A Final Project Report on

Design and Fabrication of Suspension System for All-Terrain Vehicle

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C E R T I F I C A T E

This is to certify that ***Mr. Yugesh Bhoge, Mr. Aditya Dukre, Ms. Ankita Kalhapure, Ms. Namrata Chavan*** have successfully completed the project work entitled ***"Design and Fabrication of Suspension System for All Terrain Vehicle"*** under my supervision, in the partial fulfilment of Bachelor of Engineering - Mechanical Engineering, by Savitribai Phule Pune university.

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Abstract

In this project our work is to study the static and dynamic parameter of the suspension system of an All-Terrain Vehicle by determining and analyzing the dynamics of the vehicle when driving on an off-road racetrack. Though, there are many parameters which affect the performance of the All-Terrain Vehicle, the scope of this project work is limited to design, analyze and optimize suspension systems and to integrate them into whole vehicle systems for best suitable results. The goal is to identify and optimize the parameters affecting the dynamic performance suspension systems within limitations of time, space, equipment and data from manufacturer. We have to evaluate and control Camber gain with optimized suspension geometry.

Keyword: *Camber gain, suspension geometry*

CHAPTER 1

1.1 INTRODUCTION:

The main function of any suspension system is to keep the contact of wheel to the ground by either absorbing the road load or by forcing it to maintain the contact. While driving any vehicle on the track the main problem that faced was during bump and droop due to change in suspension layout, camber is introduced in the system. This camber adversely affects the performance of vehicle as it reduces the tire contact patch with the road, which results in lowering the grip of tire on the tarmac. This issue is get even worse when it comes to racing off the track due to large wheel travel which accounts more camber change between bump and droop conditions. So this camber gain/change has to be minimized. So now the question arises how to minimize the camber gain with respect to wheel travel? The answer is very simple to optimize the suspension geometry to a greater extent that it can minimize camber change and also suitable for packing in current vehicle configuration.

Main functional parts of suspension systems are:-

- Damper (Coil over / Pneumatic standalone)
- Wishbones (Control arms)
- Knuckle
- Heim Joint/Rose Joint
- Links

1.1.1 PROBLEM STATEMENT:

While riding on off road tracks for long period of time driver may get strains in back or sometime serious back injuries too. And sometimes while taking sharp turns at high speed chances of vehicle to get roll over are quite high. The problem that was faced by many passengers while travelling is motion sickness. This discomfort arises due to motion of vehicle in different direction. So, design of suspension which has more wheel travel than conventional one and with good ride frequency and roll rate with optimized geometry to give driver more comfort while driving on off road trails.

1.1.2 OBJECTIVE:

- To ensure maximum comfort for driver without affecting or compromising any other performance parameters.
- The goal is to maintain camber gain below 0.01deg/mm of wheel travel with maximum wheel travel of 13inch (330.2mm) and with constrained wheel track of 1270mm at front and 1220mm at rear end and wheel base of 1370mm.

1.1.3 SCOPE OF THE PROJECT:

As there are no roads in many hilly areas due to their remoteness. Driver has to trails on off roads to get there. This system also can be used to evaluate contributing factors that causes motion sickness. And it will also help in lowering it down to some extent. Also, as the competition amongst off road vehicles are increasing this optimized system will be beneficial for increasing the sale also. Initially the ATVs were solely used for the transportation through the inaccessible areas, but now these vehicles have found their application in different areas as mentioned below:

a. In Defense Services like army and air force etc to carry and transport guns, ammunition and other supplies to remote areas of rough and varied terrain.
b. By railways during construction of railway tracks on mountain or on other rough terrain.
c. By police force.
d. In sport also like golf for traveling one place to other place.
e. In Antarctic bases for research things where use of conventional vehicle is impossible.
f. Now a days ATVs are also used in adventuring like mountaineering, in dirt and in snow

1.2 METHODOLOGY:

The following is the flowchart for project methodology that is used for this project.

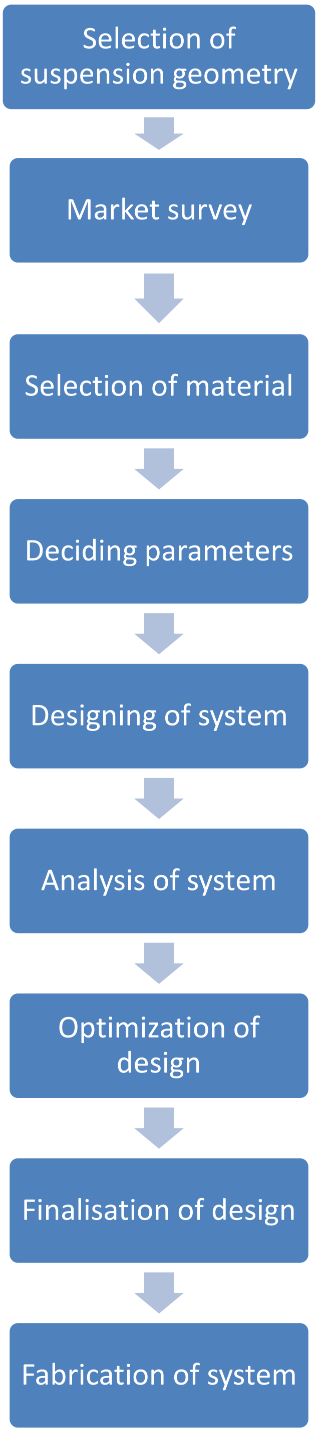


Fig 01: Flowchart for project methodology

The design of system begins with the finalizing vehicle configuration parameters such as wheel track (1270mm at front and 1220mm at rear) and wheel base (1370mm). Besides that the wheel travel plays an important role in configuring the suspension system. So after studying the track at NATRAX Pithampur site we opted to go with wheel travel of 13”. Also the design of system must in accordance with the rulebook compliances mentioned in section B.7.1.4 and B.8.7.1 of rulebook SAEINDIA BAJA 2019. The main stage in designing the suspension system is to choose best suitable type of suspension geometry from the existing available types and modify it in accordance with the system.

