REV SAE Front Drive

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Project Summary

The aim of this project is to design a front wheel drive system for a formula SAE electric car. In doing this, this project aims to promote electric cars, further the technology involved and to eventually produce a car for competition. This project was achieved firstly by identifying possible design paths based on the given constraints, these were then evaluated and an in wheel design consisting of a motor in series with a gearbox was chosen. This was then modelled in SolidWorks and tested in COSMOS. After the computer modelling was finalised construction of the proposed design commenced.

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Dear Professor Dell

I am pleased to submit this thesis entitled REV SAE Front Drive, as part of the requirements for the degree of Bachelor of Engineering.

Yours Sincerely

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Acknowledgements

I would like to sincerely thank my supervisors, previous and current students, work shop staff and my parents all of whom this project would not have been possible without.

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1. Introduction

This project aims to design a front wheel drive system for a formula electric SAE car and in doing this forms part of a larger project which ultimately aims to produce a formula electric SAE car for competition. This project is a continuation of the 2009 Formula SAE electric car, which in its first stage was designed to be a rear wheel drive car with an electric motor powering each rear wheel, this car is shown in illustration 1 below.

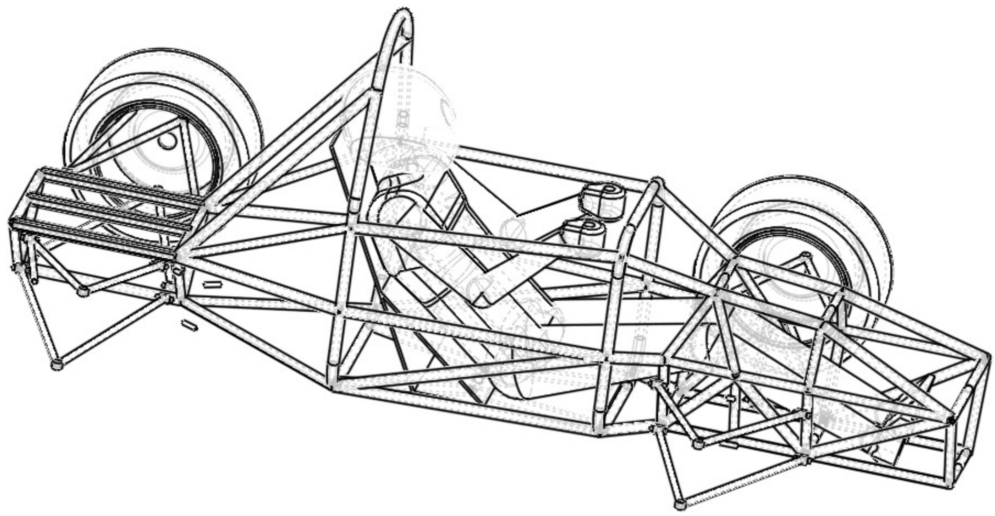


Illustration 1: Existing Formula SAE car

The original aim of this project was to implement the front wheel drive for this car, to make it a true four wheel drive car. As this project progressed it became clear that this project would not be incorporated into the original car, but a new car to be designed specifically for electric starting in the second semester of 2010. The main constants on this project originated from the original aims of producing the front wheel drive for the existing Formula SAE electric car, the existing upright is shown in illustration 2. From this existing set up come a lot of the constraints of the project, to make the project more achievable by a single thesis it was decided to design the front wheel drive system around the existing suspension design. This thesis is part of the REV project whose aims are to promote and further the technology involved in electric cars and runs in conjunction with other projects with the ultimate aim to produce a Formula SAE electric car for competition.

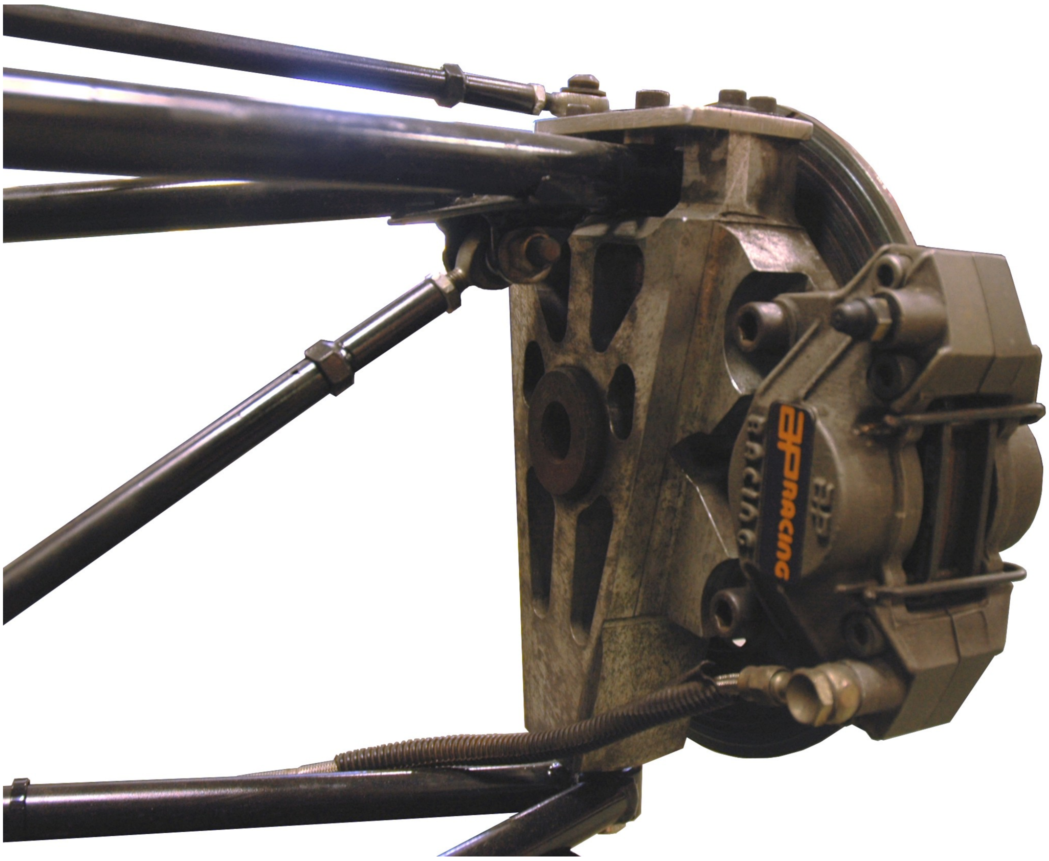


Illustration 2: Existing upright

1.1 Literature Review

In the initial emergence of motor cars there were three rival design types, internal combustion (I.C.), electric and steam. Due to the vast availability of fossil fuels and their excellent energy storage capacity I.C. cars prevailed and consequently almost all the cars on the road today are still powered by fossil fuels. The improvement in battery technology and the concern about our growing carbon emissions and the effect that this has on the environment has caused electric cars to once again be a viable option. Despite these concerns about environment impacts of I.C. cars, electric vehicle technology has been very slow to emerge on the market and in Australia there are currently no production dedicated electric cars available for sale, (Braunl 2010), only series hybrid that cannot run on dedicated electric and normal cars that have been converted to electric. In foreign markets there are a few low production electric cars as well as proposed production electric cars by many major manufacturers with Mitsubishi being the first to release one with its MiEV in Japan (Kim 2010). As well as this there are companies researching into in wheel motor technology. There are three main approaches to electric vehicle design, a single centrally mounted engine, chassis mounted motors to power two or more wheels and wheel mounted motors to power two or more wheels. The current state of the art of these individual approaches is investigated below.

1.1.1 Centrally mounted motor

Electric vehicle technology has been very slow to emerge on the market despite the concerns about the environmental impact of carbon emission from fossil fuels and the rising cost of fuel supplies. This lack of supply has caused the development of commercial and at home conversions of regular cars to electric (Smith 2010), (BEV 2010). These electric car conversions generally involve removing the original petrol engine and in its place putting an electric motor. This is on the lower scale of technological sophistication and due to cost of development doesn't take advantage of most of the benefits of electric vehicle technology.

There are a few companies offering low production electric cars that generally fill a niche market as opposed to a full sized commuter, these are generally sports cars or micro cars. The Tesla Roadster is one example of this and it consists of an electric motor powering the rear wheels (Tesla 2010). These concepts do not satisfy mainstream demands by either being too expensive or lacking features that people expect in modern cars.

There are many cars with a centrally mounted motor proposed for production, a summary with the type of car, release date and drive design is provided below.

Tesla model s. (Tesla 2010)

- Sedan.
- Planned for release in 2012 in USA.
- Single electric motor.

Mitsubishi MiEV. (Fallah 2010)

- Hatch back.
- Planned for release in 2010.
- Single electric motor.

Nissan leaf. (Nissan 2010)

- Four door hatch back.
- Planned for release in 2012.
- Single electric motor.

Chevrolet volt. (cnet 2008)

- Sedan.
- Planned for release in 2012 in Australia.
- Electric motor and petrol generator.

Ford Focus BEV. (Ford 2009)

- Hatch back.
- Planned for release in 2011 in the USA.
- Single electric motor.

This design is the most simple because it involves the continuation of conventional design practises but does not take advantage of the scalability of electric motor which can provide benefits having motors to power two or more wheels.

1.1.2 Chassis mounted multiple motors.

The next progression from a centralised motor is to have a chassis mounted motor to power two or more wheels. In doing this the centralised drive train is no longer required reducing the weight and number of moving parts which all cause an increase in over all efficiency and reliability. Having a motor on two or more wheels also adds performance and safety benefits by being able to individually control the speed of each wheel. As well as this there is also a lot more freedom in car design, no longer being constrained by the drive train and a large motor. This technology despite being fairly simple is not included in any production cars, but there are a few concept cars that propose to use this design, these are given below.

Lotus Evora. (Lotus 2010)

- Sports car.
- Concept.
- Electric motor on each rear wheel, as part of the sprung mass and a petrol generator.

Mercedes Gullwing. (SAE International 2009)

- Sports car.
- Concept.
- Motor on each wheel, as part of the sprung mass.

1.1.3 In wheel motor.

The next progression is to mount the motor in the wheel becoming part of the unsprung mass and removing even more moving parts. This concept has been around for as long as the motor car itself with the first patent being lodged in 1884 (Adams 1884) and with Porsche producing the Lohner-Porsche which had hub motors in all four wheels as well as batteries and a petrol generator (Porsche 2009) and is already seen in bicycle hub motors. This technology has the greatest potential benefit of all three that have been proposed in this literature review, these benefits are described below.

- The reduction in drive line components this includes the elimination of the following components, the central drive shaft, up to 8 C.V. joint, clutch, gearbox, differentials which can greatly decrease the weight because all these components have associated energy losses, due to this the efficiency has the potential to be a lot higher.
- Each motor can provide regenerative braking which recoups energy that would other-wise been lost to heat, again increasing the efficiency and decreasing the weight due to the need for less powerful mechanical brakes.

- A motor on each wheel allows for independent control of each wheel which has great benefits for handling, safety and performance.
- Getting rid of the centralised motor and just having them located in each wheel allows for a greater freedom of car body design.

There are many companies involved in the research and development of in wheel motor technology which aim to capitalise on the benefits of redesigning cars to suit the use of electric motors as opposed to working with the structure imposed by using a centralised motor. Below is given the current state of the art of this technology.

e-Traction

e-Traction produce hub motors that consist of a large motor directly powering the wheels, shown in illustration 3. The hub motors have a continuous power of between 6 and 32 kW, a weight of 85kg and a continuous torque of between 150-200Nm. As well they are currently producing electric cars and buses based upon these concepts to test the design, although none of these are in production, *(e-Traction 2010)* .

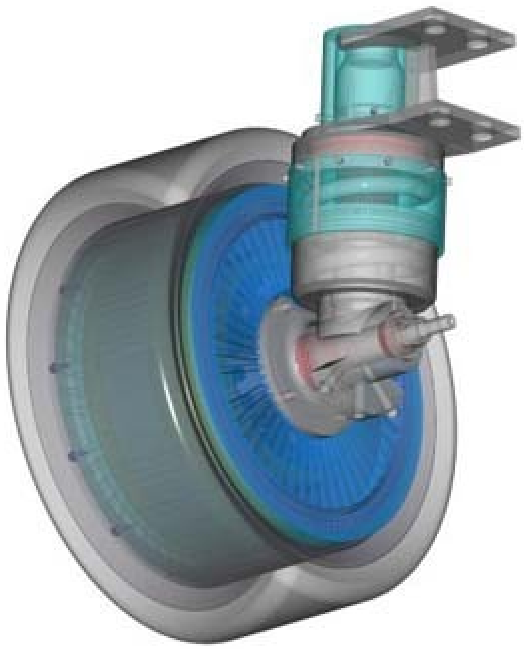


Illustration 3: e-Traction in Wheel motor

Michelin active wheel

The Michelin active wheel uses a small motor connected to the unsprung assembly via a gear box to power the wheel, shown in illustration 4. It has an unsprung mass similar to a normal car of 35kg and it produces 30kW. It is planned to be used in Venturi Voltage the production for this is in 2012, (Evans 2008).

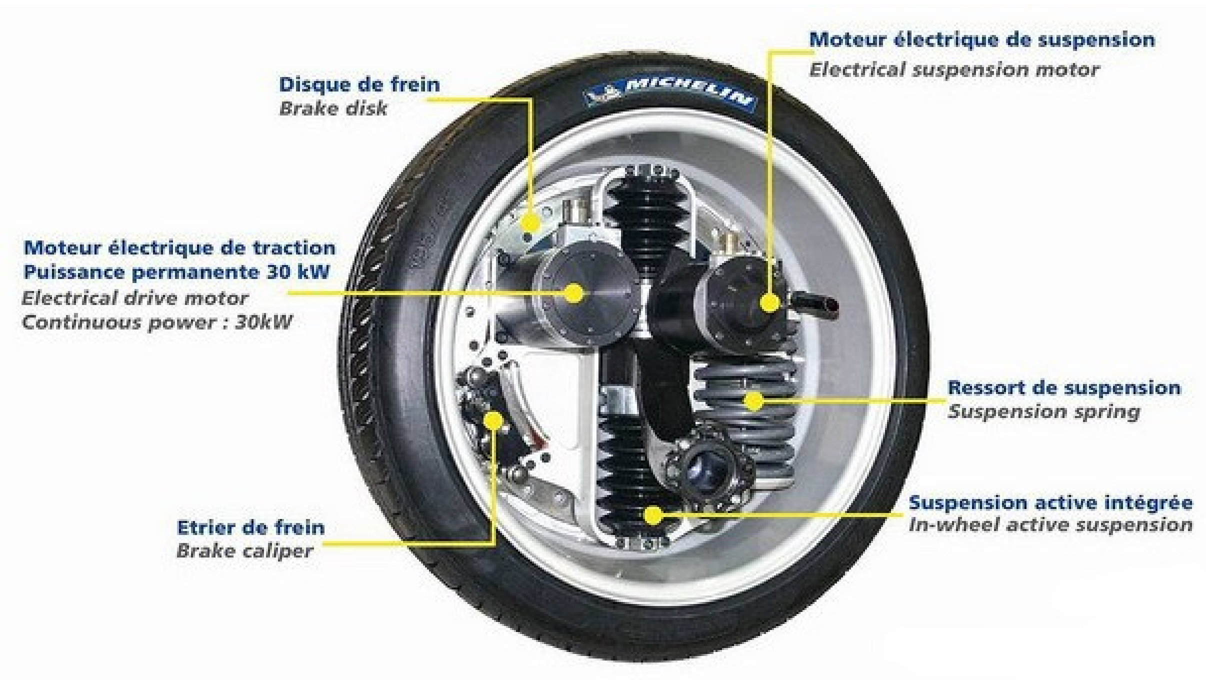


Illustration 4: Michelin active wheel concept

Hi-Pa Drive

Proten Electric produce an in wheel concept that uses a large electric motor incorporated into the wheel assembly to directly power the wheel, shown in illustration 5. The design has a peak power of 40-120kW, a weight of 18-25kg and a peak torque of 350-750Nm depending on the model. It has been demonstrated in a few cars but is not yet in production, *(Proten Electric 2010)*.

