IJCRT.ORG ISSN : 2320-2882

INTERNATIONAL JOURNAL OF CREATIVE RESEARCH THOUGHTS (IJCRT)

An International Open Access, Peer-reviewed, Refereed Journal

Design, Calculations & Analysis Of Double Wishbone Independent A-Arm Suspension System

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ABSTRACT:

The Double Wishbone Independent A-Arm Suspension System is a best choice among all other types of Independent Suspension Systems for enhancing handling, stability, and performance in high-performance vehicles. This study focuses on the design, calculations, and analysis of the suspension system, incorporating direct coil-over spring dampers mounted from the lower A-arm to the chassis at the front and from the upper A-arm to the chassis at the rear, optimizing ride comfort and dynamic response.

Lotus Suspension Analysis Software determines the suspension geometry and hard points, calculating camber gain, spring rate, bump load, caster, and roll center for optimal kinematics. The steering knuckle's structural integrity is analysed using ANSYS simulation, evaluating factor of safety (FOS), total deformation, and equivalent stress under dynamic loads. Solidworks software is employed for 3D modelling, ensuring a lightweight yet robust design that balances strength and weight optimization for improved vehicle dynamics.

This study provides detailed insights into force distribution and real-world applicability, with Lotus, ANSYS, and SolidWorks ensuring a comprehensive validation of performance parameters, making it ideal for high-performance applications.

Keywords: Double Wishbone Suspension, A-Arm Suspension, Coil-Over Spring Dampers, Suspension Geometry, Lotus Suspension Simulator, Camber Gain, Spring Rate, Bump Load, Caster Angle, Roll Center, Steering Knuckle, ANSYS Simulation, Factor of Safety (FOS), Total Deformation, Equivalent Stress, SolidWorks 3D Modelling, Vehicle Dynamics, Performance Vehicles.

1. INTRODUCTION

The suspension system plays a crucial role in ensuring both vehicle safety and occupant comfort by providing stability and control during various driving maneuvers. Without a properly designed suspension, a vehicle would be extremely difficult to handle, as every shock, bump, and vibration from the road would be directly transferred to the chassis and steering system without any damping**.** This would not only compromise driving comfort but also significantly reduce traction, making it harder for the driver to maintain control, especially on uneven or rough surfaces. A well-engineered suspension system absorbs these disturbances, ensuring a smoother ride, improved handling, and enhanced overall safety**.**

For Suspension by taking the ride stability and performance of vehicle into consideration we have decided to use Double Wishbone independent A-arm suspension system would be the optimal choice for our application. When compared with other Independent suspension types (MacPherson Strut, Multi-Link, Trailing Arm, Swing Axle Suspension, etc.), vehicles equipped with a double wishbone system offer superior driving comfort and ride quality. This advantage primarily stems from its independent suspension design, where each wheel operates separately. As a result, road disturbances such as bumps and uneven surfaces primarily affect only the contacted wheel, minimizing disruptions to overall vehicle stability and comfort, which is not the case in Dependent Suspension System.

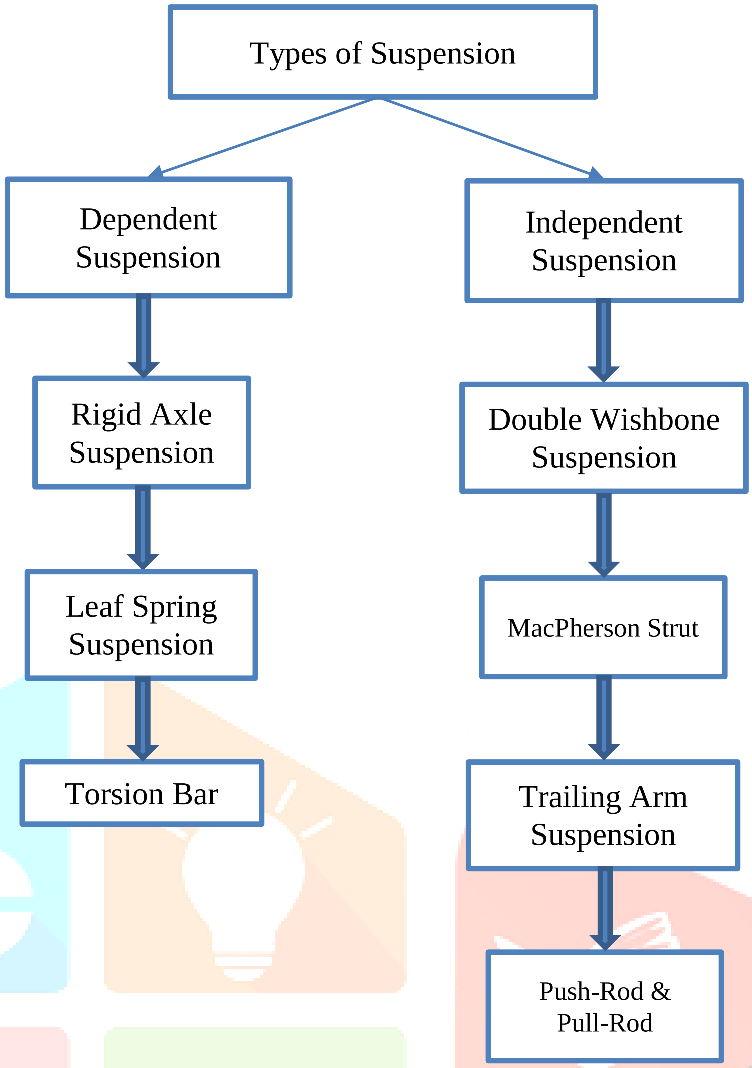
Following this, we conducted an in-depth study of Double Wishbone push-rod mechanism, pull-rod mechanism, and direct coil-over dampers mechanism, evaluating their advantages and limitations. After careful analysis, we decided to implement a direct coil-over damper setup due to its higher load-carrying capacity, simpler design, and minimal shift in the center of gravity (C.G.), making it highly suitable for performance vehicles.
Among various options, we selected the DNM RCP-2S coil-over shock absorber, considering its optimal spring rate, ride comfort, and high load-bearing capability.

To refine our suspension geometry, we utilized Racing Aspiration software, iterating through different configurations to analyse camber change rate, roll center offset, instantaneous center, kingpin inclination (KPI), and other critical parameters. Based on our simulations, we opted for a positive swing arm geometry, as it results in a lower camber change rate and a minimal roll center offset during cornering, ensuring better handling stability and increase in roll over stability.

