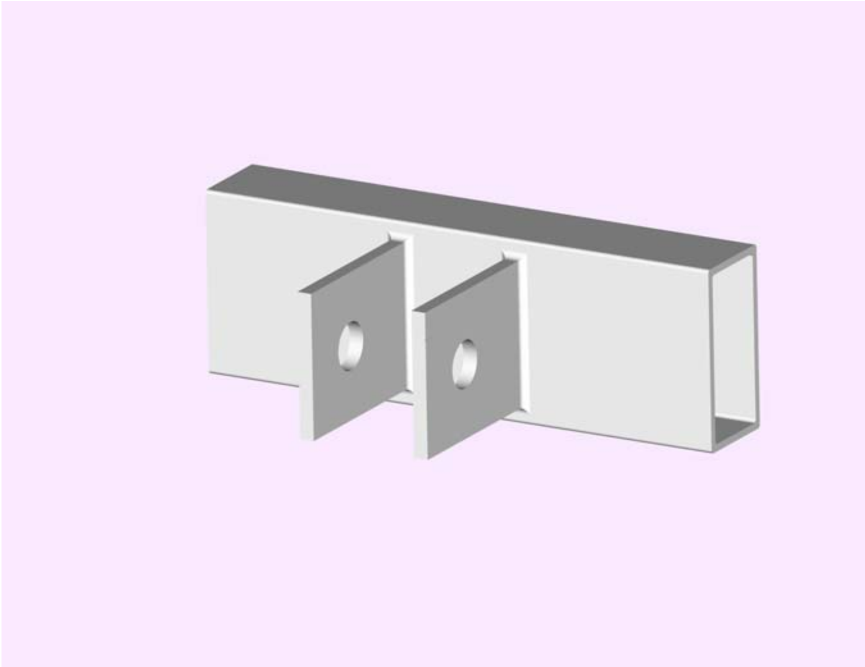


Figure 64: Typical Bracket

The forces shown above are resolved into the forces shown below on a single side of the bracket – the other side of the bracket has a component of the vertical force shown here acting in the opposite direction, with the resultant force being somewhat less than in this case.

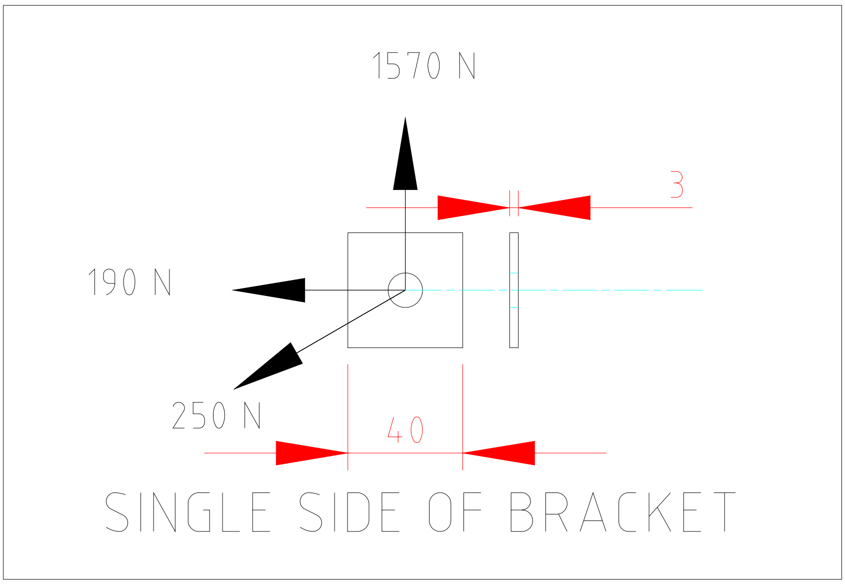


Figure 65: Forces Acting on Bracket

Shear Stress: (Double Shear)

Resultant Force = $\sqrt{(3200^2 + 400^2)}$ = **3225 N**

Bolt: τ_{ave} = F / 2 x A = 3225 / (2 x 113)

Bolt: τ_{ave} **= 14.3 MPa**

Bracket: τ_{ave} = F / 2 x A = 3225 / (2 x 2 x 42)

Bracket: τ_{ave} **= 19.2 MPa**

Tensile Stress:

Tensile stress σ = (F / A) x K (K = stress conc. factor)

σ_{ave} = (3225 / 84) x 2.5 (Pilkey, 1997)

σ_{ave} **= 96 MPa**

Bending Stress:

Bending stress σ = (M y) / I

σ_{ave} = (5000 x 1.5) / 90 MPa

σ_{ave} **= 83.3 MPa**

Steel Properties: (Beer & Johnston, p747)

Structural (ASTM-A36)

Yield	250 MPa (tension)
Yield	145 MPa (shear)
Ultimate	400 MPa (tension)

Summary on Bracket Strength

These brackets should not suffer from shear failure in normal use. The bending and tensile stresses are well within limits – about 70 % of yield strength. The only point to examine further would be if, under a combination of all these forces simultaneously, the brackets may flex enough to cause momentary binding – probably the worst time for such an occurrence.
This factor would need to be figured into the clearances for the brackets / suspension arms.

8.2. Engine Mounts

A weight of 75 kg has been assigned to the engine / gearbox assembly, as fitted to the vehicle (fuel, water, oil).
Keeping the engine in place would not be difficult, even for an engineering student. What would be more important is to keep the engine in place during an impact, a deceleration in the order of –30 g, a figure for which the SAE car, with impact attenuator attached, should achieve. This is a matter of some importance in a mid-engined vehicle, considering the already crowded cockpit and the inability (weight-wise) of the SAE car to have an armoured firewall sufficient to stop / deflect a ballistic engine.

The engine / gearbox will be held rigidly in place with a total of 8 brackets being, as it is, a stressed member of the frame.

Force = mass x acceleration or $F = ma$
$F = 75 \times 9.81 \times 30 = 22{,}100$ N

Force on each bracket = 22100 / 8 = **2760 N**

The brackets will be arranged in a triangulated fashion to facilitate the use of the engine as part of the chassis, so it would be reasonable to assume that the brackets are subject to either compression or tension loads only.

It is suggested that 12 mm x 5 mm low carbon steel be used for the engine mounts, with a maximum length of 200 mm.

Tensile stress σ = F / A = 2760 / 60 = 46 MPa

Tensile stress σ = 46 MPa

Shear stress 8 mm bolt = F / A = 2760 / 50.27 = 55 MPa

Shear stress 8 mm bolt = 55 MPa

Shear stress in bracket = F / A = 2760 / 120 = 23 MPa

Shear stress in bracket = 23 MPa

Buckling $P_{cr} = (\pi^2 E I) / Le^2 = (\pi^2 \times 200e^9 \times 1.25e^{-7}) / 0.1^2$

Buckling P_{cr} = 24 700 N

Summary on Engine Mounts

The suggested mount should not fail in shear, though it would be advisable to use a high strength (and ductile) bolt. With a good quality bolt, the engine mounts would probably withstand decelerations in the order of 100 g.

8.3. Summary

From the above calculations, it can be seen that the brackets chosen are quite capable of withstanding the loads imposed. These brackets are all fabricated from plain low carbon steel and, accordingly, are cheap to make and reliable in use.

Chapter 9

9. Conclusions

The success or otherwise of the SAE Race Car is largely dependent on the frame - as with any building, the whole is only as strong as its footings / foundations. The best suspension in the world cannot function properly on a too-flexible chassis.

It would be pertinent here to discuss how well the frame met the original objectives of:

1. **Mass less than 40 kg**
   The finished frame weighed in at 38 kg

2. **Torsional rigidity in excess of 300 N.m/°**
   The FEA analysis gave a figure of 485 N.m/° (to be physically verified) but this is without the engine, which is intended to be a structural component.

