Suspension Design and Kinematics Analysis of Suspension Hardpoints for a Formula Student Race Car

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Abstract: ***As an enthusiastic FSAE team, our objective is to design the most effective Suspension system for a Formula student race car. In this paper, we presented in detail the design procedure of the double wishbone pushrod suspension system, mechanical properties of the materials used, analytical calculations, determination of suspension hard points in CATIA V5, graphical analysis are examined for dynamic simulation of suspension system in Lotus Shark. The CAD models of the components in the suspension system are made using SolidWorks® and the Finite element analysis of the components is done using ANSYS® Workbench.***

Keywords: Double wishbone; Pushrod; Material selection; Suspension Points in CATIA V5; Dynamic Simulation Analysis of Suspension System in Lotus Shark

1. Introduction

The Suspension system is the intermediate flexible system that connects the wheels with the main frame of the vehicle. It is one of the most important systems of an automobile that deals with the dynamics of the vehicle. Suspension System of a vehicle has to maximize the contact between the vehicle tires and the road surface, provide good steering stability and provide safe vehicle control in all conditions, evenly support the weight of the vehicle, transfer the loads to springs, and guaranteeing the comfort of the driver by absorbing and dampening shock. The tuning of suspensions involves finding the right compromise of angles like camber, caster, king-pin inclination, Arms or linkages and shock absorber that comes together and enables the relative motion between the tyre and the mainframe.

The vehicle must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50 mm (25 mm jounce and 25mm rebound).

We designed our race car in such a way that forces that the tires absorb are transferred to the Upright. From the Upright, the forces are then made to transfer to the pushrod and the control arms. The pushrod then forces the Bell crank to move accordingly and then bears the forces on the damper. The damper absorbs most of the forces and the remaining forces are transferred to the chassis. This system reduces the shocks and any impact on the driver. This paper discusses the kinematics design of a double a-arm Pushrod Suspension system for an FSAE Vehicle.

The hardpoint's location can be determined using this procedure to simulate motion in any kinematic simulation software. Here, Lotus Shark is used as kinematic simulation software, and the results are verified using Analytical Calculations. The obtained values are expressed as graphs to visually understand the relationship between suspension parameters and vehicle performance. The process of iterative optimization is followed to improve the design to meet the loading conditions. The results are noted down, the designs are optimized and the most promising values are concluded.

1.2 Roll Centre Height – Roll center Height of Front suspension was kept 184.9 mm and of Rear Suspension as 184.9 mm by doing the same procedure which was in upright height. Rear Roll Centre Height is kept more to keep our car aerodynamically stable at High speed also.

1.3 Camber Angle – Camber Angle is basically based on Cornering stability so a real case value of -2.5 degrees for front wheel and -0.5 degrees for rear wheel which is considerable and also can be manufactured is kept.

1.4 Kingpin Inclination – Kingpin inclination is 6 degrees with a considerable scrub radius of 55 mm. Also it can be easy to manufacture by adjusting upright Bracket length.

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

2. Material Selection

2.1 Uprights

Our uprights are made of Al 6061 T6. Most common material used for Uprights is Aluminium. Al is used because of its properties like low weight density, resistance to corrosion and rusting etc. For manufacturing knuckle, Aluminium is the best choice as it has enough load with standing ability. In Aluminium, we have different grades like Al-6061, Al-5056, Al-7075, Al-6063 etc. Among these two, Al-7075 and Al-6061 are most common used.

However, Al-6061 is finalized for uprights. Al-6061 is chosen because:

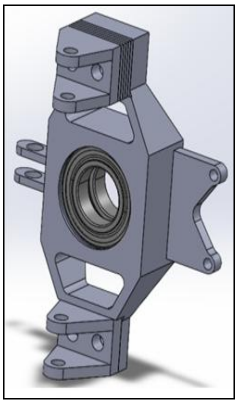
1) Its hardness is within the desirable range (Brinell Hardness no. = 95). The lower hardness allows it to be machined more easily than 7075 (BHN = 150).

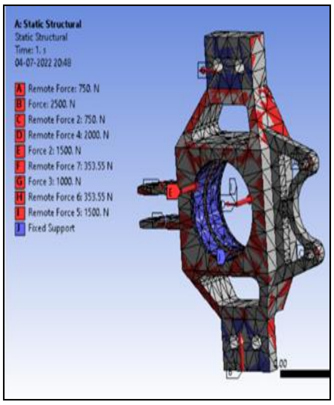
2) Easily machinable when compared to Al-7075 T6
3) Although Yield strength of Al-6061 (276 MPa) < Al-7075 (510 MPa), we choose Al-6061 because stress developed due to action of loads is not more than 270 MPa. Thus, Al-6061 can be used.
4) It has better structural strength compared to Al-7075.
5) It is tougher than many other materials including Al-7075
6) It has got better surface finish & corrosion resistance when compared to Al-7075 & Al-6082.
7) It's Density is 2700 Kg/m3and low when compared to Al-7075 (2810 Kg/m3). Thus, for given volume, mass consumption is less in the case of Al-6061. This reduces overall weight and thus also expenses.
8) Material cost of Al-6061 (₹ 275/kg) is also less than Al -7075 (₹ 600/kg) & Al-6082 (₹ 335/kg) - thus economical.
9) It has a high strength-to-weight ratio.
10) It has high fatigue strength when compared to other grades and alloys. Although Al-7075 has more fatigue strength (159 MPa), Al-6061 is opted because fatigue strength of Al-6061 (58-110 MPa) is sufficient for our requirements
11) Processing performance of Al-6061 is better than Al-7075 and other grades.
12) When dealing with fabrication, 6061 aluminium alloy has the edge over 7075 aluminium alloy.