Furthermore, we carefully estimated the roll center position, ensuring it is neither too close nor too far from the center of gravity (C.G.). A roll center positioned too close to the C.G. can lead to excessive body roll, while one positioned too far can induce vehicle jacking, affecting cornering dynamics. By selecting a balanced roll center position, we aim to enhance vehicle stability, mitigating roll-over tendencies while maintaining optimal handling performance.

This approach ensures that our suspension system achieves an optimal balance between performance, stability, and ride comfort, making it well-suited for high-performance applications.

This paper covers all the necessary technical parameters like Calculations of spring & dampers, Steering Knuckle & Hub Calculations, selection of the spring, etc. required to build a fully functional and dynamically optimized suspension system of a performance vehicle.



2. LITERATURE REVIEW

The double wishbone suspension system has been widely studied for its superior handling, ride comfort, and stability. Various researchers have analysed different aspects of suspension kinematics, roll center determination, and optimization techniques to enhance vehicle performance.

Chepkasov et al. [6] studied the suspension kinematics of a Formula SAE sports car, focusing on the concept of cornering power: defined as the ratio of cornering force to slip angle. The slip angle is the angle through which the wheel must turn to sustain lateral forces, while the cornering force acts at a right angle to the wheel plane, countering the side thrust. This study provides valuable insight into the lateral stability and steering response of a double wishbone suspension system.

In the study by Kamesh Jagtap and Yogesh Rathod, “Suspension system for an all-terrain vehicle: A review” [7]Roll center can be defined in two different ways, one based on geometric roll center (kinematic roll center) and another based on force roll center.

While designing of suspension system we consider geometric based definition, because it plays a very important role in deciding the wishbone arm lengths and the geometry of wishbones. Geometric roll center also helps in determining length of tie rods, it is expected that both upper and lower A-arms and tie rods in a suspension system follow same arc of rotation also known as Bump steer while cornering whose center is known as instantaneous center. The aim of our study is to design and analyse the coil over shock absorbers, A-arms, Steering Knuckle, Hub a double wishbone suspension system. The process includes study of the suspension parameter in LOTUS. These parameters are optimized for the desired performance through iterative procedures. The best design is then modeled in SolidWorks and analysed for strength in Ansys.

Afkar et al. [2] conducted an optimization study of a double wishbone suspension system using ADAMS software and a genetic algorithm. Their research compared three configurations:

- Standard double wishbone suspension
- Modified double wishbone suspension
- Optimized double wishbone suspension using genetic algorithms.

The study evaluated suspension geometry variations under different road conditions, steering angles, and bump impacts. The results indicated that the optimized suspension system provided the best performance in terms of stability, ride comfort, and handling characteristics.

2.1. Aim

The aim is to study different types of Suspension systems, comparing them and stating the advantages, disadvantages of each type. To do a detailed research on Double Wishbone Independent A-arm suspension system and in-depth calculations to achieve the spring rate/stiffness, Installation ratio, Critical Damping coefficient, Steering Knuckle and Hub calculations, etc. necessary to validate that the Double Wishbone suspension system is better than any other system for performance vehicles like Formula, BAJA, etc.

2.2. Research Objective

Building on these studies, our aim is to design and analyse a double wishbone suspension system with a focus on coil-over shock absorbers, A-arms, steering knuckles, and hub assemblies. The suspension parameters will be optimized using LOTUS software, followed by iterative refinements for performance enhancement. The finalized design will be modeled in SolidWorks and structurally analyzed in ANSYS to ensure durability and reliability.

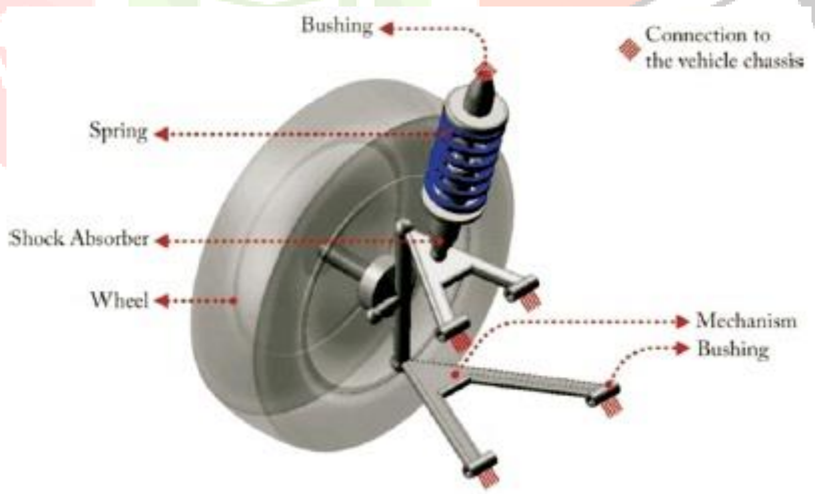


Figure No.1: Image showing Assembled Double Wishbone Suspension system

3. METHODOLOGY

3.1. Basic Parameters to be Known:

The basic parameters to be known before designing a Double wishbone suspension system are:

a. Caster Angle: It is an angle between the vertical center line of a wheel and the steering axis viewed from side view of a vehicle. If the upper point of steering axis is inclining towards the driver, it is called as Positive Caster. If the upper point of the steering axis is inclining away from the driver, it is called as Negative Caster.

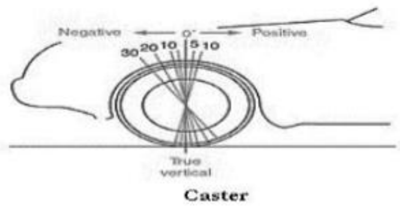


Figure No.2: Image of Caster Angles

b. Camber Angle: It is an angle between the vertical center line of a wheel and the center line of wheel when inclined inwards from the vertical line of wheel is called Negative Camber angle. If the Center line of wheel is inclined away from the vertical center line of wheel is called Positive Camber. It is viewed from front view of a vehicle.

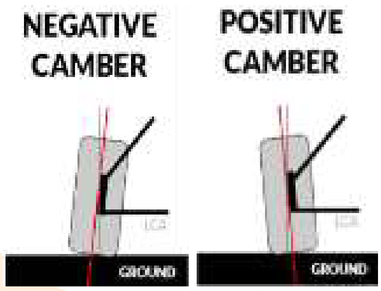


Figure No.3: Image of Camber Angles

c. Toe-Angles: When viewed from top view of a vehicle if the wheels are not perpendicular to each other but inclined away from each other at the top end, the angle is known as toe-out. If the wheels are inclined towards each other at the top end, the angle is known as toe-in.

