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# Design & Analysis of Suspension System for a Formula Student Vehicle

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**Abstract -** *The objective of presented work is to layout the optimum methodologies to design an efficient Suspension system of a formula student vehicle. This report is final output of the suspension design and analysis presentation at Formula Bharat Competition 2020. Entire work was performed under Formula Bharat rulebook restrictions and guidelines for an efficient vehicle buildup accompanying optimum speed considering excellent ride comfort. The paper enlists various methodologies adopted for design and analysis of suspension system which includes control arms, pushrod, bell crank, damper springs and knuckle. LOTUS software was used for static and dynamic analysis of suspension assembly considering various geometrical attributes such as Roll center, Wheel travel and various wheel alignment angles such as Camber, caster and toe angles. The 3D modeling of the system assembly was performed on SolidWorks whereas the analysis was carried on both Autodesk Fusion 360 & ANSYS workbench.*

***Key Words*: Double Wishbone pushrod system, Bell Crank, Suspension Damper & Spring, LOTUS software, ANSYS simulation software, Knuckle & Hub**

## 1. INTRODUCTION

The purpose of the car's suspension is to keep all four wheels in optimal contact with the ground under any and all conditions. A well-designed suspension must handle bumps and uneven surfaces as well as dynamic cornering, braking, and acceleration. The FSAE car is a racecar purpose built for a prepared track, so performance and handling will be prioritized over smoothness and suspension travel. It is generally a system of shocks (dampers), springs, uprights and arms that altogether keep a vehicle suspended above ground on its wheels. The major component involved in the system includes damper & springs, wishbones, knuckle/upright and wheels. A Double wishbone Pushrod actuated suspension design was primarily used because of the aerodynamic and adjustability advantages it gives. They consist of an inboard mounted spring a push rod and a bell crank assembly. The main requirement here is a structure that will absorb the energy and transfer it to the frame without disturbing the whole system. The study of the forces at work acting on a moving car is called vehicle dynamics, and the concepts required for designing suspension system are clearly encapsulated further.


Figure 1: CAD model of Complete vehicle

## 2. DESIGN METHODOLOGY

For the competition we used pushrod suspension system due to its high structural strength and its ease of track tuning.

The main requirements for the suspension system are:

- It should have a minimum travel of 25 mm in both jounce and travel.
- The system must be responsive and also stable enough to give best performance in cornering and straight line.
- The system should be such that it should help maintain maximum tire contact patch for better friction.
- It must be structurally strong but also lightweight to maintain low unsprung mass.

Following above assumptions after multiple iterations over LOTUS software the various basic parameters were decided & discussed upon and finally confirmed at requisite values:-

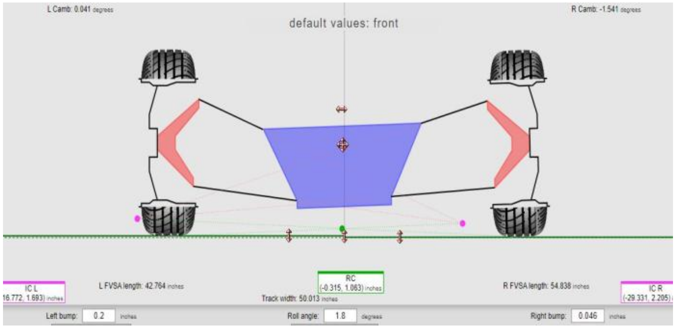
1. WheelBase : 1727 mm
2. Front Track-width: 1300 mm
3. Rear Track-width: 1300 mm
4. Camber Angle : -1.5$^0$ in front and -1$^0$ in rear
5. Caster Angle: 6$^0$ (front) & -6$^0$ (rear)
6. Toe Angle: 0.3$^0$ in front and -0.3$^0$ in rear
7. Kingpin Inclination: 2.5$^0$ in front and 2$^0$ in rear
8. Total Mass of Vehicle(with driver):310 kg