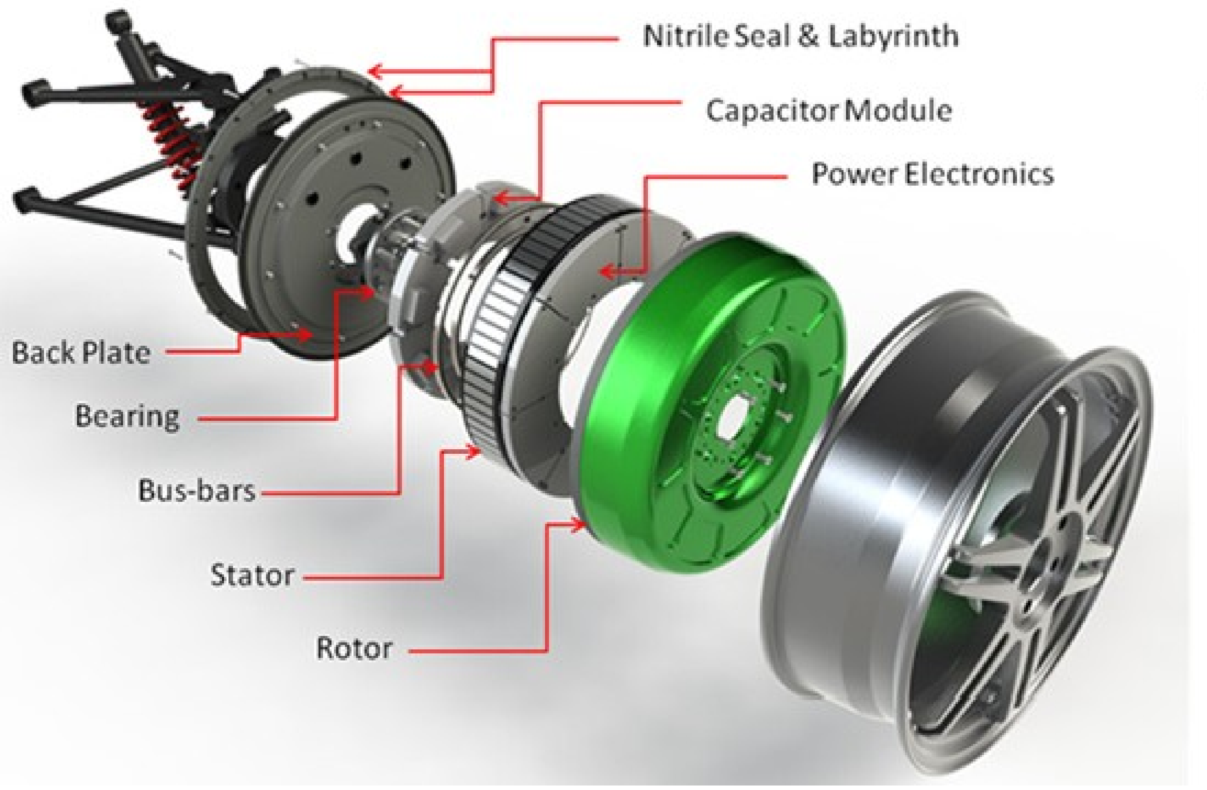


Illustration 5: Hi-Pa Drive in wheel motor concept

The technology of in wheel motors is still very much in its infancy, there are many media releases but a severe lack of demonstrated vehicle technology and production vehicles.

From this review of the technology above it can be seen that there are a lot of companies developing electric vehicle technology with a range of approaches starting at a single motor, on board motors to power more than one wheel and motors incorporated into each wheel. Despite this there seems to be a lethargy in bringing this technology to the market. All three types of technology are being evaluated by the REV project with the first two conversions the Hyundi Getz and the Lotus Elise both using the first proposed approach of having a centralised motor. The next electric car produced by the REV project was the 2009 Formula SAE electric car, this car was designed to have two chassis mounted motors powering the rear wheels. The current electric SAE car that this project is a part of aims to produce an electric SAE car powered by four in wheel motors. This effectively provides the REV project with a full spectrum of electric car designs and from this it will be possible to produce a comparison of the three approaches.

This project aims to demonstrate the viability of electric car technology mainly that associated with a motor on each wheel, by producing the drive for the front wheels of a formula electric SAE car. In doing this, this project aims to demonstrate that independent four wheel drive for electric cars is achievable and realistic goal for major automotive manufacturers to implement in their cars and has many performance, efficiency and design freedom benefits. As well this project also aims to promote electric vehicles by providing a race car that can act as an idol symbol similar to the current petrol race cars, but be an emission free alternative when charged from renewable energy sources. This is a very important step in the transition to a low carbon economy, which is very important with the rising global awareness of climate change and its impact on the planet and how we live our lives. This individual project forms part of the REV project who aim to promote electric vehicles as a viable zero tail pipe emission solution to our transportation needs. The project comes under the banner of the REV projects formula SAE car, which aims to make a four wheel drive electric car to race in a formula electric competition. Although there is only a hybrid competition at the moment it is expected to have a dedicated electric competition soon.

2. Design approach

The following section outline the design approach taken.

2.1 Constraints

The main constraints in this project originate from the fact that this project is working to put drive to the front wheels on an existing formula SAE car, this car was the original 2001 motor sports car, the layout of this can be seen in Illustration 6. Because of this there is no flexibility in the chassis design to accommodate this project, the project has a rolling shell and it is needed to redesign the current uprights to allow the mounting of a motor. The addition of these constraints like existing suspension, brakes and wheels serves to simplify the project and make it more achievable by a single thesis. More constraints come from the projects aim of creating an electric vehicle, evidently this limits the drive system to electric only. The final and one of the larger constants is the limited budget.

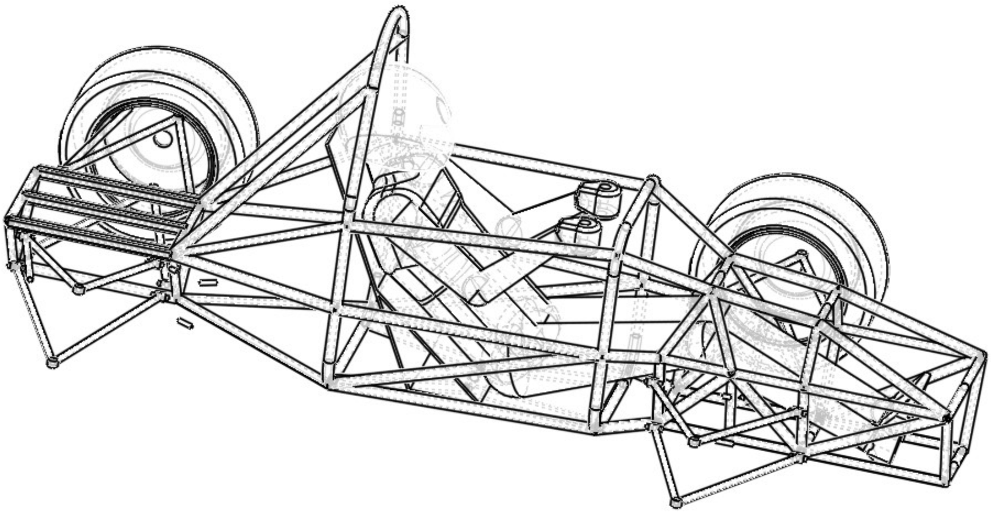


Illustration 6: Existing formula SAE car

2.2 Design criteria

These are the criteria that all design options will be judged by.

2.2.1 Weight

Weight is always an issue in race car design but due to the design being chosen to be part of the unsprung mass this is of uttermost importance because increased unsprung mass can provide a great detriment to vehicle performance as demonstrated in section 4.1.1.

2.2.2 Simplicity

If a design is simple it achieves the objectives with minimal use of materials, can also reduce manufacturing times and is often more reliable and efficient.

2.2.3 Performance

This relates to how well the design and the components selected perform in regards to efficiency of all moving parts and power of the motor.

2.2.4 Budget

This project aims to produce an electric race car on a fairly limited budget and due to this all items must be chosen at a realistic price range so the over all project can proceed.

2.2.5 Availability

This design project forms part of a larger project which aims to build a functioning electric car so all items chosen must be available so the project can be built.

3. Safety

This project involves the construction of an electric car and shares a lab with another similar project which aims to produce an electric road car. Due to this there have been many safety risks encountered by this project, an identification of these risks and the solutions that were taken to mitigate them is provided below.

3.1 Lab

- There is a swipe access door to get into the room and out of the room, this provides an extra barrier in the case of an emergency that is unnecessary.
- This issue was raised with the universities safety officer, but is still not satisfactorily solved.

- The electric cars in the lab have battery packs that have high enough voltage to brake the resistance of the skin and high enough current to kill
- The SAE car has been designed with a battery pack voltage of less than 120V DC as specified by AS 4509.1. This ensures that the voltage is not high enough to break the skin and is therefore safe for students to work with.

- When working on the cars it is often necessary to work in confined spaces which adds the risk of hitting body parts on sharp edges. As well in the lab there are many tools such as spanners, screw drivers, box cutters, wire strippers and although these are fairly safe tools there is still the risk of injury due to bad practice or inattention.

- Before having permission to work in the lab all students undergo a safety induction that familiarises them with the risks associated with working in the lab.

- If working in the lab alone and injury does happen this has the potential to increase the severity of the situation.
- To counter this it is regulated that there were always two people in the lab while anyone was working on the car.

3.2 Car

- Part of this project involves driving the existing converted electric car. This comes with many safety hazards for the driver, passengers and pedestrians.
- Anyone driving the car has to have a drivers licence with a reasonable amount of experience.

- Electric cars are nearly silent, this gives pedestrians less warning about their presence.
- In 2009 Chris Hellsten (Hellsten 2009) produced a thesis on sound generation to emulate an I.C. to alert pedestrians to the presence of the electric car.

- During a crash battery packs have the potential to leak and if this gets on the passengers or pedestrians it has the potential to cause injury. Also if it washes down the drain it will adversely effect the environment.
- This is still a risk but the battery packs are completely sealed to reduce leaking in the case of a crash, this is comparable to the risk already faced by the leaking and ignition of petrol in a crash.

- During driving the batteries have been known to give off odours which are unknown chemicals and could have the potential to harm.
- Students tested this and found nothing toxic as well the university safety department has conducted tests but the results are not known yet.

- When it comes time to test the new SAE car there is the potential that one of the parts could fail and at high speed this could be potentially dangerous.

- Physical testing of the designs will be done before they are incorporated into the car and during the initial test the driving will be sedate and gradually increased in severity, while monitoring the car to ensure that if a part does fail it is more likely to fail at low speed.

3.3 Manufacturing

- In manufacturing much heavy machinery is used, these have the potential to be very harmful if used incorrectly or through inattention.
- Before using any machinery everyone needs to be trained in its proper used by an experienced person.

- A few of the components are manufactured on the lathe, rotating machinery has the potential to rip out hair and damage anything that comes in contact with it. As well as this shards of metal that can be hot and travel fast and have the potential to damage eyes and skin.
- All hair will be tied back, protective clothing will be worn including safety glasses, enclosed foot wear, clothing that protects exposed skin.

- Assembly of the final parts is done by welding which uses high voltages which have the potential to cause electric shock, molten metal which can cause burns and extremely bright light that can damage eyes.
- Proper training will be conducted before people use the welder as well as protective clothing including an auto dimming welding mask, thick welding gloves, and clothing that will enclose all skin.

3.4 General

- Working long hours at the computer has the potential to cause spinal and other posture related damage.
- Regular breaks including stretching were conducted

- Working on such an intense project is very stressful and this potential danger really should not be understated because stress has been shown to reduce overall health and shorten your life span (Cockerham 1997).
- A balance of work and social life was attempted, but totally failed, hopefully this will be able to be improved in the future

4. Design development

This section outlines the process that was followed to achieve the final design.

4.1 Step One

The design investigation identified two main areas to place the motor either mounted on the chassis forming part of the sprung mass (see Illustration 9) or mounted directly on the upright forming part of the unsprung mass (see Illustration 11). Before these approaches are discussed it is very import to gain an understanding of the effect that unsprung mass has on the performance of a car.

4.1.1 Unsprung mass

A vehicle can be represented as a simple spring damper system with the main body of the car represented by the main mass *'m'* which is the sprung mass and the wheel assembly represented by the mass *'w'* the unsprung mass shown in illustration 7 below.

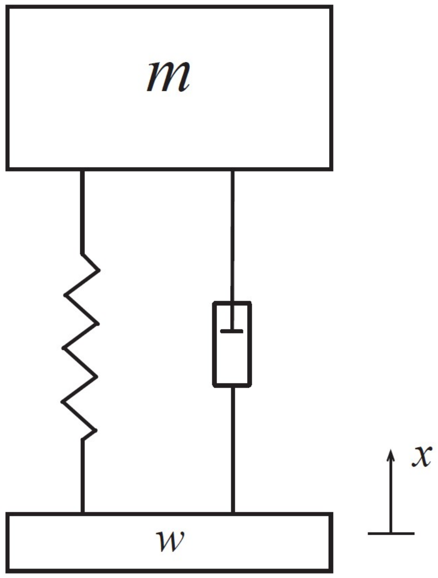


Illustration 7: Simple spring damper system

Illustration 8 shows a plot of normalised r.m.s. tyre deflections as a function of damping ratio for differing ratios of sprung mass to unsprung. From this it is clear that increasing the sprung mass increases the r.m.s. tyre deflection. This has two effects, firstly it causes a bumpier ride and secondly it causes the wheel to not track the road as well which reduces the grip of the tyres on the road, adversely affecting cornering and acceleration. From this it can be seen that for improved car dynamics it is desired that the unsprung mass is minimised , whereas the sprung mass of the car can be increased with less impact on the handling.

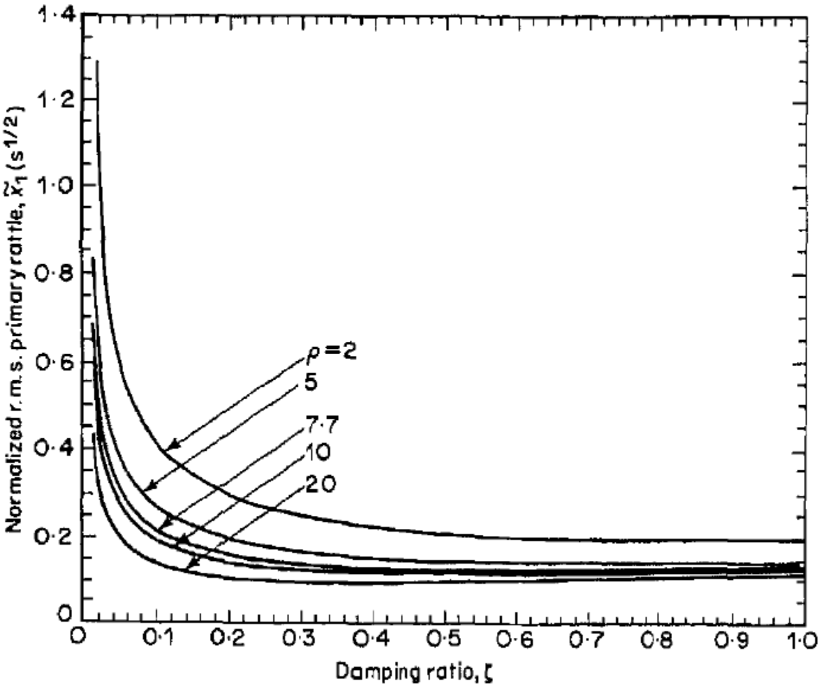


Illustration 8: Normalised r.m.s. tyre deflection as a function of damping ratio (Hrovat 1987)

4.1.2 Chassis mounted

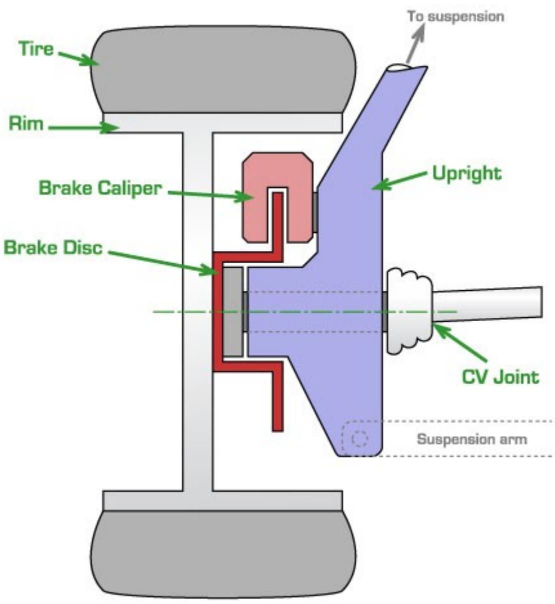


Illustration 9: Chassis mounted motor (Hooper I. 2010)

A chassis mounted motor design involves mounting the motors on the chassis shown in illustration 9, powering the wheels through a drive shaft and two CV joints. The advantages and disadvantages of this approach are investigated in the following sections.

