Introduction of Paddle Shifter in Formula Student Car Following with Design and Analysis of Wheel Assembly and Suspension System

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Abstract: **A Formula student car is designed, manufactured, etc by undergraduate and graduate students who compete in various international and national formula racing events. The project is regarding designing and manufacturing quick gear shifter known as Paddle Shifter which is to be mounted in-line perpendicularly with steering wheel so the driver can fully focus on the race track while shifting gears quickly and smoothly. Also designing and assembling of wheel and suspension units on the car following all vehicle configurations and proper wheel base, calculations are done. All the suspension & paddle shifter mounts viewed directly on body frame at technical inspection.**

This paper is divided into 3 parts: (1) Paddle Shifters, (2) Suspension System, and (3) Wheel Assembly.

Keywords: **Paddle Shifter, Wheel Assembly, Suspension, Racing Car**

1. Paddle Shifters

1.1 Introduction

Paddle shifters allow for quick gear shifting, so you will usually find them on sportier cars or sport trim levels. Typically, you will find them near the steering wheel or right behind it. One of the paddles is used to bring the car down a gear, while the other paddle brings the car up a gear.

Why to use paddle shifters?

"The paddle shifters are more about fun and engagement, rather than function because the [automatic] transmission shift logic is so good," Erich Heuschle, an engineer at Fiat Chrysler Automobiles said. "We put so much effort into making the automatic behave well."

1.2 Theory

While cars equipped with manual transmissions gradually become outdated, many cars manufacturer still integrate the feature allowing drivers to shift their own gears. How it can be implemented in automatic cars? The answer is simple – easy-to-use and efficient paddle shifters.

It is not just some sophisticated technology that looks awesome in sports cars, as today it is more and more frequently used as the method of controlling the engine power in a car. The paddle shift system is the main element in a semi-automatic vehicle control system. This mechanism can be one of the most enjoyable ways to drive a car. In this article, we will take a closer look at this remarkable technology, its design peculiarities, principle of operation, and benefits that allowed this remarkable system to gain popularity among auto enthusiasts.

1.2.1 Origins

The idea of using this handy gear shifting mechanism was firstly implemented in motorsports, namely in F-1. In the late 1980s, Ferrari became the 1st team to enjoy benefits of this technology in practice. After proving its efficiency and ease-of-use, this technology was adopted by other F-1 teams, and after a while paddle shifter appeared in conventional road cars.

1.2.2 Design and Principle of Operation

In fact, the paddle shifting system comprises of two plastic levers, by clicking on which drivers can change gears. Paddle shifters are intended for convenient and gradual upshifting or downshifting, and they are frequently labelled with a + and - sign. It can be said that paddle shifters emulate a clutch less manual gearbox.

The benefits of this technology manifest themselves during dynamic driving, for instance, when it is necessary to accelerate while overtaking other cars. Commonly, automatic transmissions shift gears in normal mode, while the use of shifters allows increasing the RPM rate, shifting gears at the peak power and consequently boosting the acceleration intensity. Considering that a lot of road accidents occur during overtaking maneuvers, paddle shifters really seem like an extremely helpful solution for auto enthusiasts.

Pressing on paddle shifters initiates transmission of a signal to the TCU unit, which actuates the solenoids of the automatic transmission, leading to changes in gear ratios. Drivers select only the moment of shifting, and the actual process of shifting gears is performed by electric drives controlled by the TCU unit. It should be noted that paddle shifters can be placed anywhere inside the cabin, but the most convenient installation place is the steering column.

The paddle shifting system in automatic transmissions can be activated in different ways. In some cars, you can start using paddle shifters instantly, in others you must firstly select the manual gear shifting mode and then use shifters. If the paddle shifting system is not used for a long time, then special systems will switch the transmission into normal operation mode.

Most commonly, the paddle shifting mechanism can be found in dual-clutch transmissions, as in standard automatics this system can be found less frequently.

Special adapters connect the transmission to the drive unit, and the decoupling moment is detected by the sensor under the clutch pedal. Engineers from Master Shift installed special systems, locking the e-drive while engaging the R gear, when the car moves forward.

1.2.3 Paddle Shifters for Manual Transmissions

As we have mentioned above, paddle shifters are mainly used in different types of automatic cars, but engineers of the American company Master Shift came up with a promising paddle shifting concept for manual cars. This invention allows replacing a standard manual gearbox to a gear shifting system with paddle shifters. At the same time, a car with this system preserves 3 pedals, but a traditional stick shifter disappears.

Special adapters connect the transmission to the drive unit, and the decoupling moment is detected by the sensor under the clutch pedal. Engineers from Master Shift installed special systems, locking the e-drive while engaging the R gear, when the car moves forward.

1.3 Material Selection

The paddle shifters are made of aluminum, and they're super satisfying to touch. Because column-mounted paddles are inherently larger than the ones on the steering wheel, they need to be strong, and plastic simply won't do.

1.4 Manufacturing Cost

The Smart Fortwo is the cheapest car with paddles, and they are actually controlling a sequential manual transmission, the slowest most dim-witted one.

The Fit is just an autobox with a manual mode. GTI and Lancer have paddles controlling their DSG gearboxes.

1.5 Cost Estimation

Paddle shifter for Aluminum Steering Wheel Shift Paddle Shifter Extension for VW Volkswagen Golf 7 MK7 GTI Black is available at $17.83 at amazon.

1.6 Blue Print

1.7 Photo of Paddle Shifter

1.8 Advantages and Disadvantages of Paddle Shifters

Paddle shifting systems integrated in automatic cars ensure the following advantages:

- The main advantage of the paddle shifting system is ease of use and faster shifting time.
- Paddle shifters are extremely helpful when it is necessary to downshift for performing the overtaking maneuver;
- As paddle shifters ensure smooth acceleration, they also help to avoid unnecessary strain on the engine during acceleration, thereby prolonging the vehicle service life;
- This system also ensures better safety in different weather conditions (rain, snow).

When it comes to disadvantages of this system, they are non-critical, but there are still a couple of them. First of all, the paddle shifting system is quite expensive. Another drawback is that with this system it is quite complex to reduce the speed smoothly when compared to manual cars. Drivers, who previously had only a manual car, frequently cannot get used to paddle shifters and stay displeased with this technology.

Paddle shifters turn automatic transmission cars into semi-automatics. Originally only available in sporty vehicles, paddle shifters are becoming a common sight in automatic cars. As major automakers gradually give up on manual transmissions, paddle shifters can become a real solution for drivers who prefer to be in charge of the gear shifting process.

1.9 Conclusion

Paddle shifter makes car last longer. Cars that drive with paddle shifters when manual/sports mode is activated have a higher chance of accelerating smoothly. Semi-automatic cars beat a fully automatic and a fully manual car from a slow start because you control the gears and power with zero sentiments.

2. Suspension System

2.1 Introduction

The Suspension system is a device connecting the body with wheels. The motion is constrained by the suspension. All kinds of forces and movements between the wheels and the ground passes to the body through the suspension.

The design of suspension system is an important part of the overall vehicle design which determines performance of the racing car

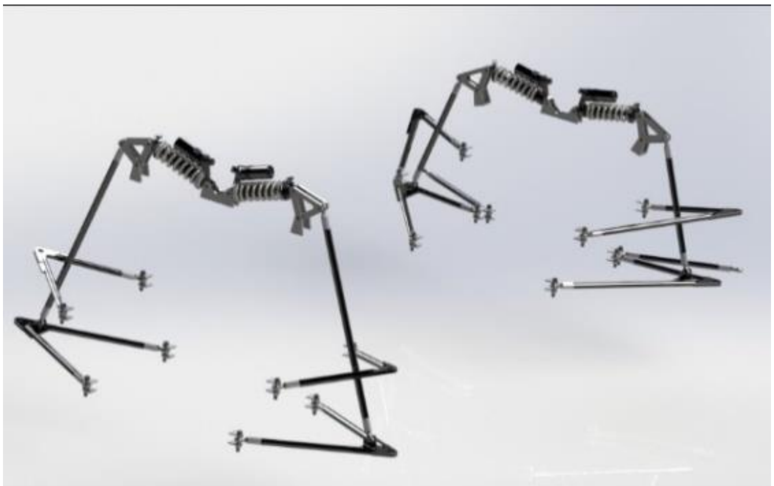


Figure 1: Suspension System

SAE Suspension should have following requirements:

(1) It must have Shock Absorbers.
(2) Suspension travel is not less than 25.4 mm (1 Inch) for both jounce and rebound.
(3) Must have appropriate attenuation vibration ability.
(4) Ensure the car has good handling and stability performance.

2.2 Problem Statement

SAE International hosts multiple Formula competitions worldwide each year. The Formula SAE Collegiate Design Competition is governed by very strict rules and regulations to allow for fair competitions and the safety of the drivers. The rules state very specific parameters in terms of the suspension and wheel assembly design and the maximum choice of the engine; but, it remains broad in other areas such as control mechanisms and aerodynamic design. In general, the rules are tailored to protect the drivers while ensuring ample space to create one's own custom designs.

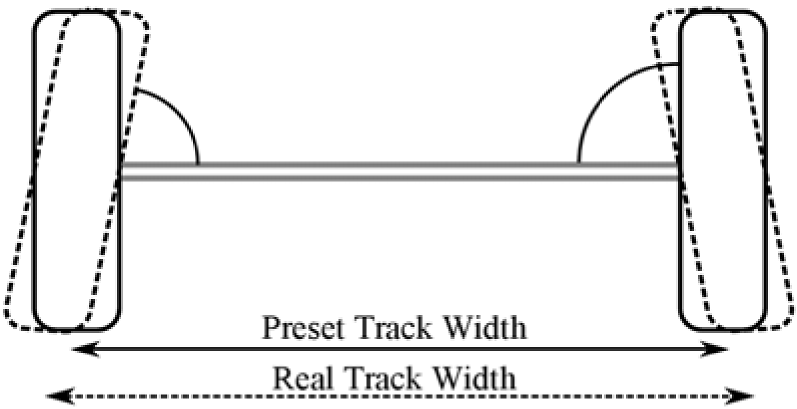
2.3 Methodology

This the very important factor by which a planned method from designing to manufacturing the final prototype. For designing purpose, use of basic hand calculations and research of design parameters referring design reports which are paper published and also certain figures to understand the basics by using internet and referring certain books like Carroll Smith's Tune to Win and Milliken & Milliken's Race car vehicle dynamics, etc.

General Terms in Suspension System and Wheel Assembly

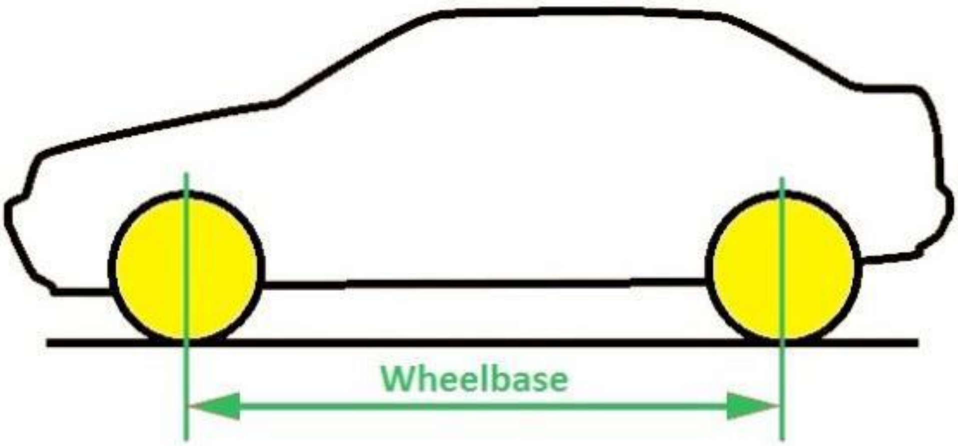
Track-width

The Distance between centre axis of tire from front view is known as track-width.



Wheelbase

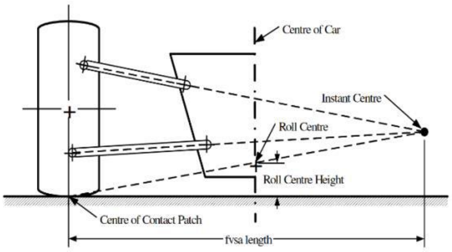
The Distance between the centre axis of Front and Rear Wheel from longitudinal direction is known as Wheelbase.