1.2.1 Material Selection:

We have selected material for different components based on strength of material, cost per kg and weight/density of the material.

1.2.1.1 Arms (H and A both):

For manufacturing of a-arms various loading conditions has to be taken into account like bump force, braking force etc. The material which has to be chosen to manufacture the arms should have high yield strength.

Properties	AISI 4130	AISI 1018	Duplex 2205 steel
Yield Strength	**665 MPa**	585 MPa	835MPa
% elongation	**20**	16	20
Cost	**Rs.635 /meter**	Rs.430 /meter	Rs.1025 /meter

Table 01: Material comparison table for Arms

1.2.1.2 Knuckle:

For manufacturing of knuckle bump force and braking torque are to be consider. The material should have high yield strength and also high hardness.

Properties	Al 6061	Al 7075	EN8
Yield Strength	270MPa	**540MPa**	465MPA
Hardness	95BHN	**150BHN**	201BHN
Density	2.70 g/cc	**2.81 g/cc**	7.85 g/cc

Table 02: Material comparison table for Knuckle

CHAPTER 2

2.1 LITERATURE REVIEW:

2.1.1 Race car vehicle dynamics:

Author: Milliken and Milliken

This book gave the insights about suspension geometry and wheel load. Selection of suspension geometry was the crucial step as every parameter depends upon it. According to wheel travel double wishbone system is selected at front of vehicle and H-arm with camber link. Also it provide with insights of camber, caster and KPI also the basic fundamentals of ride rate, roll rate.

2.1.2 Tune to win:

Author: Carrol smith

This book provided with insights of roll centre, anti dive, anti squat geometry for vehicle. This book provided with insights of recessional suspension geometry.

2.1.3 Verifications of Movable Roll Center Behaviors in Vehicle Body Frame Geometry:

By Wataru Kubota, Masayuki Ishikawa, Motohiro Kawafuku, Makoto Iwasaki

This paper provided insights of dynamic behavior of roll center. It also provided information about roll axis inclination.

2.1.4 Ride comfort analysis of Math Ride Dynamics model of Full tracked vehicle with trailing arm suspension:

By Saayan Banerjee, V. Balamurugan

Behavior of sprung mass with respect to roll center and center of gravity. The ranges of ride frequency and roll rate.

2.1.5 Dynamic effect of bump steer in wheeled tractor

By PA Simionsescu, D. Beale

This paper provided with insights of bump steer.

2.1.6 Analysis and optimization of lower control arm

By A.R. Kale and N.D. Patil

This paper provided with analysis of lower control arm. It provided with boundary Conditions for CAE analysis of lower control arm.

2.1.7 Active camber and toe control strategy for double wishbone suspension system

By B Ashoka, C. Kavitha

This paper provided the insights about active camber change for different wheel travel.

CHAPTER 3

3.1 Selection of Suspension Geometry

3.1.1 DIFFERENT TYPES OF SUSPENSION SYSTEMS USED IN AUTOMOBILES: -

Suspension systems can be broadly classified into two subgroups – Dependent and Independent**.**

3.1.1 Dependent suspension system

A dependent suspension normally has a beam or live axle that holds wheels parallel to each other and perpendicular to the axle with the help. Dependent suspension system assures constant camber, it is most commonly used in vehicles that need to carry large loads.

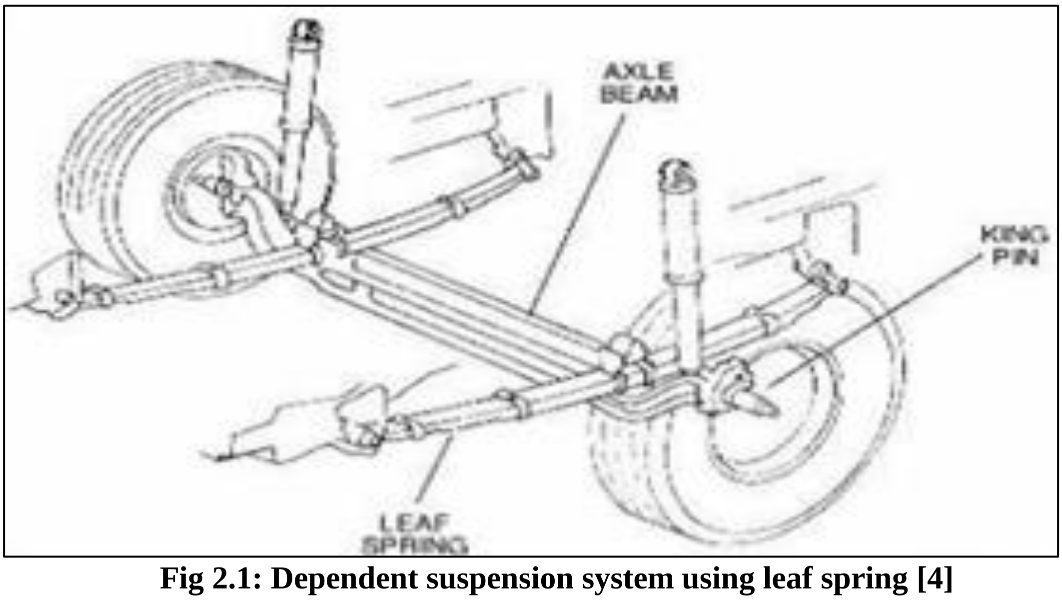


Fig 2.1: Dependent suspension system using leaf spring [4]