4.1.2.1 Advantages

- In mounting the motor on the chassis it means that a heavier motor can be used, this means that the power of the motor can be greater. This is due to the fact that more coils can be incorporated into the motor, a motor with more coils is able to handle more current which directly effects the power output of the motor. Also the diameter of the motor can be increased meaning that the torque is higher due to the force being created at a greater distance from the centre of rotation.
- Commercially there are a lot of motors available for this design criteria.
- This approach takes advantage of more conventional design approaches and due to this existing knowledge would be easier to design.

4.1.2.2 Disadvantages

- The area where the motor could be placed is shown in illustration 10.

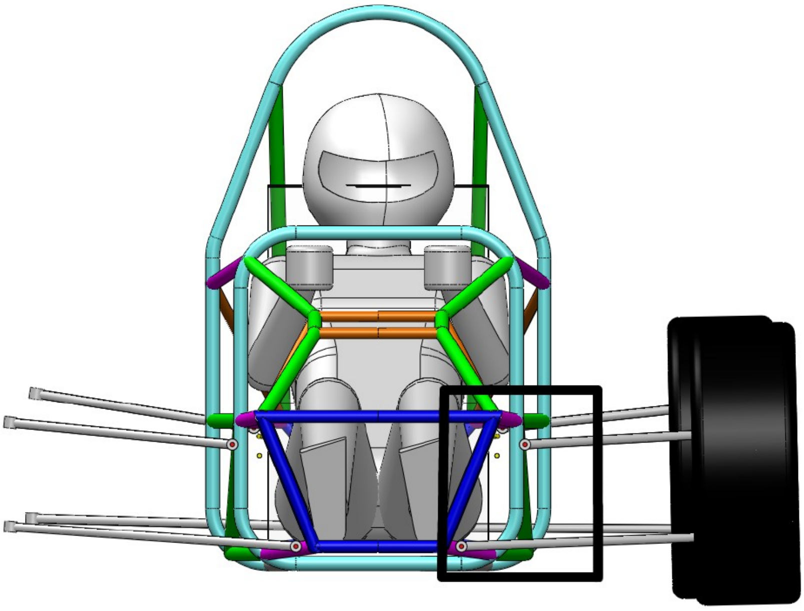


Illustration 10: Front on board motor space

From this it is clear that a motor will not fit on board due to all the space being taken up by the driver, this is the deciding factor that makes this design untenable.

- When designing an on board system the motor is generally heavier due to taking advantage of the ability to increase the sprung mass to get a more powerful motor.
- On a front wheel on board drive system there are two CV joints, a steering joint and a longer drive shaft per wheel, this increases cost, weight and decreases efficiency and simplicity.

4.1.3 Wheel Mounted

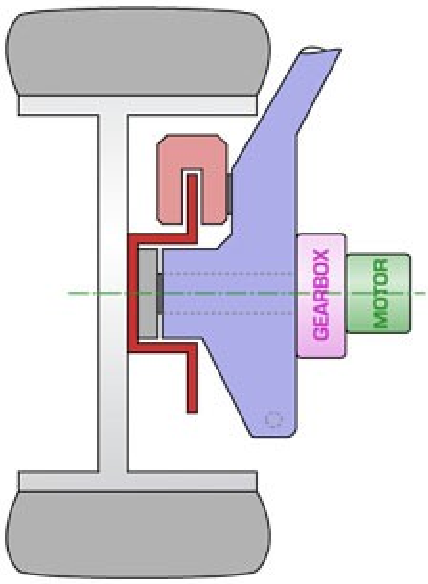


Illustration 11: Example of a wheel mounted motor (Hooper I 2010)

A wheel mounted motor involves mounting the motor into the wheel assembly negating the need for any CV joints.

4.1.3.1 Advantages

- Mounting the motor in the wheel assembly means that there will be less components. It eliminates two CV joints, a longer drive shaft and a steering joint for each wheel, CV joints are typically 98% efficient but are likely to be less than this after wear, eliminating these components makes it more efficient and reliable.

- Due to the need to reduce the weight of the unsprung assembly forces the design to be lighter weight.
- In an in wheel design there are less components, this makes the design simpler to construct.

4.1.3.2 Disadvantages

- This approach increases the unsprung mass which if exceed by too much adversely effects handling, see section 4.1.1.
- There are many manufactures already making larger motors but there is a void in the market for small high power motors.

4.1.4 Decision.

From the above analysis it can be seen that both designs satisfy the design constraints of simplicity, performance, availability and weight, but the on board design did not satisfy the constraints imposed on this project by the chassis design. Due to this fact it was decided to go with the in wheel approach, this also was better than the on board approach in terms of efficiency, simplicity.

4.2 Step 2

With this design path chosen there are still a few types of in wheel motors that have been identified, these include a small motor in series with a gearbox, pancake direct drive and rim motor, these are investigated below based on the constraints and design criteria.

4.2.1 Small motor mounted in series with gear box

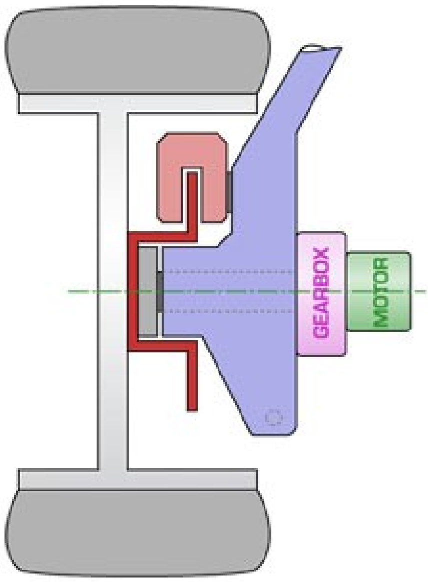


Illustration 12: Small motor mounted in series (Hooper, I. 2010)

4.2.1.1 Advantages

- Gear box and motor available off the shelf.
- Light weight due to there being fewer components and by the fact that a small motor is required.
- Simple due to there being less components used.

4.2.1.2 Disadvantages

- Smaller motors are often not as powerful because they can accept a lower current before over heating, this can be countered by active cooling but this increases complexity.
- Not as efficient use of materials than some of the other approaches, this is because it is a linear approach that can result in redundant material, but this may be compensated by having a small motor compared to the large direct drive motors in the other approaches.
- The motor and gearbox would have to extend towards the centre of the car from the

inside of the upright, to do this the damper arm would have to be moved as seen by illustration 13.

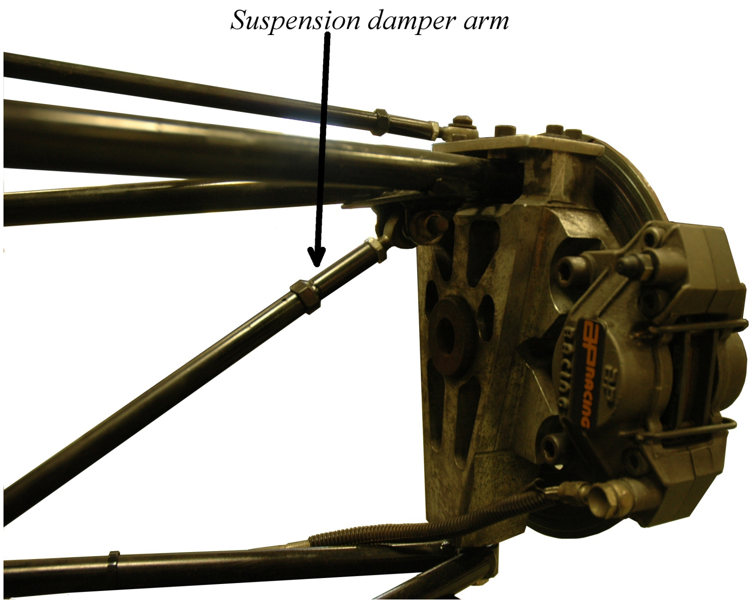


Illustration 13: Suspension damper arm

4.2.2 Pancake direct drive motor.

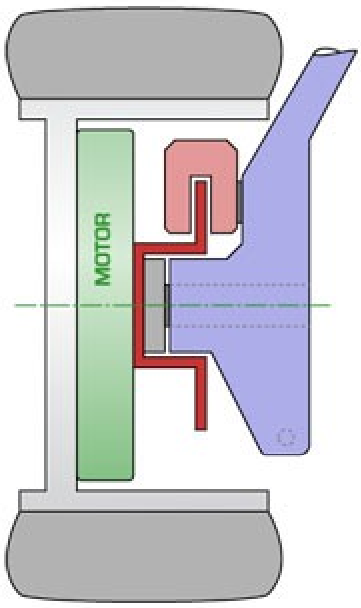


Illustration 14: Pancake motor (Hooper 2010)

A pancake direct drive system involves having an axial thrust motor connected directly to the wheel without the need for a gearbox. The need for a gearbox is removed because the larger motor generates the magnetic force at a greater distance from the axis of rotation which increases the torque. This motor can be mounted either side of the upright, the case where it is mounted on the wheel side is shown in illustration 14.

4.2.2.1 Advantages

- Due to the motor producing a greater torque a gearbox is not needed, this reduction in moving parts increases efficiency and reliability.
- The motor can be incorporated into the rim meaning that this type of in wheel motor can be easily retrofitted to existing cars.

4.2.2.2 Disadvantages

- In the research of product availability this project was not able to source a motor that would be suitable for this situation. This means that the motor would have to be custom fabricated, this increases cost and complexity.
- In direct drive applications the motor needs to be able to supply the torque without the aid of a gearbox, this means increasing the diameter of the motor and in doing this increases the material needed to build the motor, increasing the weight. As mentioned in the unsprung mass section this will lead to a greater r.m.s. tyre deflection.

4.2.3 Motor integrated into the rim

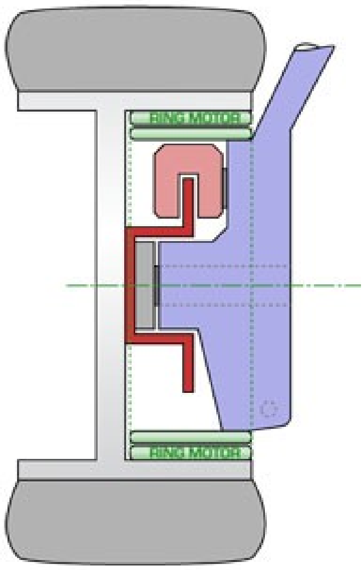


Illustration 15: Ring motor (Hooper 2010)

This approach involves building the rotor of the motor into the wheel, this uses material from the wheel to double as the outside casing for the motor, in doing this the weight is reduced.

4.2.3.1 Advantages

- This assembly is self contained in the wheel and requires minimal modification to the upright assembly which means that it can possibly bolt onto existing cars.
- Integration of the motor into the rim means that the motor casing and the rim share material, this can reduce the overall weight of the assembly.
- The motor has a greater diameter which leads to a greater torque reducing the need for a gearbox.

4.2.3.2 Disadvantages

- This involves matching the motor to the rim of the wheel, there are many companies researching this technology as mentioned in the literature review but none of these are available commercially. This means that the motor and rim would have to be made by this project, this adds to the cost and complexity.

- The rims on cars normally undergo deflection, this is not normally a problem but, due to the rim and the rotor being integrated, this alters the air gap between the rotor and stator. This is a significant problem because an air gap is typically 1mm or less.

4.2.4 Decision.

All the design paths identified above are valid and are worth pursuing and it would be very interesting to see a comparison of built examples. Due to the constraints imposed on this project the ring motor and the pancake direct drive design solutions were eliminated because they were deemed infeasible for a single final year project due to the complexity of manufacturing custom motors. This left the design decision of having a small motor in series with a gearbox. Despite being the only approach that was in the scope of this project it also offers many benefits such as being light weight due to the need for a small motor which has performance benefits due to having a lower unsprung mass. It is also the simplest approach because it does not require great redesigns of the wheel. As well a lot of the parts can be sourced off the shelf greatly reducing the time of manufacturing for the project. Now that the design methods have been decided, system design and component sourcing can begin

4.3 Upright design

The existing front upright was not designed to accept drive so this had to be redesigned. The following approaches were considered.

4.3.1 Solid block Aluminium

Illustration 16: Existing solid block upright

A solid block upright involves a solid block of aluminium and CNC machining this to the required shape.

4.3.1.1 Advantages

- This is how the current upright was manufactured and is therefore a known design that will present less challenges in its manufacture.
- CNC machining is easy to automate, this means that many can be produced quickly.

4.3.1.2 Disadvantages

- CNC machining is an expensive process.
- Solid blocks of aluminium are expensive and results in a lot of material discarded.
- This design does not allow hollow sections which results in the weight increasing.

4.3.2 Cast Aluminium

Illustration 17: Cast aluminium upright (ADR 2009)

A cast aluminium upright involves making a prototype and then making a sand mould of it, to which aluminium is poured.

4.3.2.1 Advantages

- The set up time for this can be fairly long, but once the cast is made, many uprights are able to be built quickly.
- Complex shapes are achievable.
- Cheap to manufacture.

4.3.2.2 Disadvantages

- Quite heavy, due to solid profile.

4.3.3 Thin wall steel or aluminium

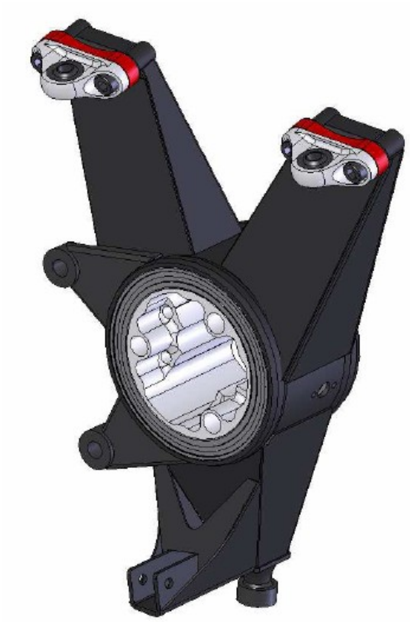


Illustration 18: Thin walled steel upright (Harrison T 2005)

4.3.3.1 Advantages

- Using sheet metal construction allows for hollow sections, this reduces the overall weight.
- All the manufacturing methods employed by this approach, laser cutting, lathe and welding are cheap to perform.
- This is the approach currently used by UWA motorsport and is functioning very well.

4.3.3.2 Disadvantages

- This approach is very labour intensive and due to this takes longer to manufacture.

4.3.4 Decision

From this evaluation of options it was decided that a thin walled design would be most suited to this project. This is due to it being able to produce an upright of the lowest mass. This is very important because the chosen design path of an in wheel motor design will be contributing to the unsprung mass which is detrimental to vehicle handling.

4.4 Material Selection

The two materials considered were steel and aluminium. A comparison of the specific alloys is presented below.

	304 Stainless steel	6061-T6 Aluminium
Yield Strength (MPa)	215	276
Density (g/c^3)	8	2.7
Specific Yield Strength (MPa c^3/g	27	102
Poissons Ratio	0.29	0.33
Shear Modulus (GPa)	86	26

Table 1: Comparison of material properties (Matweb 2010)

From the comparison between 304 stainless steel and 6061-T6 aluminium it is clear that the aluminium alloy has a much larger yield strength as a function of density by a factor of almost four. Other factors to consider are that, unlike steel, aluminium does not reach a constant yield strength when subject to fatigue. The initial corrosion layer on aluminium forms an impervious layer which halts further corrosion, unlike steel where the corrosion layer is porous causing further corrosion and the specified aluminium alloy needs heat treatment to reach the desired yield strength.

4.4.1 Decision.

Aluminium was chosen because it offered a much greater strength to weight ratio and the upright would need to be welded to the gear box mount which for weight reasons would also need to be made from aluminium so to reduce the weight the upright was made from aluminium as well. It was decided that the fatigue limit would not be an issue due to the limited life of race components.

4.4.2 Aluminium alloy selection

It was decided to go with aluminium 6061-T6 for the lathed components because this would offer the greatest strength to weight ratio with a yield strength of 276MPa (Matweb 2010). Aluminium 5005-H18 was selected for the sheet metal components of the upright due to availability, ductility and its high yield strength of 193MPa (Matweb 2010).