Instant Center and Roll Center

Instant center is the momentary centre which the suspension linkage pivot around. As the suspension moves the instant centre moves due to the changes in the suspension geometry. Instant centres can be constructed in both the front view and the side view. If the instant centre is viewed in front view a line can be drawn from the instant centre to the centre of the tyre's contact patch.

If done for both sides of the car the point of intersection between the lines is the Roll centre of the sprung mass of the car. The position of the roll centre is determined by the location of the instant centres. High instant centres will lead to a high roll centre and vice versa. The roll centre establishes the force coupling point between the sprung and the unsprung masses of the car. The higher the roll centre is the smaller the rolling moment around the roll center.



Ground Clearance and Rollover Stability

The ground clearance must be sufficient to prevent any portion of the car other than the tires from touching the ground. Intentional or excessive ground contact results in higher C.G. which decreases the rollover stability. The track and center of gravity (C.G.) of the car must combine to provide adequate rollover stability.

Motion Ratio

For packaging damper in the suspension system includes required wheel travel, jounce bump travel, desired wheel rates, strength requirements and packaging constraints. Most important is Motion ratio. Motion ratio is nothing but the ratio of wheel travel to spring travel.

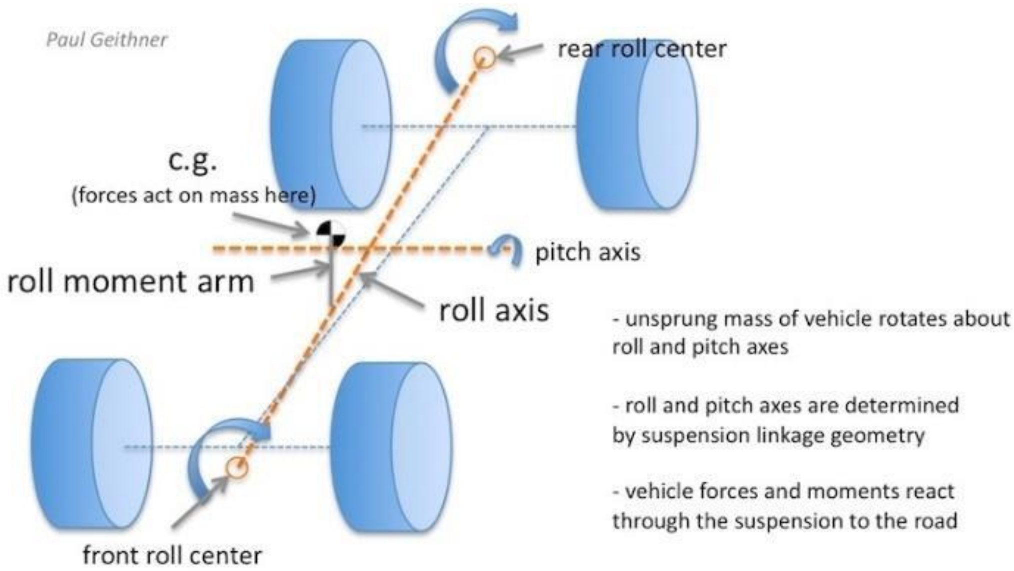
Motion Ratio (MR) = Wheel Travel / Spring Travel

C.G. Height

Center of gravity, also known as center of mass, is that point at which a system or body behaves as if all its mass were centered at that point. Where the weight, and also all accelerative forces of acceleration, braking and cornering act through it.

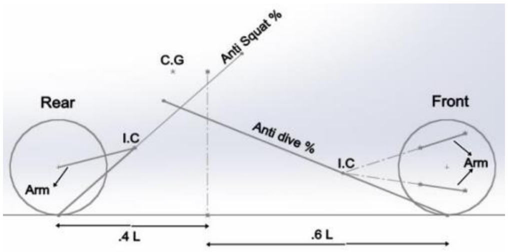
Centre of gravity location can be defined as:

- The balance point of an object.
- The point through which a force will cause pure translation.
- The point about which gravity moments are balanced.



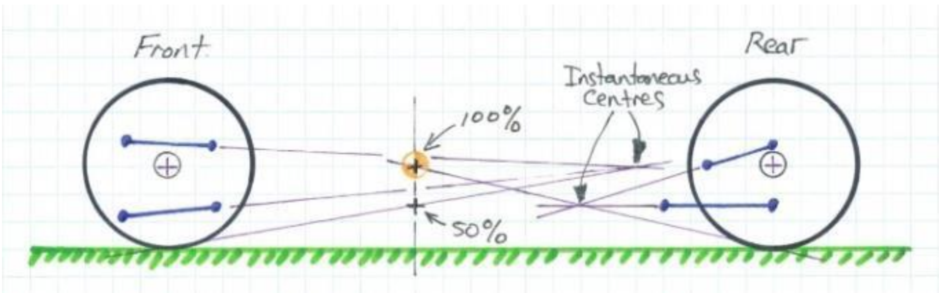
Anti-Dive

Anti-dive describes the amount of front of the vehicle dives under braking. As the brakes are applied weight is transferred to the front and that forces the front to dive. Anti-dive is dependent on the vehicle centre of gravity (C.G.), the percentage of braking force developed at the front tires vs. rear and the design of the front suspension.



Anti-Squat

Squat is a term used to refer to the amount the car tips backwards under acceleration. Over 100% of anti-squat (AS) means suspension will extend under acceleration. With 100% AS suspension would neither extend nor compress. Under 100% AS means tendency to compress under acceleration.



Suspension Geometry (Push / Pull)

Push-rod or pull-rod, the difference as the name suggests is the whether the rod push up to the rocker or pull down to the rocker. The main advantage of a pull rod lie in the possibility to make the nose lower, assemble most suspension parts lower to the ground and thus lowering the height of the centre of gravity.

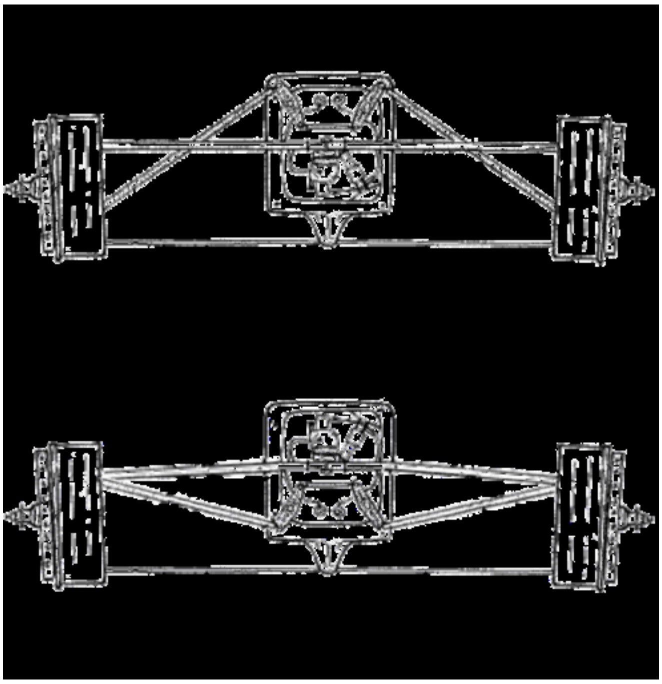


Figure 11: Suspension Geometry (Push / Pull)

Pull rod set up has a strut from the outer end of the upper wishbones that runs diagonally to the lower edge of the chassis and "pulls" a rocker to operate the spring/damper. A push rod is the opposite; the strut runs from the lower wish bone to the upper edge of the chassis.

2.4 Suspension Design Procedure

The suspension design procedure requires several terms selections and values taken into consideration as design of the FSAE starts with suspension. The procedure is as follows.

Suspension Geometry Selection

The selection of suspension geometry type is based on our research work and comparing the advantages and disadvantages of both geometries.

Difference between Push/Pull Suspension

Pull Suspension	**Push Suspension**
Gives lower Centre of Gravity	Centre of gravity is comparatively more
Less Stable at high speed	More stable at high speed
Assembly of bell crank is quite complex	Assembly of bell crank is easy
Aesthetically looks average	Aesthetically looks attractive
Mountings are compact	Mountings are easy

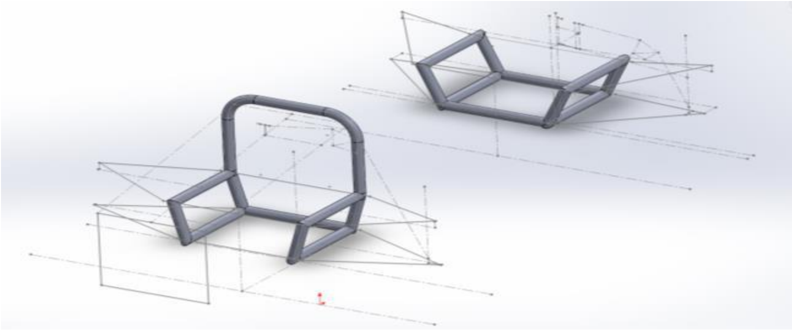
Suspension Compartment Geometry

After Fixing all the general parameters 3D sketch of suspension compartment and also the A-Arms were drafted.

A-Arms Design

A-Arms design started with CAD geometry drawing using suspension compartments and considering track-width, wheelbase, etc. parameters.

Selection of material for A-Arm was done as per Material availability and Machining cost.



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Actual A-Arm

Rod Ends and Spherical Bearing
Rod ends of POS G-8 Male as per bolt size are used for A-Arms mounting on chassis and spherical bearing of LS GE – 8E for A-Arms mounting on Upright.

Bell Crank Design
Bell crank design is quite simple and easy to manufacture using laser cut. In bell crank design firstly geometry diagram is started by which the angle between pushrod and shocks can be checked. The angle between pushrod and shocks is from 80° – 120° for better load transfer.

As one can do good weight reduction in bell crank; bell crank is first drawn by checking the angles between pushrod and shocks and geometry.

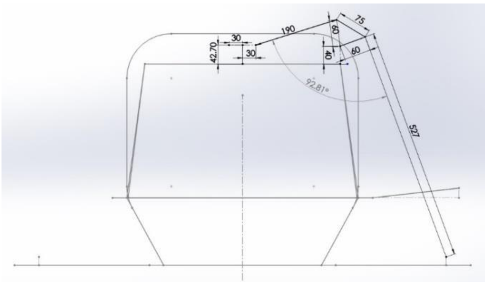

Figure: Bell Crank Geometry – Front

Figure: Actual Bellcrank

By comparing all the parameters of different machining process, it was beneficial to opt for Laser cutting.

Spacer

Spacer is nothing but a circular pipe of certain dimensions to keep distance between two bell crank plates. The final Assembly is done by bolting Bell crank on Chassis. Fasteners used in Bell Crank "Bolts".

Selection of M8 bolts at mountings of suspension Bell crank on Chassis is done and then calculated the loads on bolts and by using some formulation calculations of the bolt size were done. High grade bolts are selected for safety concerns.