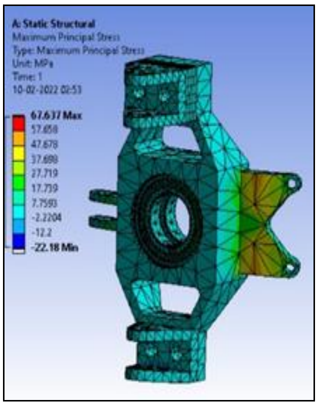
Keeping all these parameters in consideration, Al-6061 is the best choice. If at all Hardness is not meeting the expectations, hardness can be improved by heat treatment methods like Tempering. It is frequently used in automotive parts. Aluminium 6061 is a better alloy when the product is going to be welded or formed.

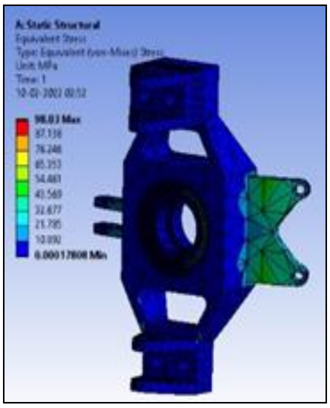
2.2 Upright Height

To decide upright height several factors were considered. Firstly, ICR's of both A-Arms and the line joining center of wheel to ICR to get Roll Centre Height were drawn. By Keeping Lower A-Arm Horizontal to get more stability and adjusting upper A-Arm we decided to keep Upright's overall height as 200 mm with eye-to-eye length of 168 mm. Distance from uprights center to Upper ball joint = 84mm and upright center to lower ball joint = 84mm





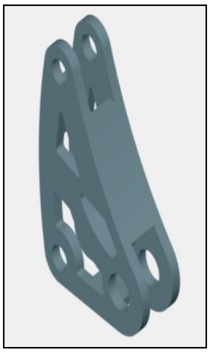


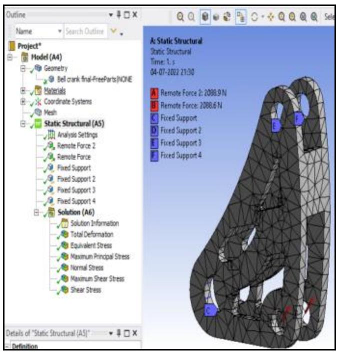


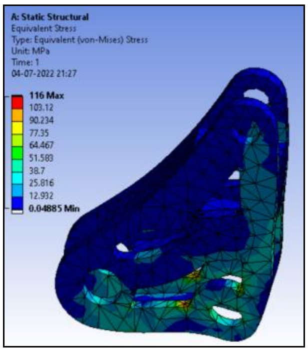
2.3 Bell Crank

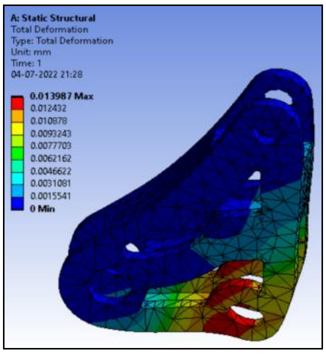
Material selected for Bell crank is Mild steel EN 1.0301. Although Mild steel is relatively inexpensive when compared to other materials like Al 6061, Al 7075.It satisfies our requirements and there is no need to compromise in mechanical properties. Hardness of mild steel is 130 BHN. The hardness is enough to withstand indentations and at the same time it is not too high so that it is machinable. It has good weldability. Mild steel contains roughly 0.05-0.30% carbon making it flexible. Mild steel has a relatively low tensile strength 440 N/mm² when compared to other materials like Al 7075(572 N/mm²), However, it is enough to withstand the stresses and forces acting on bell cranks. Hardness can be increased and improved by methods like carburizing (Case hardening). Mild steel is ductile enough to withstand forces acting on it. EN 1.0301 carbon steel contains 0.1% C+0.4% Mn+ 0.4% Si. It also contains traces of Cu, Ni, Mo, Cr-thus bridging gap between ductility, strength, and toughness.

Steel is prone to oxidizing if not prepared accordingly, resulting in rust that damages (and eventually destroys) the steel. Without the addition of any additional elements, mild steel will suffer the same fate. Chromium is a popular addition to low carbon steel due to its reaction to exposure to the atmosphere, resulting in a layer of chromium oxide that protects the steel underneath from further corrosion.The density of EN 1.0301 grade mild steel is 7850 kg/m³ which is quite acceptable. Having high tensile strength of 350-640 MPa, Ultimate yield strength of 370 MPa, Shear stress of 200-300 MPa; Mild steel is a better choice. Max Force which is applied on our bell crank is 420.29N at Front and 466.95 N at Rear. So, our selection is perfect, and it can withstand heavy loads at cornering without deflection.









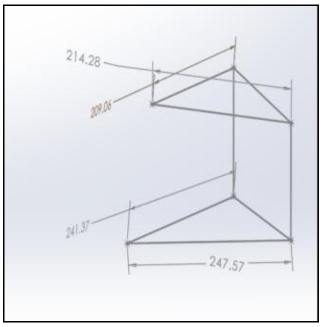
2.4 Wishbones

Material opted for wishbones is EN24. It is a popular grade of thorough-hardened alloy steel with a tensile strength of 850/1000 N/sq. mm. It is a 1.5% nickel, 1% chromium, 0.2% molybdenum alloy steel which has a long history dating back over 100 years. EN24 can be heat treated to a wide range of tensile strengths from 850-1000 N/mm² ('T' condition) up to 1550 N/mm² ('Z' condition). Heat treated EN24 offers high tensile strength combined with good ductility and resistance to shock. Owing to its excellent machinability, EN24 is used in components such as gears, shafts, bolts et cetera. It is also renowned for its wear resistant and high strength properties.