4.5 Gear box selection.

The gear box characteristics were determined by the selected motor the Predator 37/6, this produces its peak power at 6000rpm.

4.5.1 Reduction ratio.

The motor produces its peak power at 6000rpm, due to the design having a single stage gearbox, this peak power is required to occur at top speed. The equation below gives the relationship between motor rpm and output rpm from the gearbox.

$$Output(rpm) = \frac{Input(rpm)}{n} \tag{4.1}$$

$$n - reduction\ ratio$$

From this output rpm the wheel speed can be calculated via the following equation

$$Speed(kph) = 0.06 * \pi * Output(rpm) * d_w \tag{4.2}$$

$$d_w - diameter\ of\ the\ wheel$$

From these previous equations it is possible to derive an equation for the required reduction ratio as a function of the the motor rpm, wheel diameter and reduction ratio.

$$Speed\,(kph) = \frac{Input\,(rpm)}{n} * d_w * 0.06 * \pi \tag{4.3}$$

$$n = \frac{Input\,(rpm)}{Speed\,(kph)} * d_w * 0.06 * \pi \tag{4.4}$$

From this, using a desired road speed of 110kph it was determined that a reduction ratio of between 5 and 7 would be suitable.

4.5.2 Torque handling.

The motor that we selected produces a maximum torque of almost 20Nm so with a maximum reduction ratio of 7 the maximum input on the planetary gear is governed by the following simple equation.

$$Max\,gearbox\,torque\,(Nm) = n * Max\,motor\,torque \tag{4.5}$$
$$n - reduction\,ratio$$

From this equation the maximum torque is 140Nm if the maximum reduction ratio is used.

4.5.3 Gearbox type.

With the specifications of the gearbox decided upon it is now possible to evaluate the options.

4.5.3.1 Pulley

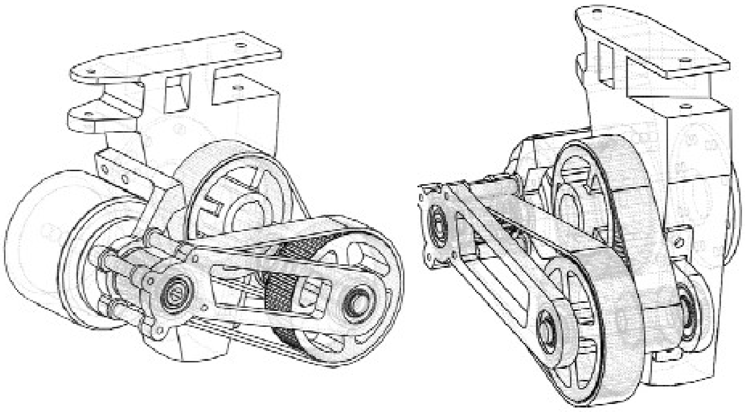


Illustration 19: Pulley as proposed by Marius (Ivanescu 2009)

A pulley consists of two gears connected by a belt as shown above. This approach was investigated by Marius Ivanescu (Ivanescu M 2009) in his 2009 honours thesis. He deemed this design to be too complicated to fit a reduction ratio 5 or greater in the confined space of an upright and due to this complication it was decided to go with an on board approach for the 2009 Formula Electric SAE car.

4.5.3.2 Chain drive.

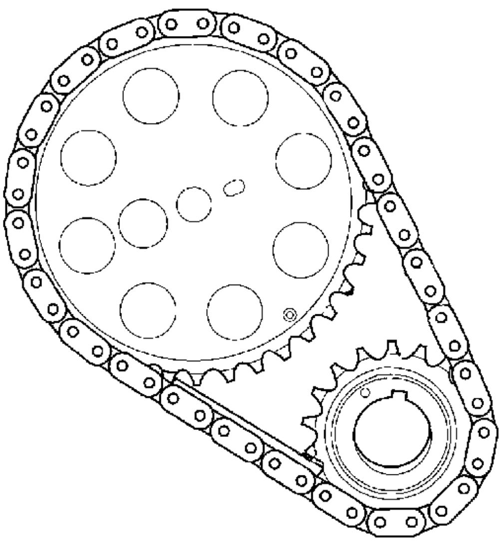


Illustration 20: Chain drive (justanswers 2009)

A chain gear was investigated as well and is much like a pulley, with two gears being connected via a chain as opposed to a belt. It was discounted for the same reasons as the pulley system being too complicated to fit in the confines of an upright.

4.5.3.3 Planetary.

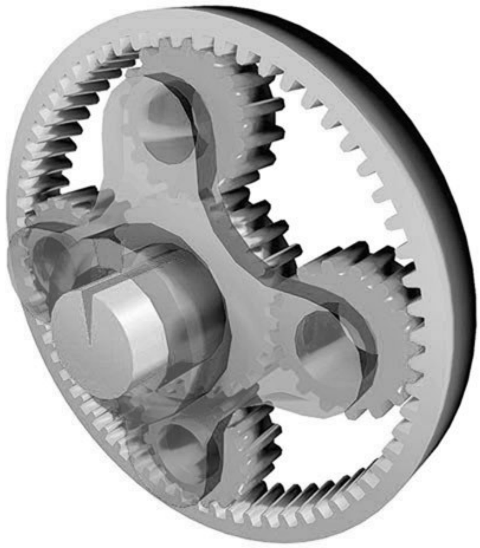


Illustration 21: Planetary gear-set (Diesel Power. 2009)

A planetary consists of three sets of gears, the ring gear on the outside, the sun gear in the middle and planetary gears rotating around the sun gear connected by the planetary carrier. From this it is possible to achieve three different gear ratios depending on which gear is held stationary. If the ring gear is held stationary the maximum ratio will be achieved by driving the sun gear and having the planetary carrier as the output, this is independent of the planetary radius and is given by;

$$n=1+\frac{r_r}{r_s} \tag{4.6}$$

$n - Reduction\ ratio$

$r_r - Radius\ of\ the\ ring\ gear\ ,\ the\ outside\ gear.$

$R_s - Radius\ of\ the\ sun\ gear\ ,\ the\ driven\ gear.$

As can be seen from equation 4.6 this achieves a greater reduction ratio than a conventional spur gear which would have a reduction ratio of $n=r_r/r_s$. As well by having the sun gear mounted inside the ring gear results in the gearbox having an outside diameter equal to the ring gear were as in the spur gear case the diameter would be $2(r_r+r_s)$.

4.5.4 Decision

From this evaluation it was decided to go with a planetary due to its light weight, high torque handling and compact size which would make it easy to incorporate into the upright. After an extensive search the Matex 120-5MHN (Matex 2010) planetary gear set, Illustration 22, was decided upon due to its ratio, torque handling, compact size and reasonable price.

Illustration 22: Matex 120-5MHN planetary gear set.

4.6 Bearing selection

It was decided to go with two single row angular contact ball bearings made for a paired mounting because this would give a greater ability to resist moments from the wheel load as opposed to a double row angular contact ball bearing. The bearing size was chosen based on the minimum required shaft diameter. The shaft diameter calculation is seen in the forces section. In minimising the shaft diameter it also allowed the outer diameter of the bearing to be minimised reducing the total weight of the bearing assembly. From this a diameter of 20mm was selected this corresponded to an external diameter of 47mm and this is shown in Illustration 23 below.

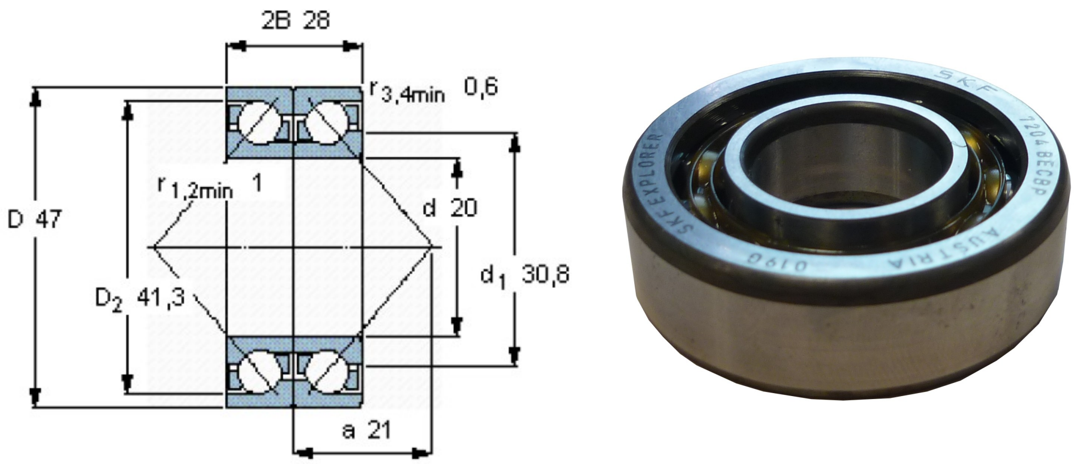


Illustration 23: SKF Angular contact ball bearing for paired mounting, 7204BECBM

4.7 Motor selection

The specifications of this project required the motor to be light weight and powerful, there are not many manufactures making motors in this category, all motors identified by this project are presented in table 2, with the Mars motor that is currently mounted to the rear wheels being included. From this analysis The Predator 37/6 was chosen because it offered the greatest power, which was the most important constraint in a race car, although its power to weight ratio was less than that of the smaller motors this increase in weight was deemed acceptable for the increase in power.

	Hacker A200 (Hacker 2010)	Predator37/6 (Plettenberg 2010)	Predator30/8 (Plettenberg 2010)	NeuMotor 2230 (NeuMotors 2010)	Mars
Peak Power	15kW	15kW	12.5kW	10kW	10kW
Max rpm	8000	7500	6400	40000	5000
Weight	2.59kg	1.9kg	1.35kg	1.36kg	11kg
Power per kg	5.8kW/kg	7.9kW/kg	9.2kW/kg	7.35kW/kg	0.9kW/kg
Type	Outrunner	Outrunner	Outrunner	Inrunner	Axial flux
Peak Efficiency (%)	Not given	Not given	89.20%	92%	82%

Table 2: Comparison of motors

4.8 Force modelling

To obtain reasonable results from the finite element analysis the forces that the assembly is subjected to must be determined, this is done for various cases below.

4.8.1 Straight line acceleration.

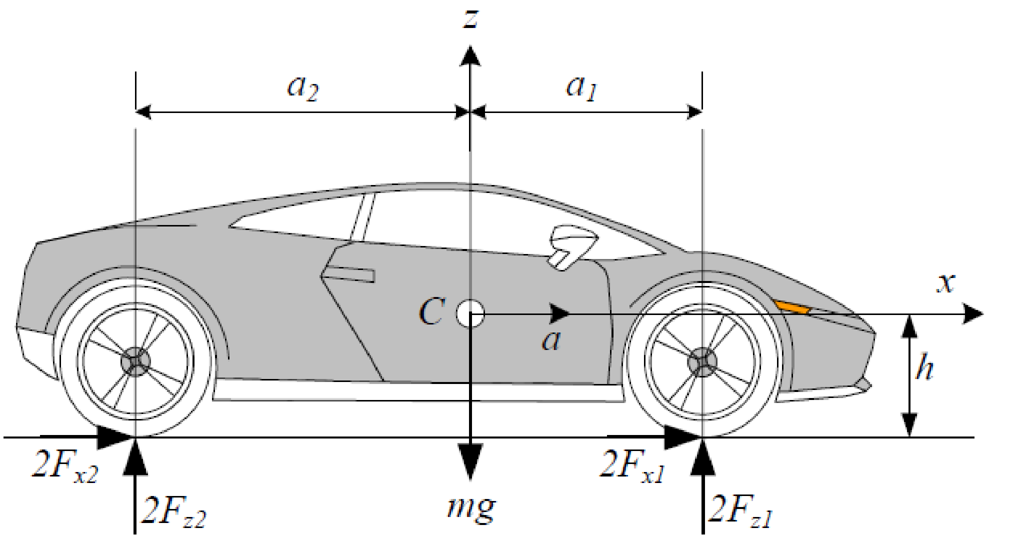


Illustration 24: Forces on car (Jazar 2008)

Under acceleration the forces on each wheel can be represented by the following formulas in reference to the diagram above (Jazar 2008).

$$F_{z1} = \frac{1}{2} mg \frac{a_2}{l} - \frac{1}{2} mg \frac{h\,a}{l\,g} \quad (4.7)$$

$$F_{z2} = \frac{1}{2} mg \frac{a_1}{l} + \frac{1}{2} mg \frac{h\,a}{l\,g} \quad (4.8)$$

For the formula SAE car an acceleration of 0.9g and a maximum braking 1.3g are required. From this the forces on the front uprights were determined to be.

	F_{z1} (N)
Braking	852
Acceleration	447

Table 3: Forces on front braking

4.8.2 Vehicle roll

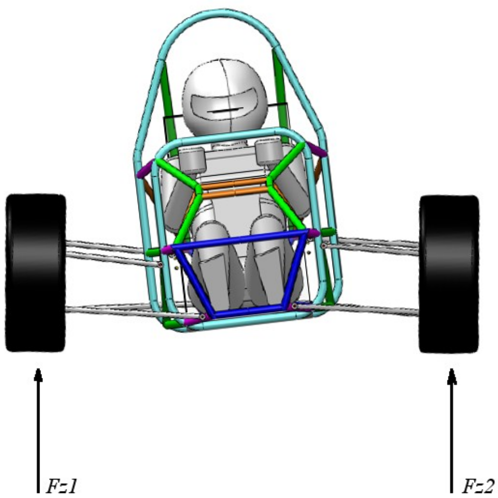


Illustration 25: Vehicle roll

Vehicle roll is the transfer of the vehicles weight to the outer wheels while cornering. This is very complex to model in 3D but it is easy to understand at its extremes. This weight transfer will increase until the inside wheels leave the ground resulting in the whole weight being split between the two outside wheels giving;

$$F_{z1}=\frac{1}{2}mg \qquad (4.9)$$

$$F_{z2}=0 \qquad (4.10)$$

4.8.3 Moment on upright and drive shaft due to weight transfer.

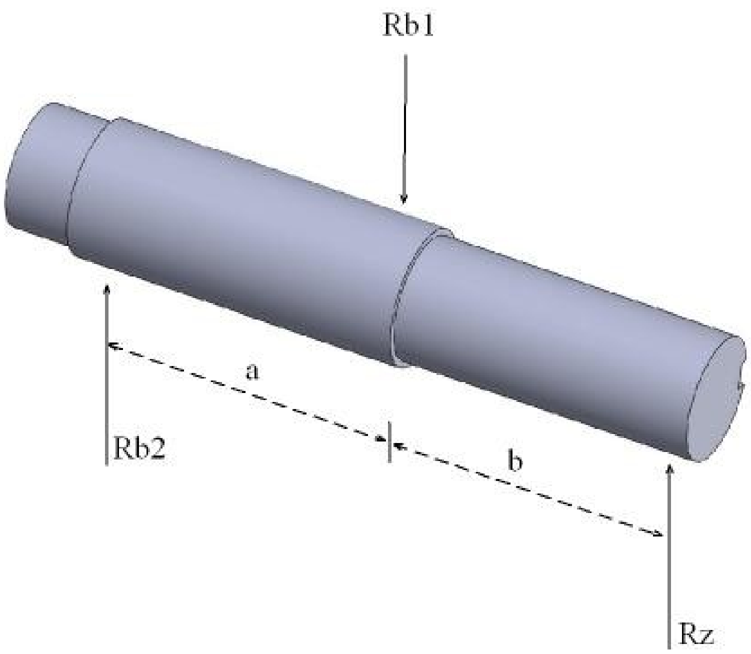


Illustration 26: Reaction forces on shaft

Simple beam bending is used to calculate the reaction forces on the bearing, in the FEA analysis this will be modelled as a distributed force. The reaction forces from the bearings on the shaft and the upright can be determined from the following equations, by taking moments about R_{b1} and R_{b2}.

$$R_{b1}=\frac{R_z*(a+b)}{a} \qquad (4.10)$$

$$R_{b2} = \frac{R_z * b}{a} \tag{4.11}$$

4.8.4 Vehicle cornering

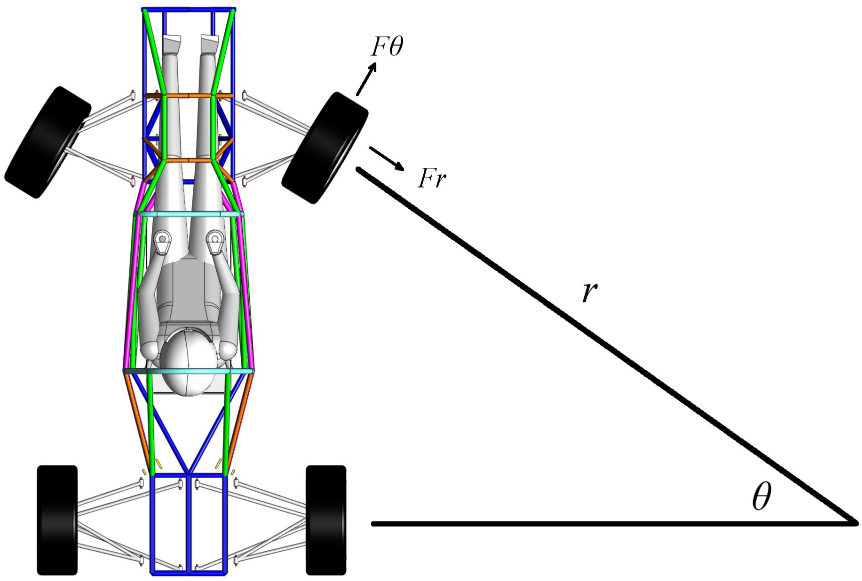


Illustration 27: Simplified model of vehicle cornering

A simplified model of vehicle cornering is presented in illustration 28, during cornering the vehicles tyres need to provide a force in the direction of the instantaneous centre of the corner, with magnitude governed by the speed and instantaneous radius. The following equations use polar coordinates to determine these forces.