2.5 Calculations

Abbreviations

Tr	=	Rare Track Width
Tp	=	Front Track Width
L	=	Wheel Base
y’	=	Lateral shift at Y-axis
W	=	Total weight of vehicle
Wp	=	Weight Of Front Wheel Where Rare is Elevated
B	=	Horizontal distance from rear axis to C.G.
r	=	Tire radius
MR	=	Motion Ratio
Fp	=	Force on push rod
BMR	=	Bell crank motion ratio
Ks	=	Spring rate
Fr	=	Ride frequency
Msm	=	Sprung mass
Kw	=	Wheel rate
kwFL	=	wheel rate front left
kwFR	=	wheel rate front right
KψF	=	Front roll rate
KψR	=	Rear roll rate
FS	=	Force on shocks
Syt	=	Tensile Yield strength
T	=	Sheer Stress
σut	=	Ultimate tensile strength
Pmax	=	Maximum Force on Shocks
Pmin	=	Minimum Force on Shocks
K	=	Spring Stiffness
D	=	Mean Diameter
d	=	Wire Diameter
C	=	Spring Index
N	=	Total number of coils
n	=	Active number of coils
Ls	=	Solid Length
Lf	=	Free Length
p	=	Pitch
α	=	Helix angle
δmax	=	Maximum spring deflection
δ	=	Spring deflection

Dynamic Load Transfer

1. Front = 1.5 × 124 × 0.254 ÷ 1.1938 = 39.57 Kg
2. Rear = 1.5 × 186 × 0.254 ÷ 1.2446 = 56.93 Kg

Masses after Transfer

1. Mfi = 62 - 39.57 = 22.43 Kg
2. Mfo = 62 + 39.57 = 101.57 Kg
3. Mri = 93 - 56.93 = 39.07 Kg
4. Mro = 93 + 56.93 = 149.93 Kg

Centrifugal Forces while Cornering = MV2R

1. Ffi = 350.46 N
2. Ffo = 1587.03 N
3. Fri = 610.46 N
4. Fro = 2342.65 N

During Impact (FOS x M x g)

F.O.S.	**Terms**	**Magnitude**
2	Ff	2432.88 N
	Fr	3649.32 N
1.5	Ff	1824.66 N
	Fr	2736.99 N
1.2	Ff	1459.73 N
	Fr	2189.59 N

STATIC

Sr. No.	Terms	Formula	Magnitude
1.	Ffi=Ff	62x9.81	608.22 N
2.	Fri-Fro	93x9.81	912.33 N

Sr. No.	Terms	Formula	Magnitude
1.	Ff	124x9.81	1216.44 N
2.	Fr	186x9.81	1824.66 N

Force Consideration

Ff=1459.73~1400 N

Fr=2189 N~2200 N

Final Data

Sr. No.	Parameter	Front	Rear
1.	Suspension Type	Double Wishbone Type	Double Wishbone Type
2.	Tire Size	145/70 R12	145/70 R12
3.	Center of Gravity Height	10”	
4.	Suspension Travel	1” jounce / 1” rebound	
5.	Roll Center Height	1.494” Below Ground	2” Above ground
6.	Roll Axis inclination	3.33°	
7.	Motion Ratio	0.7	0.625
8.	Wheel rate	12.57 N/mm	10.74N/mm
9.	Roll Rate	13852.46 Nm/rad	13684.35 Nm/rad
10.	Suspension Frequency	1.4Hz	1.7 Hz
11.	Roll stiffness	76.68	89.99
12.	Camber	0^0	
13.	Castor	-	
14.	Toe In & Out	0 deg toe,	0

		adjustable +/- 2 deg	
15.	Kingpin Inclination	0^0	-
16.	Scrub Radius	2.19”	-
17.	Antidive	49.88%	-
18.	Antisquat	-	74.24%

2.6 Results and Conclusions

2.6.1 Results

- More stability of vehicle is achieved due to negative camber angle as it provides more traction and contact patch to the wheel during cornering.
- Over-steer configuration enables good vehicle handling to the driver by reducing the required steering effort.
- Aerodynamic stability is achieved by provision of low roll center height at the front of the vehicle.
- As the C.G. height is kept near to the ground the rolling effect of vehicle is reduced.
- Anti-dive feature reduces the jerking effect at the time of braking.
- Anti-squat feature reduces the jerking effect at the time of high acceleration.

2.6.2 Conclusion

The purpose of this thesis project is not only to design and manufacture the suspension system for the car, but also to provide an in-depth study in the process taken to arrive at the final design. With the overall design being carefully considered beforehand, the manufacturing process being controlled closely, and that many design features have been proven effective within the performance requirement of the vehicle. The FEA result indicates that the suspension system is able to perform safely in real track condition as per performance requirement.

2.7 References

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3. Wheel Assembly

3.1 Introduction

In automotive suspension, a steering knuckle is that part which contains the wheel hub or spindle and attaches to the suspension and steering components. It is variously called a steering knuckle, spindle, upright or hub, as well. The wheel and tire assembly attach to the hub or spindle of the knuckle where the tire/wheel rotates while being held in a stable plane of motion by the knuckle/suspension assembly in the attached photograph of a double-wishbone suspension, the knuckle is shown attached to the upper control arm at the top and the lower control arm at the bottom. The wheel assembly is shown attached to the knuckle at its center point. Note the arm of the knuckle that sticks out, to which the steering mechanism attaches to turn the knuckle and wheel assembly. Steering knuckle is that component of a vehicle which connects the suspension system, braking system and the steering system to the chassis of the vehicle. A steering knuckle should have high precision, durability and low weight. The purpose of this study is to design a knuckle which is low in weight and has better performance with considerable factor of safety. The study is divided into two steps. The first step involves the designing of a steering knuckle with the help of designing software and by estimating the loads which are acting on the component. The second part is carrying out FEA on the component to find out the stresses induced and the deformation. This will help in optimizing the knuckle. After the analysis is done, the knuckle can be optimized by removal of materials where the induced stress is low.

The objectives for the design and optimization of the SAE Vehicle Upright Assemblies are listed as followed:

1. **Less Weight:** Compare to 2018 design.
2. **Maintains Stiffness:** Achieve the same level of stiffness when compared to the 2016 design.
3. **Maintain Serviceability/Reliability:** Achieving the same serviceability and reliability exhibited with the 2018 design.

These goals can be verified through FEA, physical testing, and actual on track performance of the vehicle. Though for the purpose of the report and due to the importance of the finished product, no actual destructive testing will be performed on the finished assemblies.

3.1.1 Design Consideration

Being a racecar, the primary goal is to achieve the best performance to weight ratio. The reduction of weight in any area will allow for better vehicle performance overall. From basic Newtonian Physics, mass force x acceleration, by reducing mass with a given amount of force capable to be exerted from the vehicle, the acceleration can be maximized. This is true not only for the obvious aspect such as straight-line acceleration based on engine power, but also cornering grip available to a vehicle. As there are a finite amount of cornering grip available from any given tire, it is just as important to reduce the vehicle weight to better exploit the available grip from the tire to achieve maximum amount of cornering acceleration possible. As such, weight is inevitably a key constraint in designing any component in the race car. Weight is also an important consideration for any components in the wheel assembly of the vehicle. As this part of the vehicle weight is defined as 'Gun-sprung weight". The importance of un-sprung weight lies in the fact it dictates the response of the suspension system to any given handling input. The higher the un-sprung weight, and more inertia there is in the given suspension system, and thereby increasing its difficulty to change direction. In the case of the wheel assembly, the spring/damper assembly of each corner is controlling the movement, with dynamic inputs from road surface variation. The goal for the spring and damper is to keep the tire firmly in contact with the road surface, in order to maximize the tire performance. If the inertia of the wheel assembly is high, it will take more time for the system to recover from a disturbance such as a bump on the track, and thereby not allowing driver to exploit the performance from the vehicle. Therefore, for any components in the wheel assembly, weight carries extra significance. Aside from un-sprung weight and inertia, another important aspect in designing any suspension components, and truly in any dynamic mechanical system, is its stiffness. Through the vehicle design process, where a set of goals has been laid out for the target vehicle to achieve, the only way for the vehicle to stay true to its design intent is to ensure that all the key variables that the designer wants gets translated and dynamically maintained in the final product. In the case of the suspension system that means the geometries on paper has to be maintained by the components in the great loads are applied to them. If there are excessive amount of deflection then all the key geometries will not be where the designer intended them to be in a given situation. This is crucially important especially when dealing with various adjustable parameters available on the racecar. As any adjustment an engineer makes he expects to see certain effect on the vehicle. If the stiffness is not there then the desired results cannot be obtained, as the system will not be in the state where the engineer expects it to be. With the above points in mind, the design goal for the vehicle suspension upright assemblies then have to achieve an optimized stiffness to weight ratio. Such that the un-sprung weight in the assemblies will keep its effect to the wheel movement to a minimum and that adequate stiffness is present in the system so that vehicle behavior remains predictable and repeatable in the vehicle development process.

3.1.2 Design Constraints

As with any design, there are a number of constraints that limits the physical layout, material choice, and manufacturability of the upright. The constraints are highlighted in the following sections:

3.1.3 Physical Limits

As the upright assembly exists entirely enveloped by the wheel, its size obviously cannot be bigger than that of the space available in the wheel. Also, the primary driving factors of the upright layout are the designed layout and geometries of the suspension system. The upright has to incorporate all the pivots needed by the suspension and allows the system to move within its designed range of travel without obstructions and cause binding. Other physical limitation includes the choice of the bearing sizes and the amount of suspension adjustments needed to be built-in to the design. The finalized design needs to be within these limits while maintaining the functional requirements of the upright.

Figure 3.1.3.1: Front Assembly Model

In this report the design process for a Formula SAE vehicle will be explored. As many challenges that must be overcome. Several factors will be taken into account. Including vehicle dynamics, chassis rigidity, ponet packaging and overall manufacturing and performance. This project will be split into several phases: manufacturing, testing and validation of the 2017 SAE vehicle and the design. analysis, manufacturing, and finally the test and validation of the 2018 SAE chassis and suspension. All decisions for design were based on all pros and cons from previous SAE testing and competition results. The 2017 vehicle will be tested to acquire data in order to establish which of the design assumptions and past decisions were valid and which were not. It is possible that after design and testing. The components do not perform as expected.The interaction between the various components on the vehicle could be of greater importance than originally estimated. During the second phase, design of the vehicle start a free body diagram, analyzing all of the forces and moments that the vehicle will be exposed to. It will be shown that race cars are practically built from the tires up, due to the fact that maximizing vehicle performance is directly correlated to maximizing tire performance. Tires will be analyzed using the tire data from a test consortium, an, the specific tire coefficients will be plotted to simulate the performance of the tire with the aid of MATLAB.

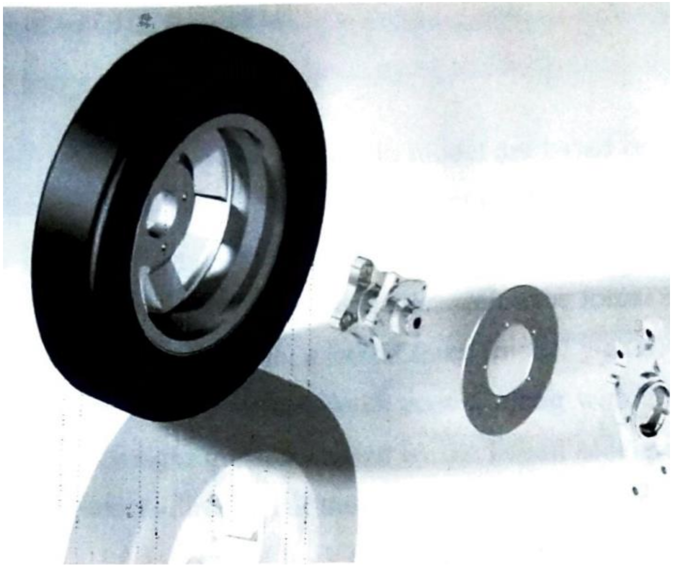

Figure 3.1.3.2: Rear Assembly Model

This allows for a better analysis of the tires, which will ultimately lead to a much better vehicle design. From the tire analysis, the suspension pick-up points are determined, and the chassis is built around it. In the analysis phase, various programs will be used to determine the stresses and the fatiguing of the chassis and suspension. This will prove that reliability isn't as much of a concern to race cars as it is to production vehicles, and that the vehicles are designed more for material stiffness than strength. Manufacturing will be done in workshop, with greater focus on preparation and machining quality than before. In the test and analysis phase, the same procedures used for the 2017 vehicle will be used and improved upon for the 2019 vehicle. Ideally, there are a base set of characteristics of the vehicles that are desired, and it is up to the engineers to allow for the adjustability in the chassis and suspension to attain it. Various methodologies will be followed in order to ensure the stability and performance of the vehicle at different track configurations.

3.2 Literature Review

Ashish Kumar Singh et. al [1] purpose for the investigation was to design and manufacture the SUPRA vehicle upright. The purpose of an upright was to provide a physical mounting and links from the suspension arms to the hub and wheel assembly, as well as carrying brake components. It is a load bearing member of the suspension system and is constantly moving with the motion of the wheel.