3.1.2 Independent suspension system

In an independent suspension system wheels are allowed to rise and fall on their own without affecting the opposite wheel by using kinematic linkages and coil springs. Suspensions with other devices, such as anti-roll bars that link the wheels are also classified in independent suspension system. The various independent suspension systems are:

a. Double wishbone suspensions
b. McPherson struts and strut dampers
c. Rear axle trailing-arm suspension
d. Semi-trailing-arm rear axles
e. Multi-link suspension

In this type of suspension system, the wheels are not constrained to remain perpendicular to a flat road surface in turning, braking and varying load conditions; control of the wheel camber is an important issue.

In double wishbone and multi-link system we can have more control over the geometry of system than swing axle, McPherson strut or swinging arm because of the cost and space requirements.

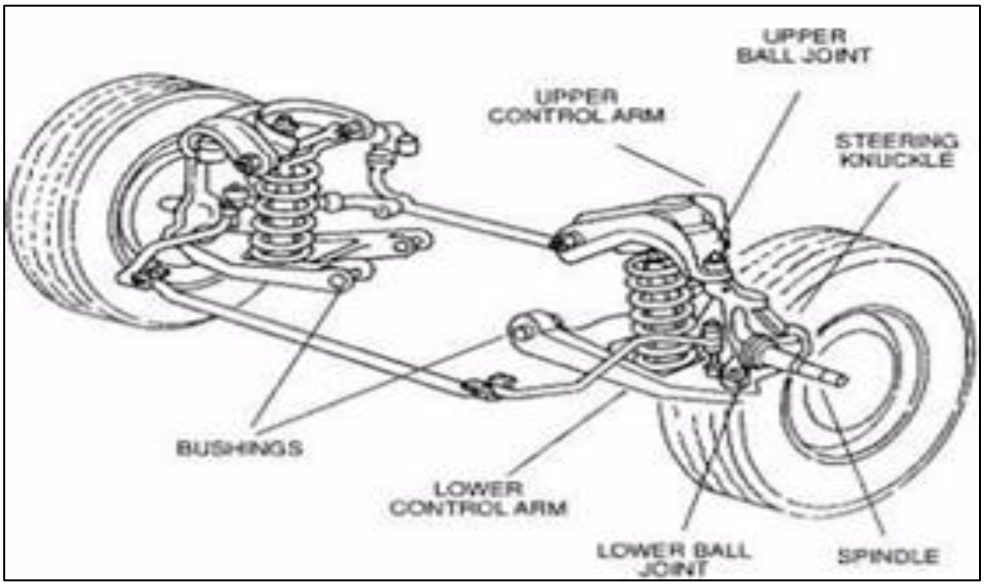


Fig 2.2: Independent suspension system using Double wishbone [1]

3.1.3 Multi-Link Suspension

Multi-link suspension is installed on vehicles with rear- and front-wheel drive layout. It has a more complex device, so it is used on expensive cars. For the first time, multi-link suspension was installed on the Jaguar E-Tour in the early 60s. Over time, it has modernized and is now actively used on Mercedes, BMW, Audi and many others.

The multi-link rear suspension, in contrast to the semi-independent beam, has an anti-roll bar in its design. The name itself speaks for the purpose of the element. This part reduces rolls when cornering at speed. Also, the stiffness of shock absorbers and springs affects this parameter. The presence of a stabilizer significantly reduces the risk of skidding during cornering, as it provides continuous contact of the wheels with the road surface.

Advantages of Multi-Link Suspension

Multi-link suspension allows the auto designer the ability to incorporate both good ride and good handling in the same vehicle. In its simplest form, multi-link suspension is orthogonal—i.e., it is possible to alter one parameter in the suspension at a time, without affecting anything else. Advantages also extend to off-road driving. A multi-link suspension allows the vehicle to flex more; this means simply that the suspension is able to move more easily to conform to the varying angles of off-road driving.

Disadvantages of Multi-Link Suspension

Multilink suspension is costly and complex. It is also difficult to tune the geometry without a full 3D CAD analysis. Compliance under load can have an important effect and must be checked using a multibody simulation software**.**