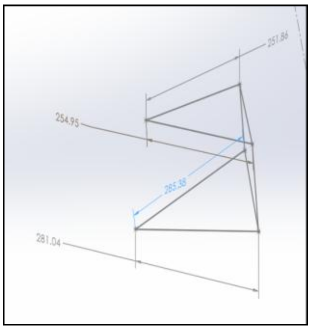
Components like propellers, connecting rods, aircraft landing gears and also automotive and general engineering applications have been known to be made of EN24 because they tend to be put under high stress. EN24 is commonly supplied as EN24T or EN24U. For EN24V, EN24W, EN24X, EN24Y & EN24Z material should be fully annealed before heat treating to any of these conditions. For these reasons, EN24 is used as a parent material for pushrods in the manufacturing of our race car. Solid Wishbones of diameter 12 mm are selected to have better strength withstanding ability.

Table 1: Typical Mechanical Properties for EN24

Condition	*Tensile Strength (N/mm²)*	*Yield Strength (N/mm²)*	*Elongation %*	*Izod KCV J*	*Hardness Brinell BHN*
T	850-1000	650	13	35	248-302
U	925-1075	755	12	42	269-331
V	1000-1150	850	12	42	293-352
W	1075-1225	940	11	35	311-375
X	1150-1300	1020	10	28	341-401
Y	1225-1375	1095	10	21	363-429
Z	1550	1235	5	9	444

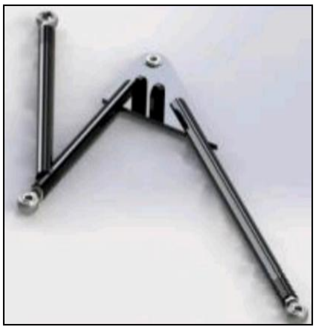


Rear wishbone lengths



Front Wishbone lengths


Front A-Arm


Rear A-Arm

2.5 Pushrod

We chose push-rods that are made of Stainless Steel. Stainless steel is the name of a family of iron-based alloys known for their corrosion and heat resistance. One of the main characteristics of stainless steel is its minimum chromium content of 10.5%, which gives it superior resistance to corrosion in comparison to other types of steels. Like other steels, stainless steel is composed primarily from iron and carbon, but with the addition of several other alloying elements, the most prominent being chromium.

Stainless steel has numerous properties that make it desirable to use in the widespread manufacture of parts and components. The mechanical properties of one of the frequently used grades of stainless steel, the 304 grade is as below:

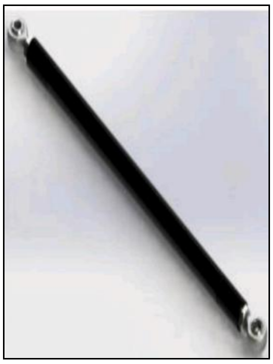
Table 2: Mechanical Properties

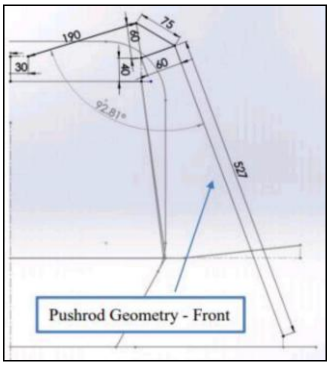
Properties	*Value*
Density (x1,000 kg/m³)	8
Poisson's ratio	0.27- 0.30
Elastic modulus (GPa)	193
Tensile strength (MPa)	515
Yield strength (MPa)	205

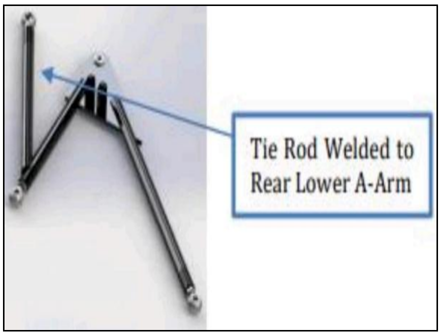
Elongation (%)	40
Hardness (BHN)	88
Thermal expansion (10°C)	17.2
Thermal conductivity (W/m-K)	16.2

One of the favourable properties is its high strength that allows consumers to use it in numerous applications. Its durability, high and low temperature resistance, increased formability and easy fabrication, low maintenance, long lasting and attractive appearance are the pros of Stainless Steel that attracts users.

As we are going with Pushrod type suspension, we are using Pushrod. Diameter of pushrod = 12 mm & length of Pushrod = 150 mm.





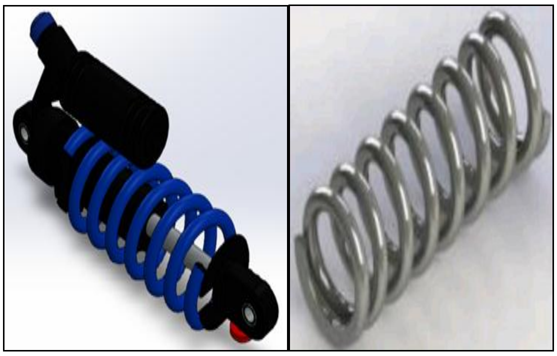


2.6 Damper

We are going with DNM RCP2S shock absorbers because of better quality and excellent shock absorbing abilities. Vibrations can be damped easily.

Damper calculations are done and eye to eye length of 210 mm is fixed with spring size = 12 mm. Spring travel = 61 mm and mass of shock absorber = 0.41 kg.

Body is made up of dark hard anodized Al 7075 material to have higher strength with low weight.