Vrushabh Jijode et. al [2] Assembly is an important part of an automobile and its failure is hazardous endangering human life. Therefore, is required to design the Wheel Assembly and its components considering all the factors leading to the failure by developing a safe Design. It must also be noted that, the components must be designed in such a way that they have a minimum weight at the same time care must be taken that they do not cross a certain limit of stress value. In this Paper the Complete Design Procedure of the Wheel Assembly for R12 Rims with wet Tires (165×60) has been presented along with optimization of the same components.

Carroll Smith [3] A more hands on, practical approach to the same subject as outlined by the Milliken text, as well as other sub-systems of the vehicle. As the author is a longtime competitor in various form of racing, many of the points illustrated are more experience based. Again, this is more centered towards the initial design of the suspension system.

Milliken & Milliken [4] provides the much needed science and engineering content into the racing vehicle design. This book is used in the initial suspension system design stage to help analyze and determine the overall design direction of the suspension system in terms of kinematics and dynamic control of the system. The results will the drive the design of the mechanical subsystem such as the upright assemblies.

Carroll Smith [5] book in the "To Win" series of books by the same author. This one Focuses more on the designing of mechanical sub-system of a racing vehicle. This book also devotes a portion of it to material selection.

3.3 Research Procedure

3.3.1 Steps Followed

3.3.1.1 Model Loading

The loads applied to model are based on the data collected in the previous years from the vehicle data acquisition system. The system records the maximum cornering force and this information is used in conjunction with the vehicle layout and weight distribution to determine the forces on the front and rear tires. For the cornering scenario, a lateral force (model y-axis) of 4001bf is applied to the front upright at the contact patch centre, along with a 8001bf of combined bump and lateral weight transfer caused by the lateral acceleration of the vehicle, applied to the vertical direction at the contact patch centre (model z-axis). For the rear upright, the load is scaled back to account for the smaller loads experienced by the rear tire.

3.3.1.2 Model Constraints

The upright model is constrained at the upper and lower ball joint plus the steering/rear toe pickup points. Since all the joints are made with spherical bearing, they do not offer any resistance to moment; their rotational constraints are all left to be free. For the lower ball joint on the race car, it is connected to the lower a-arm and also the pushrod. Under load, the a-arm will resist the movement in lateral and longitudinal direction, while the pushrod will resist the load in the vertical direction. Therefore the lower ball joints are constrained in the model in the displacement in x, y and z axis. For upper ball joint, since there are no pushrod connection, it resists movement only in longitudinal and lateral direction, therefore it is assigned with constraints in x and y axis. For the steering/toe-link pickup, the only link that connects to this joint is either the steering link or toe link. They only resist movement in the lateral direction, so only y-axis is constrained in the model.

3.3.1.3 Model Stress

The FEA package allows for the computation of stresses in different ways, the stresses can be represented in principle stress, component stress, or Von Mises stress. Since it is important to know the yield and material limit, as well as the computation of safety factor, Von Mises stress is used in presenting the stress results. The FEA results are compared against the fatigue strength of the material corrected for a known service life. The correction factors followed that of a standard fatigue calculation and takes into account of load factor, size factor, surface quality, operating temperature, and reliability.

3.3.1.4 Optimization Parameters

The deflection of the upright assembly will be the basis for the optimization process. With stiffness being the performance standard and weight being the concern, the design goals are defined to be reduction in weight over the 2013 design with comparable stiffness. To optimize for weight, thickness for different faces of the upright are changed iteratively based on the previous run's stress distribution and deflection value, the material thickness were reduced in the areas where stresses are low. The limiting factor being stresses cannot exceed the material limit. With available thickness value based on available stock material, a number of combinations were analyzed and the optimum front and rear upright designs were selected as the final designs.

3.3.1.5 Results

The finalized designs and their associated FEA results can be seen in the figures below. Knowing the aforementioned issue with FEA results interpretation in the boundary region of the mating edges between solid and shell element, the focus then is on the region that's around the boundary. As such, the stresses In those region combined with calculated endurance limit resulted in the fatigue safety factor of 1.07 for the front. The value may sound to be too risky, but knowing the conservative estimate for the fatigue cycle, as well as the actual joint design being more robust with multiple weldments, these values should be more than adequate. Based on the FEA model, maximum deflection of the upright assembly based on the given loading condition for cornering and bump is 0.0021" which is better than 0.005" of 2013 design. The gain can be contributed to the closer proximity of the bearing support housing to the outer perimeter of the upright body, since this where the maximum deflection occurs. The resulted design also weighs less in the model from than the 2013 design, due to the material reduction in the less critical area along the upper ball joint. The front upright is 2.031b in the model compared to 2013 model's 2.391b.

Background in the recent years there has been a big improvement in the field of additive manufacturing, structure optimization and surface modeling. It is important to understand each one of this subject, how they work, and when to apply them correctly. This section will discuss the technical details involved and effects they have on each other. This information sets the foundation for the rest of this thesis. In Figure 2-1 the design cycle is shown from my senior design.

3.3.2 Structure Optimization

Structure Optimization is a mathematical approach known as the material distribution method, spreading material in a layer in a given design space for a given set of loads and boundary conditions. This set of optimizations is split into sizing, shape and topology optimization as can be seen in Figure 2-2. Structure optimization is nicely stated by N. G. lyengar "Optimization techniques play an important role in structure design, the very purpose of which is to find the best solution from which a designer or a decision maker can derive a maximum benefit from the available resource".

3.3.2.1 Sizing Optimization

Sizing Optimization is one of the three different types of structure optimization that are used to create a conceptual design. Sizing optimization defines ideal parameters to a component. The parameters are a set of constraints that the components must comply with while at the same time trying to achieve a set goal and or goals. These components are usually defined as a beam. shaft, and plates and so on. Some examples 5 of goals can be lower weight, lower compliance and reduction of cost. The main objective of the sizing optimization is to determine the ideal thickness of any of these components to achieve its goal or goals. The sizing optimization generally comes after shape and topology optimization because it needs initial geometry of the component to be defined before it can be run. Sizing is the last step in the structure optimization process.

3.3.2.2 Shape Optimization

Shape Optimization is one of the three different types of structure optimization that are used to create a conceptual design. Shape optimization defines ideal parameters to a section. The parameters are a set of constraints that the section must comply with while at the same time trying to achieve a set goal and or goals. These sections are usually defined as members, walls or shapes. Some examples of goals can be lower weight, lower compliance and reduction of cost. The main objective of the shape optimization is to determine the ideal shape of any of these sections to achieve its goal or goals. The shape optimization generally comes after topology optimization because it needs initial spread of the material to be defined in a design space before it can be run. Shape is the second step in the structure optimization process.

3.4 Theory

1. Axle Hub

The forging or casting at the end of an axle shaft to which a wheel is bolted onto. The end of the hub usually features splined teeth that will mate with those at the end of the axle shaft. The axle hub rotates with the wheel, and delivers driving force to the wheel bolted onto it. These are also described as wheel hubs.

2. Axle Hub Bearings

A type of roller bearing assembly that fits between the axle shaft and axle hub in order to allow non-drive wheels to rotate freely.

3. Axle Nut

A specially designed nut that fits onto the end of a wheel spindle or axle shaft to secure brake rotors or other hardware in place. Depending on vehicle application, an axle nut may feature grooves that allow cotter pins to be used for attachment purposes. If the nut's sides, or flats, have become damaged these are inexpensive to replace.

4. Axle Shaft

A shaft on which a wheel revolves, or that revolves with a wheel. Also known as a half shaft. Axle shafts come in many varieties, depending on vehicle application.

5. Axle Shaft Bearing

A type of roller bearing assembly located at the point where an axle shaft goes into the vehicle's differential or transmission. These keep an axle shaft supported and aligned properly while allowing it to rotate without friction buildup.

6. Axle Shaft Seal

A round seal located at the point where an axle shaft goes into the vehicle's differential or transmission. It prevents fluid from spilling out of the gearbox as the axle shaft rotates. In some vehicles, the axle shaft seal also helps to keep the axle shaft in proper alignment. These are natural wear items and should be replaced any time they begin leaking, or when a shaft is removed for other reasons. Also known as axle gaskets.

7. Axle Shims

These are thin pieces of metal that are placed between axle components to adjust free play. It may be typical that the shims are available in a variety of thicknesses in order to accommodate the required free play.

8. Axle Support Bushing

These are rubber bushing pieces designed to dampen vibrations that occur naturally when an axle shaft rotates within an axle housing assembly. Your vehicle may or may not have them - but if so, replace them any time axle work is done to ensure vibrations don't creep into your driveline over time.

9. Axle Vent

Also known as axle breather tubes, these check valves allow for pressure changes inside a differential or transaxle that the axle shaft goes into. As gear oil heats up during driving, pressurized air and moisture is allowed to escape.

This relieves excess pressure buildup when that could blow out axle seals. Depending on vehicle application, axle vents may locate directly on the differential housing, or very close to it on the axle housing assembly. If the vent piece becomes clogged or cannot function, moisture can become trapped inside the differential or transaxle - also contributing to breakdown of gear oil. It's important to replace these pieces whenever differential gear oil is changed, or when other related repairs are made.

10. **CV Joint (Constant Velocity Joint)**

A universal joint found on one or both ends of axle shafts fitted to vehicles with independent suspension setups. A ball-shaped piece on the axle shaft end(s) sits within cup-shaped piece attached to shorter shafts leading to the wheels or center mounting point. Ball bearings situated in between allow the CV Axle to change angles - pivoting at the joints as necessary when wheels articulate up and down over bumps. CV joints are mostly found on frontwheel-drive vehicles because they can best accommodate steering angle variations in addition to up-and-down wheel travel. CV joints are fully encased in a rubber boot in order to prevent the lubricating grease they are submerged in from escaping. Many modern rear-wheel-drive cars with independent rear suspension setups use CV joints at the ends of rear axle halfshafts as well as driveshaft.

11. **CV Axle**

An axle assembly that is sold complete with a CV Joint already attached to one or both ends. These may also be described as CV Axle Assemblies, or CV Drive Axles. They're found on virtually all modern front-wheel-drive cars as well as rear-wheel-drive ones with independent rear suspension.

12. **Differential / Axle Plenum**

A specially shaped piece within some Jeep "Vari-Lok" (a.k.a. "Hydra Lok") 4 wheel-drive differentials that allows progressive lockup of the differential. When one axle spins faster than the other, a pump produces pressure that activates a clutch that partially locks the differential while allowing some degree of slippage to avoid driveline binding.

13. **Full-Floating Axle**

An axle design where the weight of the vehicle is supported entirely by a rigid, non-flexible axle housing assembly that surrounds the axle shafts. Two roller bearings support the weight of the vehicle, and the axle shafts themselves do not support any weight. This allows the axle shaft to be removed without unbolting the wheel. These are usually found on the rear of 4x4s. This type of axle can be identified by a protruding hub that the axle shaft flange is bolted to. A full-floating axle setup is considerably stronger than a similarly sized semi-floating axle.

14. **Semi-Floating Axle**

A type of axle setup that's more common on the rear of many half-ton (and lighter) 4x4s where the axle shaft supports the weight of the vehicle in addition to being the means of propulsion. Unlike a full-floating axle, in which each wheel hub is fully supported by bearings on both sides of the hub, a semifloating axle uses a single bearing. And unlike full-floating designs, a wheel must be unbolted in order to remove the axle. Additionally, a broken axle shaft will result in the wheel coming off the vehicle.

15. **Solid Axle**

A suspension setup where both front and rear wheels are connected by a single, rigid axle housing assembly that runs across the entire width of the vehicle. This does not allow left-and right-side wheels to articulate independently of one another the way an independent suspension setup does. These are commonly found on the rear of older cars as well as in the front and rear of most traditional 4x4s. Also known as beam axles or rigid axles.

16. **Three-Quarter Floating Axle**

An axle design rarely used today where surrounding axle housing supports all vehicle weight, but the axle itself is subject to torque loads only some of the time. Bearings are located between the hub and the axle housing.

17. **Wheel Spindle**

A short shaft on which a wheel rotates. These are typically located on front wheels of rear wheel drive cars but are also found at the rear on vehicles with independent rear suspension setups. Also known as stub axles.

18. **Wheel Spindle Bearing**

Roller type bearing assemblies that fit onto and around wheel spindles. They're located in the wheel hub on disc-brake wheels, and inside the brake drum assembly (which forms part of the hub) on drum-brake wheels.