Velocity of the car in polar coordinates is given by.

$$v = \frac{d\mathbf{r}}{dt} = \dot{r}\,\hat{\mathbf{r}} + r\,\dot{\theta}\,\hat{\boldsymbol{\theta}} \tag{4.12}$$

$$v - velocity$$
$$\hat{\mathbf{r}} - radial\ unit\ vector$$
$$\hat{\boldsymbol{\theta}} - angular\ unit\ vector$$
$$r - corner\ radius$$
$$\dot{r} - represent\ devivitive\ with\ respect\ to\ time\ ,\ likewise\ for\ \theta$$

The force needed to change the direction of the whole car is given by.

$$F=m\boldsymbol{a}=m(\ddot{r}-r\dot{\theta}^2)\hat{\boldsymbol{r}}+m(r\ddot{\theta}+2\dot{r}\dot{\theta})\hat{\boldsymbol{\theta}} \tag{4.13}$$

For the force on the upright in the radial direction, the radial component is singled out.

$$F_r=m(\ddot{r}-r\dot{\theta}^2) \tag{4.14}$$

Assuming the car is travelling around a circle of constant radius this means that $\ddot{r}=0$ & $\dot{r}=0$, therefore F_r simplifies to.

$$F_r=-mr\dot{\theta}^2 \tag{4.15}$$

As well equation 4.12 simplifies to

$$\boldsymbol{v}=\frac{d\boldsymbol{r}}{dt}=r\dot{\theta}\hat{\boldsymbol{\theta}} \tag{4.16}$$

Taking the magnitude gives.

$$|\boldsymbol{v}|=s=r\dot{\theta} \tag{4.17}$$

Assuming that the velocity of the car is known the radial force can be determined by substituting 4.17 into 4.15, giving.

$$|F_r|=m\frac{s^2}{r} \tag{4.18}$$

Normally this would be split between all four wheels, but in the worst case it is envisaged that while cornering the inside wheels have left the ground leaving this force to be provided by two wheels giving the force in the radial direction that each wheel must provide as being;

$$|F_r| = \frac{1}{2} m \frac{s^2}{r} \quad (4.19)$$

4.8.5 Force on the steering mount.

The force on the steering mount is determined by the torque the driver can apply on the steering wheel and then by the size of the pinion gear in the rack and pinion steering set-up.

$$T_s = r_s * F_s \quad (4.20)$$

$$T_s - \textit{torque on the steering wheel}\,(Nm)$$

$$r_s - \textit{radius of the steering wheel}\,(m)$$

$$F_s - \textit{force the driver is able to exert on the steering wheel}\,(N)$$

From the torque on the steering wheel the force on the steering mount can be determined via the pinion radius;

$$F_{sm} = T_s * r_p \quad (4.21)$$

$$F_{sm} - \textit{is the force on the steering mount}\,(N)$$

$$r_p - \textit{is the pinion radius}\,(m)$$

4.8.6 Force on the brake mount

The force on the brake mount is determined by the braking acceleration, the distance of the brake mount from the centre of the drive shaft and the wheel radius. The force from deceleration is given by;

$$F = m\,a \quad (4.22)$$

$$m - \textit{mass of the car}\,(kg)$$

$$a - \textit{is the required deceleration}\,(m/s^2)$$

From this the force on each wheel, assuming all wheels are contributing equally, is;

$$F = \frac{1}{4} m\, a \qquad (4.23)$$

Converting this to a torque, via the radius of the tyre

$$T = \frac{1}{4} m\, a\, r_w \qquad (4.24)$$

r_w *is the radius of the wheel.*

Substituting the following values into this equation gives

$$m - 250\text{kg}$$
$$a - 1.3\text{g}\, m/s^2$$
$$r_w - 0.25\text{m}$$
$$T = 199\text{Nm}$$

Now this can be converted to a force on the brake mount, based on the distance the brake mount is from the centre of the drive shaft;

$$F_{bm} = T\, r_b \qquad (4.25)$$

r_b *– is the distance of the brake mount from the centre of the drive shaft.*

4.8.7 Torsional Force on drive shaft and wheel mount

The gearbox is rated to a maximum 200 Nm of torque and the brakes can provide 199Nm of torque. Due to this the drive shaft and wheel mount will be designed to withstand a maximum torque of 200Nm.

4.9 Safety factors

Safety factors account for variation in imposed loads, material properties, corrosion and operating temperature to mention a few. As well the safety factor prevents sudden failure of the part which is especially important in the race car environment where failure can have serious consequences. Due to the forces in automotive dynamics not being fully quantified it would be expected to have a larger safety factor. For well know materials in uncertain environments and stresses it is recommended that the safety factor is 3-4 and for well known material subjected to well determined loads the safety factor should be 1.5-2 (Wright 2001).

4.10 Naming of sections of the upright assembly

To make the below discussion clearer the following diagram has been provided to explain the names given to the individual components.

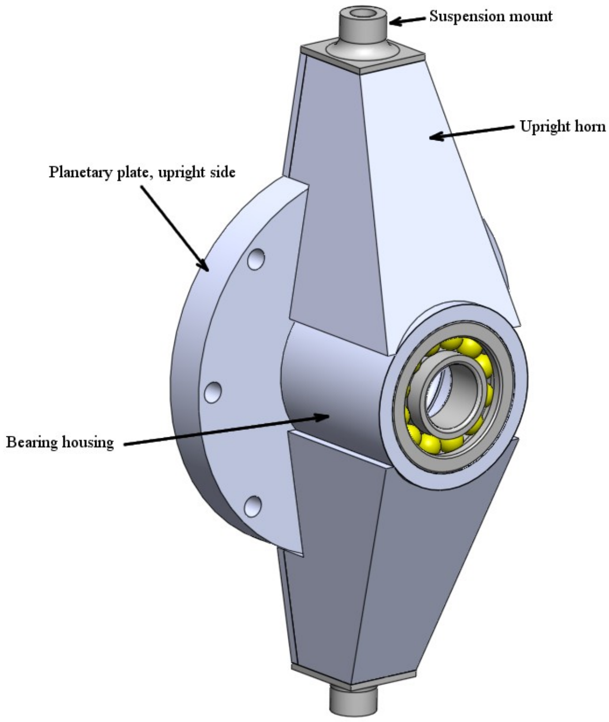


Illustration 28: Names of upright parts

4.11 FEA

After all the load conditions were determined FEA was able to start on the proposed design, Cosmos was used for this due to its user friendly interface. For each component a picture of the safety factor distribution and the Von Mises stress is presented. The safety factor diagrams show regions that are below a certain safety factor and the Von Mises diagrams show the stress distribution via a colour gradient. These either validated the model or showed up limitations that needed modification. From these diagrams changes were made to the model to ensure that it satisfied the desired safety factors. As well as this an illustration of the imposed forces and restraints is provided for each case. The results for the individual components are presented below;

4.11.1 Upright

The design of the upright was formed by the constraints of the existing suspension set-up, chosen bearings and the planetary gear set. It was designed around these constrains to minimise weight while maximising the strength of the assembly. The FEA of the final model is shown below.

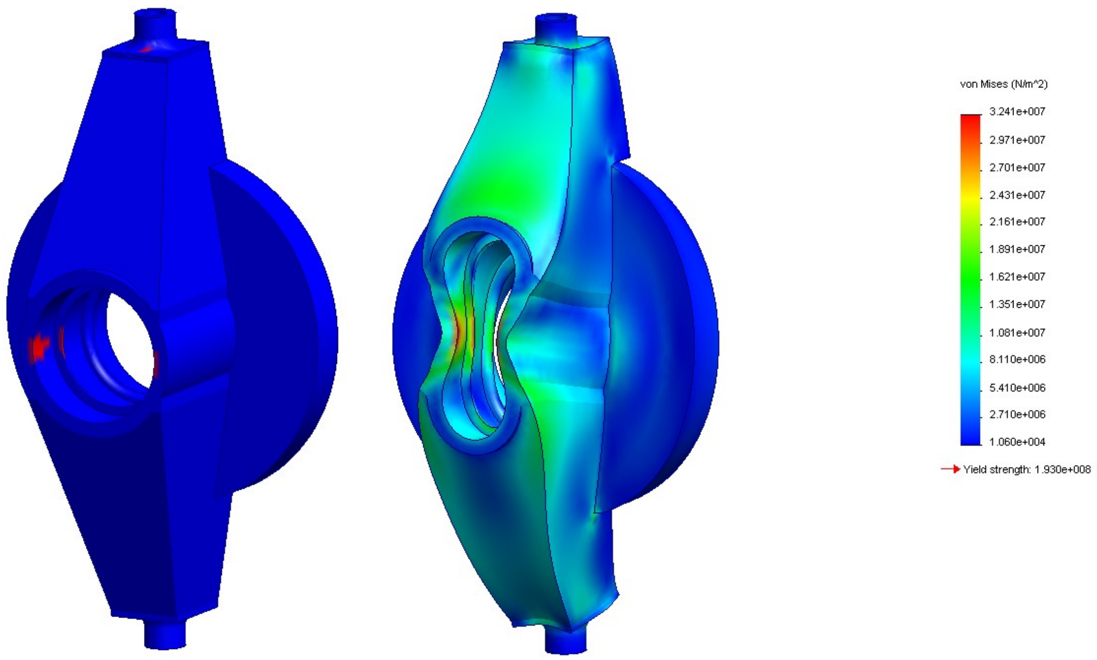
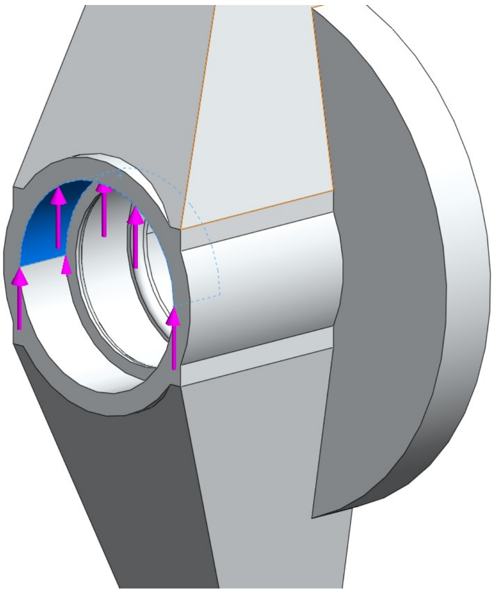


Illustration 29: Upright safety factor (red shows below 8) and Von Mises stress distubution



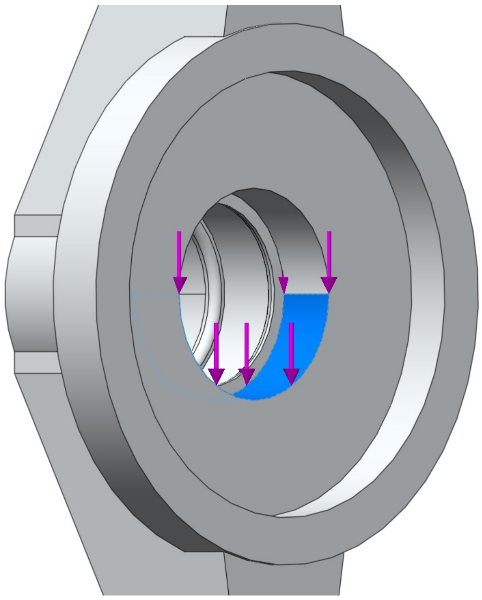


Illustration 30: Forces on upright due to maximum wheel loading

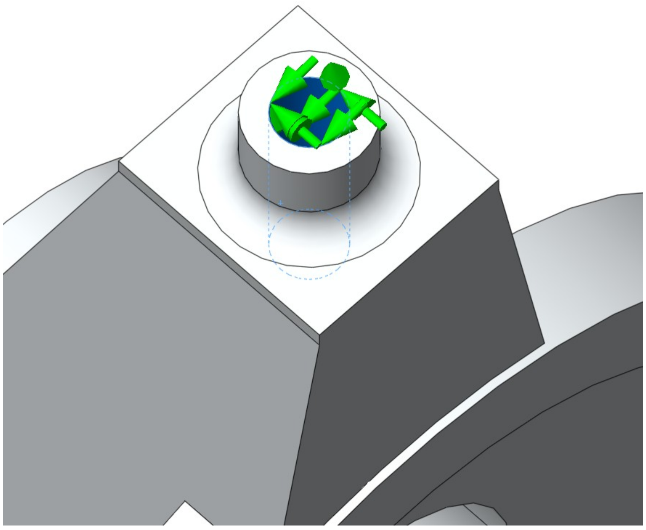


Illustration 31: Upright restraint

4.11.1.1 Safety factor

The lowest safety factor occurring in the upright is nearly 6, the reason this is so large is due to the uncertainty associated with the forces that the upright is subjected to as well as the consequences of failure and exceed the recommended range of 3-4.

4.11.1.2 Areas of low stress

The sheet metal sections were shown to have a lower stress concentration and due to this the sheet metal thickness was decreased from the original 4mm to 3mm. As well there are areas of low stress in the planetary plate on the upright side but due to the constraints of this part having to mate with the planetary gear-set there was little freedom to modify this.

4.11.1.2 Areas of high stress.

The sides of the bearing casing and the base of the suspension mount had the highest levels of stress but with a safety factor of almost 6 this is greater than what is recommended by the range of 3-4 (Wright 2001).

4.11.2 Steering

The steering mount was designed to have Ackermann steering geometry, FEA of the final design incorporated into the upright assembly is presented below, with the restraints being the same as for section 4.12.1.