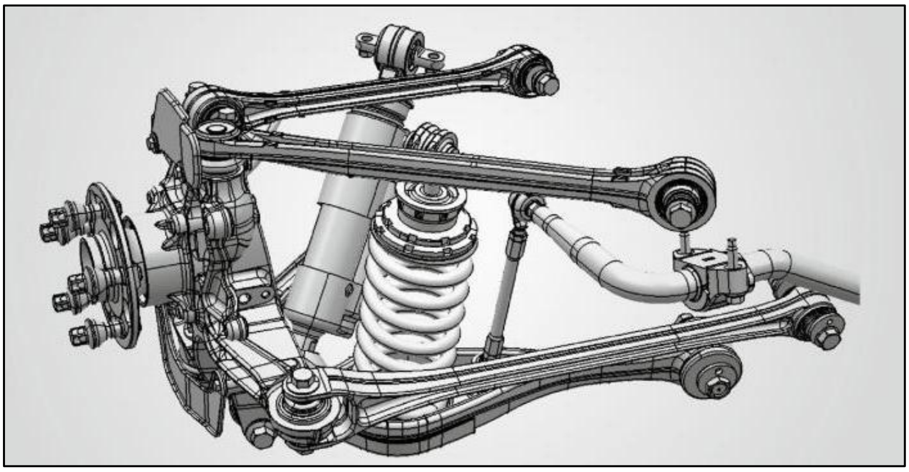


Fig 2.3: Multi-Link Suspension System [4]

3.1.4 MacPherson Strut Suspension System

A MacPherson strut suspension's upper bolt is connected directly to the body of the vehicle, with its lower end bolted to the vehicle's hub carrier. The strut itself is part of the steering geometry and provides the steering axis inclination. It usually is designed with the coil spring on top of the shock absorber cartridge and is designed for single body vehicles rather than body/frame designs

Advantages of MacPherson Strut Suspension System

MacPherson struts have the advantage of lower cost and a simpler design, as well as forgiving ride quality on the highway. The entire strut assembly is lighter as well, which has payoffs in un-sprung weight and the vehicle's overall power/weight ratio. They take up less space than a double wishbone setup and allow for a wider engine compartment, as well. Struts are sturdy and simple but should be checked every 50,000 miles.

Disadvantages of MacPherson Strut Suspension System

MacPherson struts are tall, effectively raising the center of gravity and making it difficult to lower a vehicle's profile and ride height for performance and handling. They also change camber angle whenever the suspension moves, making it more difficult to keep all four wheels in solid contact with the road while cornering, affecting control. On older vehicles that are starting to show some wear, struts can also transmit more noise through the body of the car.

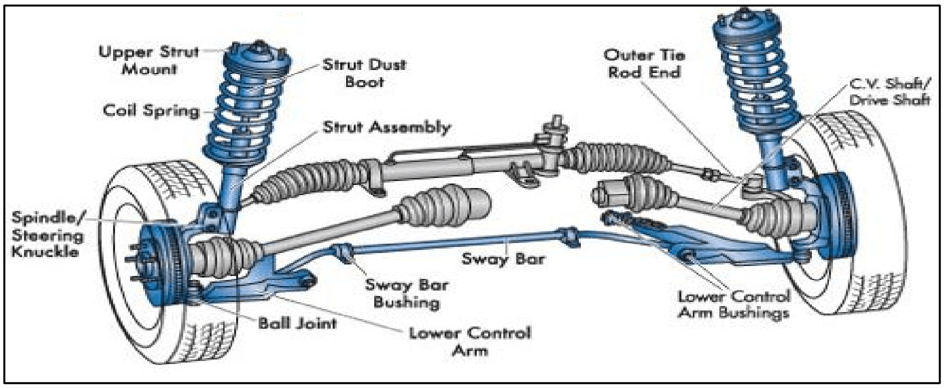


Fig 2.4: MacPherson Strut Suspension System [2]

3.1.5 Double Wishbone Suspension System

Double wishbone suspensions have been around for a long, long time and are featured in everything from a performance coupe to your grandpa's '72 Buick sedan. In a double wishbone setup, the shock absorbers and coil springs connect upper and lower control arms, with the steering knuckle and hub carrier on the lower control arm and the upper control arm attached to the frame. It's designed mostly for frame/body-type vehicles and is inherently more rigid than a MacPherson strut suspension, with less of a camber change while cornering or negotiating bumps.

Advantages of Double Wishbone Suspension System

The double wishbone suspension provides the engineer more design choices than some other types do. It is fairly easy to work out the effect of moving each joint, so the kinematics of the suspension can be tuned easily and wheel motion can be optimized. It is also easy to work out the loads that different parts will be subjected to which allows more optimized lightweight parts to be designed. They also provide increasing negative camber gain all the way to full jounce travel, unlike the MacPherson strut, which provides negative camber gain only at the beginning of jounce travel and then reverses into positive camber gain at high jounce amounts.

Disadvantages of Double Wishbone Suspension System

Double wishbone suspensions may take up less space but are more complex, and thus more expensive, than other systems like a MacPherson strut. Due to the increased number of components within the suspension setup, it takes much longer to service and is heavier than an equivalent MacPherson design.

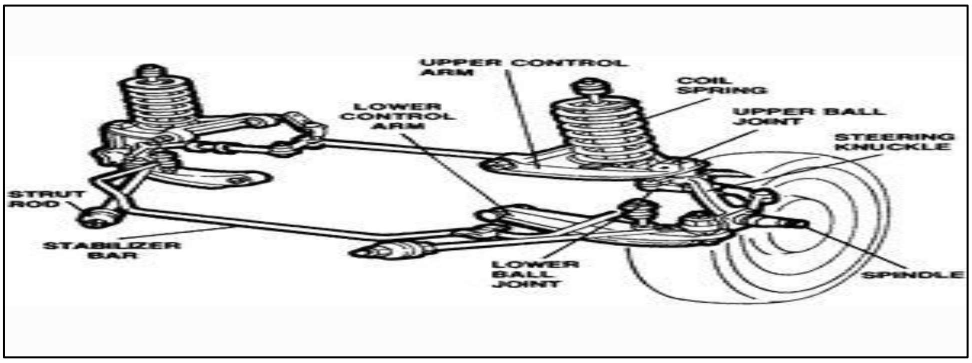


Fig 2.5: Double Wishbone Suspension System [1]