9. Sprung weight: 210 kg
10. Weight distribution : 45:55 (front: rear)

### 2.1 BASIC GEOMETRICAL SETUP

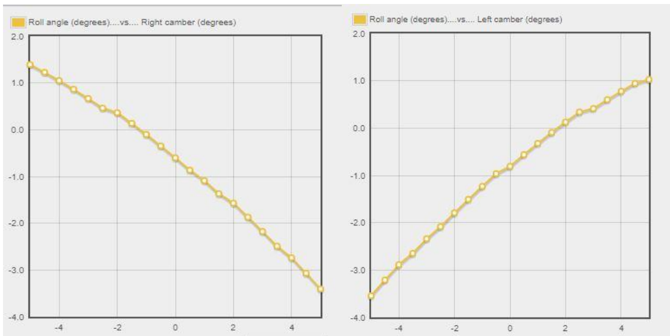
The process of assembly designing was initiated with at first geometrical parameter such as Roll Centre and Roll Height setup followed by proper Wheel alignment angles and plotting its graphical analysis.

The initial setup affixed with some parameters:




**The Right Wheel Camber** was adjusted to approx. $-1.5^0$ as stated already before with requisite values of Roll Angle, Roll center so as to stabilize the camber value.

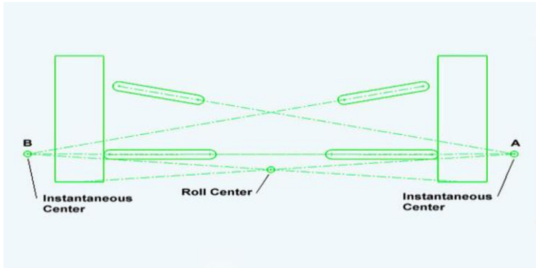
Some geometrical attributes related to camber procure certain plots as:-



Right Camber vs Roll Angle Left Camber vs Roll Angle

**Roll Center adjustments:** The roll centre positions of your front and rear suspension geometry are key features affecting the lateral load transfer rates of your front and rear axles. Their position and difference in height front to rear can be used to tune the roll stiffness distribution of your car in a similar way that you would adjust the stiffness of your front and rear roll bars to tune understeer and oversteer.

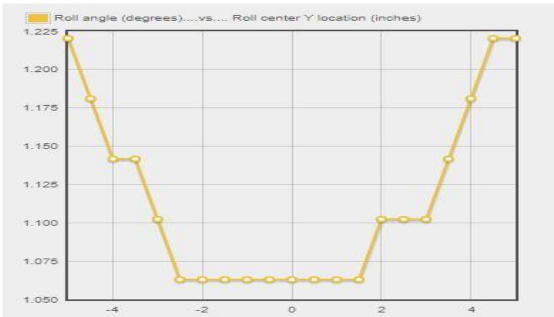
If the roll centre is located above the ground the lateral force generated by the tire generates a moment about the instant centre, which pushes the wheel down and lifts the sprung mass. This effect is called Jacking. If the roll centre is below the ground level the force will push the sprung mass down. The lateral force will, regarding the position of the roll centre, imply a vertical deflection. If the roll centre passes through the ground level when the car is rolling there will be a change in the movement direction of the sprung mass.




Geometrical setup to get Roll center

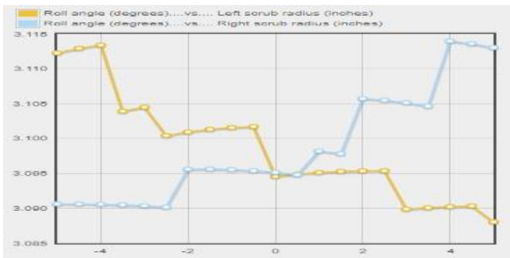
*Decision:*

Roll centre Height of Front suspension was kept 22mm and of Rear Suspension as 58mm. Rear Roll Centre Height is kept more to keep our car aerodynamically stable at High speed also. The graphical comparison with camber angles were plotted as :-