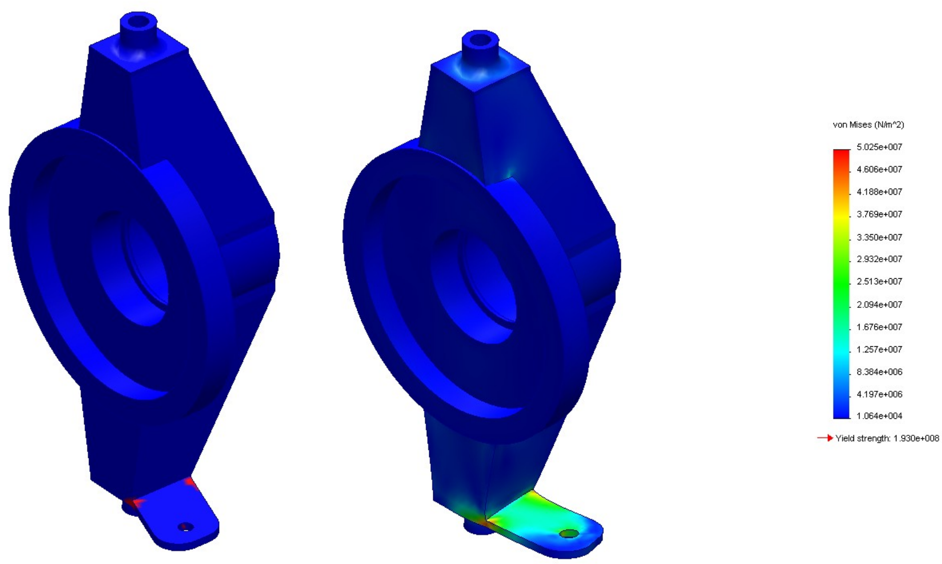


Illustration 32: Steering mount safety factor (red shows below 6) and Von Mises stress distribution

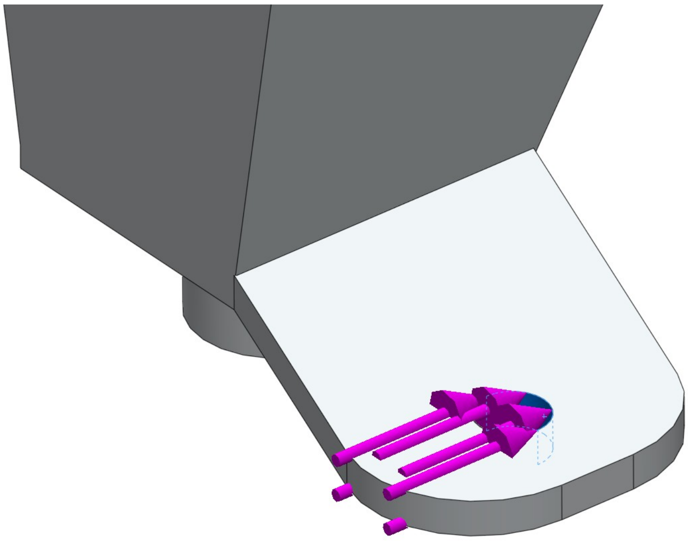


Illustration 33: Force on steering mount

FEA showed the lowest safety factor to 3.84, this occurred where the suspension mount joins the upright and where the suspension arm mounts onto the mount. Due to the forces being well understood and the material properties known the desired safety factor range was 1.5-2, the achieved safety factor of 3.84 exceeded this range and was deemed acceptable.

4.11.3 Brake

The brake mount was designed around the existing AP racing brake calliper, the specifications of this are shown in the appendix and the forces that this would be subjected to are derived in section 4.9.6. FEA of the final design is presented below, with the restraints being the same as for the case presented in section 4.12.1.

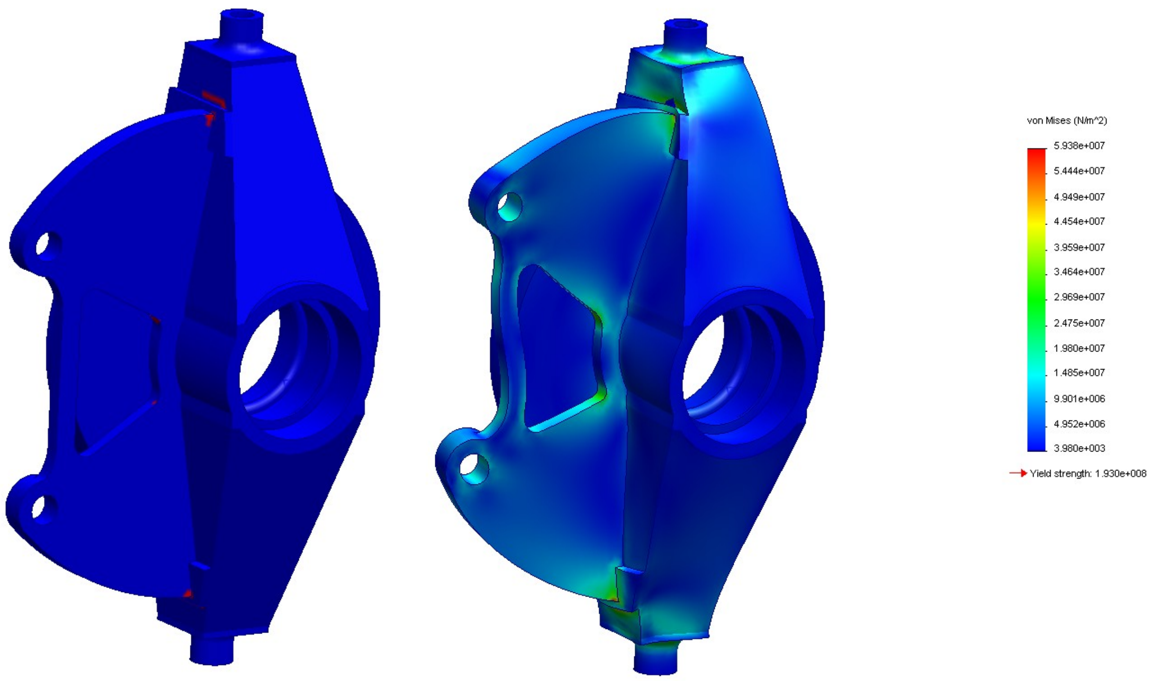


Illustration 34: Brake mount safety factor (red shows below 6) and Von Mises

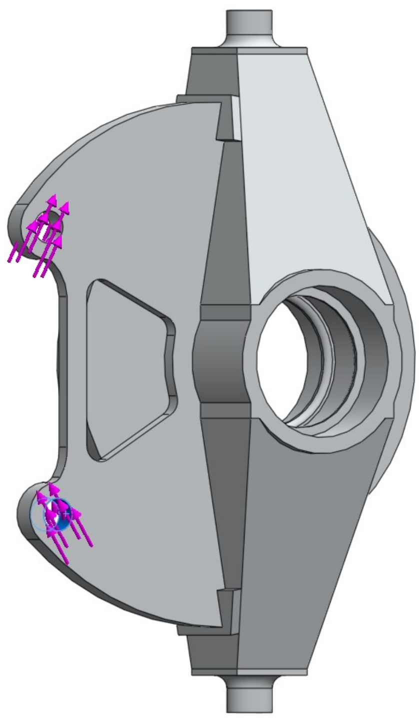


Illustration 35: Forces on brake mount

4.11.3.1 Safety factor

The lowest safety safety factor of 3.25 occurred where the top and bottom of the brake mount meet the upright and in the corners of the cutaway section. The Illustration 35 above shows where the safety factor is below 6. The forces on the brake mount are well known and determined by the deceleration that the brake calliper can apply to the car, so the desired safety factor fell within the range of 1.5-2, due to this the achieved safety factor is acceptable.

4.11.3.2 High stress

The main area that was shown to have a high stress concentration was at the top of the brake mount where it joined to the upright, quite a few changes were made to reduce this stress concentration. Firstly this was extended further up the upright, this was a good improvement but further improvement was still needed. So to distribute the load on the thin upright wall more evenly a thin plate was incorporated at the top and the top of the brake mount was thickened. This all served to reduce the stress concentration to an acceptable limit.

4.11.3.3 Low stress.

The area in the middle of the brake mount contributed very little strength to the design and due to this a cut away of the section was created, although this in itself caused a slight increase in stress concentration it was deemed acceptable due to the saving in weight.

4.11.4 Wheel mount and shaft

The wheel mount and shaft were designed around the need to mount the wheel to the planetary gear box output, as well as this it incorporates a mount for the brake mount and provides enough clearance so the brake calliper does not interfere with the rim. FEA of the final design in presented below.

4.11.4.1 Bending

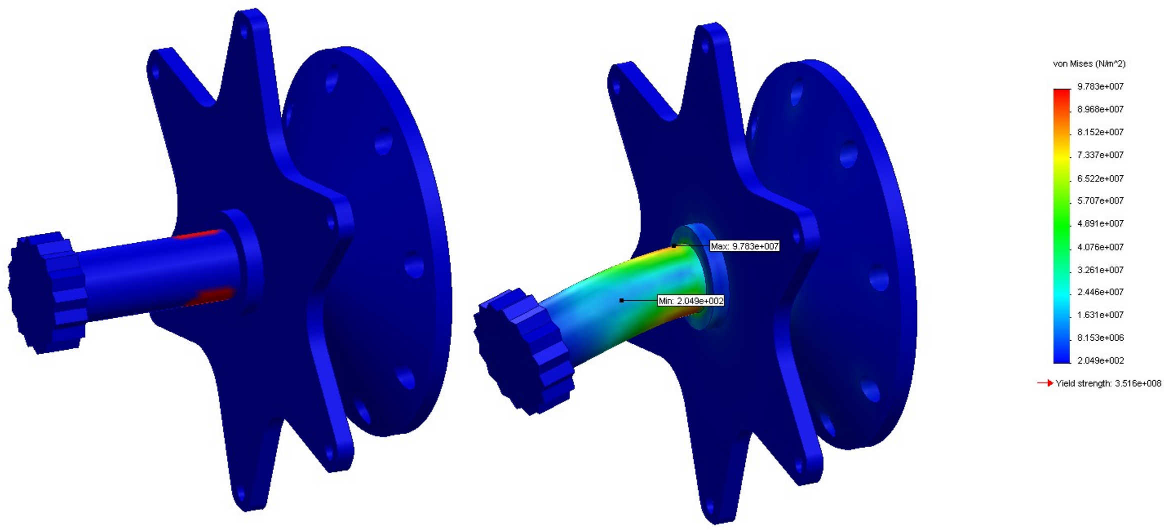


Illustration 36: Wheel mount safety factor (red shows below 5) and Von Mises

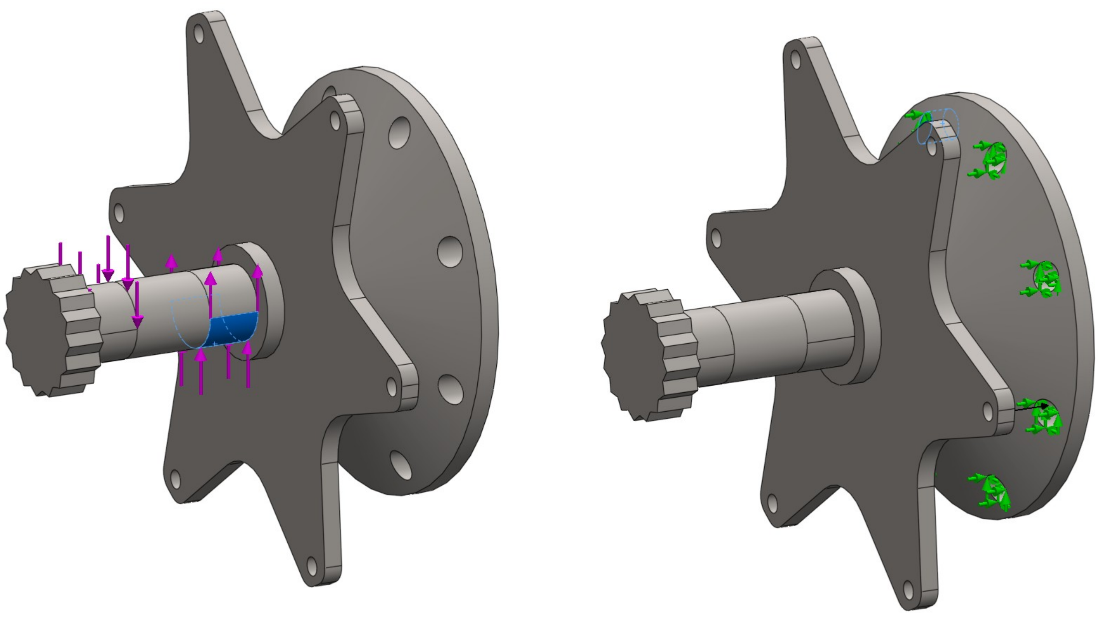


Illustration 37: Forces and restraints on drive-shaft during bending

4.11.4.1.1 Safety factor

The lowest safety factor in the drive shaft and wheel mount under bending due to the maximum expected load on the wheel was found to be 3.59 and the diagram shows in red safety factors below 5. The forces that are on the upright and the forces causing bending in the drive-shaft are the same and due to this have the same uncertainty so fall under the range of 3-4, the achieved minimum safety factor of 3.59 satisfies this.

4.11.4.1.2 High stress

As mentioned before the highest levels of stress occur in the shaft where it mounts to the front bearing. Due to the limitation imposed on the shaft there was nothing that could be done to reduce this level of stress except to change the material.

4.11.4.1.3 Low stress

There are no significant regions of low stress from where material could be removed.

4.11.4.2 Torsion

This section provides torsional analysis of the wheel mount and the drive shaft due to the forces of acceleration and braking with these forces having been derived in section 4.9.6 & 4.9.7.

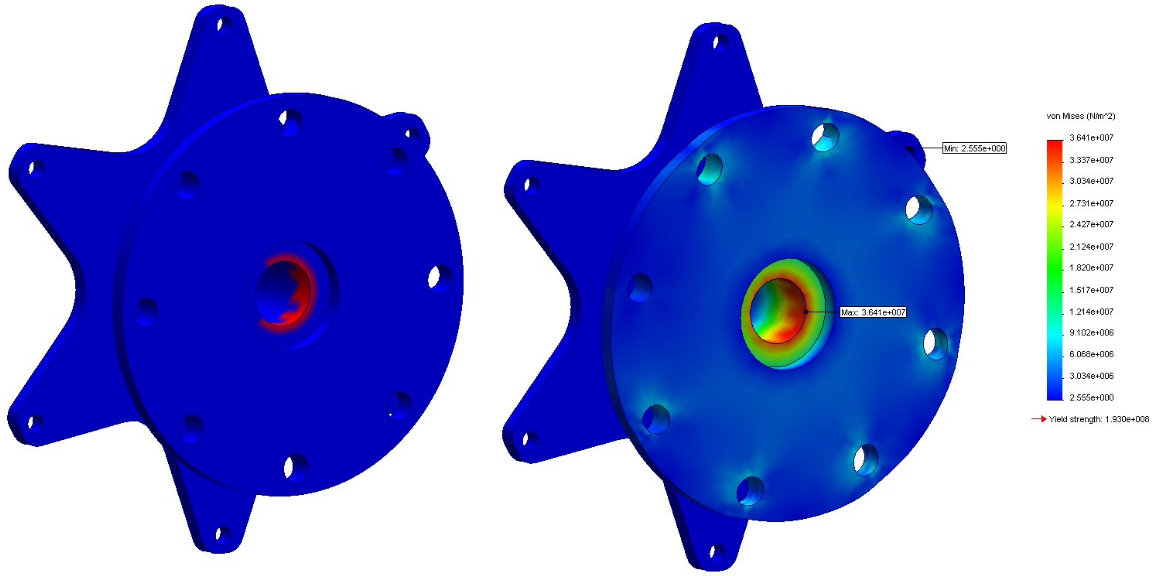


Illustration 38: Wheel mount safety factor (red shows below 6) and Von Mises

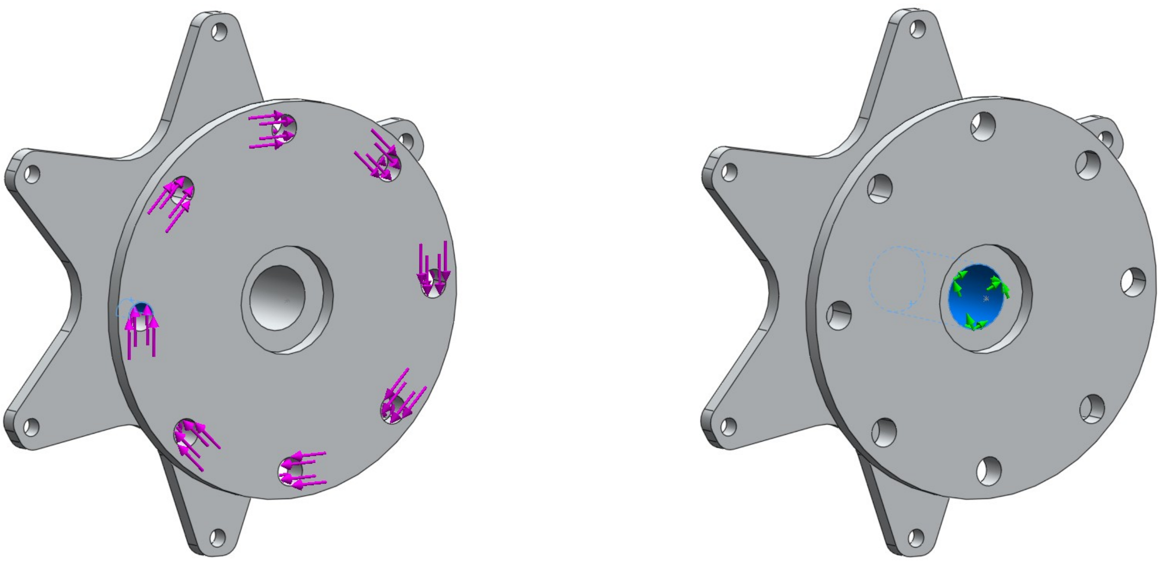


Illustration 39: Forces and restraints on wheel mount

4.11.4.2.1 Safety factor

The lowest safety factor of 5.3 occurs where the aluminium wheel mount attaches to the steel shaft, the Illustration 39 shows where the safety factor is below 6 to highlight these stresses. The torsion in the wheel mount is due to the torque transmitted through the shaft from the gearbox, the maximum torque that the gearbox is rated for is 200Nm. Due to the torsional force on the wheel mount being well understood a correspondingly lower safety factor is acceptable.