Shock Absorbers

3. Calculations

Force = 180 x 9.81 = 1765.8 N
Longitudinal Force = 882.9 N
Vertical load transfer on each wheel [Front] = 662.17 x 3 = 1986.53 N
Vertical load transfer on each wheel [Rear] = 809.325 x 3 = 2427.975 N

3.1 Load Calculations

Longitudinal Forces during Braking:

While Braking, the weight of the rear side tends to come in the front side of the vehicle so there is a load transfer that is taking place from rear to front.

It interns affects the knuckle as these forces act on the A – arm mounting points through the A – arms.

Considering Maximum acceleration of 1g = 9.81m/s^2

Force at the front side = mass at the rear side of the vehicle x acceleration.

Let the mass at the rear side of the vehicle be 0.6 times the total weight

Mass at the rear side of the vehicle = 0.6 x 300 = 180kg

Force = 180 x 9.81 = 1765.8 N

Now force on each wheel = 1765.8/2 = 882.9N

Longitudinal Force = 882.9 N

3.2 Lateral Forces

Lateral Forces are because of two reasons – centrifugal force and lateral load transfer from outside to inside while turning. The centrifugal force is considered as follows.

Let the vehicle take a turn of 3.2m turning radius and at a speed of 28.8kmph = 8m/s
r = turning radius = 3.2m
1km = 1000m; 1hr = 3600 sec.
1km/hr = 5/18 m/sec

To convert km/hr into m/sec, multiply the number by 5 and then divide it by 18.

Centrifugal Force = $mv^2/2$ = 2400 N

Now consider if all the weight at the front side comes on the wheel assembly the force will be Force due to lateral load transfer = 0.4 x 300 x 9.81 = 1175.5 N
Vertical Force at Bump,
The vertical load transfer occurs at bump and as per load theories 3g of weight applies on vehicles at the time of bump.
Vertical load transfer on each wheel [Front] = 662.17 x 3 = 1986.53 N
Vertical load transfer on each wheel [Rear] = 809.325 x 3 = 2427.975 N

3.3 Knuckle calculations:

Friction force on front wheel (Ff) = 256.168 N
Braking force per each wheel (Fb) = 102.467 N
Vertical force per wheel (Fv) = 384.252 N
Lateral force per wheel (Fl) = 256.168 N
Force on Upper Ball Joint in side view (FUS) = 29.325 N
Force on Lower Ball Joint in side view (FLS) = 131.792 N
Force acting on Steering Ball Joint (Fsb) = 86.194 N
Force acting on Upper Control Arm (Fuc) = 35.026 N
Force on UC Arm in Horizontal Direction (Fuch)= 24.763 N
Force on UC Arm in Vertical Direction (Fucv) = - 145.211 N
Force on LC Arm in Horizontal Direction (Flch)=409.015 N
Force on LC Arm in Vertical Direction (Flcv) = 409.015 N

3.4 Wishbone Calculations:

First moment of area of the wishbone (Q) = Qout - Qin
Qout = A Y' = x ro
Qin = A Y' = x ri
Where,
ri = inner radius of the wishbone = 5 mm
x (5) = 83.33 mm^3
Qin = 83.33 mm^3
First moment of area of the wishbone (Q) = 258 mm^3
Mass varies as 0.4 times total weight at Front wishbone Mass at Front side of the vehicle = 0.4 x 300 = 120 kg
Load at Front Wishbone (Vf) = 1177.2N
Moment of Inertia of the wishbone (I) = 2724.735 mm^4
Thickness of wishbone pipe (t) = 3 mm
Shear stress offered on the front wishbones are (τf) = 37.156 N/mm4

3.4.1 To choose Wishbone Material

A prototype material "Chromoly" is taken for consideration. Based on the shear strain (ϕ) value of chromoly, ϕ = 80 G Pa (AISI 4130) = 80 x 10 3N/mm^2

Shear Modulus (G) = Shear Modulus of the front wishbone (Gf) = 4.645 x 10^{-4} (too small). Hence the material selected undergoes much less deformation with a given load. The material chosen is perfectly suitable for control arms.

Rear wishbone Shear stress offered on the wishbone (τ) = First moment of area of the wishbone (Q) = Qout - Qin
Qout = A Y' = x ro

Where, ro = outer radius of the wishbone = 8 mm = x (8) = 341.33 mm^3
Qout = 341.33 mm^3
Qin = A 'Y = x ri
Where, ri = inner radius of the wishbone = 5 mm
x (5) = 83.33 mm^3
Qin = 83.33 mm^3

First moment of area of the wishbone (Q) = Qout - Qin = 258 mm^3
First moment of area of the wishbone (Q) = 258 mm^3
Load on the Rear wishbone Load (V) = mass x acceleration
Mass varies as 0.6 times total weight at Rear wishbone Mass at Rear side of the vehicle = 0.6 x 300 = 180 kg
Load at Rear Wishbone (VR) = 1765.8N
Moment of Inertia of the wishbone (I) = 2724.735 mm^4
The Thickness of wishbone pipe (t) = 3 mm
Shear stress offered on the rear wishbones is (τR) = 55.733 N/mm

4. Determination of suspension points:

4.1 Designing

Step-1: Consider the front part of the vehicle (as a rectangle) with the Centre of Gravity. For that we develop a 3D model placing the tire on one side at a distance of half of the track width and assume scrub radius as 40% of tire width.

Step-2: Designing Front view Geometry
We get Front view Swing arm Length by formula:
Camber Change Rate = tan-1 (1/ FVSA L)
Where CCR = KPI / Wheel Travel of Jounce and Rebound
Assuming KPI and Wheel Travel, we get CCR and thereby FVSA L

Step-3: Having KPI and Scrub Radius, we can get the position of knuckle mounting points. Locate Both Upper and Lower Mounting Points of Knuckle.