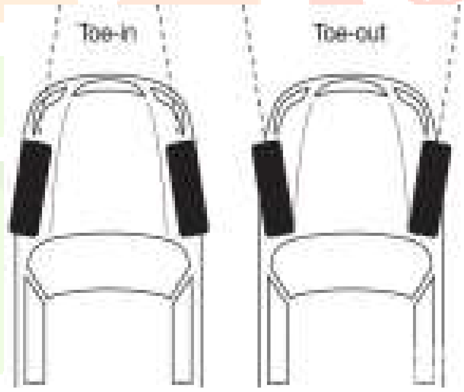


Figure No.4: Image of Toe Angles

d. Kingpin Inclination (KPI): It is an angle between two upper and lower mounts of the steering knuckle on which the upper and lower A-arms are mounted, when viewed from front view of a vehicle.

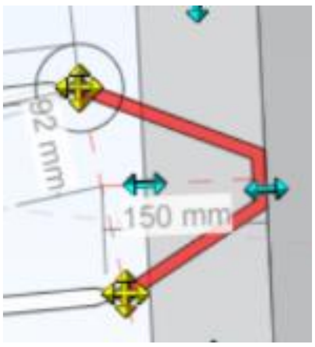


Figure No.5: Image of Kingpin Inclination line

3.2. Vehicle Specifications:

Table No.1: Technical Specifications of a Vehicle

Sr.No	Parameters	Specifications	
		Front	Rear
1	Caster	4˚	0˚
2	Camber	0˚	0˚
3	Toe Angle	0˚- 2˚	0˚
4	KPI Angle	6˚	0˚
5	Scrub Radius(Sr)	26.2mm	0mm
6	Track Width(t)	1125mm	1125mm
7	Wheelbase(WB)	1720mm	
8	Height of Center of Gravity(C.G.)	463.906mm	
9	Height of Roll Center	105mm	115mm
10	Ground Clearance(G.C.)	190mm	200mm
11	Suspension System	Double Wishbone Independent A-arm	
12	Type of Suspension(Direct Suspension)	Damper to lower A-arm	Damper to Upper A-arm
13	Type of Jounce Damping	hysteretic	hysteretic
14	Type of Rebound Damping	hysteretic	hysteretic

3.3. Design of Spring:

a. For designing the spring using a spring rate/ spring stiffness calculators (online), the parameters are required are wire diameter, outer diameter of the coil, pitch, number of coils, active number of coils, type of coils whether(open, closed & square ended, closed & grounded, Double closed ended), free length, solid length, material of the coil, etc.

Table No.2: Front Suspension Spring Specifications

Diameter of spring wire, d:	**8.000 mm**
Outer diameter of spring, D_{outer} :	**51.000 mm**
Inner diameter of spring, D_{inner} :	**35.000 mm**
Mean diameter of spring, D_{mean} :	**43.000 mm**
Free length of spring, L_{free} :	**200.000 mm**
Number of active coils, N_a :	**13**
Number of total coils, N_T :	**15**
Solid height, L_{solid} :	**128.000 mm**
Type of ends:	**closed & squared**
Spring index, C :	**5.375**
Distance between coils, $Coil_{pitch}$:	**13.538 mm**

Table No.3: Rear Suspension Spring Specifications

Diameter of spring wire, d:	**10.000 mm**
Outer diameter of spring, D_{outer} :	**60.000 mm**
Inner diameter of spring, D_{inner} :	**40.000 mm**
Mean diameter of spring, D_{mean} :	**50.000 mm**
Free length of spring, L_{free} :	**200.000 mm**
Number of active coils, N_a :	**13**
Number of total coils, N_T :	**15**
Solid height, L_{solid} :	**160.000 mm**
Type of ends:	**closed & squared**
Spring index, C :	**5.000**
Distance between coils, $Coil_{pitch}$:	**13.077 mm**

b. Suspension System Calculations:

Along with calculating the spring rate from the above derived values of the spring design, for a vehicle to be dynamically optimized and achieving all the desired goals i.e. for formula the track is a smooth track with minimal obstruction, unevenness, etc. but while designing a spring or an entire suspension system for an all-terrain vehicle like BAJA in our case needs to have maximum wheel travel to overcome the obstruction and climb anywhere as per the requirement on the track, the track is full of obstruction and unevenness in that case the terms like motion ratio, wheel rate, spring rate. ride frequency, roll rate, tire rate, damping coefficient, etc. are equally important for making the ride comfortable as well as dynamically robust.

Following are the detailed Calculations of Double Wishbone Suspension System:

- **Motion Ratio (MR):-**

1) Front

(MRf) = X1/X2

(MRf) = 212.845/280.06

MRf = 0.76

2) Rear

(MRr) = X1/X2

(MRr) = 178.92/235.42

MRr = 0.76

- **Installation Ratio(IR)=1/MR**

1) Front

IR=1/0.76

(IRf) = 1.3

2) Rear

IR=1/0.76

(IRr) = 1.3

- **Wheel Rate(Kw):-**

1) Front

Kwf = Static load/Wheel Travel

Kwf = 65kg/40mm

Kwf = 1.625kg/mm

Kwf = 16305.166N/m

2) Rear

Kwr = Static load/Wheel Travel

Kwr = 120.25kg/40mm

Kwr = 3.00625kg/mm

Kwr = 30156.853N/m

- **Wheel Travel(WT):-**

1) Front

WTf = Load acting on wheel/Wheel Rate

WTf = 65/1.6305Kg/mm

WTf = 40.625mm

2) Rear

WTr = Load acting on wheel/Wheel Rate

WTr =120/3.0156Kg/mm

WTr = 40.23mm

- **Maximum Wheel Travel:-**

1) Front

WTf(max)= wheel travel at static load *(Solid Length of spring/ spring travel at static load)

WTf(max) = 40*(63.75/17.41)

WTf(max) = 146.467mm

2) **Rear**

WTr(max) = wheel travel at static load*(Solid Length of spring/ spring travel at static load)

WTr(max) = 40*90/24.165

WTr(max) = 148.976mm

- **Spring Rate(Ks):-**

1) **Front**

Ksf = Kwf/(MR)^2*cos45

Ksf = 16305.166/(0.76)^2*0.71

Ksf = 39922.068N/m

2) **Rear**

Ksr = Kwr/(MR)^2*cos30

Ksr = 30156.853/(0.76)^2*0.866

Ksr = 60287.6288N/m

- **Spring Travel (ST):-**

1) **Front**

STf = Force acting on spring/Spring Rate

STf = (65-15)/sin45/4.06953Kg/mm

STf = 17.41mm

2) **Rear**

STr = Force acting on spring/Spring Rate

STr = (120-15)/sin45/6.145Kg/mm

STr = 24.165mm

- **Maximum Spring Travel:-**

1) **Front**

STf (max) = 63.75mm

2) **Rear**

STr (max) = 90mm

- **Ride Rate(Kr):-**

1) **Front**

Krf = (Kwf*Kt)/(kwf+Kt)