3. **Ease of manufacture**
   The prototype was cut and tacked together by 2 final year students, the finish welded by a tradesman welder.

4. **Ease of maintenance**
   Plain low carbon steel – can be welded at home with safety; easily painted and repaired.

5. **Low cost**
   Cost price under $1,000.

Areas for Improvement

It would also be timely to discuss possible future areas for improvement.

The basic frame dimensions need to be determined and agreed on by all early in the Project – then there must be no arguments over wheelbase, track etc afterwards.

A more in-depth Finite Element Analysis to look at each member in turn, and determine the minimum size (and mass) for each tube would be appropriate, as would the subsequent timely ordering of the various sizes calculated above (many would not be locally available 'off the shelf').
This process would, of course, be more expensive but could pare off a few more kilograms and add more strength.

It would be important for future frame builders to source their fabricator early and get to know them, ask pertinent questions and *listen and learn*.
Though the FEA above may show a certain size tube (e.g. 0.7 mm wall thickness) will be sufficient, the fabricator (who, in true USQ Motorsport Club tradition, will most likely be doing the job for nothing) may be a little less enthusiastic about welding it.

It is absolutely imperative that the design process includes input from the fabricator. There is no point in designing something that cannot be made.

Summary

An interesting project that has achieved the objectives originally set out – light in weight and torsionally rigid (to be verified), easy to build and maintain as well as having the added advantage of low cost.

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Appendix A - Project Specification

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR: **Anthony Michael O'NEILL**
TOPIC: Chassis Design for SAE Racer
SUPERVISOR: Chris Snook

PROJECT AIM: This project aims to design a rigid and lightweight chassis for the SAE Racer.

PROGRAM: **Issue A, 15th March 2005.**

1. Research SAE rules to determine safety and design requirements.
2. Review and critique designs used by other teams.
3. Determination of layout, suspension type and dimensions in consultation with Team.
4. Selection of materials to be used.
5. Determination of work processes (including quality control) for construction of frame.
6. Determination of imposed loads – suspension, engine, torsional etc.
7. Research and design a suitable mounting bracket for suspension, engine etc.
8. Testing of joint strength of selected material in configurations used in chassis.
9. Determination of optimal frame design (with regards to weight, deflection and torsional stiffness) by Finite Element Analysis.
10. Liaise with Team and Faculty Workshop in the construction of the frame.
11. Testing (and modification, if necessary) of frame to ensure compliance with design and safety objectives.

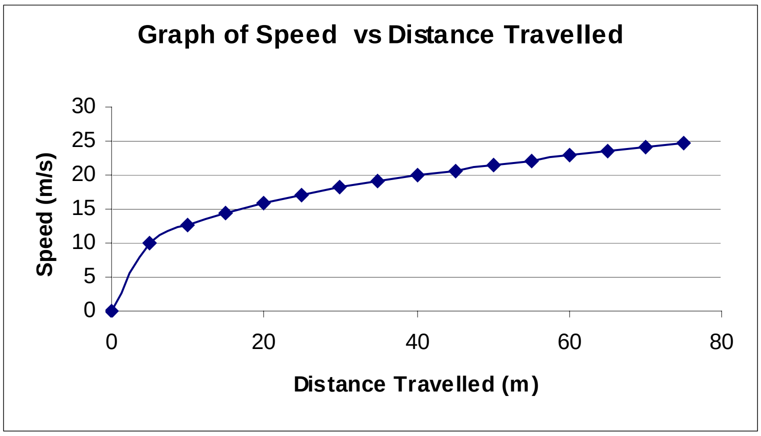
AGREED:____________________(Student) ______________________(Supervisor)

(date) ____/____/____

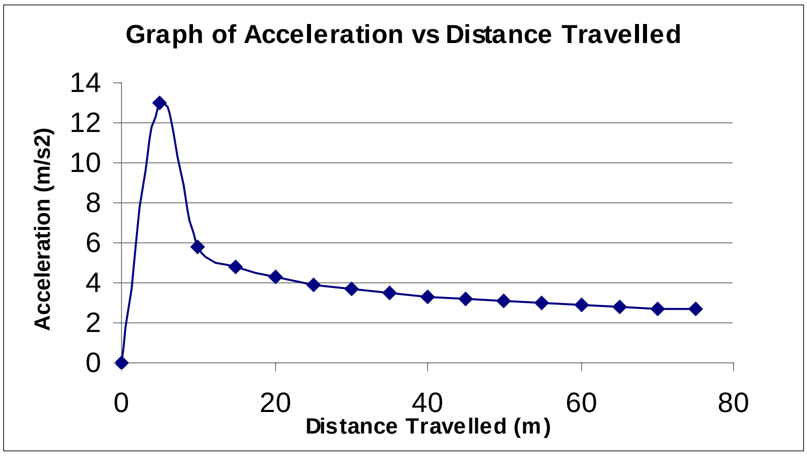
Appendix 1: Performance Graphs - 300 kg & 50hp

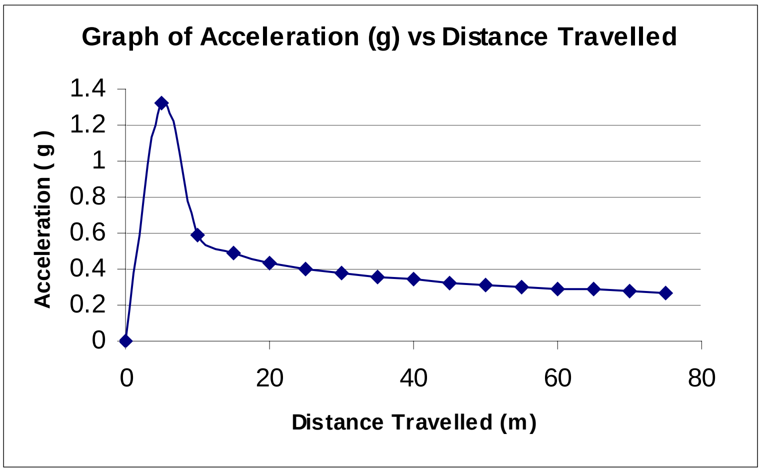
Weight	300	kg							
Horsepower	50	hp		speed	speed				
d	n	k	time	mph	m/s	dt	dv	a = dv/dt	a (g)
0	0	0	0	0	0	0	0	0	0
5	80.467	0.034	0.77	22.29	9.97	0.77	9.97	13.02	1.33
10	40.234	0.136	1.22	28.09	12.56	0.45	2.59	5.76	0.59
15	26.822	0.306	1.59	32.15	14.37	0.38	1.82	4.82	0.49
20	20.117	0.544	1.93	35.38	15.82	0.34	1.45	4.30	0.44
25	16.093	0.849	2.24	38.12	17.04	0.31	1.22	3.95	0.40
30	13.411	1.223	2.53	40.51	18.11	0.29	1.07	3.69	0.38
35	11.495	1.665	2.80	42.64	19.06	0.27	0.95	3.49	0.36
40	10.058	2.175	3.06	44.58	19.93	0.26	0.87	3.33	0.34
45	8.941	2.752	3.31	46.37	20.73	0.25	0.80	3.19	0.33
50	8.047	3.398	3.55	48.02	21.47	0.24	0.74	3.07	0.31
55	7.315	4.111	3.79	49.58	22.16	0.23	0.69	2.97	0.30
60	6.706	4.893	4.01	51.03	22.81	0.23	0.65	2.88	0.29
65	6.190	5.742	4.23	52.41	23.43	0.22	0.62	2.81	0.29
70	5.748	6.659	4.45	53.72	24.02	0.21	0.59	2.73	0.28
75	5.364	7.645	4.66	54.97	24.58	0.21	0.56	2.67	0.27