Step-4: Assuming Roll Centre Height
- Roll Centre should be near the CG to avoid body roll
- Optimum Jacking Force

Step-5: Intersect the axis passing from bottom of the tire contact patch and the roll center to the perpendicular axis to get IC.

Step-6: Joining UBJ with IC, we will get the Upper Control Arm axis and joining the LBJ with IC, we will get the Lower Control Arm axis.
So, in the front view we will see UCA and LCA (Hard points 1, 2 and 3, 4)

Step-7: Find the coordinates (x, y, and z) of these hard points.
In the front view, we get only y, z coordinates. For x-coordinate we should see the side view of the tire.

Step-8: Assume Castor Angle for side view. Castor line is basically the side view of kingpin axis.

Step-9: Extending UBJ up to Castor line, we will get x - coordinate, similarly for LBJ.

Step-10: Assuming Anti-dive% in Side View, which will give SVSA angle.

Step-11: Keep SVSA L as short as possible because it determines the pitching action of the vehicle, responsible for applying moment at IC of side view.
Line passing from SVSA L will intersect SVSA at IC.

Step-12: Add some distance to the center line in the right direction because the front of the vehicle is towards the right.

Step-13: Now join that point with IC, we will get the UCA axis.

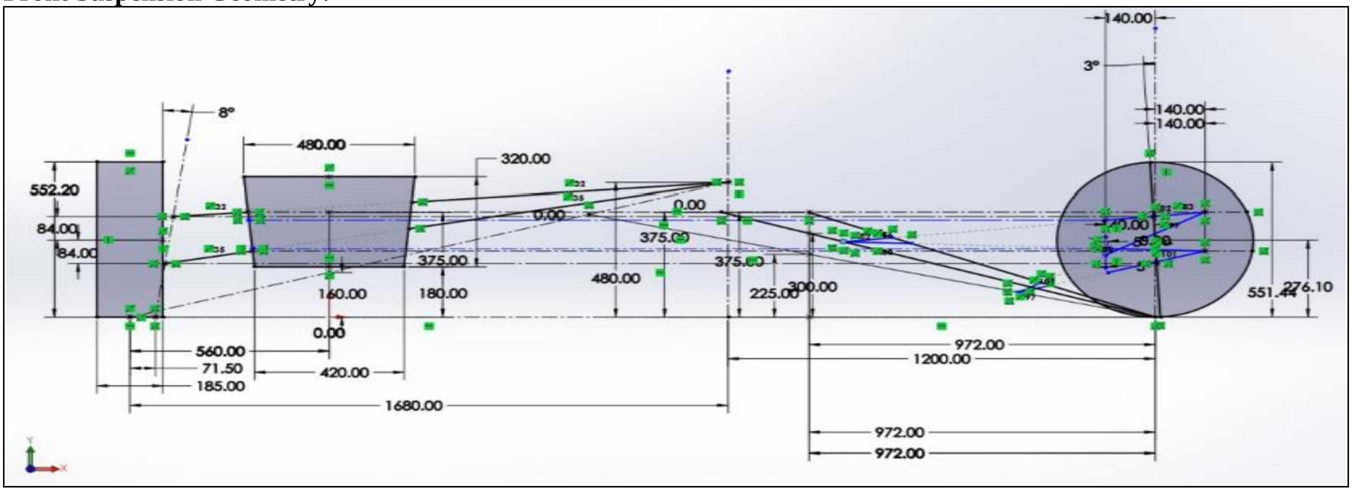
Step-14: As we know the other pivot point of UCA is located on the axis, similarly for LCA.

Step-15: follow the same approach for rear (just mirror it).

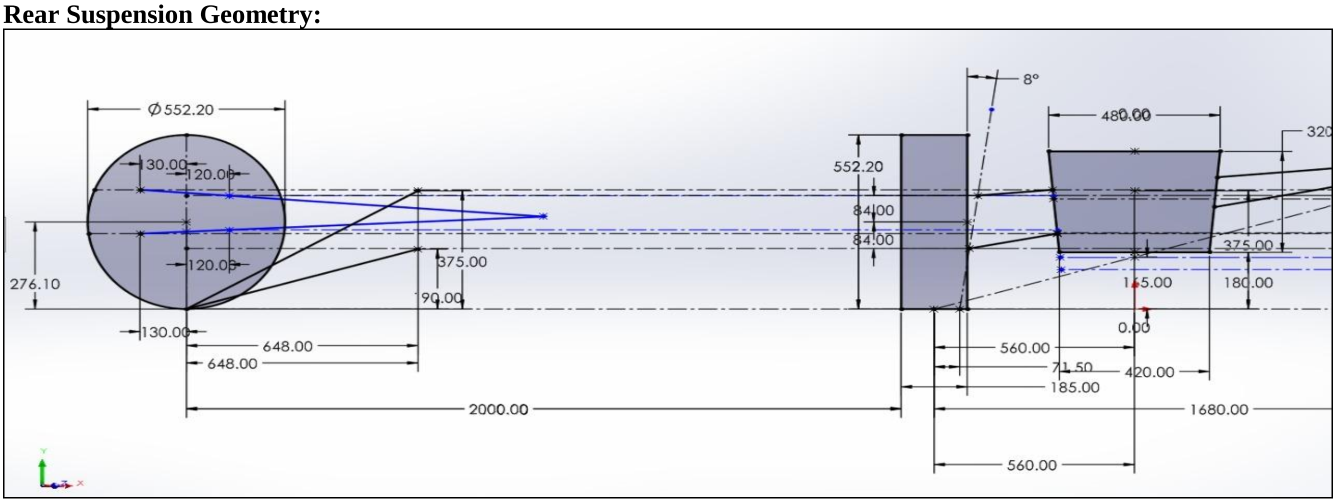
Step-16: Put all these suspension mounting points in lotus shark. We also need to give basic inputs like type of Suspension system, approach of our suspension, tire data, Wheelbase, Dive, Weight, etc and get respective graphs.