CHAPTER 4

4.1 MATHEMATIC MODEL: -

4.1.1 Suspension Geometry (Roll center Height):-

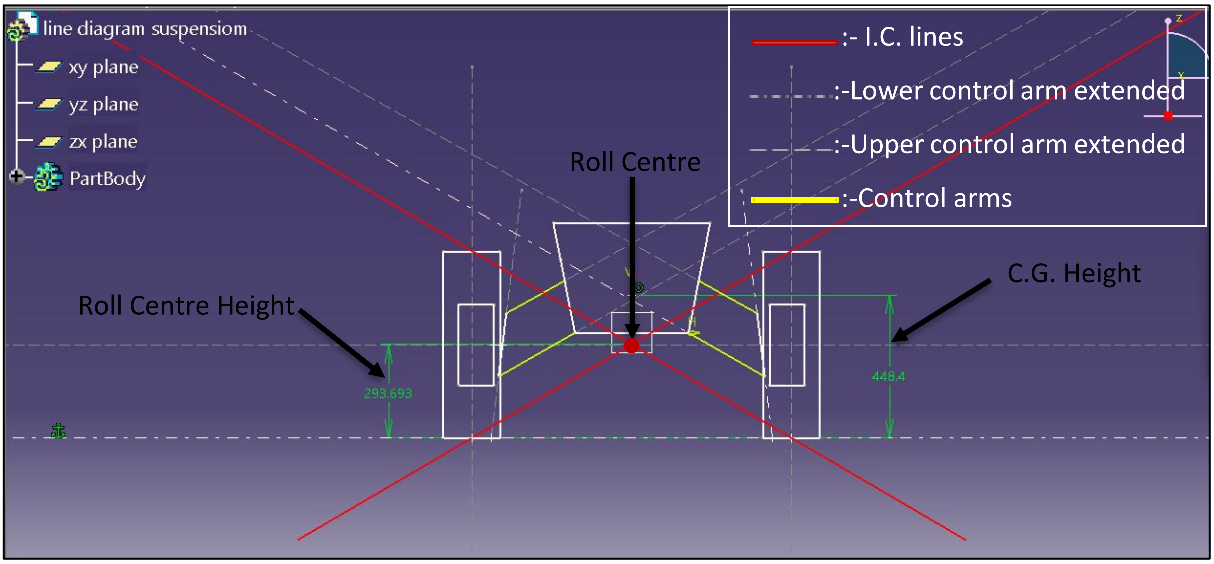


Fig 03: Line Diagram for front wheel

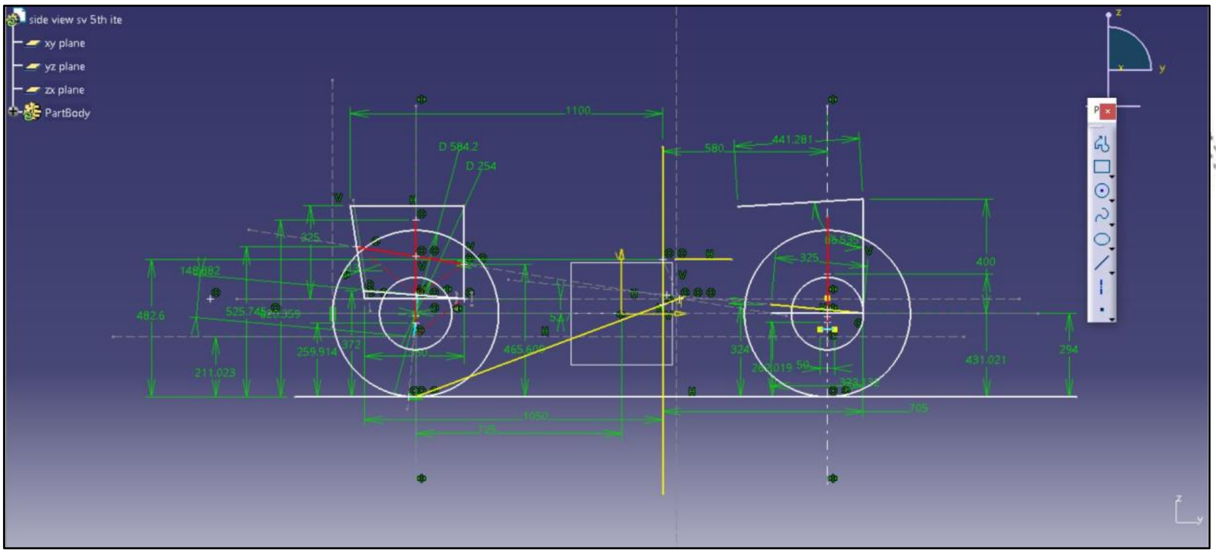


Fig 04: Side view roll center

From various available system we choose double wishbone type suspension at front and a modification of double wishbone system (H-arm with camber link) due to the packaging constrains. This geometry is evaluated in catia V5 software using line diagram to obtain coordinates of various joint point and evaluation of roll center in accordance with center of gravity. Because the rolling moment generated while turning or rolling over depends upon the distance between C.G. and R.C.