Table of Time, Speed and Acceleration for 75 metre Standing Start



Performance Graphs for 300 kg & 50 hp (cont)

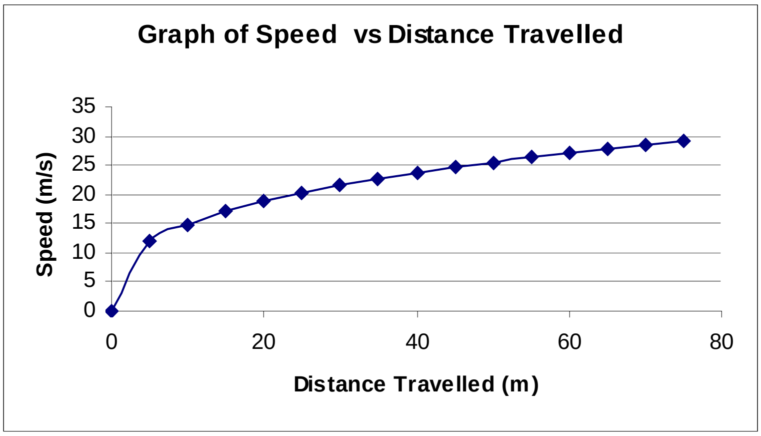




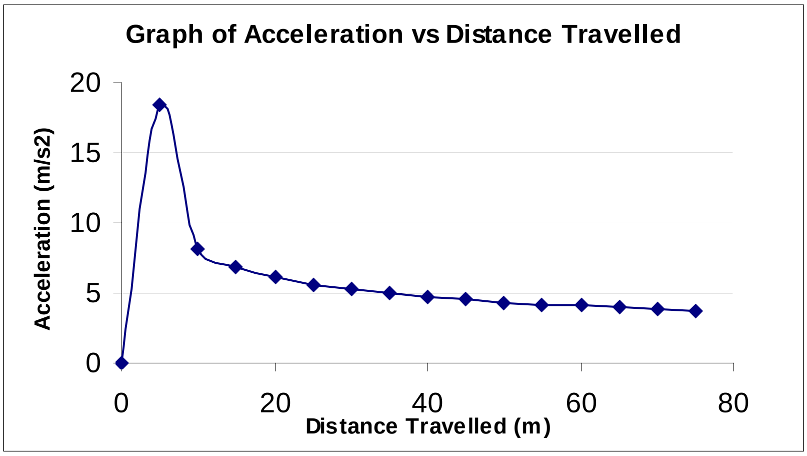
Appendix 2: Performance Graphs for 250 kg & 70 hp

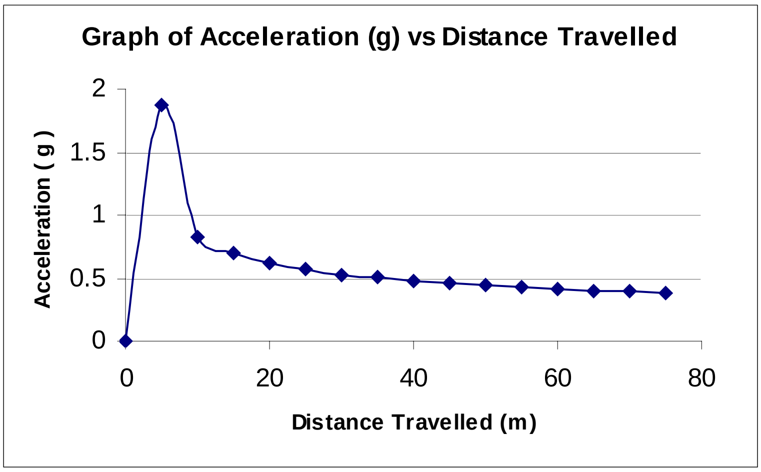
Weight	250	kg							
Horsepower	70	hp		speed	speed				
d	n	k	time	mph	m/s	dt	dv	a = dv/dt	a (g)
0	0	0	0	0	0	0	0	0	0
5	80.467	0.034	0.64	26.50	11.85	0.64	11.85	18.40	1.88
10	40.234	0.136	1.02	33.39	14.93	0.38	3.08	8.14	0.83
15	26.822	0.306	1.34	38.22	17.09	0.32	2.16	6.81	0.69
20	20.117	0.544	1.62	42.07	18.80	0.28	1.72	6.07	0.62
25	16.093	0.849	1.88	45.31	20.26	0.26	1.45	5.58	0.57
30	13.411	1.223	2.13	48.15	21.53	0.24	1.27	5.22	0.53
35	11.495	1.665	2.36	50.69	22.66	0.23	1.13	4.93	0.50
40	10.058	2.175	2.58	53.00	23.69	0.22	1.03	4.70	0.48
45	8.941	2.752	2.79	55.12	24.64	0.21	0.95	4.51	0.46
50	8.047	3.398	2.99	57.09	25.52	0.20	0.88	4.34	0.44
55	7.315	4.111	3.18	58.93	26.35	0.20	0.82	4.20	0.43
60	6.706	4.893	3.37	60.67	27.12	0.19	0.78	4.08	0.42
65	6.190	5.742	3.56	62.31	27.85	0.18	0.73	3.96	0.40
70	5.748	6.659	3.74	63.87	28.55	0.18	0.70	3.86	0.39
75	5.364	7.645	3.92	65.35	29.22	0.18	0.66	3.77	0.38

Table of Time, Speed and Acceleration for 75 metre Standing Start



Performance Graphs for 250 kg and 70 hp (cont)





Appendix 3: Costing Data & Calc's for SAE Frame

Chassis Costing Quantities

Section Name	RHS 50x25 x2.0 # off	Circular 27x2.7 # off	Circular 25.4x1.65 # off	length metres	# cuts	length of cut	# drilled holes 1"	# bends	# end preps	# welds	weld lengths
Main Hoop		1		2.70	2	0.03		4			
Front Cockpit Hoop		1		1.00	2	0.03		2	2	2	0.09
		2		0.30	4	0.03			4		
			1	0.44			2		2	2	0.09
Front Bulkhead Hoop		1		0.64	2	0.03		2	2	2	0.09
		2		0.30	4	0.03			4		
			1	0.44			2		2	2	0.09
Main Hoop Internal			1	0.68			2		2	2	0.09
			1	0.84			2		2	2	0.09
			1	0.75			2		2	2	0.09
			4	0.44			8		8	4	0.105
										4	0.125
Main Hoop Braces			2	1.40			4		4	4	0.105
Front Hoop Braces			2	0.84			4		4	4	0.105
Front Side Braces			2	0.49	4	0.07			4	4	0.15
			2	0.35	4	0.05			4	4	0.12
			2	0.34	2	0.07	2		4	2	0.15
										2	0.12
Front Centre Struts			2	0.16	4	0.03			4	4	0.09
			2	0.34	2	0.03	2		4	4	0.09
			1	0.20			2		2	2	0.09
			1	0.44	2	0.03			2	2	0.09
Cockpit Floor			2	0.79			4		4	4	0.09
			2	0.76			4		4	4	0.09
Cockpit Sides			2	0.76			4		4	4	0.12
			2	0.77			4		4	4	0.09
			2	0.75			4		4	4	0.09
Engine Compartment			2	0.65			4		4	4	0.105
			2	0.64			4		4	4	0.105
			2	0.67			4		4	2	0.105
			2	0.70			4		4	2	0.15
Rear Drive Hoops		2		0.43	4	0.12			4	2	0.12
		2		0.34	4	0.12			4	2	0.12
		4		0.28	8	0.12			8	4	0.12
Drive Side Bracing			4	0.26	8	0.075			8	8	0.105
Longitudinal Bars	4			0.85	8	0.05	8		8	8	0.15
	4			0.45	8	0.05	8		8	8	0.15
Suspension Brackets 40x40x2 flat	24			0.05	48	0.04	24		24	24	0.04