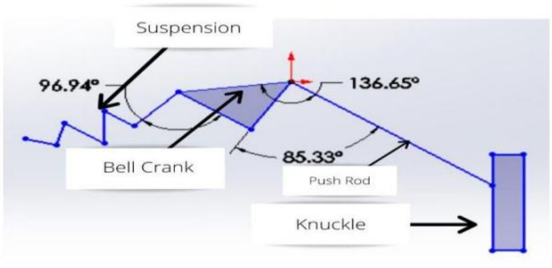

Roll Centre vs Right Camber

**Kingpin Inclination:** As stated above the KPI was setup to $2.5^0$ in front and $2^0$ in rear the requisite *scrub radius* was further calculated and plotted alongside Roll angle variations under dynamical aspect of vehicle. The graph is observed for both bump and droop conditions which is on left and right side plot respectively.



### 2.3 SUSPENSION COMPONENTS DESIGN

The initial iteration of wheel alignment and geometrical setup ascertain certain suspension hard points both in chassis and knuckle for different parts accommodation in between such as control arms, bell crank, pushrod and coilover (dampers) in an aesthetic manner.




Geometrical setup of suspension parts

**Material Selected for suspension components:**

The material selected for suspension design components including A-arm and other mountings & platforms for the vehicle was ***AISI 4130 Mild Steel*** which embodies a good range of supportive features to be ideal for a Vehicle system.

**Suspension Control Arms**: A-Arms or Control Arms design started with CAD geometry drawing using suspension compartments and considering track width, wheelbase and other similar parameters. Selection of material for A-Arm was done as per Material availability and Machining cost.

The CAD model of control arms with its other sections is depicted as:-

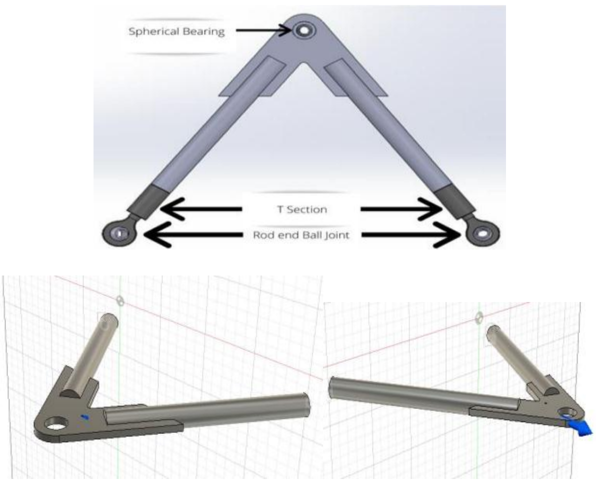
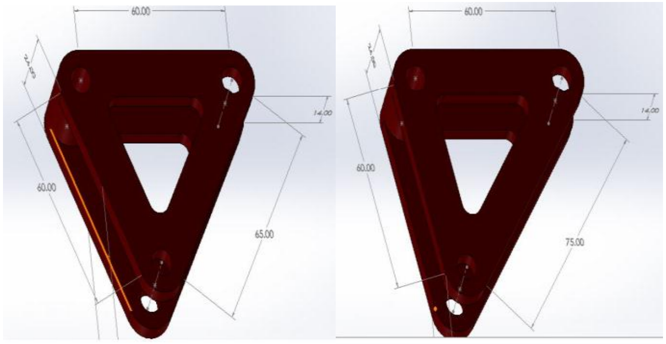



Figure 3: Lower and Upper Control Arms

**Bell Crank**: A bell crank sometimes depicted as rocker arm aids in transferring bump and droop motion during vehicle translation from wheels via pushrod to suspension dampers in order to ascertain healthy ride comfort. To overcome the unequal load distribution which occurs with the reactive balance beam suspension when either driving or braking, a non-reactive bell crank lever and rod linkage has been developed which automatically feeds similar directional reaction forces to both axle rear spring end supports. After proper dimensioning and load calculations the CAD model of bell crank was setup.