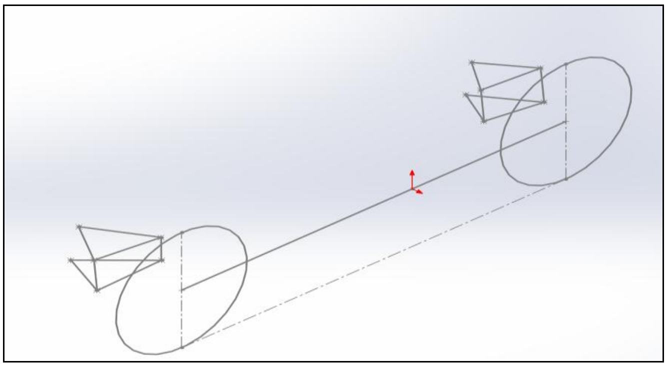
Step-17: Analyse the Graphs, whether it is feasible or not for our requirements at static and dynamic conditions
Finding suspension points is an iterative process and below is the image of the line diagram of suspension points.

Front Suspension Geometry:

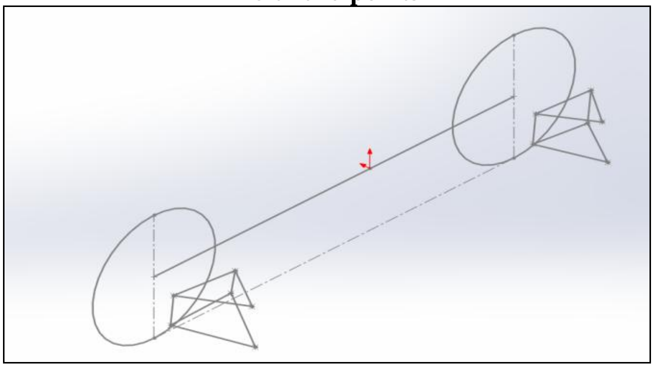


Rear Suspension Geometry:





Left hard points



Right hard points

Front suspension points:

S.NO	X	Y	Z
1	-4.4	467.89	360.1
2	4.4	491.5	192.1
3	140	258.5	376.22
4	140	245.34	236.91
5	-140	255.65	346.94
6	-131.15	246.17	245.77

Upper wishbone ball joint - 1
Lower wishbone outer ball joint - 2
Upper wishbone front point - 3
Lower wishbone front point - 4
Upper wishbone rear point - 5
Lower wishbone rear point - 6
Anti-dive percentage: 80%
SVSA height: 270.08 mm
SVSA length: 875.05 mm
Roll center height: 160 mm
Distance from front end to side-view CG: 972 mm
Distance from rear end to side-view CG: 648 mm
CG height: 375 mm

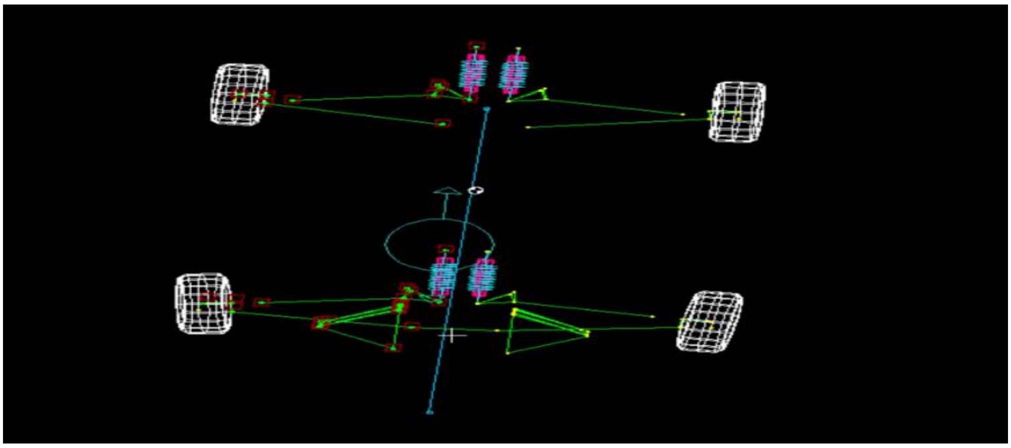
Rear suspension points:

S. No	X	Y	Z
7	1620	427.8	360.1
8	1620	551.5	192.1
9	1750	258.41	377.15
10	1750	345.11	234.47
11	1500	256.7	358.5
12	1500	242.07	202.03