19. **Computer Software**

1. LOTUS

A suspension kinematics design program used to design the overall vehicle suspension geometry in terms of pivot location. The program also outputs the parameter variation throughout the suspension travel in order to allow the designer to understand the suspension geometry change under dynamic conditions. The upright design parameters are established from the geometries in this program.

2. **CATIA v5 r20**
3D Solid CAD modeling program, used to model the actual upright assemblies based on the Susprog 3D parameters. With accurately modeled assemblies in the program important clearances can also be checked to ensure the design meets the packaging requirement.

3. **ANSYS VI 5.0**
With properly defined loads and constraints this can simulate the stresses and deformation experienced by the assembly. This provides objective basis to analyze the design.

3.4.1 Design of Upright

An upright is one of the key components in vehicle dynamics which connects all suspension components between the wheel and the car. The uprights provide a link between the upper and lower ball joints. The upright connects components for example, the control arms, steering arms, springs, shock absorbers, brakes, tires and at the rear it connects the axles. Since it is a key component it must withstand all forces that the suspension will encounter. The uprights must be strong enough to withstand those forces, sum may occur simultaneously for example during braking into a corner. As for this design, the goal is to design a light, yet sufficiently strong uprights that can withstand the forces that the new FS car will encounter.

The design of the upright involves the consideration of many aspects of the car's suspension. It defines the outer half of the mounting locations of the suspension including both upper and lower a-arms and the toe control. The upright defines kingpin inclination, and at least partly controls Ackerman angle, the scrub radius, and wheel-mounting offset. The front brake assembly, including rotor, hat, and calliper must also be considered during the design of the upright. In addition, all the loads into the car are transferred though the upright so, it must be strong but since it is un-sprung weight, it is also beneficial to have it lightweight.

For these reasons, the design of the upright staffed with the design of other parts of the suspension. Once a general idea of the parameters where established, the upright could be designed. Features of the uprights include: the use of light weight 356-T6 Al, sand casting to lower manufacturing cost, one design for all four wheels, direct mounting of brake callipers, desired kingpin inclination of 70, double shear at lower ball joint mount, large distance between bearings, to reducing required bearing size, and separate steering mount allowing adjustability of Ackerman.

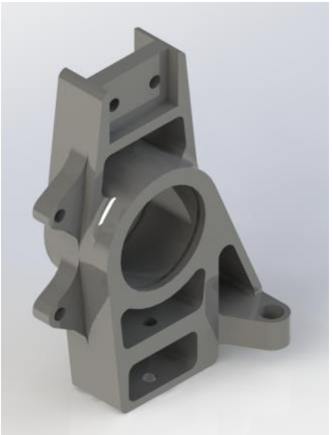

Figure 3.4.l.1: Front Upright Model

The analysis of the upright involved calculating loads at both the suspension mounting points as well as at the bearing mounting points, to select the appropriate bearings. A Statics analysis was used in determining these loads. Once the loads were determined, parameters such as the distance between the bearings, the distance from wheel mount to bearings, the distance between suspension mounts, etc., could be varied and the results noted, choosing the best compromise in the design.

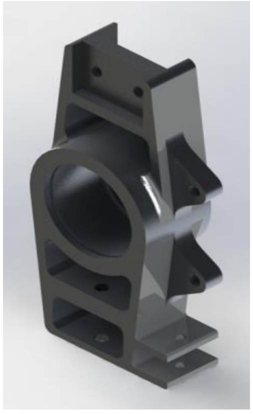

Figure 3.4.1.2: Rear Upright Model

Next, solid models were created using CAT IA v5 r20 and FEA performed using Algor. Bearing life calculations were also performed in the bearing selection.

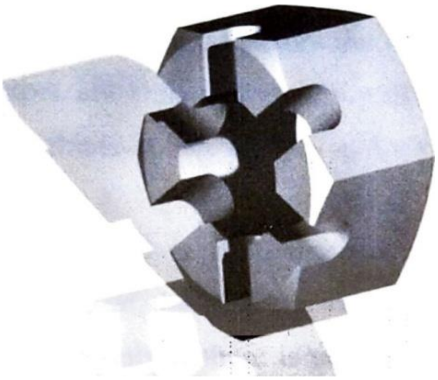

Figure 3.4.l.3: Castle Nut or Central Locking Nut

3.4.1.1 Methods

In order to start the upright design the tire and wheel size must be settled. As mentioned before, the car is aimed to be running on 12" wheels with 4.5" wide at the front and 4.5" at the rear and running on keily tires which are 19" in diameter. Those parameters where set by the suspension team that is conducting an analysis for the suspension using LOTUS software. The brake disc position is the absolute farthest location for the LBJ. In race cars the height of the LBJ is placed as low as possible but since it is placed inside the wheel it has to clear the wheel under all travel and load conditions.

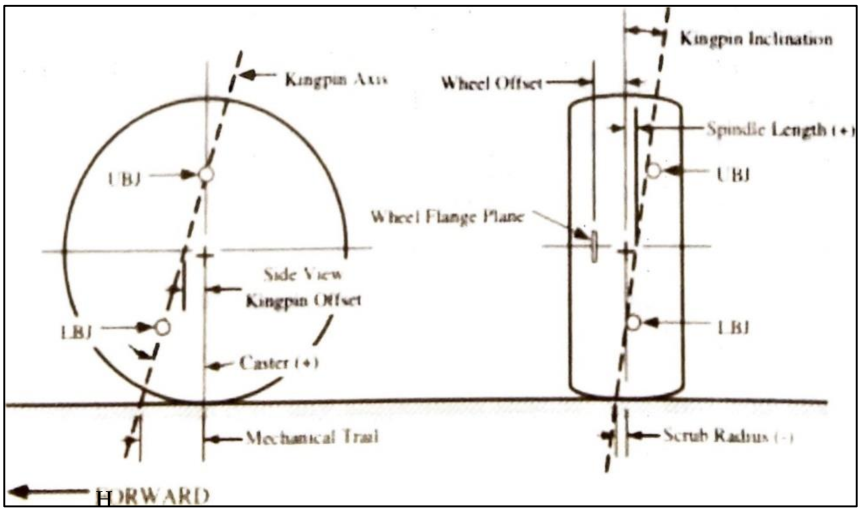

Figure 3.4.1.1.1: Kingpin Geometry, Side View and Front View

The UBJ location is determined by the kingpin axis. The kingpin axis is a line drawn through the UBJ and LBJ down to the tire. The distance that the line is offset from the tire center line is called scrub radius, measured horizontally. If we look at this in front view, the angle that the two lines make is called Kingpin inclination. The spindle length is the distance measured horizontally from the kingpin axis to the wheel center at axle height. These factors are shown in figure and all are interrelated and each affects the behavior of the vehicle in its own way and a compromise is needed. These effects are the following:

1. **Spindle Length:** If the spindle length is positive the car will lift when the wheels are turned. This results an increase of the steering moment at the steering wheel. This effect is symmetric side to side only if there is no caster angle. This raise also has a self-aligning effect that aids centering of the steering at lower speeds.

2. **Kingpin Inclination:** Kingpin angle also effects lift during steering. The more the kingpin angle the more the car lifts when steered. The kingpin inclination affects the camber of the wheel when steered. The camber is a function of the kingpin angle and the caster angle.

As kingpin is increased the wheel loses camber with more steering input thus giving positive camber on the outside wheel. When the wheel is turned it will lean outwards at the top towards positive camber if the kingpin is in normal direction. The amount of this is small but significant if the track includes tight turns. Scrub radius: The braking forces introduce steer torques that are proportional to the spindle length. If there is a difference between braking force on left and right tires there will be a net steering torque felt by the driver at the steering wheel. With zero scrub radiuses this is not true since there is no moment arm for the drive forces to generate torque around the kingpin. In the side view in figure there is a term called Mechanical trail and Caster angle. If the Kingpin axis doesn't go through the wheel center then there is a kingpin offset. The distance between the points where the kingpin axis touches the ground and the wheel centerline is called mechanical trial. The angle that is between them is called Caster angle. These factors affect the handling of the vehicle and are of great importance in any suspension design. The trial and caster produce more self-centering effects and are the primary source of self-centering moment about the kingpin at high speeds. The larger the trial, the more torque is needed for steering.

The caster angle causes the wheel to rise and fall with steer like the kingpin angle. Although unlike the kingpin inclination, the effect of the caster angle is the opposite from side to side and causes weight transfer and roll. This behavior leads to an oversteering effect. The caster angle has positive effect on the steer-camber that is with positive caster angle the outside wheel will camber in a negative direction while the inner wheel cambers in positive direction. This causes both wheels to lean into a turn Camber angle to the road surface is one of the fundamental variables in suspension design but it is the angle between the vertical plane and a tilted wheel plane. Positive camber is defined when the wheel is tilted outwards at the top and negative if the wheel is tilted inwards at the top. The camber angle affects tire performance, along with load, slip angle, pressure and temperature. Camber also works like a steer that is when a tire is in camber angle; it tends to pull the vehicle in the same direction as the top of the tire is leaning. Toe settings can be used to adjust and improve handling difficulties in the car. For example with rear toe-out can be used to improve corner turn-in. It is possible to adjust the load that transfers to the outside wheel as the car turns in and gives the effect of an over-steer direction. In the front the static toe depends on many factors such as Ackerman geometry, ride and roll.

3.4.2 Design of Hubs

Wheel hubs are one of key features in the suspension. Wheel hubs support the lugs of the vehicle and houses the wheel bearing. The purpose of the wheel hub is to connect the wheel to the suspension and keep the wheel spinning freely. If the hub is placed on a drive axle, the power from the motor transfers through the hub to the wheels. The hubs also hold the brake rotor in order to transfer the braking force to the wheels and slow the car. In modern race vehicle, depending on the rules and regulations, the connection of the wheel may be restricted. For example, in a normal street legal car, the lug bolt pattern varies from 4-6 bolts mainly. However in NASCAR racing it is mandatory to use 5 lug bolt pattern. In Formula I teams utilize a central lock wheel hub with only one nut fastening the wheel to the hub. This is done because of the quick pit stops that teams need to perform during a race and also to save more un-sprung weight. Design of hub the vehicle is a rear engine rear wheel drive that indicates that the design of front and rear hubs will be different from each other as the rear hub has to support the driving torque from the axle half shafts through splines in addition to all the impact loads. The hubs also undergo the braking torque the driver hits the brake pedal.

3.4.2.1 Methods

The wheel hub has to be strong enough to withstand the forces acting on it. During a race, there are four main forces acting on the wheel hub:

1. Force due to acceleration or deceleration
2. Cornering
3. Wheel travel or bump
4. Brake torque or torque from the axles

Although central locking wheels use only one lug nut to fasten the wheel, there are tough lug bolts that are used to guide the wheel to the hubs and to receive the torque from the drive train. Regardless of the forces, there are also number of numerical dimensions that the hubs must confirm on. Those numerical dimensions constrain the design and it must under all circumstances match the following:

1. The inner diameter of the wheel bearing
2. The inner diameter of the CL wheel (axle stub diameter)
3. The outer diameter of the CL wheel center (for control pins)

4. Position of the brake rotor and calliper
5. Offset of the wheels

First order of business was to acquire information for the inner diameter of the CL wheels. According to spec. sheet about the CL wheels, the axle stub diameter is — 40mm. The outer diameter of the wheel center is ¯ 12 inch. The dimension that the stub axle needs to clear depends on the offset of the wheel due to the packaging of the brake caliper.

3.4.2.1.1 Front Hubs

The front hubs provides mounting holes for both wheel rim and brake rotor. The rim fits with 4 holes PCD 100 mm and 12mm bolt diameter. A central shaft connecting these mounting points is designed with groves cut for inserting the roller bearings at the two ends of the hub. Front Hub modeled in CATIA v5 r20, After modeling of the hub in Solid Works, the model was meshed in ANSYS WB. For the impact analysis, the central axial shaft was given a roller support and Force of 3g was applied on the mounting holes of the wheel rim in upward direction. Von Mises stress gives the appropriate value of amount of stress induced in component with respect to the force applied. For the simulation of the braking torque, the mounting holes of the brake disc were fixed using rigid support and the braking torque was applied at the mounting holes of the rim along the shaft axis.