Appendix 3 (continued)

Materials

Supplier's Name & Ph#		Quantity (m)	$/unit	Cost
	50x25x2 RHS	5.20	10.00	$52.00
	27x2.7 CHS	8.19	4.00	$32.76
	25.4x1.65 CHS	27.41	3.00	$82.23
	40x40x2 flat	1.20		

Operations Costs

Supplier's Name & Ph#		Quantity	$/unit	
	Cuts (m)	4.60	16.00	$73.60
Toowoomba Specialised Welding	# Drilled Holes	84.00	0.35	$29.40
Ph 0422 576 460	# bends	8.00	0.75	$6.00
	# end preps	138.00	0.75	$103.50
	Welds (m)	14.15	14.00	$198.10

Operations Labour

Supplier's Name & Ph#		Hours	$/hr	
Toowoomba Specialised Welding	Setout	2.00	35.00	$70.00
Ph 0422 576 460	Assembly	10.00	35.00	$350.00

Total Cost **$997.59**

Appendix 4: Formula SAE-A

University of Southern Queensland

USQ Motorsport

Car 13

Design Report

1. Overview

Due to the great luck from last year, the car's number (13) has been retained. The Formula SAE vehicle is a product of the USQ Motorsport Club, members of which have designed and built the car throughout 2005. In this, the Team has had much assistance from the USQ Mechanical Workshop in the production / fabrication of many of the components. The USQ Motorsport Club is based at the USQ, a regional University in a city with only a small industrial base – a factor which severely limits sponsorship opportunities. Since this was only the second year in the competition for the USQ, the design philosophy was to improve the car by making small improvements in all components. The basic aim was to reduce weight wherever possible whilst increasing the useable power of the engine through inlet and exhaust tuning. The USQ Motorsport Team is still small (but growing) at this stage, so mutual help and encouragement has been a major factor in its success. With time, the membership base will grow in numbers and experience, as more first and second year engineering students become involved, and more improvements will flow from this.

2. Chassis

The vehicle uses a simple space frame chassis which was designed using ProEngineer solid modelling software and full-scale timber mock-ups for verification of ergonomic dimensions. The frame consists of 31.75mm x 2.1mm ERW tubular steel with a yield strength of 250 MPa and was fabricated using the TIG – GTAW process.
The chassis design has been analysed using non-destructive testing and Finite Element Analysis (FEA) using the ANSYS package. The non-destructive testing consisted of torsional and bending tests – and these confirmed that the FEA model was giving a reasonable approximation of the true stresses in the chassis.

3. Suspension and Steering

The 2005 USQ Motorsport car uses unequal length A-arms with an in-board coil over damper unit which is actuated by a pushrod. The lengths and angles are designed to give minimal change, both front and rear, to the track and roll centres, in order to keep the handling predictable and safe.

The car's uprights and hubs are student designed and have been manufactured from steel. The uprights have a 'built-in' 5.7° of castor and a 3.5° king pin inclination which reduces the offset to 20 mm. This is intended to give good feedback to the driver whilst minimising steering kickback.

The steering system used is a modified rack and pinion assembly from a 1983 Honda Civic sedan. The geometry is 100% Ackermann with 18° steering angle. The tie rods are parallel with the upper A-arms to minimise bump steer. The turning radius is 5m for 210° steering wheel rotation.

4. Brakes

The braking system on the car features production items from a 1994 Yamaha YZF600 motorcycle. This consists of a cross-drilled rotor and four-pot calliper acting on each of the front wheels and a single rotor and two-pot calliper acting on the rear axle. The system is operated by two identical ¾ inch master cylinders that provide pressure to the callipers through a pedal with mechanically adjustable bias control for brake force distribution.

5. Drivetrain

The car's drivetrain consists of a chain and sprocket drive to a solid rear axle, which transmits torque to the wheels through equal length constant velocity (CV) shafts.

The final drive ratio was increased by using a 13-tooth front sprocket and a custom-made 60-tooth rear sprocket. This will increase the final gear ratio to 4.61 and improve acceleration and torque.

The centre rear axle was specifically designed to minimise weight whilst maintaining reliability, and for that reason was manufactured from hollow bar steel. The decision not to integrate a differential into the car may have some negative effects on the cornering performance, however a narrower rear track and stiff spring rates will minimise these effects.

The advantages of a solid rear axle are:

1. Cost effective solution offering excellent strength and high reliability.
2. Can offer superior traction in straight-line acceleration.
3. Can further reduce weight by using a single brake assembly.
4. Requires no maintenance.

Also, to minimise weight, small CV assemblies from a Suzuki SS80V were sourced. These CV joints also feature an internal spline, which reduces the manufacturing costs of the CV shafts and rear axle as only mating external splines are required to be machined into these shafts and are much less demanding.

6. Engine

USQ Motorsport's 2005 entry is powered by a 600 cc Yamaha motorbike engine. This engine was sourced from a 1994 model YZF600, and is naturally aspirated with a 16-valve, DOHC, inline 4-cylinder configuration.

Inlet
The inlet manifold is student-designed and built, and features a streamlined design with four long intake runners meeting at a single plenum. The runners are constructed from 25 mm bore mild steel, and are each 275 mm long. This critical diameter is designed to provide optimum flow velocity (and cylinder filling) in the desired rpm range, while the gently swept, pipe-bent runners provide a direct path to each inlet, and are tuned to take advantage of reflected pressure waves for increased intake charge.

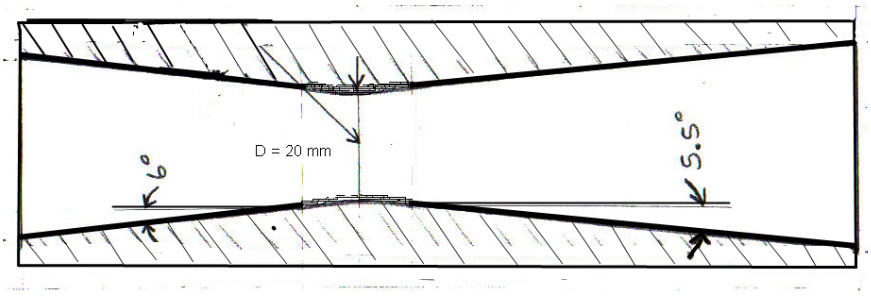
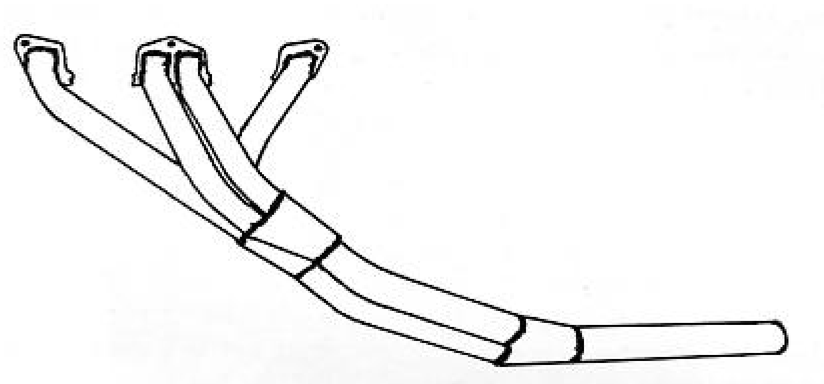


Figure 1: Intake Restrictor

The plenum offers interchangeable spacers to provide a variety of plenum volumes, from 60 cc to 500 cc. A dividing plate is incorporated into the plenum to group companion cylinders (cylinders 2 & 3, 1 & 4) and assist with prevention of inter-cylinder charge robbery as well as maximise throttle response. The mandatory inlet restrictor features a converging-diverging conical shape, machined from a single piece of aluminium, tapering to the specified 20 mm circular diameter between the nozzle and diffuser (Figure 1).