Instantaneous Center (I.C.): The point along which the wheel rotates in circle when seen from front view.

Roll Center (R.C.): The point along which the body (un-sprung mass) is supposed to be roll.

Centre of Gravity (C.G.): The point where the total mass of vehicle is concentrated (448.4mm).

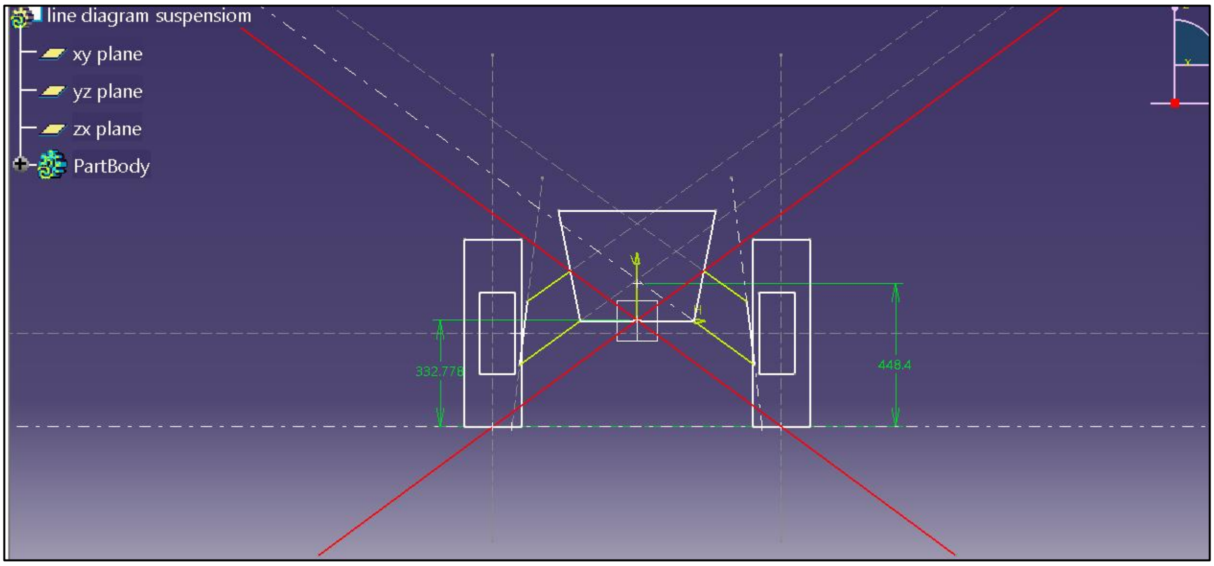


Fig 05: Rear view roll Centre

4.1.2 Forces and Calculation:

The main forces that should be consider for safety of vehicle considering the worst possible scenario are as follows,

1. Bump force as the vehicle is to be dropped from the height of 7ft on single wheel (Drop test).
2. The worst dynamic force that will coming upon the suspension system is combination of braking while cornering with bump or droop.(calculating the forces separately and applying them combined in FEA software)
3. Considering the above case scenarios the minimum factor of safety as per manufactures norm are restricted not below 2.5.

4.1.2.1 Calculations:

1. Weight and weight distribution of vehicle:

   Total weight of vehicle (kerb weight including driver) = $230kg \times 9.81\, m/s^2$ = 2256.3N

   Front wheel load = $0.4 \times 2256.3N$ = **902.52N**

   Rear wheel load = $0.6 \times 2256.3N$ = **1353.78N**

   Stationary load on individual front wheel = **451.26N**

   Stationary load on individual rear wheel = **676.89N**

2. The bump force (Drop force): - Vehicle is dropping from the height of seven feet and vehicle is landing of a single tire at angle of 45°.

$$\text{Bump force} = \frac{mass\ of\ whole\ vehicle \times \nabla v}{\Delta t}$$

$$= \frac{230kg \times \sqrt{(2\times 9.81 m/s^2 \times 2.1336m)}}{0.2}$$

$$= \mathbf{7440.529\ N}$$

3. Cornering force: - Maximum force coming up on the wheel while negotiating turn is known as cornering force. The maximum force will be generated on front right wheel while negotiating left turn and vice versa for front left one.

$$\text{Cornering force} = \frac{mass\ of\ vehicle \times (velocity)^2}{radius\ of\ turn}$$