Krf = (16.3052*144.9)/(16.3052+144.9)

Krf = 14.656N/mm

Krf = 14656N/m

2) **Rear**

Krr = (Kwr*Kt)/ (kwr+Kt)

Krr = (30.156*144.9)/(30.156+144.9)

Krr = 24.962N/mm

Krr = 24961.728N/m

- **Roll Rate(KΦ):-**

1) **Front**

KΦf = (t)^2*Krf/2

KΦf = (1.125)^2*14656/2

KΦf = 9274.5Nm/rad

2) **Rear**

KΦr = (t)^2*Krr/2

KΦr = (1.125)^2*24961.728/2

KΦr = 15796.09Nm/rad

- **Ride Frequency(ωr):-**

1) **Front**

ωrf = (1/2π)√ (Ksf/m)

ωrf = (1/2 π) √ (39922.068/370)

ωrf = 1.65Hz

2) **Rear**

ωrr = (1/2π)√ (Ksr/m)

ωrr = (1/2 π) √ (60287.628/370)

ωrr = 2.03Hz

- **Tire Rate**:-

1) Front = **144.9N/mm**

2) Rear = **144.9N/mm**

- **Damping Coeff.(C):-**

1) Front = **1320.92N-s/m**

2) Rear = **1990.382N-s/m**

- **Damping Ratio(ζ):-**

ζ=C/Ccr

1) **Front**

ζf = Cf/Ccrf

ζf = 1320.92/3221.76

ζf = 0.41………..underdamped

2) **Rear**

ζr = Cr/Ccrr

ζr = 1990.382/5379.41

ζr = 0.37………..Underdamped

- **Critical Damping(Ccr):-**

1) **Front**

Ccrf=2√(Ksf*m)

Ccrf=2√(39922.068*65)

Ccrf= 3221.76N-s/m………one wheel

2) **Rear**

Ccrr=2√(Ksr*m)

Ccrr=2√(60287.628*120)