Fuel Delivery
The engine is naturally aspirated and introduces fuel through a carburettor. USQ Motorsport is currently developing an electronic fuel injection (EFI) system to achieve improved power characteristics and better fuel economy. The design of the inlet manifold makes provisions for both carburetion and injection systems, with a throttle body adaptor plate, and mountings for injectors, utilising a multi-point configuration.

Exhaust
Custom headers are student-designed and built to increase power by taking advantage of wave scavenging characteristics. Free-flow joiners, linking the four primary pipes and the secondary pair, have been fabricated using 180° mandrel bends, cut and welded. A ‘4-into-2-into-1’ configuration offers superior low- and mid-range torque and a more even delivery of power, and utilises 750 mm-long primaries and 500 mm-long secondaries, each tuned for specific rpm bands. Companion cylinder grouping is again utilised, similar in principle to the intake system. An ‘off-the-shelf’ muffler, rated to meet FSAE-A noise level requirements, is used.



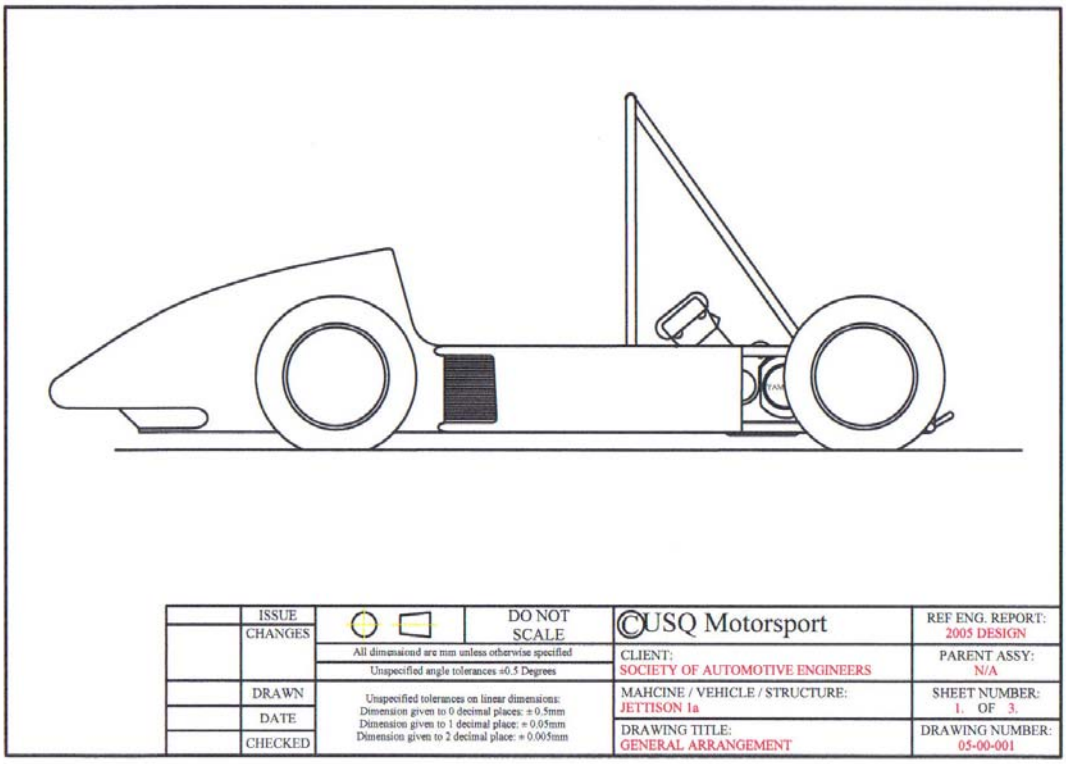
7. Bodywork

The external body will consist of 4 sections:

1: Side pods. These will be moulded from 'E' glass cloth and vinyl ester resin (probably Dow Chemicals "Derakane" 510A)
2: Scuttle panel. Forms the instrument panel and runs forward to the steering gear.
3: Nose. Runs from the scuttle forward.
Other bodywork;
Radiator inlet ramps- run from the front wishbone attachments to the inner end of the radiator core between the upper and lower longitudinal rails.
Cooling pipe cover (across cockpit floor) covers the pipe connecting the radiators together.
Anti-fouling plates to protect drivers' feet from the front suspension elements.
"Stressed skin" shear panels in cockpit sides.
Upper wishbone arrestor plates.

8. Cockpit Design

The vehicle’s cockpit has been designed fully in accordance with the anthropometric requirements with excellent adjustability and simplicity. The pedal box is constructed for good adjustability as the seat is in a fixed position.


ISSUE
CHANGES
DO NOT SCALE
©USQ Motorsport
REF ENG. REPORT:
2005 DESIGN
All dimensiond are mm unless otherwise specified
Unspecified angle tolerances ±0.5 Degrees
CLIENT:
SOCIETY OF AUTOMOTIVE ENGINEERS
PARENT ASSY:
N/A
DRAWN
DATE
CHECKED
Unspecified tolerances on linear dimensions:
Dimension given to 0 decimal places: ± 0.5mm
Dimension given to 1 decimal place: ± 0.05mm
Dimension given to 2 decimal place: ± 0.005mm
MAHCINE / VEHICLE / STRUCTURE:
JETTISON 1a
SHEET NUMBER:
1. OF 3.
DRAWING TITLE:
GENERAL ARRANGEMENT
DRAWING NUMBER:
05-00-001

Appendix 5: Formula SAE-AUS Rules - Frame

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 must 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 (4) 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

3.2.2.1 Wheels

The wheels of the car must be 203.2 mm (8.0 inches) or more in diameter.

Any wheel mounting system that uses a single retaining nut must incorporate a device to retain the nut and the wheel in the event that the nut loosens.

3.2.2.2 Tires

Vehicles may have two types of tires as follows:

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

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

> **(i)** The tread pattern or grooves were molded in by the tire manufacturer, or were cut by the tire manufacturer or his appointed agent. Any grooves that have been cut must have documentary proof that it was done in accordance with these rules.
>
> **(ii)** There is a minimum tread depth of 2.4 mms (3/32 inch).
>
> **Note:** Hand cutting, grooving or modification of the tires by the teams is specifically prohibited.

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

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 system free play is limited to 7 degrees total measured at the steering wheel. Rear wheel steering is permitted only if mechanical stops limit the turn angle of the rear wheels to ± 3 degrees from the straight ahead position.

The steering wheel must be mechanically connected to the front wheels, i.e. "steer-by-wire" of the front wheels 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 must have two independent hydraulic circuits such that in the case of a leak or failure at any point in the system, effective braking power is maintained on at least two wheels. Each hydraulic circuit must have its own fluid reserve, either by the use of separate reservoirs or by the use of a dammed, OEM-style reservoir. "Brake-by-wire" systems are prohibited. A single brake acting on a limited-slip differential is acceptable. The braking system must be protected with scatter shields from failure of the drive train or from minor collisions. Unarmored plastic brake lines are prohibited.

3.2.5.1 Brake Over Travel Switch

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

3.2.5.2 Brake Light

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

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 Impact Attenuator, and side protection.