**Front Bell Crank Rear Bell Crank**

The well dimensioned CAD model of Bell Crank is depicted here which outlines its geometrical structure.

**Knuckle & Hub**: Knuckle is the non-rotating part in the wheel assembly. The stub axle is welded on the knuckle. Any change in the shape of knuckle can lead to the change in the suspension geometry. The rear hubs were designed in parallel with the front hubs, since they are the main interface between the suspension and the tire. In order to simplify manufacturing, the rear hubs were created identical to the front hubs, with the exception of the wheel bearing preload hardware and the addition of a torque transmission method. However, the hub is completely encased in this region by the wheel bearing which is Angular contact roller bearing, so no yielding should occur.

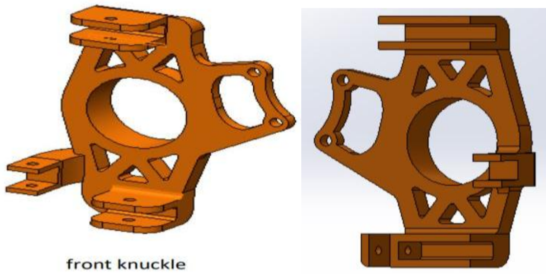



Figure 4: Front and Rear Knuckle

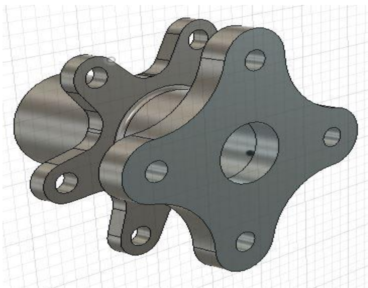


**Figure 5: CAD model of Wheel HUB**

*Assembly:* Firstly the knuckle is taken. The bearing is press fitted on the knuckle. Then the spacer between the knuckle and hub is press fitted on the bearing in order to accommodate with tire. After wards the inner race of the bearing is press fitted on the hub. The knuckle is taken and the outer race of the bearing is press fitted in the knuckle.

*Bearings:* The bearing installed with the above system was an Angular contact roller bearing with the purpose of its selection to withstand both axial and longitudinal loads during dynamic motion of the vehicle.

**Suspension Dampers:** Suspension shockers/dampers are the prime functional part of suspension assembly which dampens the incoming loads through the wheel encountered during vehicle's translation. The suspension damper employed in the system was a coilover structure type in which damper containing fluid is coiled over with the spring to bear the compressions. The aforementioned geometrical assembly of suspension quite well establishes the various linkages where dampers are linked to bell crank which in turn receives the forces from knuckle through pushrod assembly. The CAD model of suspension damper and spring was designed on SolidWorks after multiple dimensioning over chassis for proper adjustments in order to maintain desired stroke of the coilover and the spring was designed after enough calculations.

*Design of Spring (Calculation involved)*: The designing of coilover spring enlists calculation involved in finding out the coilover spring attributes such as free length, mean diameter, Number of active turns and Total number of turns.

At first we consider weight acting on single wheel keeping the view of sprung and unsprung mass and move further as:

Total weight of vehicle: 310 kg

Sprung weight: 210 kg

Unsprung weight: 100 kg

Weight distribution (front: rear): 45:55

Front weight: 139.5kg

Front sprung weight: 94.5 kg

Rear weight: 170.5 kg

Rear sprung weight: 115.5 kg

Front sprung mass on 1 wheel: 94.5/2 = 47.25 kg

As we know ride frequency, ride rate and sprung mass are related as:

$\omega_{rf} = \sqrt{\frac{K_r}{w_1}}$ where $K_r$ is ride rate & $w_1$ is sprung wt.-1wheel

Assuming **ride frequency ($\omega_{rf}$) of 2.5** Hz as in the case of most FSAE sport vehicles and using weight as 47.25 kg as sprung mass over 1 wheel at front:

Ride rate $= \omega_{rf}{}^2 \times w_1$

$= 2.5^2 \times 47.25 = 295.31$ kg-Hz²

The dimension of the selected tire for our FSAE vehicle was 20"-7.5"-13" with a tire rate of 19.16 N/mm

As we know:

$K_r = \frac{K_w \times K_t}{K_w + K_t}$, where $K_w$ &$K_t$ are wheel rate and tire rate respectively:

Putting values of tire and ride rate from above, we get:

Wheel rate ($K_w$) = 18.2 N/mm

Spring rate/stiffness = Wheel rate /Motion ratio

We have kept installation ratio, i.e.

Motion ratio (=$\frac{springtravel}{wheeltravel}$) to be 1.

So spring rate/stiffness ($K_s$) is 18.2 N/mm in front suspension assembly.

*Design of spring* : Initially we first use deflection of spring formula which relates spring rate and shear modulus of spring material as:

$K_S = \frac{G \times d^4}{8D^3 \times N}$, G=shear modulus; d=wire diameter; D= mean diameter; N=no. of active coils

Keeping $K_s$=18.2 N/mm and G= 10169 psi

we get $\frac{d^4}{D^3 \times N} = 1.441$ ……………..(i)

Later from Tresca theory we find maximum shear stress as half of ultimate tensile strength and relate as:

$\tau_{\max} = \frac{8 \times D \times F_S}{3.14 \times d^3} \times K$, $\tau_{\max}$ (shear stress) = 56 psi

$F_s$= w1/Motion ratio = 47.25 N & K=8(Wahl's factor)

We get $D/d^3$ from above and later we solve with (i) to get **D= 45.13 mm, d=8.8 mm and N= 5 turns and $N_t$= 7**

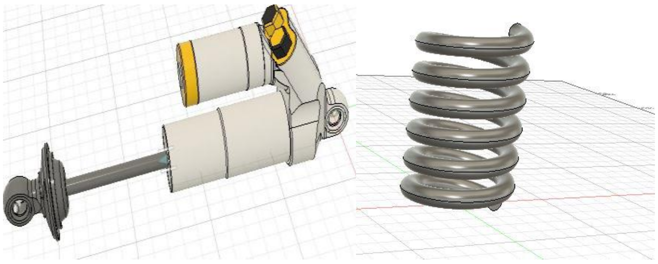


Figure 5: CAD model of damper and spring

**3. ANALYSIS**

The analysis was performed for various suspension assemblies like control arms, bell crank, suspension spring after proper load applications in order to ascertain stress and displacement factor of parts during action of load dynamically during vehicle in motion.

**Geometry Considered:**

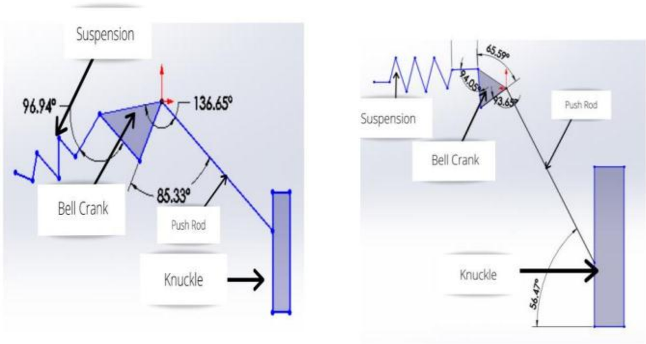



Figure 4: Front and rear geometrical figures

**Analysis of Control Arms:** For control arms analysis we start with load calculations in front and rear suspension's A-arms both upper and lower during knuckle movement encountering cornering or braking and the same with the chassis movement.