$$= \frac{0.4 \times 0.5 \times 230kg \times 10^2}{2.1m}$$

$$= \mathbf{2190.47N}$$

4. Braking torque (front) = **423.79N.m**

5. Braking torque (rear) = **395.53N.m**

4.2 EXPERIMENTAL SETUP:

4.2.1 Simulation in Lotus Shark software:

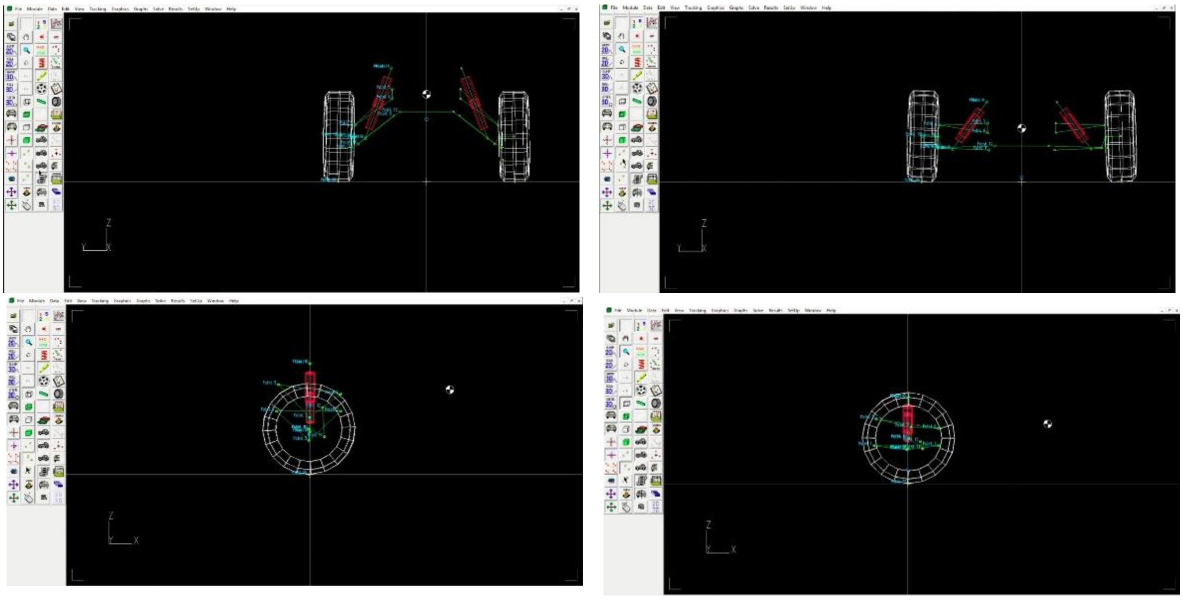


Fig 06: Lotus Simulation

This fig. shows the interface of Lotus Shark Simulation software. This software is used for performing multi-body dynamics simulation. In this software we have to put the co-ordinates of suspension geometry from the line diagram we have previously done in the CATIA software. After entering co-ordinates, we simulated the geometry inside the software. After simulation we get various results amongst which we have to consider the result of camber gain v/s wheel travel.