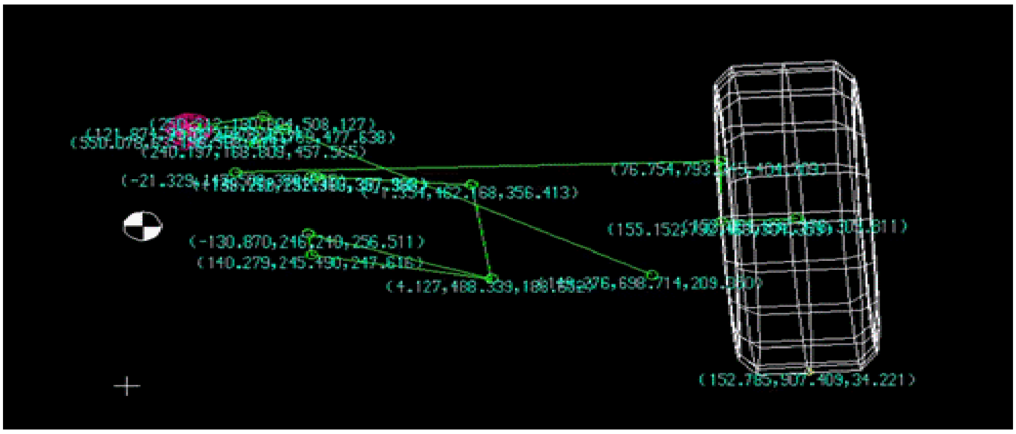
Upper wishbone ball joint - 7
Lower wishbone outer ball joint - 8
Upper wishbone front point - 9
Lower wishbone front point - 10
Upper wishbone rear point - 11
Lower wishbone rear point – 12
Anti-squat percentage: 50.67%
SVSA height: 293.21 mm
SVSA length: 1000 mm
Roll center height: 162.85 mm
Distance from front end to side-view CG: 972 mm
Distance from rear end to side-view CG: 648 mm
CG height: 375 mm
FVSA length: 1680 mm

5. Dynamic Simulation Analysis of Suspension System

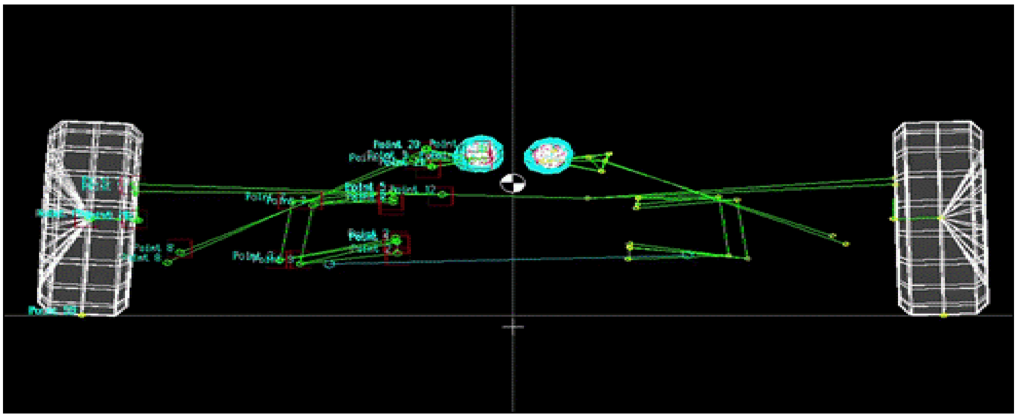
Dynamic analysis of suspension is done by using Lotus Shark suspension analysis software. In that coordinates of suspension points to arrange the geometry of the suspension system and then by using the software the Bump, Roll and Steering effect on dynamic conditions were checked.



Suspension Geometry in Lotus Shark



Half Simulation

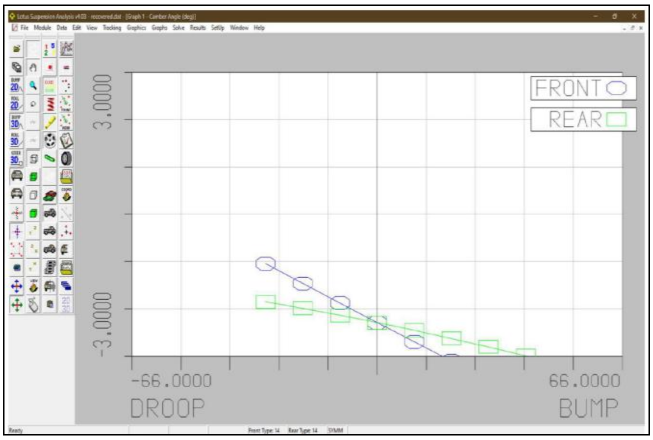


Full Simulation

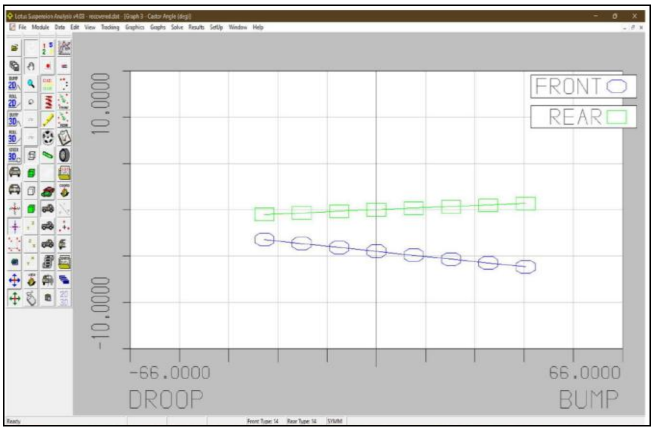
The layout of the suspensions hard point positions is interposed and the required kinematic behaviour is achieved.

We finalized points after performing many iterations and analysing graphs in Lotus Shark software.