Ccrr=5379.41N-s/m……….one wheel

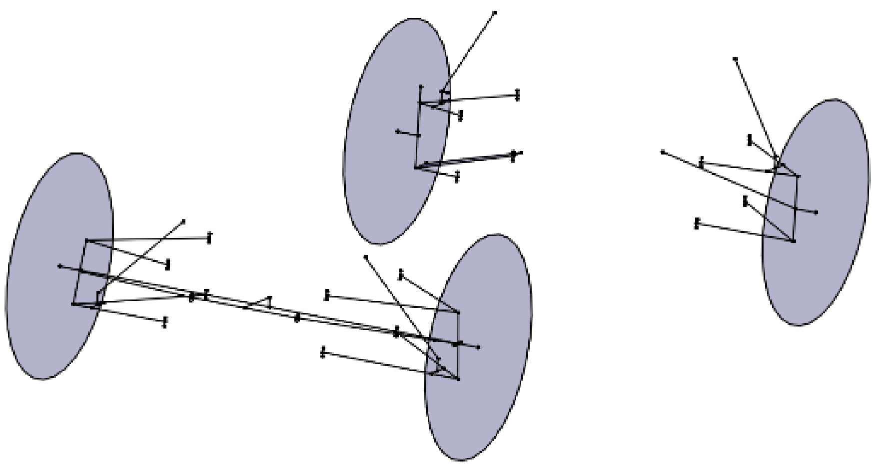


Figure No.6: Image of CAD of 3D Suspension Geometry

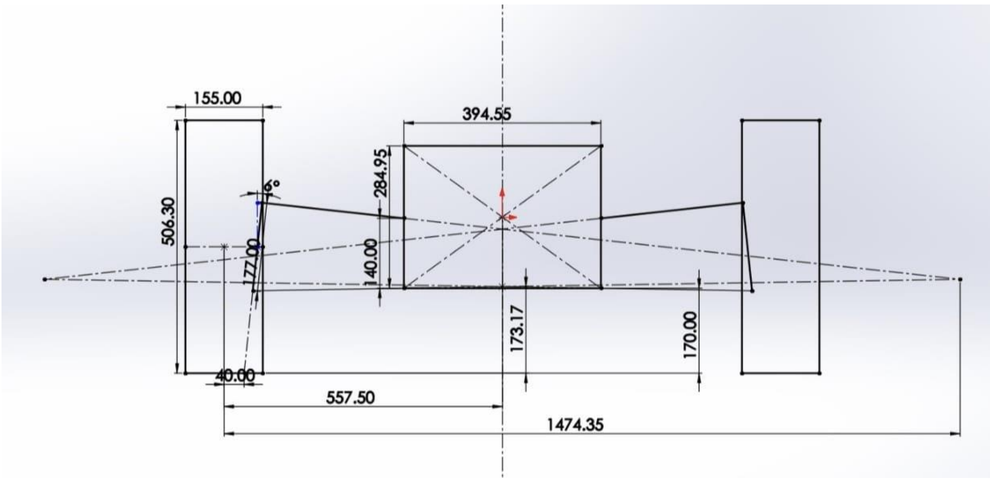


Figure No.7: Image of CAD of Front Suspension 2D Geometry

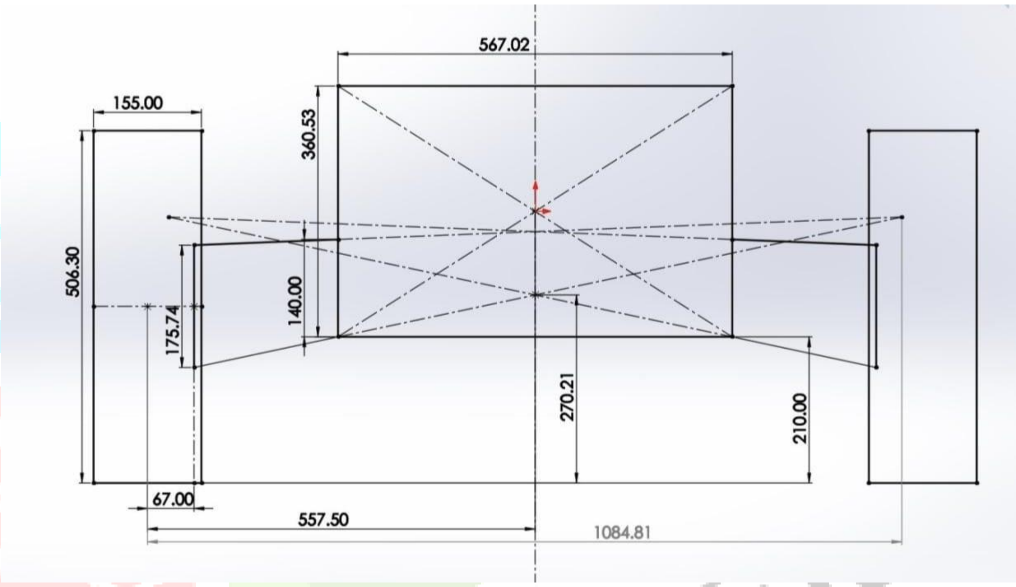


Figure No.8: Image of CAD of Rear Suspension 2D Geometry

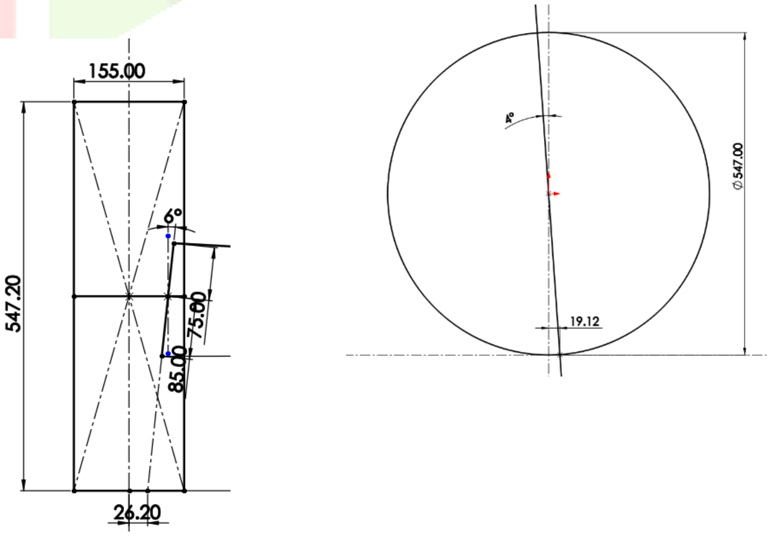


Figure No.9: Image of CAD of 2D Wheel Geometry

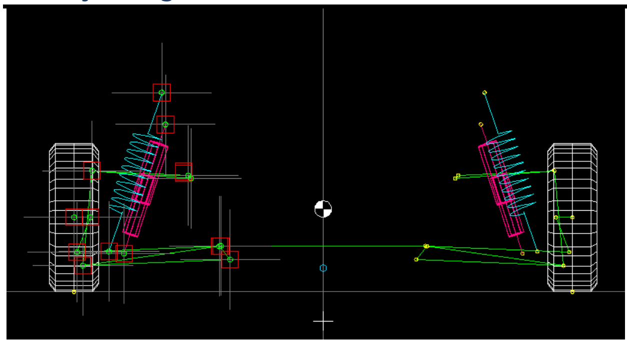


Figure No.10: Image of LOTUS Sim Front System

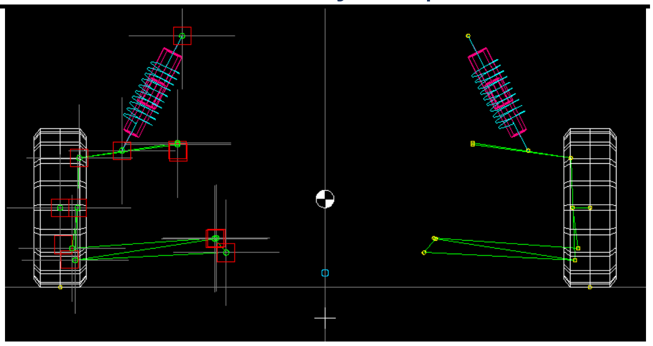


Figure No.11: Image of LOTUS Sim Rear System

Figure No.12: Images showing CAD model of Front & Rear Shock-Absorbers (Coil over Dampers)

c. Steering Knuckle & Hub Calculations:

While designing a Steering knuckle & Hub for a performance vehicle having a significant amount of weight of around 300kg (max), careful study and detailed calculations are required for calculating the loads acting on the Knuckle & Hub, the distribution of weight, the load transfer due to cornering and force acting on the knuckle and hub during static condition, bump, braking, etc.
Following are the formulas and calculations of the Steering Knuckle & Hub:

- **Static Load(SL):-**
  SL = load acting on one wheel(kg)*acceleration due to gravity(m/s^2)

1) Front
   SLf=65*9.81
   SLf= 637.65Non one wheel
2) Rear
   SLr=120*9.81
   SLr=1177.2N on one wheel
   1) Front= 637.65N on one wheel
   2) Rear= 1179.2N on one wheel

- **Bump Load(BL) = mx(V1-V2)/t**
  Where :-
  m= mass of the vehicle
  V1= initial velocity of vehicle before hitting bump
  V2= final velocity of vehicle after hitting bump
  t= total time the vehicle was in contact with bump
  BL = 370*(11.11-5.55)/2
  BL = 1028.6N…………….. on whole vehicle
  Bump on one wheel :-

1) Front(BLf)= 65*(11.11-5.55)/2 = 180.7N
2) Rear(BLr)= 120*(11.11-5.55)/2 = 333.6N

- **Longitudinal Load Transfer During Acceleration**:-

FL = (Fr*b)/h
Fr= (m*a*h)/b
Where :-
m= vehicle mass
a= vehicle acceleration
h= height of centre of gravity
b= distance of centre of gravity from rear axle
Fr= Dynamical vertical Force on Rear axle due to acceleration
Fr = (370*0.75*463.906)/602
Fr= 213.84N
FL=(213.84*602)/463.906
FL= 277.5N
: 28.28kg load gets transferred on rear axle while acceleration
Therefore the load acting on rear axle while acceleration is=> 28.28+240kg=268.787kg

- **Longitudinal Load Transfer During Braking**:-

1) At Front=2004.68N
2) At Rear= 1625.072N

1. Normal Force at Front and Rear Axle: -

Where, c is weight distribution,
N(front) = mass * g*c
= 370*0.35*9.81
= 1270.395 N
N(rear) = mass * g*c
= 370*0.65*9.81
= 2359.305 N