4.11.4.2.2 High stress

The highest levels of stress occur where the aluminium wheel mount joins to the steel shaft, this will either be a combination of an interference fit and a keyed fit or a geared fit.

4.11.4.2.3 Low stress

There are no significant regions of low stress that can be removed .

4.11.5 Drive shaft

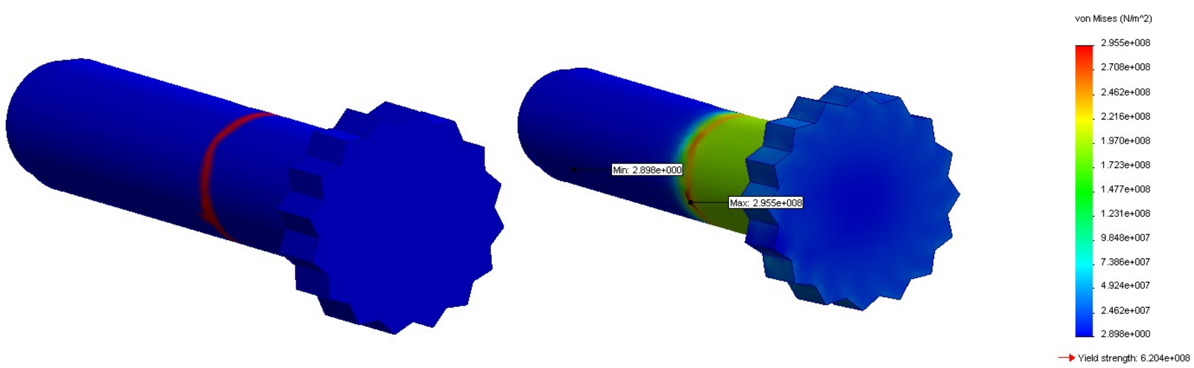


Illustration 40: Shaft safety factor (red shows below 3) and Von Mises stress distribution

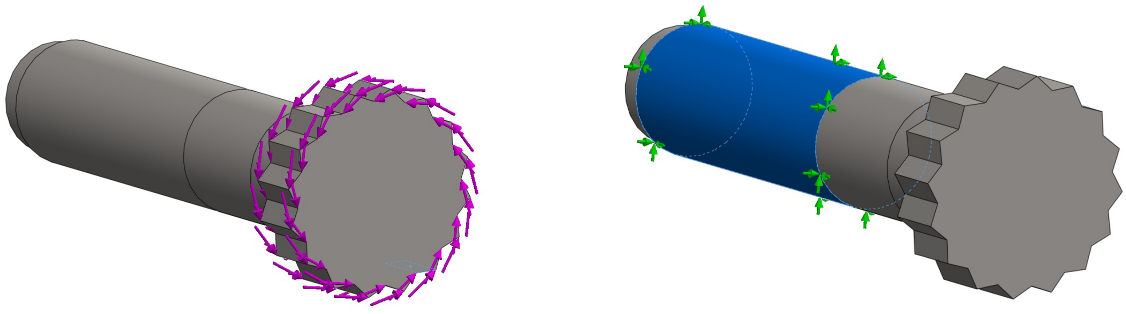


Illustration 41: Forces and restraints on shaft during torsion

4.11.5.1 Safety factor

\

The lowest safety factor of 2.1 occurs where the drive shaft joins with the wheel mount and illustration 41 shows in red regions where the safety factor is below 3. As stated previously the forces acting in torsion are well understood so the correspondingly lower safety factor of 1.5-2 is acceptable.

4.11.5.2 High/ Low stress

There are no regions of high or low stress that could be modified to improve the safety factor, except from changing the type of material.

4.12 Final Model

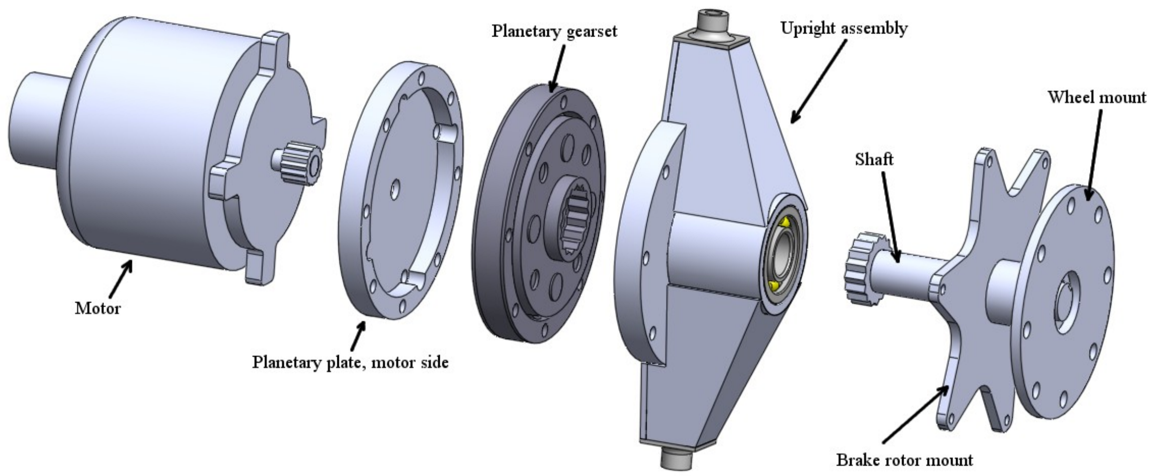


Illustration 42: Final assembly exploded view

Illustration 42 shows the exploded view of the final design and Illustrations 43 & 44 show how this design will mate with the existing suspension and wheels of the 2001 formula SAE car.

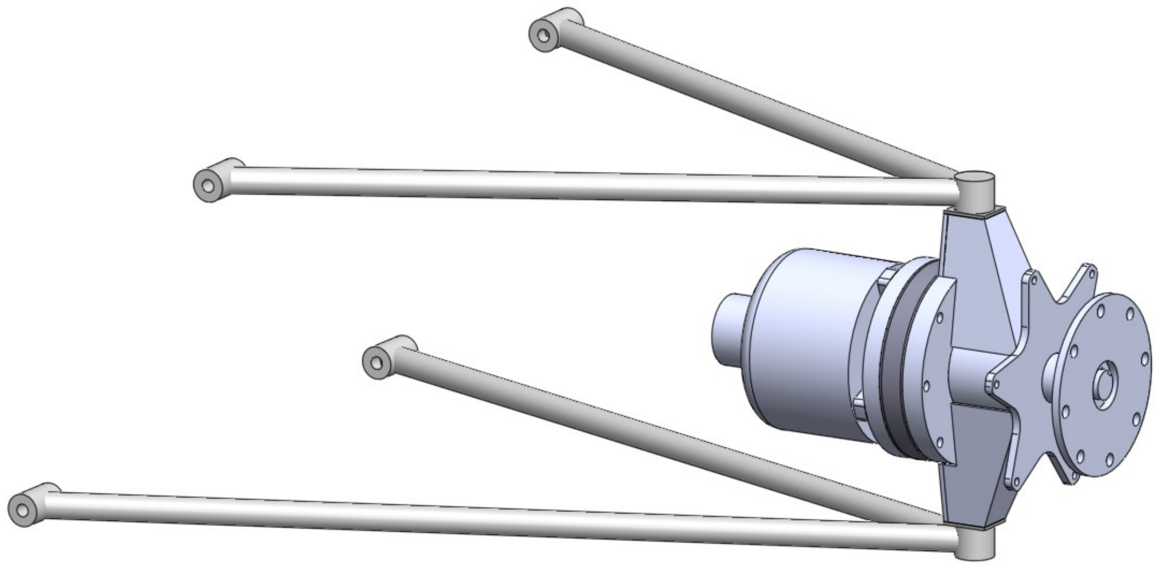


Illustration 43: Final assembly incorporated into suspension

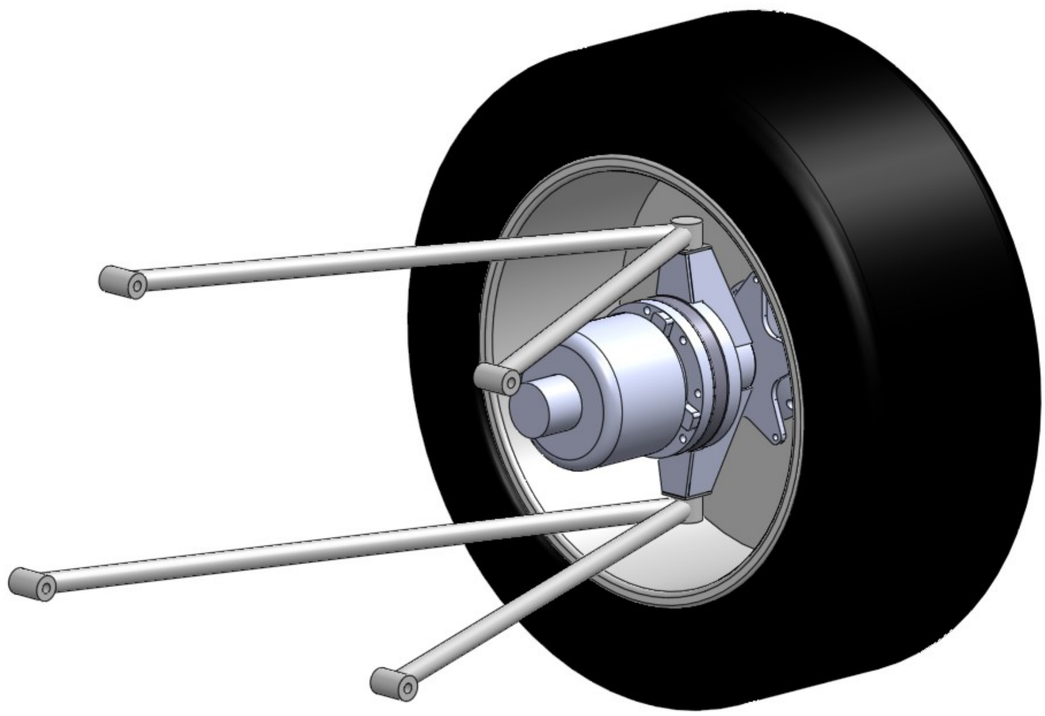


Illustration 44: Final assembly shown with suspension and wheel

5 Manufacturing

All the components were designed in mind to how they would be manufactured, the processes selected are a combination of machining on the lathe, laser cutting and welding, the components manufactured by each approach are listed below.

5.1 Lathe

- Bearing housing.
- Planetary plates.
- Suspension mounts.
- Wheel mounts.

5.2 Laser cutting

- Upright horns.
- Assembly jig

5.3 Assembly guide

An assembly guide was designed to ensure the correct alignment of all the components during welding. The jig shown in illustration 46 was designed to bolt into the planetary plate and align the upright horns as well as the suspension mounts.

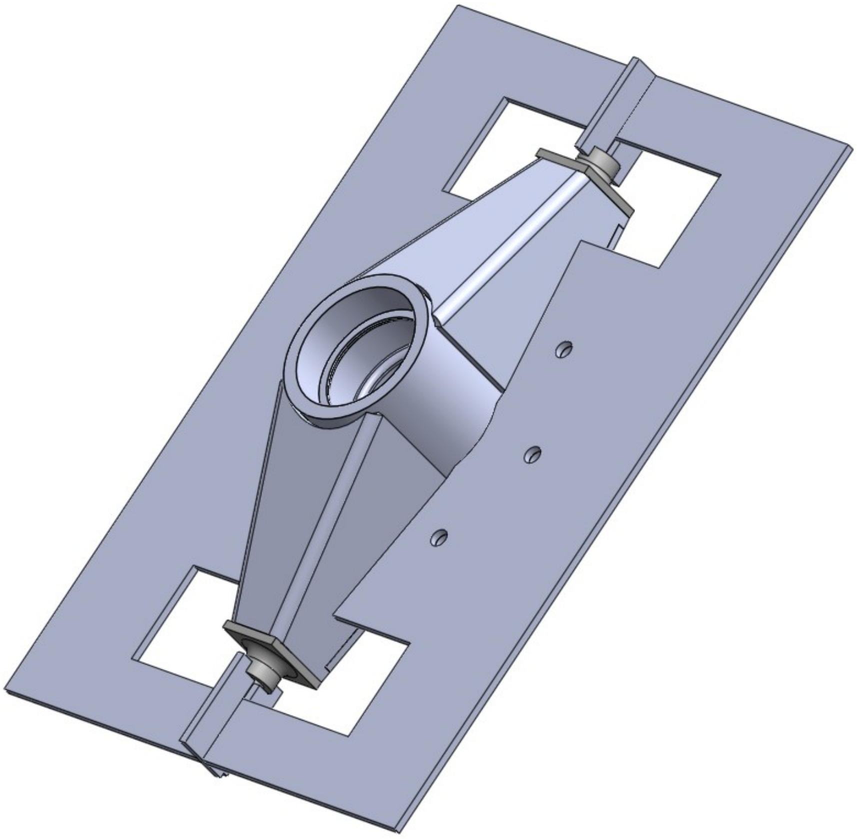


Illustration 45: Assembly jig

5.4 Assembly procedure & construction

- Planetary plate is machined in the lathe.
- Bearing housing is machined in the lathe, but spare material is left inside to allow for distortion during welding.
- Suspension mounts are machined in the lathe and then made square with a band saw.
- Bearing housing is welded onto the planetary plate
- Using the assembly jig the upright horns are welded onto the bearing casing and planetary plate
- This assembly is put back in the lathe and the inside surface of the bearing housing is machined to its final size.
- Using the assembly jig the suspension mounts are welded onto the upright horns.