**Load Calculations:**

Max cornering force= 3720N

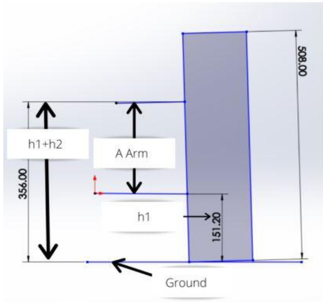
Total lateral weight transfer

= (cornering force) x (cg height) / (trackwidth)

= (3720 x 288)/1300 = 824.12N

The calculation was performed based on A-arm position in reference to knuckle mounted to the wheel hub.

*Front outer:*




Vertical wheel load = 1305.6N

On applying dynamic multiplication factor

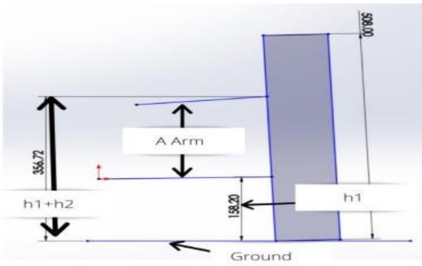
$F_{vertical}$ = 1.3x1305.6 = 1697.28N

$F_{lateral}$ = 1.2x 1697.28N = 2036.7N

Force on upper A-arm = (Flat xh1)/h2 = 1503.62N

Force on lower A-arm = (Flat xh1+ h2)/h2 = 3542.3N

**Rear A-arm:**




Outer vertical wheel load = 1068.35

On applying dynamic multiplication factor

Fvertical = 1.3x1068.35 = 1388.8N

Flat = 1.2x1388.8 = 1666.67N

Force on upper A-arm = (Flat xh1)/h2 = 1328.3N

Force on lower A-arm = (Flat xh1+ h2)/h2 = 2994.8N

*Output:*

| COMPONENTS | LOAD | CONSTRAINTS DOF arrested | DEFLECTION (mm) | F.O.S |
|---|---|---|---|---|
| Front upper A-arm | 1328N | Outward direction of mounting point | 0.006mm | 25 |
| Front lower A-arm | 2768N | Along the action line of push rod | 1.25mm | 1.13 |
| Rear Upper A-arm | 1503N | Outward direction | 0.0058mm | 23 |
| Read lower A-arm | 3542N | Inner side | 0.015 | 8.5 |

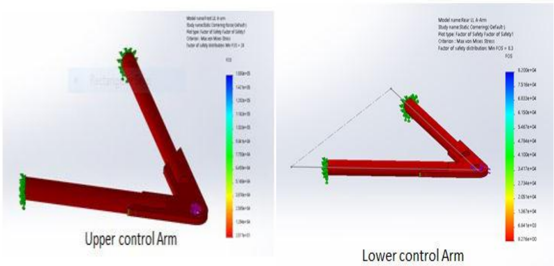
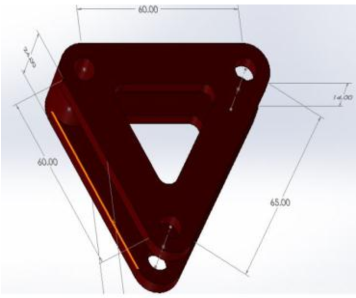



Figure 6: ANSYS output of Upper and Lower A-arm

**Bell Crank Analysis:** The bell crank is in attachment to pushrod acted over by respective loads which is calculated as under:

*Load Calculations:*




Maximum vertical load per side = 4557N

Rear wheel design vertical load = 2506.35N

Front wheel design vertical load = 2050.65N

$F_{pushrod}$ front = vertical load/sin(ϴ) = 2050/sin(66.1)

=2242.8N

$F_{pushrod}$ rear = vertical load/sin (ϴ) = 2506/sin(56.47)

= 3019.2N

Max force which is applied on the bell crank is 2242.8N for front & 3019.2N for rear.