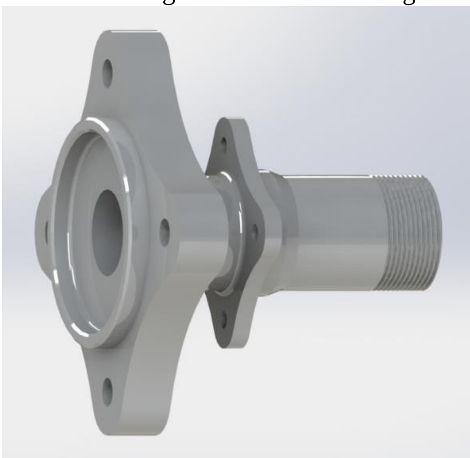

Figure 3.4.2.l.1.1: Front Hub Model

3.4.2.1.2 Rear Hubs

The design of the rear hub is different from the front hub as it is only supported by the splined shaft which is inserted in the rear upright. Although the mountings of disc and wheel rim remain the same. In addition to the impact load during a bump and the braking torque ,the rear hub also supports the propulsion torque from the axle half shafts. This was applied to the inner splined shaft and the mounting holes of the wheel rim were fixed using rigid support The biggest challenge was to use Aluminium instead of steel. Steel is a more common material choice, mostly due to its superior fatigue properties over aluminium. The problem with steel in this application is twofold; first, it is heavy, nearly three times as dense as 6061-T6 Al. The second down side is it is too hard to machine after it is heat-treated, it would require the surface grinding, which cannot be done in the student shop at CSUS. Because of these to drawbacks, aluminium was the desired material.

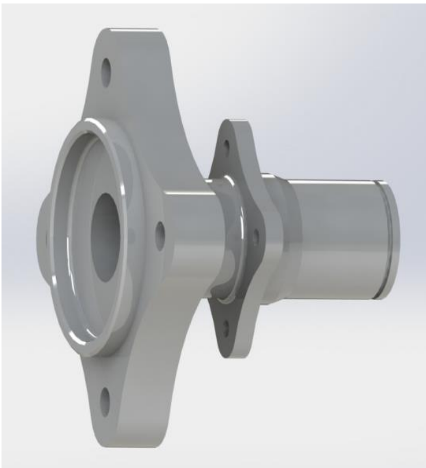

Figure 3.4.2.1.2.1: Rear Hub Model

In order to use aluminum, it had to be shown it could hold up in this application. Fatigue was the biggest concern in the design of the hubs. Fatigue life analysis was performed and it was determined that if the shaft was notched to seat the bearings, it would fail, therefore the a spacer was design to hold the inner race of bearing instead .0nce determined that 6061-T6 Al could hold up in fatigue, a FEA was performed to be sure the deflections were minimized. 7075T6 Al was initially the material of choice but it is more than twice the cost of 6061-T6, so the design was altered slightly for the use of 6061-T6.For the rear hubs, the analysis of splines was also necessary. A spreadsheet was created to analyze the splines, both statically and in fatigue. Comparison of 6061-T6 Al, 7075-T6 Al, and 4340 steel was performed.

3.4.2.3 Sizing Optimization

Sizing Optimization is one of the three different types of structure optimization that are used to create a conceptual design. Sizing optimization defines ideal parameters to a component. The parameters are a set of constraints that the components must comply with while at the same time trying to achieve a set goal and or goals. These components are usually defined as a beam, shaft, and plates and so on. Some examples 5 of goals can be lower weight, lower compliance and reduction of cost. The main objective of the sizing optimization is to determine the ideal thickness of any of these components to achieve its goal or goals. I he sizing optimization generally comes after shape and topology optimization because it needs initial geometry of the component to be defined before it can be run. Sizing is the last step in the structure optimization process.

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Shape Optimization is one of the three different types of structure optimization that are used to create a conceptual design. Shape optimization defines ideal parameters to a section. The parameters are a set of constraints that the section must comply with while at the same time trying to achieve a set goal and or goals. These sections are usually defined as members, walls or shapes. Some examples of goals can be lower weight, lower compliance and reduction of cost. The main objective of the shape optimization is to determine the ideal shape of any of these sections to achieve its goal or goals. The shape optimization generally comes after topology optimization because it needs initial spread of the material to be defined in a design space before it can be run. Shape is the second step in the structure optimization process.

3.5 Material Selection and Analysis

3.5.1 Material Selection

One of the first key design parameter when designing a part for FS competition is the material selection. The factors that we need to consider during our design are cost, strength and weight. In our case for the upright the aim is to design light yet strong parts. The most common material to use is aluminium or steel for the uprights but since this year we are taking a huge step and going to 12" wheels, that leads us to smaller yet lighter uprights but they have to withstand higher forces due to suspension geometry and remain fairly stiff. To limit the unsprung mass as much as possible it is intended to use aluminium 6061 for the uprights because it strength is similar to steel and it is easily to machined.

3.5.1.1 Aluminium 6061-T6

Subcategory: 6000 Series Aluminum Alloy; Metal; Nonferrous Metal Key Words: a16061, UNS A96061; ISO AIMglSiCu; Aluminum 6061-T6, AD-33 (Russia); AA6061-T6, 6061 T6, UNS; ISO AIMgl SiCu; Aluminum 6061-T651, AD-33 (Russia); AA6061-T651

Component	**Wt. %**
Al	95.8 - 98.6
Cr	0.04 - 0.35
Cu	0.15 - 0.4
Fe	Max 0.7
Mg	0.8 - 1.2
Mn	Max 0.15
Other Each	Max 0.05
Other Total	Max 0.15
Si	0.4 - 0.8
Ti	Max 0.15
Zn	Max 0.25

Material Notes

Information provided by Alcoa, Star met and the references. General 6061 characteristics and uses: Excellent joining characteristics, good acceptance of applied coatings. Combines relatively high strength, good workability, and high resistance to corrosion; widely available. The T8 and T9 tempers offer better chipping characteristics over the T6 temper.

Applications: Aircraft fittings, camera lens mounts, couplings, marines fittings and hardware, electrical fittings and connectors, decorative or misc. hardware, hinge pins, magneto parts, brake pistons, hydraulic pistons, appliance fittings, valves and valve parts; bike frames.

1. Physical Properties

Density	2.7 g/cc

2. Mechanical Properties

Ultimate Tensile Strength	310 MPa
Tensile Yield Strength	276 MPa
Modulus of Elasticity	68.9 GPa
Poisson's Ratio	0.33
Fatigue Strength	96.5 MPa
Shear Strength	207 MPa

3.5.1.2 Aluminium 7075

Subcategory: 7000 Series Aluminium Alloy; Aluminium Alloy; Metal; Nonferrous Metal A Zr + Ti limit of 0.25% maximum may be used with this alloy designation for extruded and forged products only, but only when the supplier or producer and the purchaser have mutually so agreed. Agreement may be indicated, for example, by reference to a standard, by letter, by order note, or other means which allow the Zr + Ti limit.

Component	**Wt. %**
Al	87.1 - 91.4
Cr	0.18 - 0.28
Cu	1.2 - 2
Fe	Max 0.5
Mg	2.1 - 2.9
Mn	Max 0.3
Other Each	Max 0.05
Other Total	Max 0.15
Si	Max 0.4
Ti	Max 0.2

Material Notes

General 7075 characteristics and uses (from Alcoa): Very high strength material used for highly stressed structural parts. The T7351 temper offers improved stress-corrosion cracking resistance.

Applications: Aircraft fittings, gears and shafts, fuse parts, meter shafts and gears, missile parts, regulating valve parts, worm gears, keys, aircraft, aerospace and defence applications; bike frames, all terrain vehicle (ATV) sprockets.

1. Physical Properties

Density	2.81 g/cc

2. Mechanical Properties

Ultimate Tensile Strength	572 MPa
Tensile Yield Strength	503 MPa
Modulus of Elasticity	71.7 GPa
Poisson's Ratio	0.33
Fatigue Strength	159 MPa
Shear Strength	331 MPa

We selected the material AL6061 for upright and hub analysis.

3.5.2 Analysis

Finite Element Analysis or FEA is the method used to optimize the design of the 2018 upright. The FEA package of choice for this project is ANSYS v15.0, the FEA suite for the CAD software CATIA v5 r20. This arrangement allows for easy integration between the CAD model to the FEA software and quick changes and analysis can be performed in the design process to optimize the design. The accuracy of the FEA results is largely dependent on the constraints and setup for the analysis. Since real world loading conditions and constraint can be incredibly complex, simplified representative conditions are often used to model the real world constrain. "Fake" parts are usually including in the FEA model to replace those parts in which may exist on the real assembly but their performances are not important to the model of interest. The results of the FFA then require the designer to interpret with that knowledge in 111 ind.

3.5.2.1 Front Upright and Steering Arm

A FEM analysis was conducted on the arm. Dominant method was used with 1 mm mesh. Two analyses were conducted, one fixed at the bolt holes and other with fixed bolt holes and fixed body that faces the upright. The reason for that is if the front upright weren't stiff enough and would flex a bit, the Ackerman bracket needs to be able to flex with it. A dummy pin was added to simulate the tie rod that connects to the bracket and 1000 N of force (used 600 N last year) as the maximum steering output. The new steering wheel is designed to have larger diameter and therefore can produce more torque which transfers to more linear force at the steering rack. This force is determined by the steering team.

A FEA analysis was conducted on the ANSYS v15.0. The worst case scenario for the upright is when the car is braking into a turn and the simulation was based upon that criteria. The Hex dominant method was used with 1 mm mesh since the ANSYS license capacity was maxed with that setting. The maximum shear stress is found at the inside of the lower brake calliper mount. On that edge there is no fillet that could improve the results but due to the license that could not be evaluated at this moment in time.

Load Consideration

Table 3.5.2.1.1: Load on Front Upright

Direction	**Magnitude**	**Situation**
Steering Rack	1000 N	Steering Effort
Brake Moment	1700 Nm	Braking
X	6278 N	Acc/Braking
Y	9418 N	Bump
Z	6278 N	Corner

Constrain Points

Loads are applied on the A arm mounting points from all three directions and bearing seat is kept constrain or fixed.

Solutions

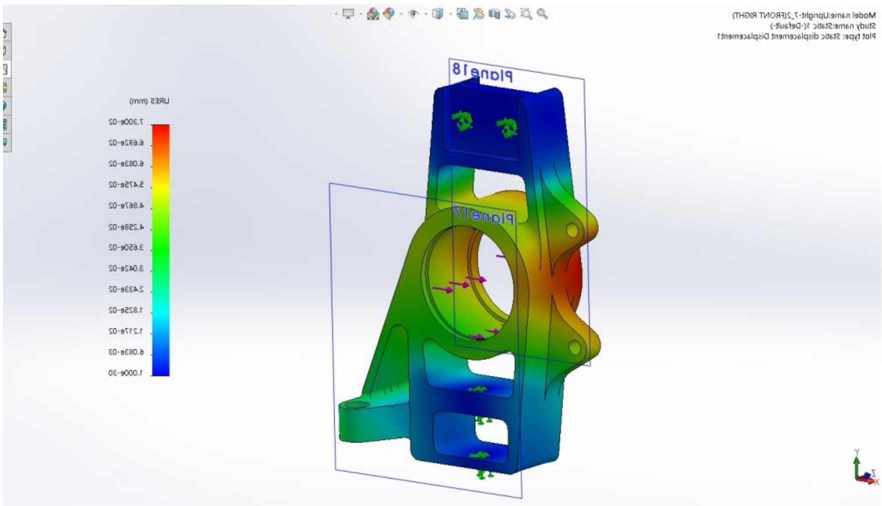

Figure 3.5.2.1.1: Forces on Front Upright

In the first analysis were the arm is only fixed by its mounting holes the shear stress value is 121.16 MPa. The locations of the maximum shear stress is at the edge is facing the upright. The maximum equivalent stress (von-Mises) is also located at the front edge of the mounting hole facing the upright. The maximum stress is 237.38 MPa. The total deformation of the Ackerman arm is at the edge that is furthest away from the upright facing the car where the tie rod dummy rod is located. The deflection is 0.055 mm

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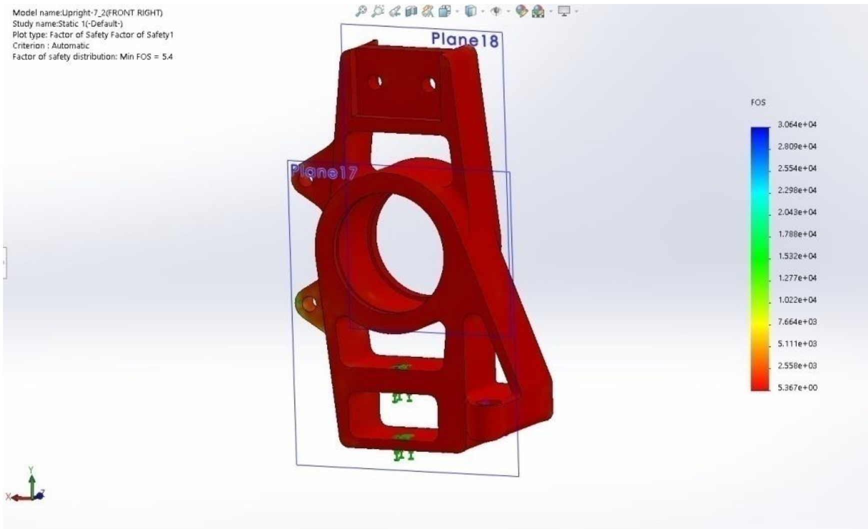

Figure 3.5.2.1.2: Forces on Steering Arm

The total deformation under the loads is at the inward facing edges of the bracket case scenario with a value of 0.240 mm. This deflection is a bit of concern since the suspension needs to have low flex in its components and let the damper take those loads. The maximum shear stress is in the edge where it mounts in the bracket. The value is 149.60 MPa, The factor of safety we get is 1.85 during force analysis.