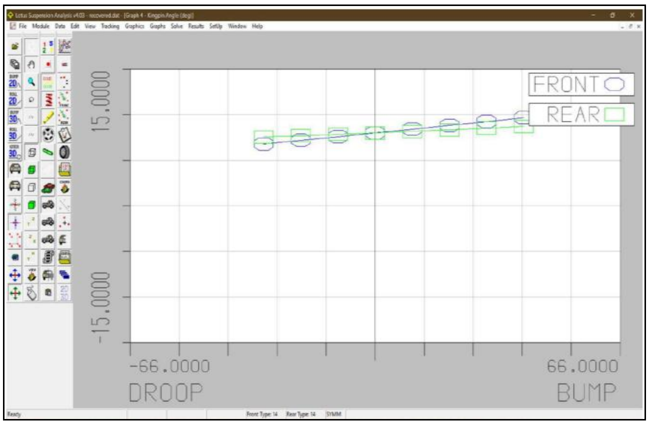
Graphs of final suspension points are as follows:



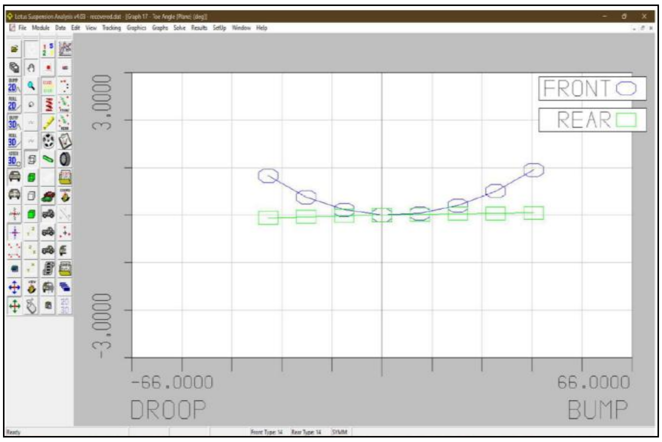
Camber Angle



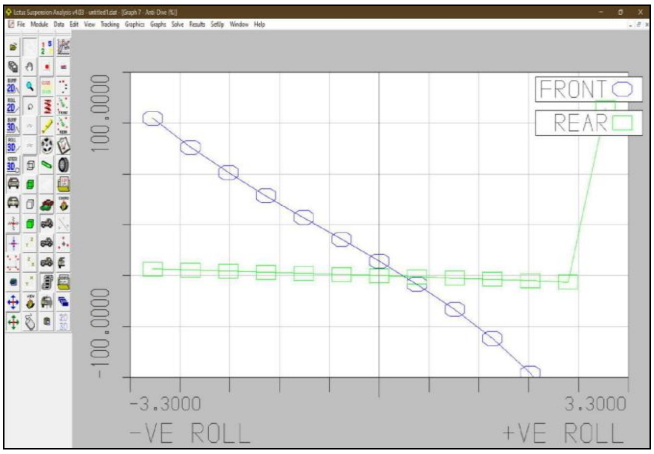
Castor Angle



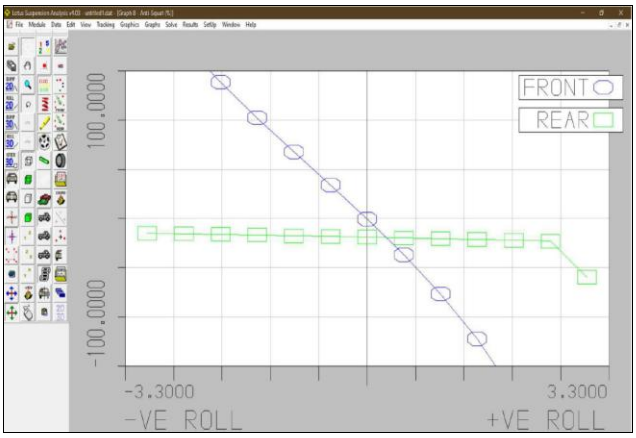
Kingpin Angle



Toe Angle



Anti Dive



Anti Squat

6. Results

- 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
- 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.
- Oversteer configuration enables good vehicle handling to the driver by reducing the required steering effort.
- Oversteer configuration enables good vehicle handling to the driver by reducing the required steering effort.
- More stability of vehicle is achieved due to negative camber angle as it provides more traction and contact patch to the wheel during cornering.
- It is clear from the graphs, there is a small or negligible change in the toe angle when the vehicle faces the condition of bump and rebound.

Specifications	Front/Rear
Spring rate [N/mm]	86/77
Roll rate*104 [Nm/deg]	1477.54/1540.86
Ride rate [N/mm]	0.645/0.647
Wheel rate [N/mm]	140.15/146.53
Suspension travel [mm]	50 / 50
Max Damper stroke [mm]	76
CG height [mm]	375
Ground clearance [mm]	180
Castor [deg]	3
KPI [deg]	8
Scrub radius [mm]	55
Sprung mass [N]	392.266
Unsprung mass [N]	2353.596
Motion ratio	1.275 / 1.356
Roll center height [mm]	160
Roll gradient [rad/g]	-0.068

7. Conclusion

The purpose of this 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.

The design is first conceptualized based on personal experiences during the previous projects under SAE competitions. Engineering principles and design processes are then used to verify and create a vehicle with optimal performance, safety, manufacturability and ergonomics.

- High ground clearance as 180mm and the shock travel is up to the maximum of 76mm, whereas the wishbone hard points were mounted to the nodes of the triangulated chassis where point can bear the peak number of stresses, all the 8 hard points of the front suspension were mounted to the nodes of the chassis systematically executed.
- The distance between upper A-arm and lower A-arm is 168 mm and the distance between the knuckle upper A-arm mounting point and lower A-arm mounting point is 168 mm (parallel double wishbone suspension system). The camber and toe angles can be controlled so that the stress on the tie rod and steering arm will be less and less chances of failure.

Therefore, it can be considered that the optimized set of values will render a very comfortable ride. The FEA result indicates that the suspension system is able to perform safely in real track condition as per performance requirement.

With the overall design being carefully considered beforehand, the manufacturing process being controlled closely, and many design features have been proven effective within the performance requirement of the vehicle.

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