Illustration 46 & 47 show a labelled view of the upright assembly in the jig prior to welding

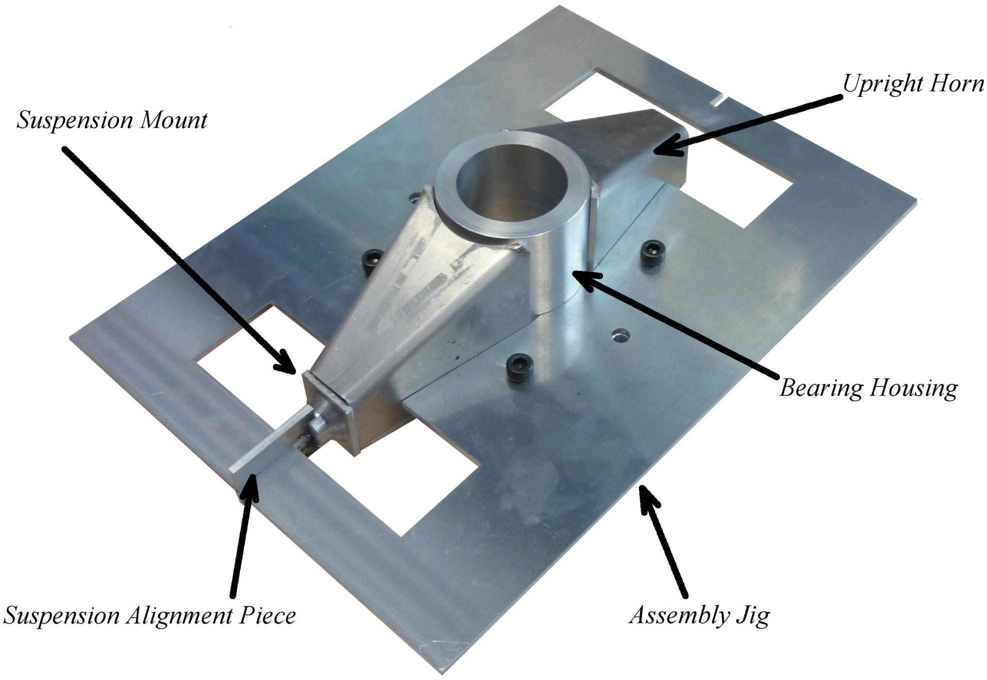


Illustration 46: Assembly before welding front

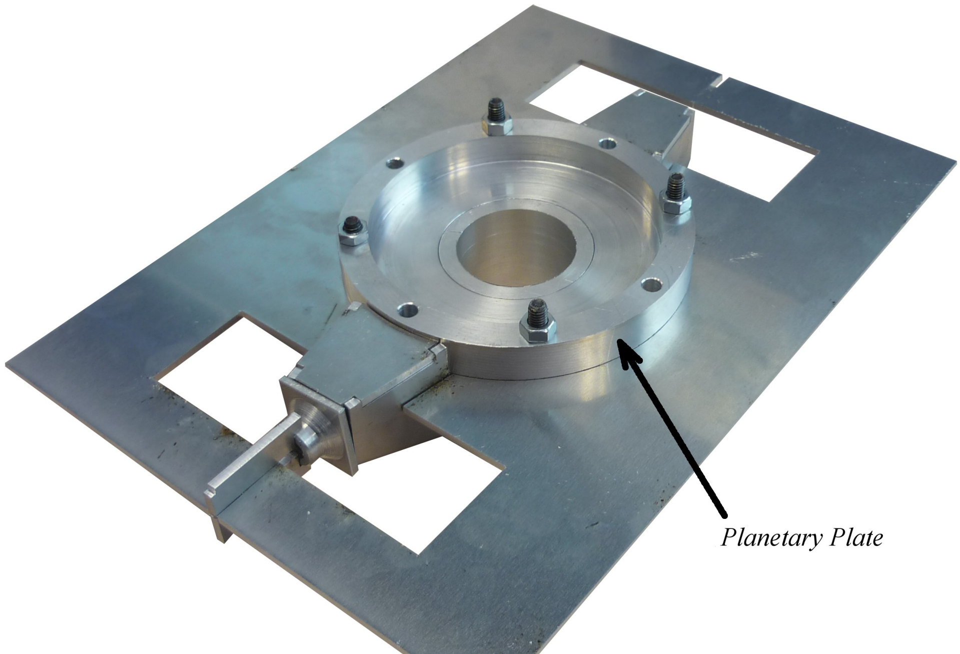


Illustration 47: Assembly before welding rear

Illustration 48: Assembly after welding

6 Testing

The design was not constructed in time to allow testing, so testing will begin next semester.

7. Conclusions and Future Work

This project aimed to design the front wheel drive system for a formula electric SAE car that would eventually form part of a car for competition. In doing so this project also aims to promote electric vehicle technology and build a car that is able to attain the idol status that petrol race cars currently have. This project has successfully designed the front wheel drive system for a formula electric SAE car and the upright assembly has been built. This provides the REV project with a full spectrum of design types from a centralised motor, on board and in wheel motors. As well during this project has seen the introduction of recharging stations in Perth for electric vehicles with the help of Thomas Bräunl (UWA 2009). Following on from this project there is still more work to be done to get a functioning Formula SAE car for competition, firstly advocacy work needs to be done to establish an electric Formula SAE competition, without this a car can only be entered once into the hybrid competition. With this project forming part of the yet to be constructed 2010 Formula SAE electric car there is still a lot of work to be done with Paul Holmes working on the torque control, Ian Hooper working on an evaluation of the different in wheel motor designs not covered by this thesis and a new influx of students to take up the rest of the tasks.

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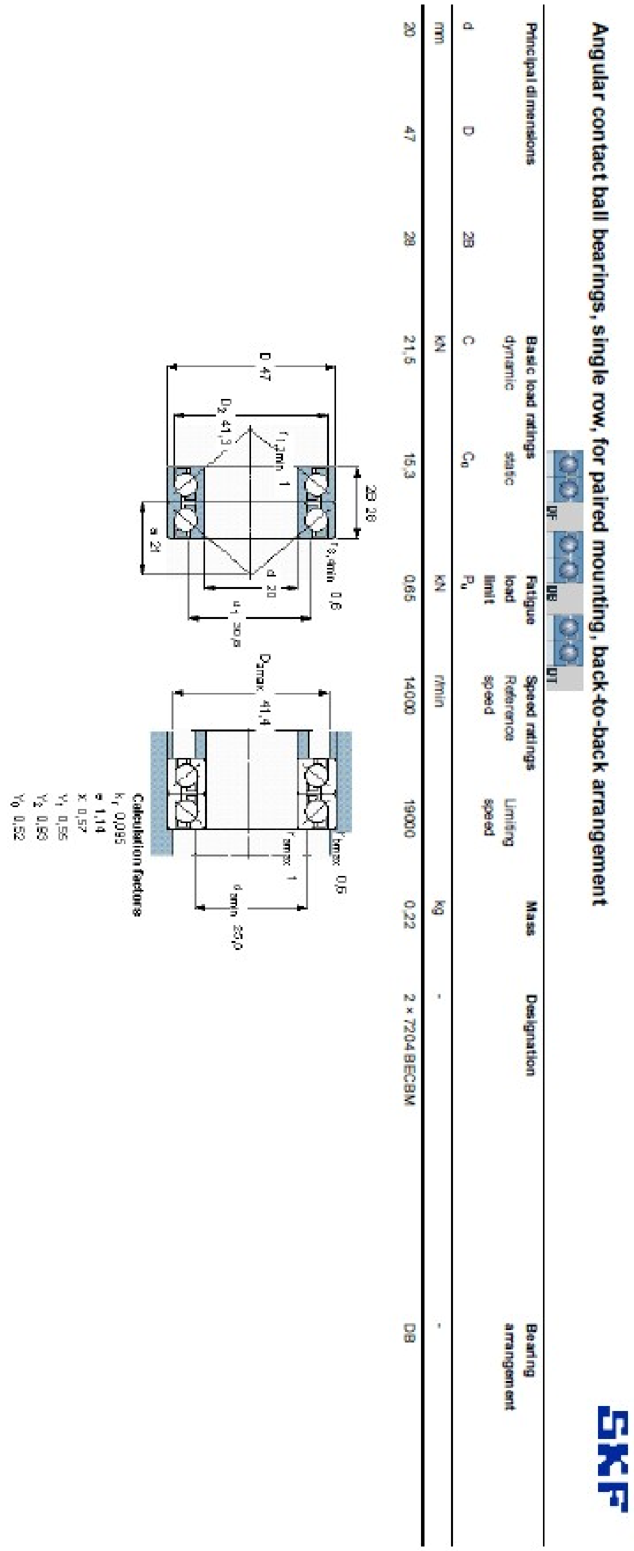
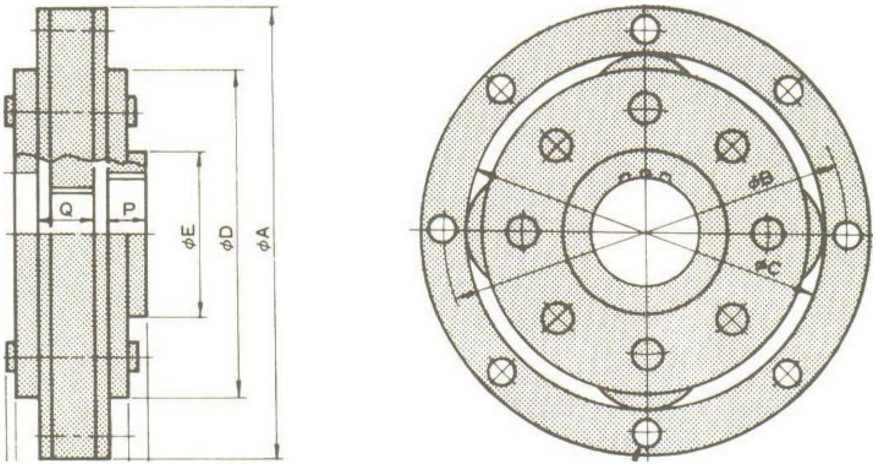
9. Appendices

<u>bearing data</u>

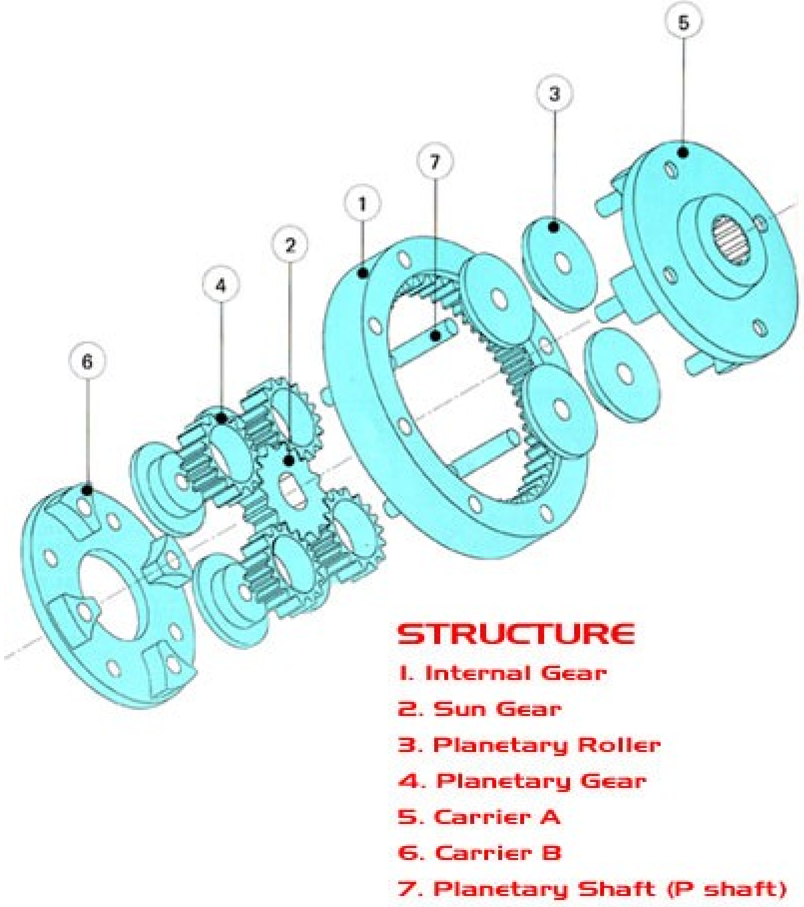


Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Bearing arrangement
d	D	2B	C	C_0	P_u					
mm			kN		kN	r/min		kg	-	-
20	47	28	21,6	15,3	0,65	14000	19000	0,22	2 × 7204 BECBM	DB

planetary data

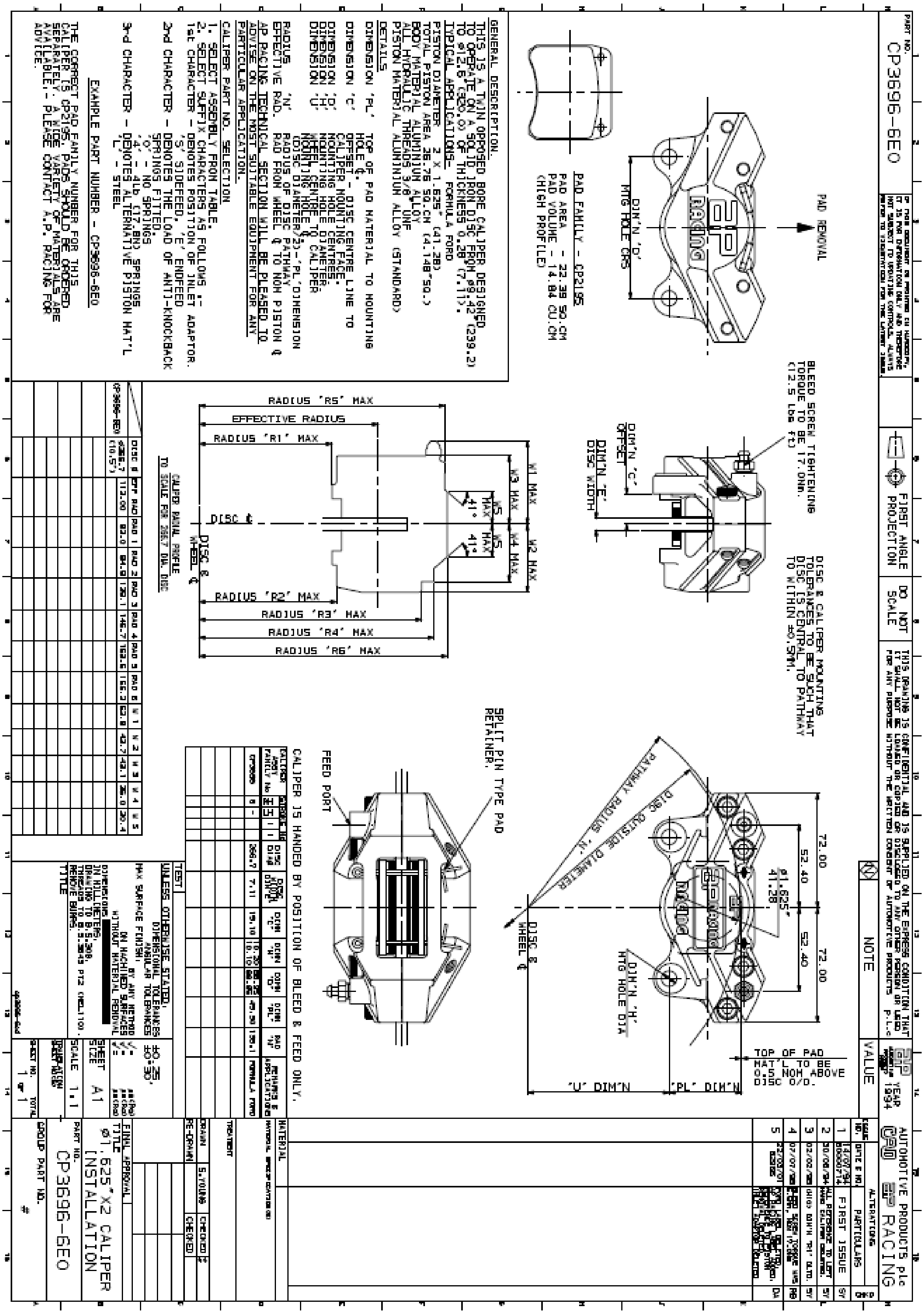


Specifications	
Maximum Torque	2000 kg-cm
Maximum Torque	1739 lb-in.
Nominal Reduction Ratio	5:1
Actual Reduction Ratio	5:1
Dimension - A	120.00 mm
Dimension - B	109.00 mm
Dimension - C	99.00 mm
Dimension - D	94.00 mm
Dimension - E	40.00 mm
Dimension - F	32.00 mm
Dimension - J	8 x 6.5 mm
Dimension - L1	29.80 mm
Dimension - L2	16.00 mm
Dimension - L3	26.30 mm
Dimension - M	10.30 mm
Dimension - N	3.50 mm
Dimension - O	3.20 mm
Dimension - P	10.00 mm
Dimension - R	1.00 mm
Weight	1330 grams
Moment of Inertia	51.79 kg-cm^2
Female Spline - Size (Input)	17.00 mm
Female Spline - Teeth Number (Input)	15
Female Spline - Module (Input)	1.00 mm
Female Spline - Size (Output)	30.00 mm
Female Spline - Teeth Number (Output)	16
Female Spline - Module (Output)	1.67 mm
Dimension - Q	13.00 mm


5
3
7
1
2
4
6
STRUCTURE
1. Internal Gear
2. Sun Gear
3. Planetary Roller
4. Planetary Gear
5. Carrier A
6. Carrier B
7. Planetary Shaft (P shaft)

<u>brake</u>


AP RACING


CP3696-6E0
PAD REMOVAL
PAD FAMILY - CP2195
FIRST ANGLE PROJECTION
DO NOT SCALE
NOTE
VALUE
AUTOMOTIVE PRODUCTS plc
AP RACING
Ø1.625"X2 CALIPER
INSTALLATION
CP3696-6E0