3.3.1 Definitions

The following definitions apply throughout the Rules document:

(A) Main Hoop - Rollover protection (roll bar) located alongside or just behind the driver's torso.

(B) Front Hoop - Rollover protection (roll bar) located above the driver's legs, in proximity to the steering wheel.

(C) Frame Member - A minimum representative single piece of uncut, continuous tubing.

(D) Frame - The Frame is the fabricated structural assembly that supports all functional vehicle systems. This assembly may be a single welded structure, multiple welded structures or a combination of composite and welded structures.

(E) Safety Structure – The Safety Structure is comprised of the following Frame components: 1) Main Hoop, 2) Front Hoop, 3) Side Impact Structure, 4) Roll Hoop Braces, 5) Front Bulkhead and 6) all Frame Members, guides and supports that transfer load from the Driver's Restraint System into items 1 through 5.

(F) Major Structure of the Frame – The portion of the Frame that lies within the envelope defined by the Safety Structure. The upper portion of the Main Hoop and the Main Hoop braces are not included in defining this envelope.

(G) Front Bulkhead – A planar structure that defines the forward plane of the Major Structure of the Frame and functions to protect the driver's feet.

(H) Impact Attenuator – A deformable, energy absorbing device located forward of the Front Bulkhead.

3.3.2 Safety Structure Equivalency

The use of 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, is allowed, provided they have been judged as such by a technical review. Approval of alternative material or tubing sizes will be based upon the engineering judgment and experience of the chief technical inspector or his appointee.

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 must 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) or 25.0 mm x 2.50 mm metric
Side Impact Protection, Front, Bulkhead, Roll Hoop Bracing & Safety Harness Attachment	25.4 mm (1.0 inch) x 1.65 mm (0.065 inch) or 25.0 mm x 1.75 mm metric

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.

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

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 must 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) must 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 must 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. The Frame must include both a Main Hoop and a Front Hoop as shown in Figure 1.

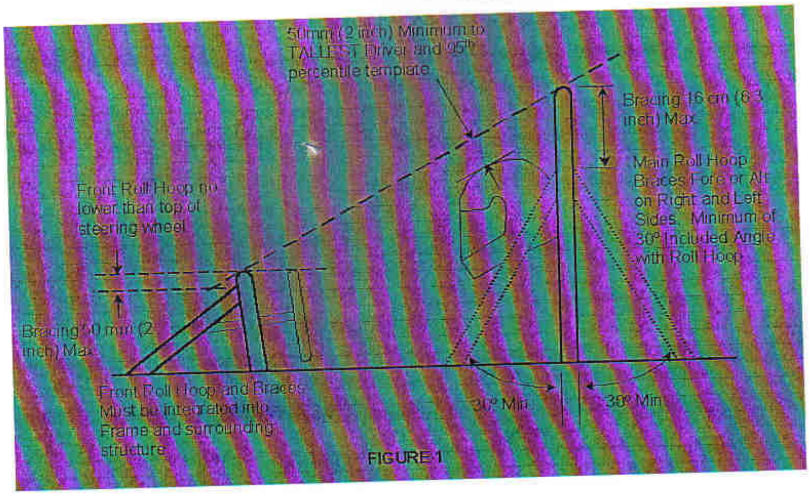


FIGURE 1

3.3.4.1 Main and Front Hoops – General Requirements

(A) When seated normally and restrained by the Driver's Restraint System, 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 used to represent the 95th percentile male is made to the following dimensions: -A circle of diameter 200 mm (7.87 inch) will represent the hips and buttocks. -A circle of diameter 200 mm (7.87 inch) will represent the shoulder/cervical region. -A circle of diameter 300 mm (11.81 inch) will represent the head (with helmet). -A straight line measuring 600 mm (23.62 inch) will connect the centers of the two 200 mm circles. -A straight line measuring 150 mm (5.9 inch) will connect centers of the upper 200 mm circle and the 300 mm head circle.

With the seat adjusted to the rearmost position, the bottom 200 mm circle will be placed in the seat, and the middle 200 mm 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) The minimum radius of any bend, measured at the tube centerline, must be at least three times the tube outside diameter. Bends must be smooth and continuous with no evidence of crimping or wall failure.

(C) The Main Hoop and Front Hoop must be securely integrated into the Safety Structure using gussets and/or tube triangulation.

(D) 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 a single piece of uncut, continuous, closed section steel tubing per Section 3.3.3.

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

(C) The Main Hoop must extend from the lowest Frame Member on one side of the Frame, up, over and down the lowest Frame Member on the other side of the Frame.

(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 Frame must be within 10 degrees of the vertical.

(E) In the front view of the vehicle, the vertical members of the Main Hoop must be at least 380 mm (15 inch) apart (inside dimension) at the location where the Main Hoop is attached to the Major Structure of the Frame.

monocoque

(F) On vehicles where the Safety Structure is not made from steel tubes, the Main Hoop must be continuous and extend down to the bottom of the Frame. The Main Hoop must be securely attached to the monocoque structure using 8 mm Grade 8.8 (5/16 in Grade 5) bolts. Mounting plates welded to the Roll Hoop shall be at least 2.0 mm (0.080 inch) thick steel. Steel backup plates of equal thickness must be installed on the opposing side of the monocoque structure such that there is no evidence of crushing of the core. The attachment of the Main Hoop to the monocoque structure requires an approved Safety Structure Equivalency Form per Section 3.3.2. The form must demonstrate that the design is equivalent to a welded Frame and must include justification for the number and placement of the bolts.

3.3.4.3 Front Hoop

(A) The Front Hoop must be constructed of closed section metal tubing per Section 3.3.3.

(B) The use of composite materials is prohibited for the Front Hoop.

(C) The Front Hoop must extend from the lowest Frame Member on one side of the Frame, up, over and down to the lowest Frame Member on the other side of the Frame. With proper gusseting and/or triangulation, it is permissible to fabricate the Front Hoop from more than one piece of tubing.

(D) The top-most surface of the Front Hoop must be no lower than the top of the steering wheel in any angular position.

3.3.5 Roll Hoop Bracing

3.3.5.1 Main Hoop Bracing

(A) Main Hoop braces must be constructed of closed section steel tubing per Section 3.3.3.

(B) The use of aluminum alloys, titanium alloys or composite materials is prohibited for the Main Hoop braces.

(C) The Main Hoop must be supported by two braces extending in the forward or rearward direction on both the left and right sides of the Main Hoop. In the side view of the Frame, the Main Hoop and the Main Hoop braces must not lie on the same side of the vertical line through the top of the Main Hoop, i.e. if the Main Hoop leans forward, the braces must be forward of the Main Hoop, and if the Main Hoop leans rearward, the braces must be rearward of the Main Hoop.

(D) The Main Hoop braces must be attached as near as possible to the top of the Main Hoop but not more than 160 mm (6.3 in) below the top-most surface of the Main Hoop. The included angle formed by the Main Hoop and the Main Hoop braces must be at least 30 degrees.

(E) Main Hoop braces must be straight, i.e. without any bends.

3.3.5.2 Front Hoop Bracing

(A) Front Hoop braces must be constructed of material per Section 3.3.3.

(B) The Front Hoop must be supported by two braces extending in the forward direction on both the left and right sides of the Front Hoop.

(C) The Front Hoop braces must be constructed such that they protect the driver's legs and should extend to the structure in front of the driver's feet.

(D) The Front Hoop braces must be attached as near as possible to the top of the Front Hoop but not more than 50.8 mm (2 in) below the top-most surface of the Front Hoop.

(E) Monocoque construction used as Front Hoop bracing requires an approved Safety Structure Equivalency Form per Section 3.3.2.