*Output:*

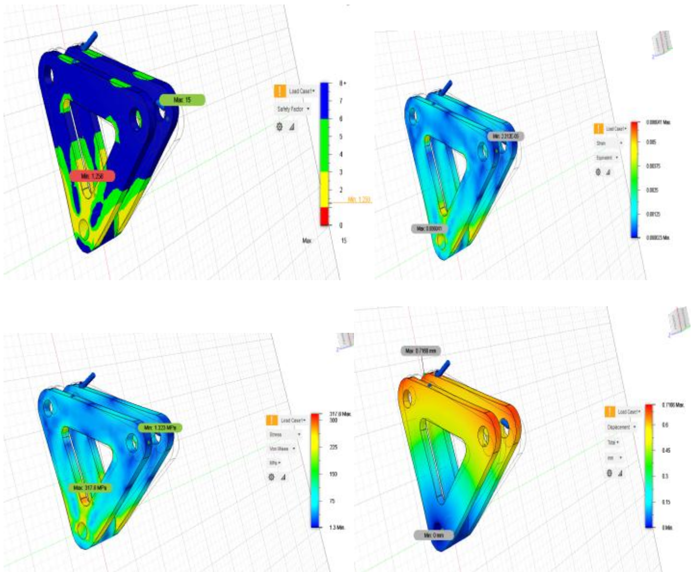


Figure 7: Analysis w.r.t various parameters

**Suspension spring simulation**:

*Load Calculation:*

The load will be transmitted parallely without any aberration from pushrod to suspension spring via bell crank so as to compress it towards chassis.

So,

$F_{front\ suspension}$=2242.8N

$F_{rear\ suspension}$= 3019.2 N

Fixing the other end as it is to be mounted on chassis and applying loads on the another end as calculated above , we get the output over Fusion 360 as:

*Output:*

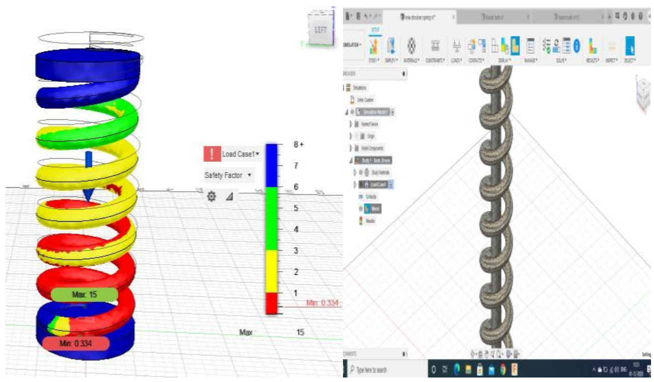



Figure 8: Analysis of spring

### 3.1 ANALYSIS OF COMPLETE SETUP ON LOTUS

The complete setup of suspension assembly ranging from wheel & knuckles to control arms arranged altogether were iterated finally on LOTUS suspension simulation software so as to ascertain Pitch and Roll Analysis for an efficient suspension system. The different values obtained as calculated above were input and the results of setup were iterated for both static and dynamic condition of Vehicle.

TYPE 14 Double Wishbone, Push Rod to damper

INCREMENTAL GEOMETRY VALUES

| BUMP TRAVEL (mm) | CAMBER ANGLE (deg) | TOE ANGLE (deg) | CASTOR ANGLE (deg) | KINGPIN ANGLE (deg) | DAMPER RATIO [-] | SPRING RATIO [-] |
|---|---|---|---|---|---|---|
| -30.00 | -1.2765 | -0.6583 | 5.9925 | 2.4980 | 1.735 | 1.735 |
| -20.00 | -1.2652 | -0.5933 | 5.9924 | 2.4798 | 1.579 | 1.579 |
| -10.00 | -1.2821 | -0.5550 | 5.9925 | 2.4927 | 1.444 | 1.444 |
| 0.00 | -1.3265 | -0.5414 | 5.9927 | 2.5357 | 1.321 | 1.321 |
| 10.00 | -1.3981 | -0.5507 | 5.9930 | 2.6083 | 1.204 | 1.204 |
| 20.00 | -1.4970 | -0.5814 | 5.9935 | 2.7104 | 1.086 | 1.086 |
| 30.00 | -1.6236 | -0.6325 | 5.9942 | 2.8423 | 0.956 | 0.956 |