2. Weight transfer: -

(Wtrans) = (W*Hcg*a)/WB
= (370*463.907*0.75g)/1720
= 734.233 N
= **74.84 kg**

3. Dynamic weight transfer: -

Front= N(front) + (Wtrans)
=1270.395+734.233
= 2004.682N
= 204.345 kg

Rear = N(rear)- (Wtrans)
= 2359.305-734.233
= 1625.072N
=165.654 kg

- **Lateral Load Transfer due to lateral acceleration:-**

Wf = Ay(W/Tf)[(H*KΦf)/((KΦf+KΦr)+(a/l)*Zrf)]
Wr = Ay(W/Tr)[(H*KΦr)/((KΦf+KΦr)+(b/l)*Zrr)]
Where**:-**
Ay = Lateral Acceleration
W = weight of vehicle
Tf=Front track width
Tr=Rear track width
H=Distance Between C.O.G and Roll center
KΦf=Front Roll Rate
KΦr=Rear Roll Rate
a= distance of C.O.G from front axle

b=distance of C.O.G from Rear Axle
l= wheelbase
Zrf= Roll Centre Height Front
Zrr= Roll Center height Rear
Wf=0.65(370/1125)[(273.9*9274.5)/((9274.5+15796.09)+(1118/1720)*105)]
Wf=21.602kg=211.92N
Wr=0.65(370/1125)[(263*15796.09)/((9274.5+15796.09)+(602/1720)*115)]
Wr=35.367kg=346.95N
Force on wheel:-

1) **Front Outer= 86.6kg=849.56N**
2) **Front Inner= 43.398kg=425.734N**
3) **Rear Outer= 155.367kg=1524.15N**
4) **Rear Inner= 84.633kg=830.24N**

- **Force During Turning=47.87kg=478.7N**

- **HUB CALCULATIONS**:-

- **Braking Torque**:-

1) **Front:-**
   T= F(front)*Reff……….. T = torque
   T= 3.99*81.45
   T= 324.98Nm
2) **Rear**:-
   T = F(Rear)*Reff
   T= 3.99*66.022
   T= 299.64Nm

- **Lateral Load Transfer due to lateral acceleration:-**

Wf=Ay(W/Tf)[(H*KΦf)/((KΦf+KΦr)+(a/l)*Zrf)]
Wr=Ay(W/Tr)[(H*KΦr)/((KΦf+KΦr)+(b/l)*Zrr)]

1) **Front**
   Wf=21.602kg=211.92N
2) **Rear**
   Wr=35.367kg=346.95N

Force on wheel:-

1) **Front Outer = 86.6kg=849.56N**
2) **Front Inner = 43.398kg=425.734N**
3) **Rear Outer = 155.367kg=1524.15N**
4) **Rear Inner = 84.633kg=830.24N**

- **Bump Load(BL)**
  BL= m*(V1V2)/t=1028.6N ……on whole vehicle

1) **Front**
   BLf= 65*(11.11-5.55)/2 = 180.7N…..on one wheel
2) **Rear**
   BLr= 120*(11.11-5.55)/2 = 333.6N….. on one wheel

- **Static Load(SL):-**

1) Front= 637.65N….. on one wheel
2) Rear= 1177.2N….. on one wheel

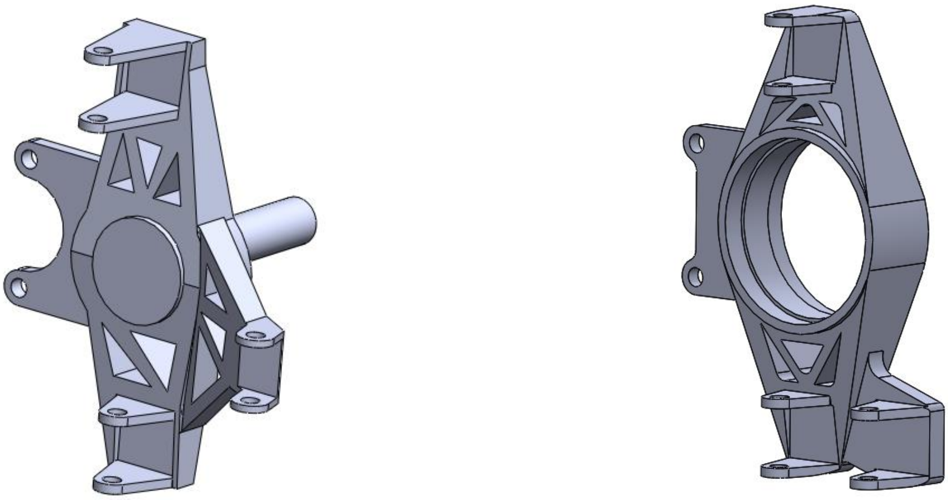


Figure No.13: Images Showing CAD of Front & Rear Steering Knuckles

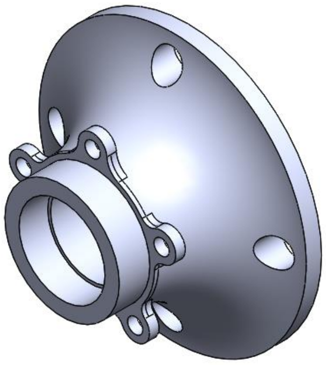


Figure No.14: Image of CAD of a HUB

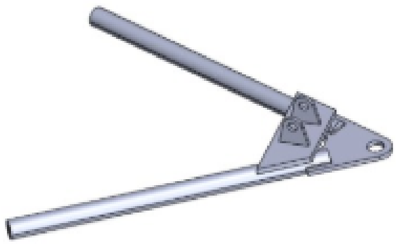


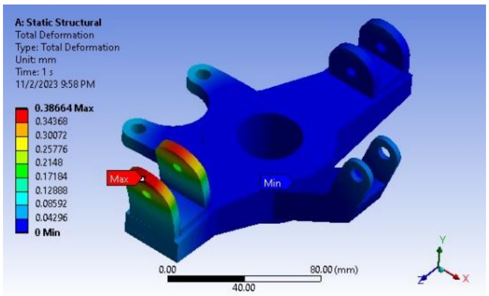
Figure No.15: Image of CAD of A-arm

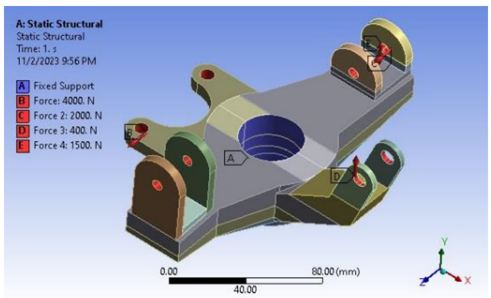
4. ANALYSIS & VALIDATION

After calculating and Designing the components of a Double Wishbone Independent A-arm Suspension System using SOLIDWORKS software.
It is important to analyse and validate the calculations and to check the structural integrity of the Component designed. The Softwares used for analysing the components and the suspension system are:

- ANSYS
- LOTUS SUSPENSION SIMULATION SOFTWARE
- RACING ASPIRATION

a. FEA of Front Steering Knuckle using ANSYS:





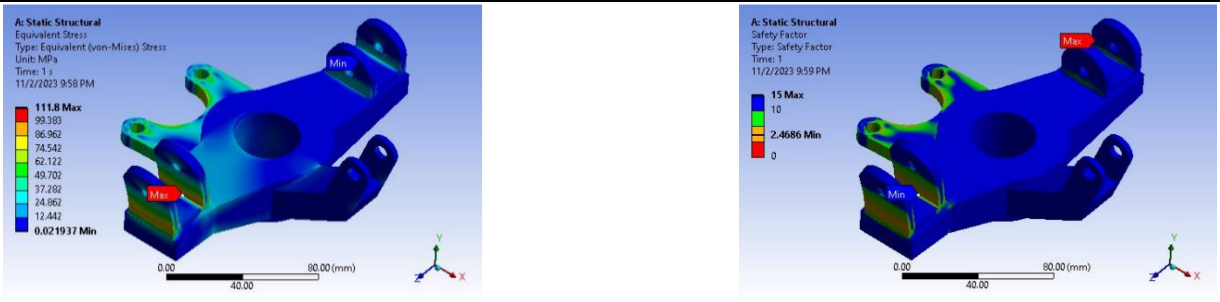


Figure No.16: Images showing the result of analysis of tests performed on Front Knuckle

b. FEA of Rear Steering Knuckle using ANSYS:

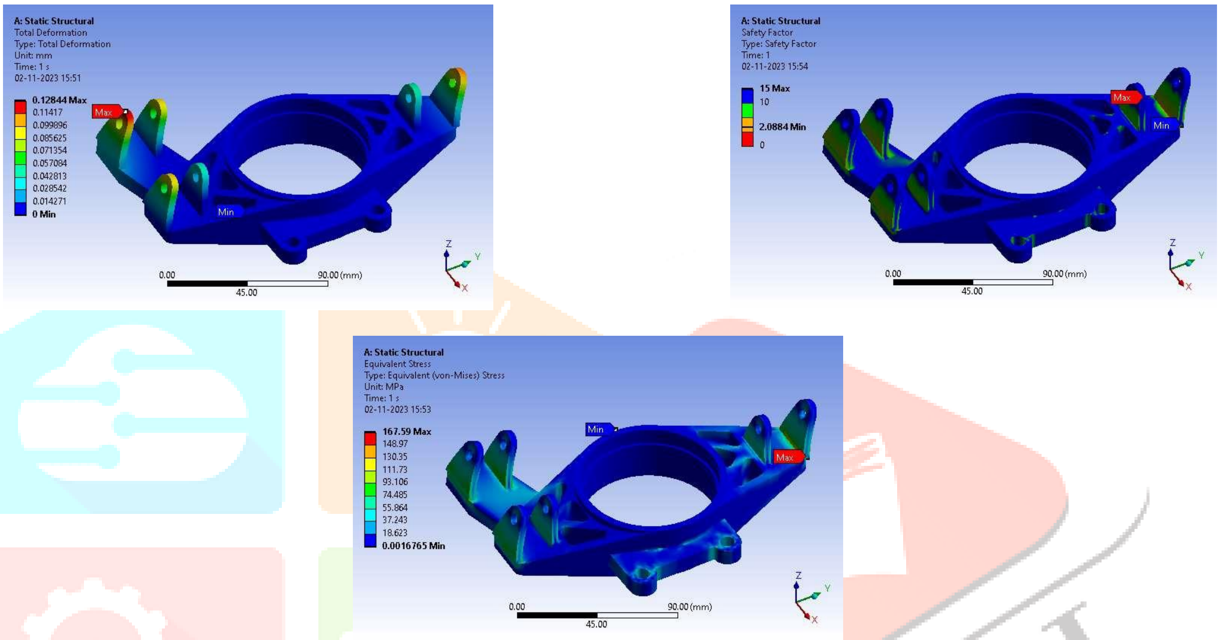


Figure No.17: Images showing the result of analysis of tests performed on Rear Knuckle

c. Analysis of Suspension system using LOTUS Simulation Software:

For simulating the suspension system on LOTUS we have first select which type of suspension system is to be simulated, in our case I have selected Double Wishbone Independent A-arm (Dampers to lower Wishbone) for the front & (Dampers to upper Wishbone) for the rear.

You can select according to the type of your vehicle and after simulating we can view the result of the simulation.