3.3.5.3 Other Bracing Requirements

(A) Where the braces are not welded to steel Frame Members, the braces must be securely attached to the Frame using 8 mm Grade 8.8 (5/16 in Grade 5), or stronger, bolts. Mounting plates welded to the Roll Hoop braces must be at least 2.0 mm (0.080 in) thick steel.

(B) Where Main Hoop braces are attached to a monocoque structure, backup plates, equivalent to the mounting plates, must be installed on the opposing side of the monocoque structure such that there is no evidence of crushing of the core. The attachment of the Main Hoop braces to the monocoque structure requires an approved Safety Structure Equivalency Form per Section 3.3.2. The form must demonstrate that the design is equivalent to a welded Frame and must include justification for the number and placement of the bolts.

3.3.5.4 Other Side Tube Requirements

If there is a roll hoop brace or other frame tube alongside the driver, at the height of the neck of any of the team's drivers, a metal tube or piece of sheet metal must be firmly attached to the Frame to prevent the drivers' shoulders from passing under the roll hoop brace or frame tube, and his/her neck contacting this brace or tube.

3.3.5.5 Removable Roll Hoop Bracing

(A) Roll Hoop bracing may be removable. Any non-permanent joint must be either a double-lug joint as shown in figures 2 and 3, or a sleeved butt joint as shown in Figure 4. The threaded fasteners used to secure non-permanent joints are considered critical fasteners and must comply with paragraph 3.7.2.2. No spherical rod ends are allowed.

(B) For double-lug joints, each lug must 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 double lug joints must include a capping arrangement (figure 2) and/or a doubler (figure 3), fabricated of at least 1.65 mm (0.065 in) steel. If a doubler is used, it must extend at least 120 degrees around the frame member. The pin or bolt must be 10 mm Grade 9.8 (3/8 in. Grade 8) minimum. The attachment holes in the lugs and in the attached bracing must be a close fit with the pin or bolt.

(C) For sleeved butt joints, the sleeve must have a minimum length of 76 mm (3 inch), 38 mm (1.5 inch) either side of the joint, and be a close-fit around the base tubes. The wall thickness of the sleeve must be at least that of the base tubes. The bolts must be 6 mm Grade 9.8 (1/4 inch Grade 8) minimum. The holes in the sleeves and tubes must be a close-fit with the bolts.

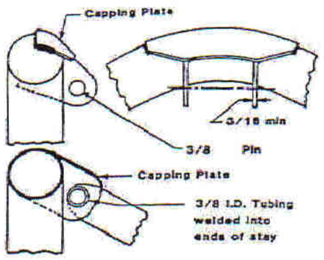


Figure 2

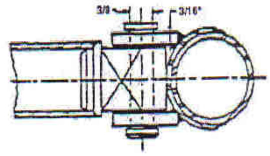


Figure 3

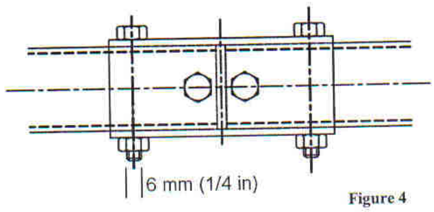


Figure 4

REMOVABLE ROLL BAR BRACES ATTACHMENT DETAILS (FIGURES 2, 3 & 4)

3.3.6 Frontal Impact Protection – Drivers

In order to provide protection from frontal impact, the driver's feet must be contained within the Major Structure of the Frame. While the driver's feet are touching the pedals, no part of the driver's feet can extend above and/or outside of the Major Structure of the Frame. Forward of the Front Bulkhead must be an energy-absorbing Impact Attenuator.

3.3.6.1 Bulkhead

(A) The Front Bulkhead must be constructed of closed section tubing per Section 3.3.3.

(B) The Front Bulkhead must be located forward of all non-crushable objects, e.g. batteries, master cylinders.

(C) The Front Bulkhead must be located such that the soles of the driver's feet, when touching but not applying the pedals, are rearward of the bulkhead plane. (This plane it defined by the forward-most surface of the tubing.) Adjustable pedals must be in the forward most position.

(D) The Front Bulkhead must be securely integrated into the Frame. As a minimum, the Front Bulkhead must be supported by Frame Members on both the left and right sides of the Frame within 50.8 m (2 in) of the top-most surface of the Front Bulkhead.

(E) Monocoque Frames require an approved Safety Equivalency Form, per Section 3.3.2. The form must demonstrate that the design is equivalent to a welded Frame in terms of energy dissipation, yield and ultimate strengths in bending, buckling and tension.

3.3.6.2 Impact Attenuator

(A) The Impact Attenuator must be capable of decelerating the car within an acceptable limit.

(B) The Impact Attenuator must be installed forward of the Front Bulkhead.

(C) The Impact Attenuator must be at least 150 mm (5.9 in) long, with it's length oriented along the fore/aft axis of the Frame.

(D) The Impact Attenuator must be at least 100 mm (3.9 in) high and 200 mm (7.8 in) wide for a minimum distance of 150 mm (5.9 in) forward of the Front Bulkhead.

(E) The Impact Attenuator must be attached securely and directly to the Front Bulkhead such that it cannot penetrate the Front Bulkhead in the event of an impact. The use of adhesive tape and/or Dzus type fasteners is prohibited. The Impact Attenuator shall not be attached to the vehicle by being part of non-structural bodywork.

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 impact attenuator zone.

3.3.7 Frontal Impact Protection – Others

People must 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, must have forward facing radii of at least 38 mm (1.5 inches). This minimum radius must 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. The Side Impact Protection must meet the requirements listed below.

3.3.8.1 Tube Frames

The Side Impact Protection must be comprised of at least three (3) tubular members located on each side of the driver while seated in the normal driving position, as shown in Figure 5. The three (3) required tubular members must be constructed of material per Section 3.3.3. The locations for the three (3) required tubular members are as follows:

(A) The upper Side Impact Protection member must connect the Main Hoop and the Front Hoop at a height between 300 mm (11.8 inch) and 350 mm (13.8 inch) above the ground with a 77kg (170 pound) driver seated in the normal driving position. The upper frame rail may be used as this member if it meets the height, diameter and thickness requirements.

(B) The lower Side Impact Protection member must connect the bottom of the Main Hoop and the bottom of the Front Hoop. The lower frame rail/frame member may be this member if it meets the diameter and wall thickness requirements.

(C) The diagonal Side Impact Protection member must connect the upper and lower Side Impact Protection members forward of the Main Hoop and rearward of the Front Hoop.

With proper gusseting and/or triangulation, it is permissible to fabricate the Side Impact Protection members from more than one piece of tubing.