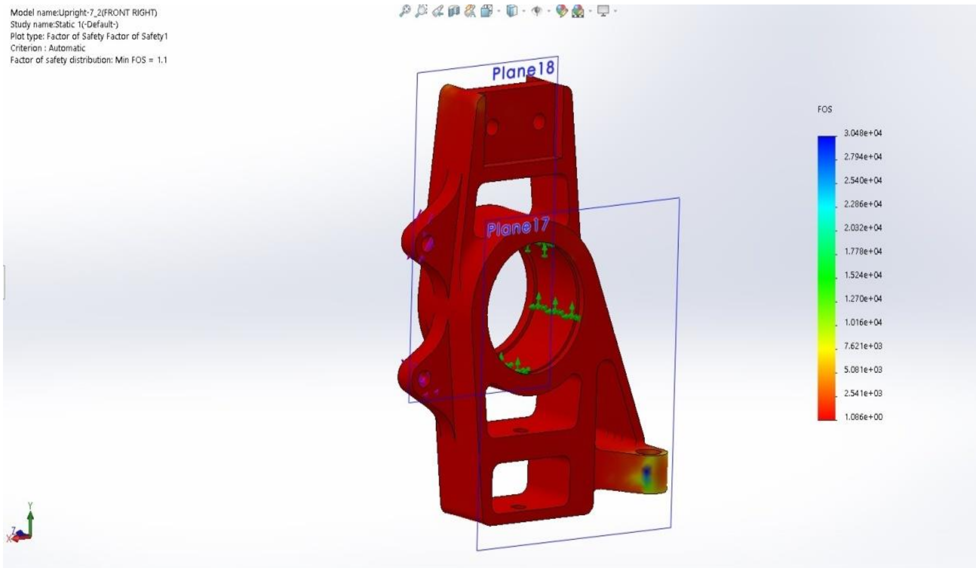

Figure 3.5.2.1.3: Braking Moment Analysis

The maximum shear stress is in the edge where it mounts in the bracket. The value is 150 MPa, the maximum von-Mises equivalent stress is found at the inside of the lower brake calliper mount, same place as the maximum shear stress with the value of 237.38 MPa. The safety factor for the front upright under this worst case scenario load is 1.16.

3.5.2.2 Rear Upright Analysis

FEM analysis was conducted on the rear uprights. The analysis used the same parameters as for the front without the force due to steering input from the driver. The torque from the engine is 37 Nm which is less than the brakes can produce. Therefore only the worst case scenario was simulated, braking into comer. Figure shows the deformation in the rear upright.

The analysis was conducted in two parts, first analysis with only the mounting holes fixed and the second with the body that is facing the upright also fixed. For the first analysis the maximum deformation is in the top edge inward facing with a value of 0.0258 mm.

Table 3.5.2.2.l: Loads on Rear Upright

Direction	**Magnitude**	**Situation**
Brake moment	1700 Nm	Braking
X	6278 N	Acc/Braking
Y	6278 N	Corner
Z	9418 N	Bump

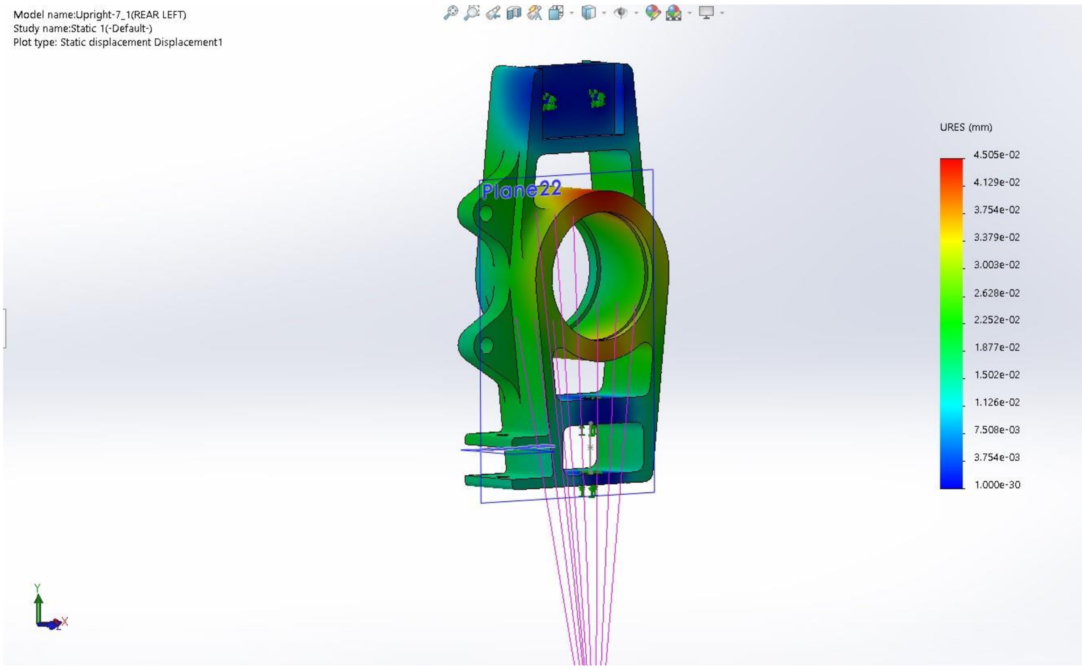

Figure 3.5.2.2.1: Forces on Rear Upright

In the first analysis were the bearing seat only fixed by its mounting holes the shear stress value is 121.16 MPa. The locations of the maximum shear stress is at the edge is facing the upright. The maximum equivalent stress (von-Mises) is also located at the front edge of the mounting hole facing the upright. The maximum stress is 164.06 MPa. The total deformation of the upright is 0.429 mm. It gives safety factor of 1.68.

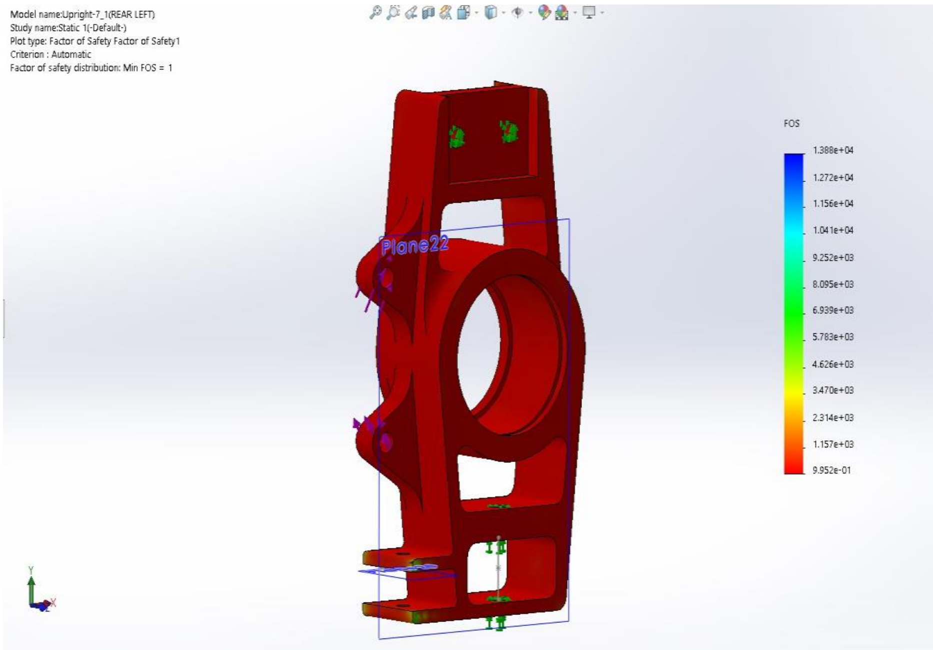

Figure 3.5.3.2: Braking Moment Analysis

The second analysis with the body facing the upright also fixed the maximum deformation is in the mounting holes for the linkage with a value of 0.2013 mm.The maximum equivalent stress (von-Mises) is 164.03MPa with safety factor of 1.03.

3.5.2.3 Front Hub Analysis

A suspension model In software called ANSYS was to be programmed beside this project. ANSYS can evaluate real time forces on many aspects of a vehicle, including the wheel hubs. Due to late changes in the design and complications by the suspension team, the model was not finished in time to use in the analysis of wheel hubs. Because of this complication, forces for worst case scenario were estimated to be the following per tire: 2g in longitudinal direction, 2g in lateral direction and 3g in bump. The maximum forces are fisted in table 3.2. The wheel hub is constrained by several of parameters set by other design evaluations: The inner diameter of the wheel bearing comes from the selected wheel bearing in previous section, ID40 mm and OD
68 mm.

The FEM analysis is done in two parts. The first part utilizes the forces in x, y and z direction. In the second analysis the forces that act on the hub while braking is conducted. The forces that were used during the both analysis are listed in table. In ANSYS, Hex dominant was used for meshing with 1 mm of mesh for both analyses.

Table 3.5.2.3.1: Loads on Front Hub

Direction	**Magnitude**	**Situation**
Brake Moment	1700 Nm	Braking
X	6278 N	Acc/Braking
Y	6278 N	Corner
Z	9418 N	Bump

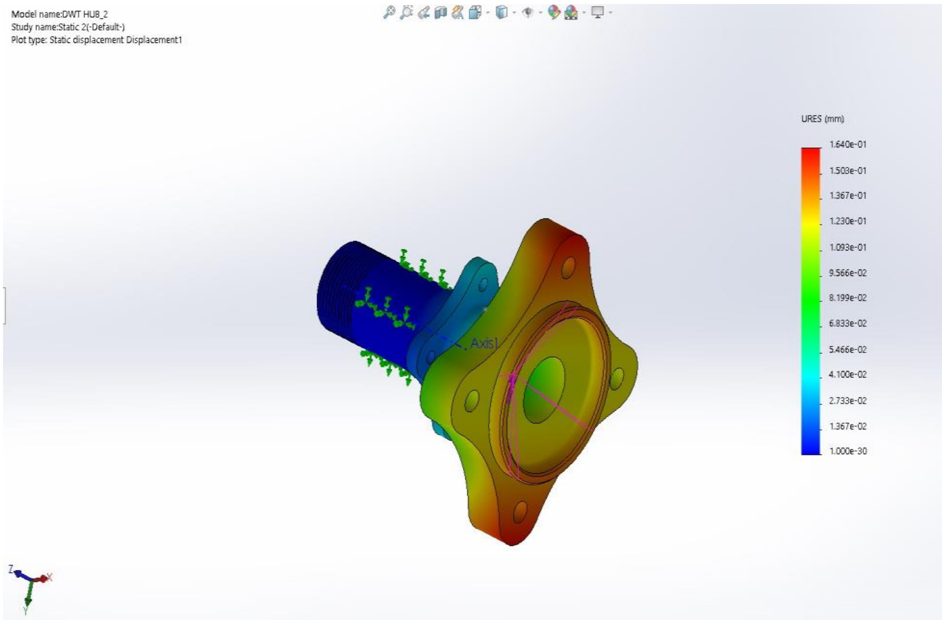

Figure 3.5.2.3.1: Forces on Front Hub Analysis

In the first part the forces in x, y and z direction are due to Acceleration, Braking, cornering and bump. The bearing seat was constrained as fixed and force applied to the shaft where the tire rests. The maximum shear stress is 56 MPa and is located the groove by the bearing seat. The most deflection is at the front of the hub (the most outer point) at magnitude of 0.0714 mm. The maximum von-Mises equivalent stress is located at the groove where the bearing seat. The magnitude of the stress is 187.09 MPa. The factor of safety is 1.47 for the worst case scenario of forces.