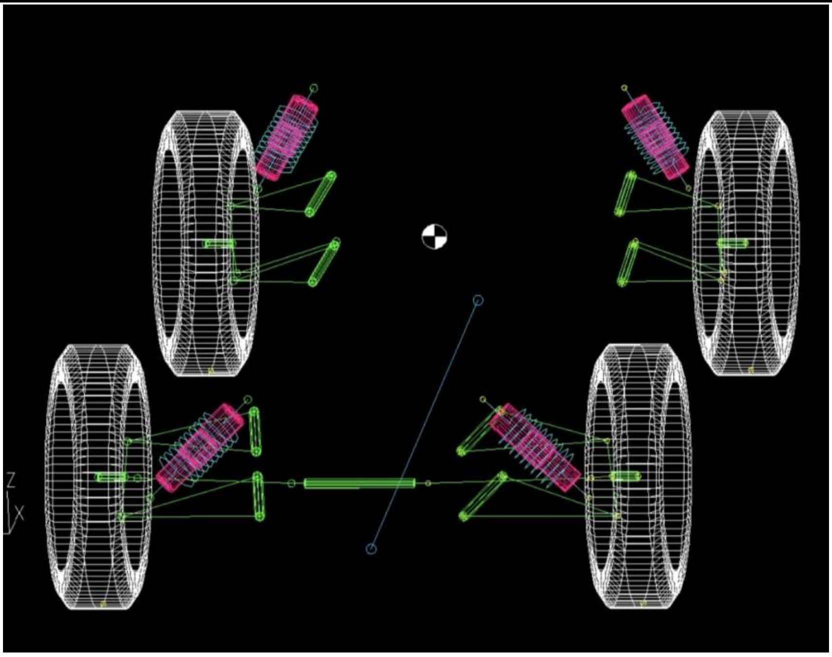


Figure No.18: Image of Suspension System on LOTUS

INCREMENTAL SUSPENSION PARAMETER VALUES

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)	ROLL CENTRE HEIGHT TO BODY (mm)	ROLL CENTRE HEIGHT TO GRND (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
-40.00	11.64	0.00	140.78	180.78	-10.16	-1.18	21.72	21.72
-30.00	10.11	0.00	130.96	160.96	-7.07	-0.72	16.37	16.37
-20.00	8.58	0.00	121.44	141.44	-4.36	-0.37	10.97	10.97
-10.00	7.05	0.00	112.16	122.16	-2.00	-0.13	5.51	5.51
0.00	5.49	0.00	103.09	103.09	0.00	0.00	0.00	0.00
10.00	3.91	0.00	94.18	84.18	1.66	0.02	-5.58	-5.58
20.00	2.28	0.00	85.39	65.39	2.98	-0.07	-11.22	-11.22
30.00	0.61	0.00	76.69	46.69	3.97	-0.27	-16.93	-16.93
40.00	-1.11	0.00	68.03	28.03	4.63	-0.58	-22.72	-22.72
50.00	-2.90	0.00	59.40	9.40	4.95	-1.02	-28.59	-28.59
60.00	-4.77	0.00	50.75	-9.25	4.94	-1.57	-34.54	-34.54

Figure No.19: Image showing the Values of the incremental suspension parameters on simulating

d. Analysis of Double Wishbone Suspension System using RACING ASPIRATION:

RACING ASPIRATION is an open source software typically used for placement and measuring the dimensions of the bulkheads and the components used in the suspension system to check whether the designed component with specific dimensions give the accurate dynamic results and the values like roll center offset, camber change, bump steer, height of the C.G., Instantaneous Centres, change in track width, etc.
Also proving whether the springs are accurately designed according to their required stiffness.

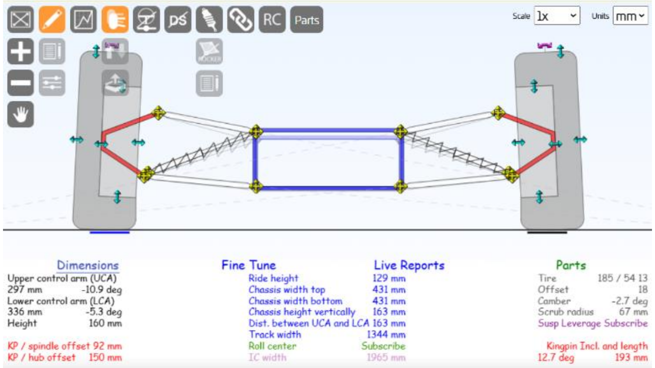


Fig No.20: Image showing Dimensions of parts

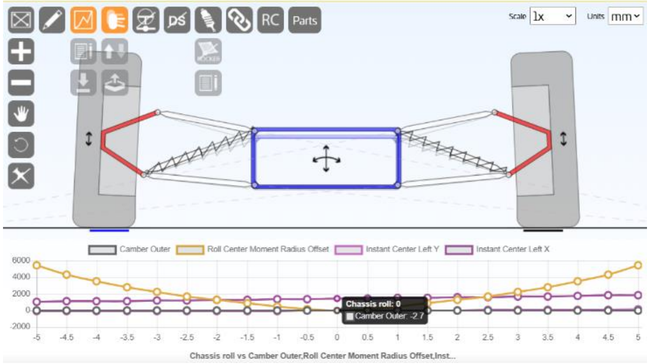


Fig No.21: Image showing graph of Camber change

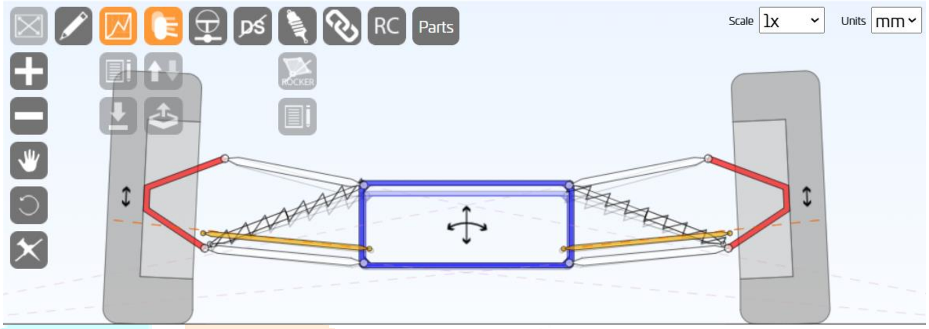


Figure No.22: Image showing Bump Steer

5. RESULT

The design, calculation, and analysis of the Double Wishbone Independent A-arm suspension system were successfully carried out, ensuring optimal performance and reliability. Key components, including coil-over dampers, A-arms, steering knuckles, hubs, and associated linkages, were meticulously designed to meet performance and structural integrity requirements.

Comprehensive calculations were performed to determine critical suspension parameters such as camber angle, caster angle, kingpin inclination, scrub radius, roll center height, bump steer, etc. These calculations ensured an optimal balance between handling stability, ride comfort, and mechanical efficiency.

Finite Element Analysis (FEA) of the front and rear steering knuckles was conducted using ANSYS, evaluating stress distribution, deformation, and factor of safety under dynamic and static loading conditions. The results verified structural robustness and highlighted necessary design optimizations.

Dynamic simulation of the entire suspension system was performed using Lotus Suspension Analysis**,** validating the suspension kinematics and ensuring proper articulation during different driving conditions.

Additionally, Racing Aspiration software was employed for precise measurement of designed component dimensions and placement, replicating real-world mounting conditions. The software also facilitated graphical representation of camber change rate and roll-over rate, providing insights into vehicle behaviour under lateral and longitudinal load transfers.

6. CONCLUSION

The overall results demonstrate a well-optimized suspension system, balancing performance, durability, and manufacturability, making it suitable for high-performance and motorsport applications.
Thus, the modified suspension system has been optimized to meet Indian road standards through extensive analysis and validation. Various simulation and testing methods confirmed its feasibility, ensuring enhanced performance, durability, and adaptability to real-world driving conditions.
The above designed was used in a performance vehicle which participated in SAE REEV 2023 which took place in Christ University Bengaluru.

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