<b> REAR SUSPENSION - BUMP TRAVEL</b>
RHS WHEEL (+ve Y)

TYPE 14 Double Wishbone, Push Rod to damper

INCREMENTAL GEOMETRY VALUES

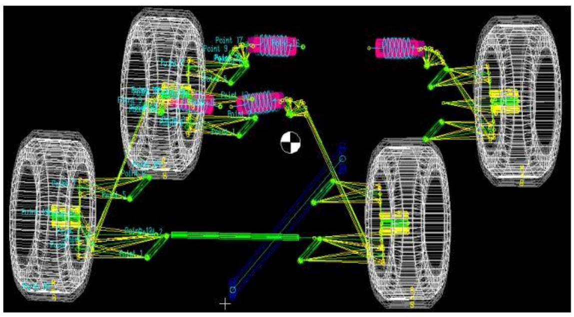
| BUMP TRAVEL (mm) | CAMBER ANGLE (deg) | TOE ANGLE (deg) | CASTOR ANGLE (deg) | KINGPIN ANGLE (deg) | DAMPER RATIO [-] | SPRING RATIO [-] |
|---|---|---|---|---|---|---|
| -30.00 | -0.3223 | 0.3776 | -5.9964 | 1.2758 | 1.281 | 1.281 |
| -20.00 | -0.5602 | 0.3492 | -5.9970 | 1.5107 | 1.276 | 1.276 |
| -10.00 | -0.8151 | 0.3245 | -5.9977 | 1.7630 | 1.272 | 1.272 |
| 0.00 | -1.0882 | 0.3034 | -5.9987 | 2.0339 | 1.271 | 1.271 |
| 10.00 | -1.3810 | 0.2865 | -5.9998 | 2.3249 | 1.278 | 1.278 |
| 20.00 | -1.6950 | 0.2739 | -6.0012 | 2.6376 | 1.298 | 1.298 |
| 30.00 | -2.0324 | 0.2664 | -6.0029 | 2.9742 | 1.342 | 1.342 |

*The bump and roll simulation output was confirmed after bump force calculations on wheel:*

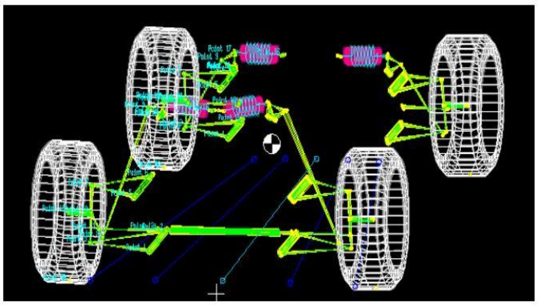
At the time of a bump in the surface a force will act on the portion of the bearing which is inside the knuckle. This is because the hub is bolted directly to the wheel. This force is obtained from the wheel rate. For design purpose the wheel rate is kept as $45 N/mm^2$.

Also it is considered that there will be no bump more than 30mm as the track is extremely flat.

Bump Force = Wheel rate × Travel due to bump = 45× 30 =1350 N



Bump simulation



Roll simulation

Figure 9: Bump and Roll simulation

### 4. CONCLUSIONS

This paper sums up the basic design and analytical concepts of the suspension system used in student formula car. After designing wishbones, knuckle and spring 3D model built with the help of CATIA and analysis done on ANSYS workbench. Integral assemblies were simulated on LOTUS software. Results obtained by simulation match with designed parameters. And after analyzing various suspension components, the paper lays down a methodology for analysis of different components. The FSAE guidelines have been thoroughly followed while working on this paper. Besides, it is to be noted that "Iteration is the key to perfection".

### ACKNOWLEDGEMENT

We would like to thank Team Alpha4ZE, Jamia Millia Islamia for their continuous help and support throughout the project.

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