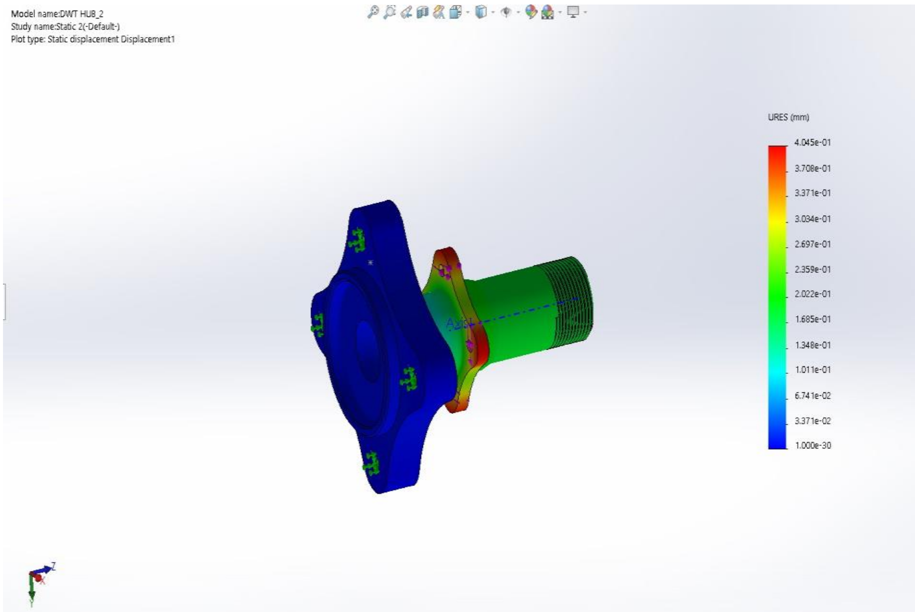

Figure 3.5.2.3.2: Braking Moment Analysis

The same mesh set up was used in the second analysis and brake torque that is listed in table applied to the cylinder where the wheel is located. The mounting holes for the brake rotor are fixed during this simulation and the bearing seat becomes loose. The moment of the brake is applied at the chamfer where the wheel attaches to the hub.

The maximum shear stress is 561 VLa and is located the groove by the bearing seat.

The most deflection is at the front of the hub (the most outer point) at magnitude of 0.04 mm. The maximum von-fViises equivalent stress is located at the groove where the bearing seat. The magnitude of the stress is 162.64 MPa. The factor of safety is 1.54 for the worst case scenario of forces.

3.5.2.4 Castle Nut or Central Locking Nut

A Central lock wheel nut is the fastener that tightens the wheel to the wheel hub. In a normal road car there is usually a 4 to 6 lug bolt pattern where the wheel is guided and bolted to the wheel hub. The CL wheel nut needs to fit the wheel hub and be strong enough to withstand the forces that it endures during a race. Two options were considered, make a custom nut for each wheel or to purchase off the shelf castle nut that would fit. A standard steel castle nut is heavy and steel nut can damage aluminium threads on the wheel hub while a custom made aluminium nut could prevent thread damaging and offers more customization that better suits this application. The first option was modeled in Inventor using standard M30x3.5 castle nut as a reference guide and modify it in a way that it fits the CL wheel. The nut has to have chamfer on the area that presses up against the wheel.

The nut has to have a fail/safe safety mechanism to prevent that if the nut comes lose it would fall off and the wheel also. To prevent that, a circular pattern was modeled at the wheel hub end for standard R-clip that goes through a slot on the nut and through holes on the wheel hub preventing possible lose nut falling off as stated in the SAE rules.

Only comering force of 6278N is applied on the threads of nut and rear face is constram.

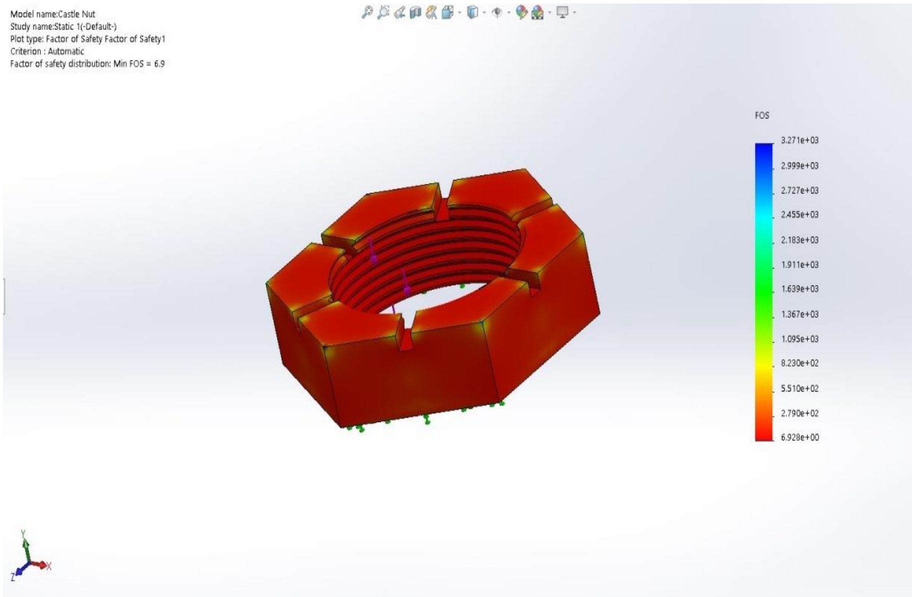

Figure 3.5.2.4.1: Central Locking Nut Analysis

The maximum shear stress is 50MPa and is located the thread end. The most deflection is at the rear of the nut (the most outer point) at magnitude of 0.0016mm. The maximum von-Mises equivalent stress is located at the rear face edge. The magnitude of the stress is 38.08MPa. The factor of safety is 7.2 for the worst case scenario of forces.

3.5.2.5 Rear Hub Analysis

Since our vehicle is rear wheel drive the rear hubs must withstand the torque of the motor and connect the axles to the wheels. In sake of symmetry and machine setup it was decided to base the rear hubs on the same design as the front with the addition of connecting the axles with CV-joints to transfer the power from the motor to the wheels. The design goals for the rear wheel hub are the following:

1. Same external geometry as front wheel hub
2. Connect the CV joint to rear wheel hub

A FEM analysis was also conducted on the rear wheel hub similar to the front hubs. The torque output of the engine is way less than the 1700 Nm analysis conducted. The maximum torque from the motor is 37Nm at the gearbox output shaft.

Table 3.5.2.5.1: Loads on Rear Hub

Direction	**Magnitude**	**Situation**
Brake moment	1700 Nm	Braking
x	6278 N	Acc/Braking
y	6278 N	Bump
Z	9418 N	Engine Torque
Torque	37 Nm	

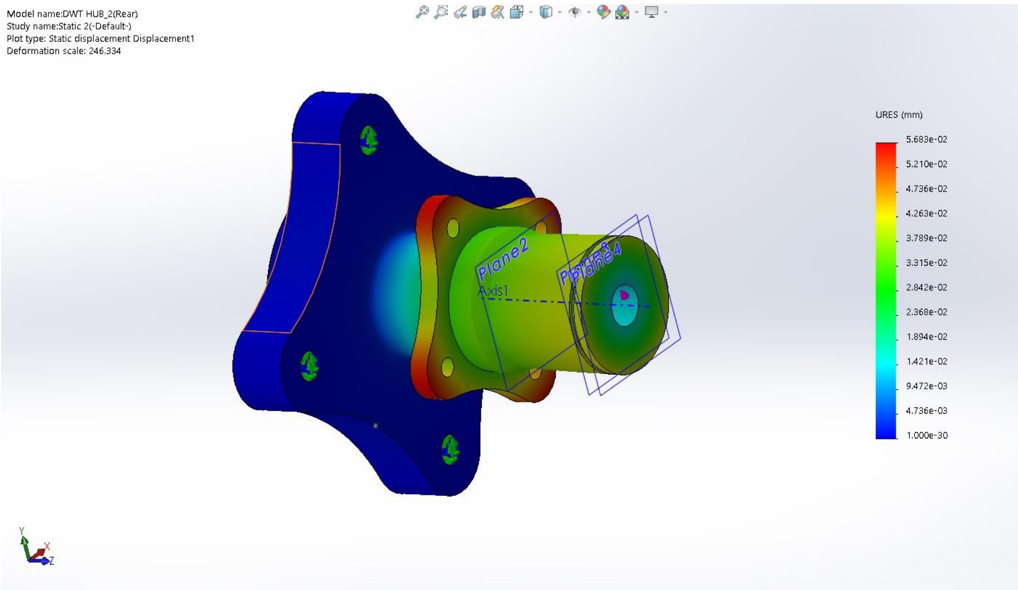

Figure 3.5.2.5.1 Rear Hub Forces Analysis

In the first part the forces in x, y and z direction are due to Acceleration, Braking, cornering and bump. The bearing seat was constrained as fixed and force applied to the shaft where the tire rests. The maximum shear stress is 56MPa and is located the groove by the bearing seat. The most deflection is at the front of the hub (the most outer point) at magnitude of 0.056 mm. The maximum von-Mises equivalent stress is located at the groove where the bearing seat. The magnitude of the stress is 199.09MPa. The factor of safety is I .38 for the worst case scenario of forces.

3.6 Cost Estimation

3.6.1 Material Cost

It includes raw material required manufacturing of components and also the components or parts already available in the market such as nuts and bolts.

Table 3.6.1.1: Material Cost

Components	Material	Quantity	Amount (in Rs.)
Front Upright	AL6061-T6 slab	2	5440
Rear Upright	AL6061-T6 slab	2	2988
Front Hub	AL6061-T6 round bar	2	4866
Rear Hub	AL6061-T6 round bar	2	4217
Castle Nut	AL6061-T6 round bar	1	340
Struts	Steel	16	250

Total cost of material is Rs. 18,101.

3.6.2 Manufacturing Cost

It includes the total machining cost of each component or parts required to be manufacture and also the quantity of the parts.

Table 3.6.2.1: Material Cost

Component	Process	Quantity	Amount (in Rs.)
Front Upright	Milling, CNC Machining	2	8,000
Rear Upright	Milling, CNC Machining	2	8,000
Front Hub	Lathe Machining, Milling	2	5,000
Rear Hub	Lathe Machining, Milling.	2	5,000
Castle Nut	CNC Machining, treading	2	1,200

Total manufacturing cost is Rs. 27,200.
Total cost of the project is Rs. 45,301.

3.7 Conclusion

The deflection of the upright assembly will be on the basis of the optimization process. With stiffness being the performance standard and weight being the concern. the design goals are defined to be reduction in weight over the 2018 design with comparable stiffness. To optimize for weight, sheet metal thickness for different races of the upright are changed respectively based on the previous runs stress distribution and deflection value. the material thickness was reduced in the areas where stresses are in the limiting factor being stress cannot exceed the material limit. With allowable thickness value based on available stock material, a number of combinations were analyzed and the optimum front and rear upright and hub designs were selected as the final designs.

3.8 References

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