Alternative geometry that does not comply with the minimum requirements given above requires an approved 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

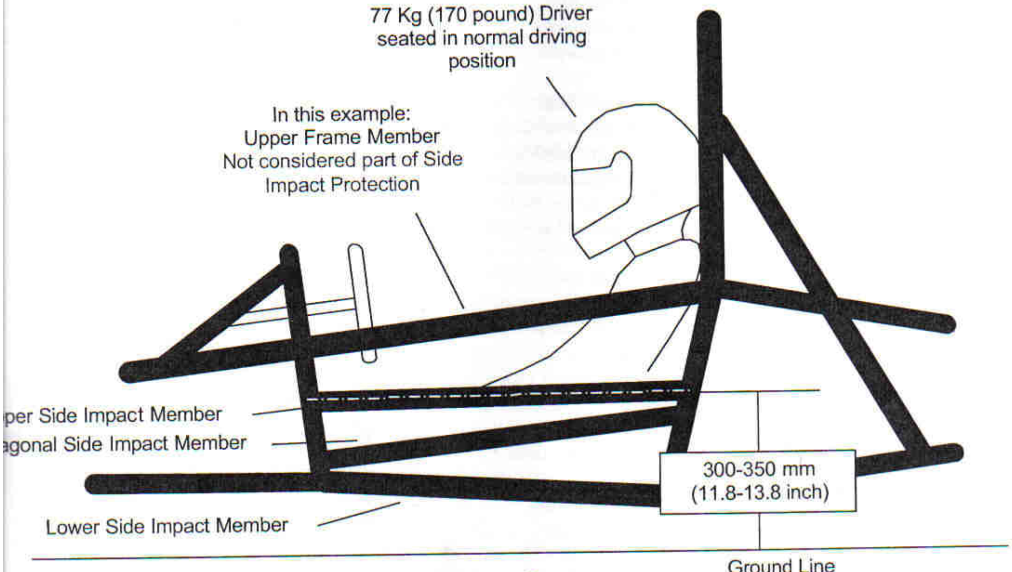


Figure 5

3.4 SAFETY - DRIVER RULES

3.4.1 Driver's Restraint System

All drivers must use either a five or six-point restraint harness meeting the following specifications. Arm restraints are also required. The restraint system installation is subject to approval of the Chief Technical Inspector. The restraint system must be worn as tightly as possible at all times.

(A) 5 Point System
A five-point system consists of a 76 mm (3 inch) wide lap belt, approximately 76 mm (3 inch) wide shoulder harness straps and a single, approximately 51 mm (2 inch) wide anti-submarine strap.

The single anti-submarine strap of the five-point system must have a metal-to-metal connection with the single release common to the lap belt and shoulder harness.

(B) 6 Point System
A six point system consists of a 76 mm (3 inch) wide lap belt, approximately 76 mm (3 inch) wide shoulder harness straps and two, approximately 51 mm (2 inch) wide leg or anti-submarine strap.

The double leg straps of the six-point system may be attached to the Safety Structure, or be attached to the lap belt so that the driver sits on them, passing them up between his or her legs and attaching to the single release common to the lap belt and shoulder harness. The leg straps may also be secured at a point common with the lap belt attachment to the Safety Structure, passing them under the driver and up between his or her legs to the harness release.

(C) Material Requirements
The material of all straps must be Nylon or Dacron polyester and in new or perfect condition. There must be a single release common to the lap belt and shoulder harness using a metal-to-metal quick-release type latch. All driver restraint systems must meet either SFI Specification 16.1, or FIA specification 8853/98. The belts must bear the appropriate dated labels, and be no more than five years old. It is recommended that driver restraint systems be replaced every three years.

(D) Belt and Strap Mounting
The lap belt, shoulder harness and anti-submarine strap(s) must be securely mounted to the Safety Structure. Such structure and any guide or support for the belts must meet the minimum requirements of 3.3.3. Bolting through aluminum floor closeout panels, etc. is not permitted.

The attachment of the Driver's Restraint System to a monocoque structure requires an approved Safety Structure Equivalency Form per Section 3.3.2.

(E) Belt Position Requirements
The lap belt must pass around the pelvic area below the Anterior Superior Iliac Spines (the hip bones) (Figure 6a). Under no condition may the lap belt be worn over the area of the intestines or abdomen. The lap belts should come through the seat at the bottom of the sides of the seat to maximize the wrap of the pelvic surface and continue in a straight line to the anchorage point. The centerline of the lap belt at the seat bottom should be approximately 76 mm (3 inch) forward of the seat back to seat bottom junction (see Recommended Location in Figure 6). The lap belts should not

be routed over the sides of the seat. The seat must be rolled or grommeted to prevent chafing of the belts.

(F) Shoulder Harness
The shoulder harness must be the over-the shoulder type. It must be mounted behind the driver and above a line drawn downward from the shoulder point at an angle of 40 degrees with the horizontal to minimize spine compression injuries under high "g" deceleration. Only separate shoulder straps are permitted (i.e. "Y"-type shoulder straps are not allowed). "H"-type configuration is allowed. It is mandatory that the shoulder harness, where it passes over the shoulders, be 76 mm (3 inch) wide, except as noted below. The shoulder harness straps must be threaded through the three bar adjusters in accordance with manufacturers instructions.

When the HANS device is used by the driver, FIA certified 51 mm (2 inch) wide shoulder harnesses are allowed. Should a driver, at anytime not utilize the HANS device, then 76 mm (3 inch) wide shoulder harnesses are required.

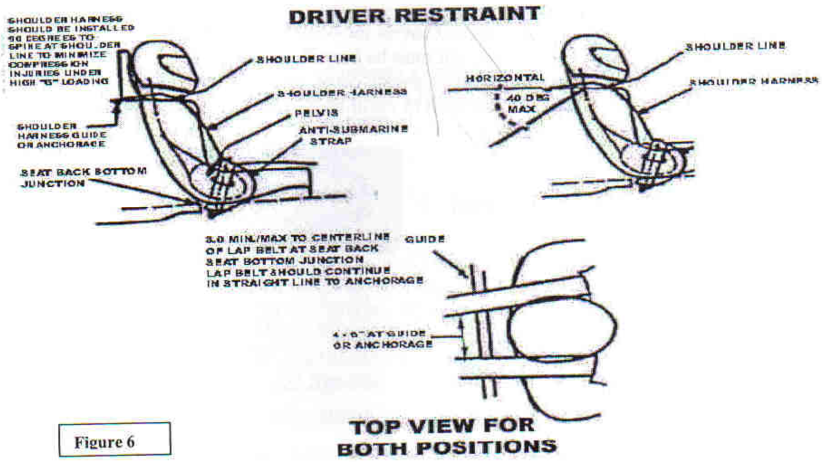


Figure 6

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2005 Formula SAE® Rules

Appendix 6: Data from ProEngineer

(courtesy of N Arvind Doss)

VOLUME = 4.5944516e+06 MM^3
SURFACE AREA = 4.7314475e+06 MM^2
DENSITY = 7.8000000e-09 TONNE / MM^3
MASS = 3.5836722e-02 TONNE

CENTER OF GRAVITY with respect to _ARV1 coordinate frame:
X Y Z 5.0703858e-01 1.1318377e+02 1.2415124e+03 MM

INERTIA with respect to _ARV1 coordinate frame: (TONNE * MM^2)

INERTIA TENSOR:
Ixx Ixy Ixz 7.6338034e+04 -1.6107711e-01 -2.1317352e+01
Iyx Iyy Iyz -1.6107711e-01 7.5621078e+04 -5.1333637e+03
Izx Izy Izz -2.1317352e+01 -5.1333637e+03 4.3569448e+03

INERTIA at CENTER OF GRAVITY with respect to _ARV1 coordinate frame:
(TONNE * MM^2)

INERTIA TENSOR:
Ixx Ixy Ixz 2.0641908e+04 1.8955399e+00 1.2416733e+00
Iyx Iyy Iyz 1.8955399e+00 2.0384032e+04 -9.7621725e+01
Izx Izy Izz 1.2416733e+00 -9.7621725e+01 3.8978469e+03

PRINCIPAL MOMENTS OF INERTIA: (TONNE * MM^2)
I1 I2 I3 3.8972688e+03 2.0384596e+04 2.0641922e+04

ROTATION MATRIX from _ARV1 orientation to PRINCIPAL AXES:

-0.00007	-0.00734	-0.99997
0.00592	0.99996	-0.00734
0.99998	-0.00592	-0.00003

ROTATION ANGLES from _ARV1 orientation to PRINCIPAL AXES (degrees):
angles about x y z 90.245 -89.580 90.584

RADII OF GYRATION with respect to PRINCIPAL AXES:
R1 R2 R3 3.2977372e+02 7.5420067e+02 7.5894609e+02 MM