4.2.2 CAD design in CATIA V5 software:

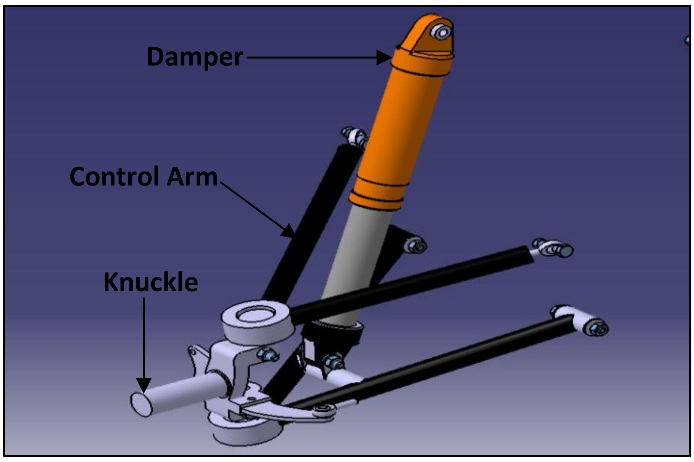


Fig 07: Assembly of Front suspension system

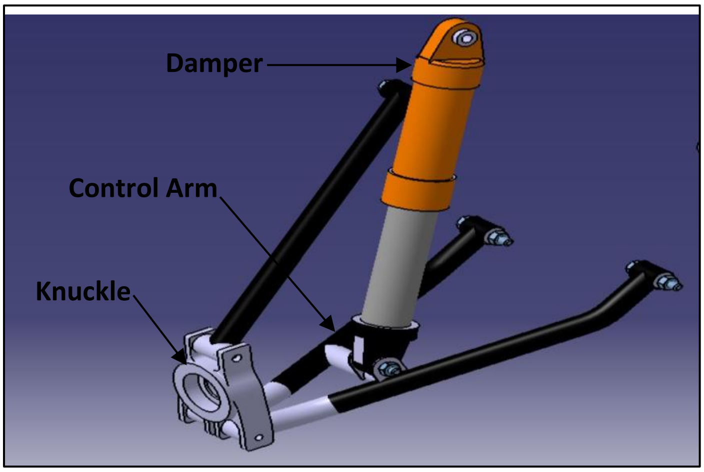


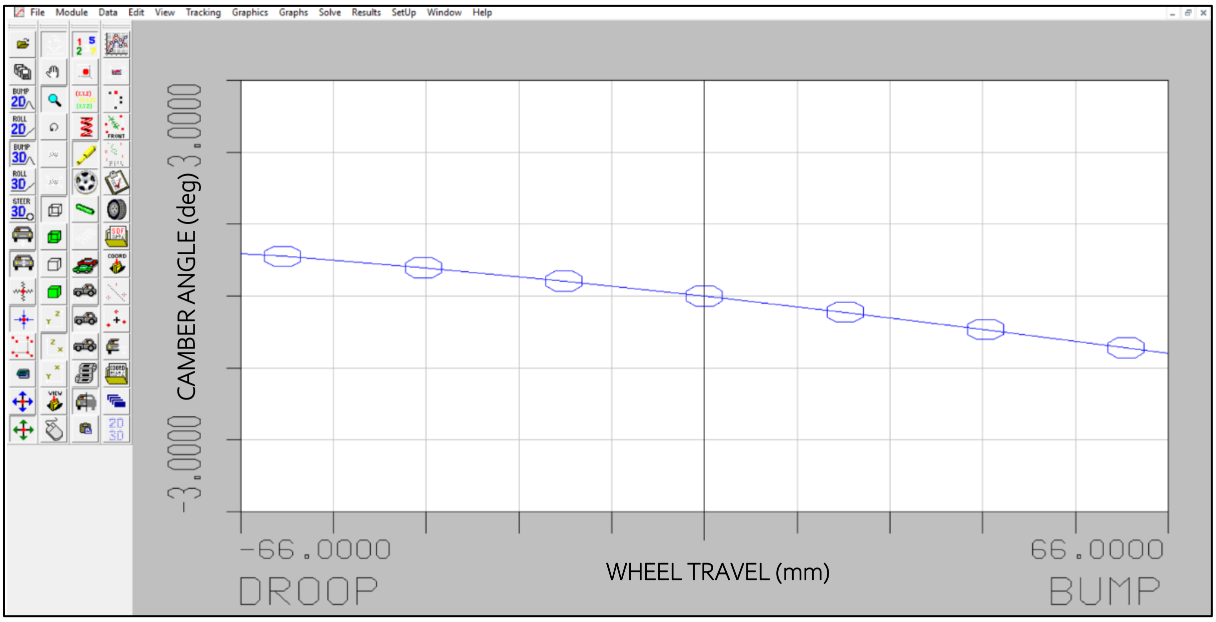
Fig 08: Assembly of rear suspension system

CHAPTER 5

5.1 Result

5.1.1 Simulation Results

5.1.1.1 Iteration 1:-



Graph 01. Wheel travel (mm) vs camber gain (deg) (1st Iteration)

This graph is obtained from Lotus Shark simulation software after putting the respective co-ordinates from suspension geometry like IBJ, OBJ, CG, RC etc.

IBJ: - Inner Ball Joint

OBJ: - Outer Ball Joint

C.G.: - Center of Gravity

R.C.: - Roll Center

This is the graph of camber angle (deg.) v/s wheel travel (mm).

FRONT SUSPENSION - BUMP TRAVEL
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

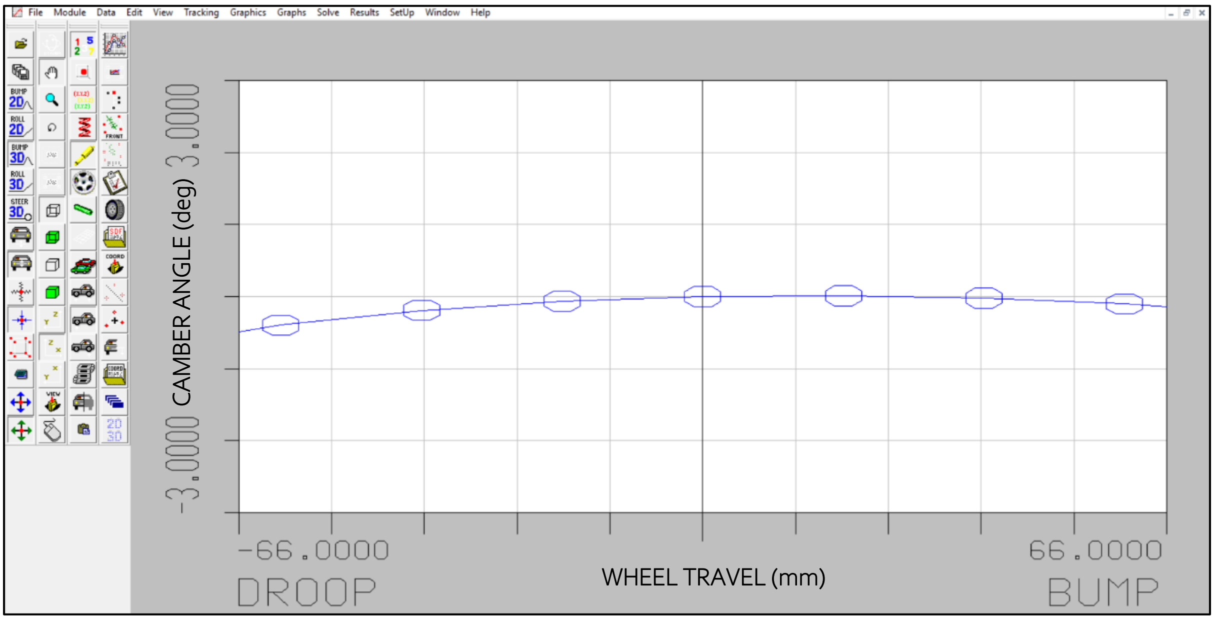
INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-120.00	4.7745	3.3669	-0.1779	2.2217	1.693	1.693
-110.00	4.5391	3.3838	0.2633	2.4311	1.724	1.724
-100.00	4.2326	3.1921	0.7010	2.7132	1.745	1.745
-90.00	3.8825	2.9117	1.1363	3.0435	1.759	1.759
-80.00	3.5038	2.5914	1.5697	3.4083	1.767	1.767
-70.00	3.1047	2.2543	2.0016	3.7994	1.772	1.772
-60.00	2.6905	1.9128	2.4323	4.2117	1.774	1.774
-50.00	2.2643	1.5734	2.8620	4.6417	1.774	1.774
-40.00	1.8281	1.2400	3.2908	5.0871	1.771	1.771
-30.00	1.3830	0.9150	3.7189	5.5465	1.768	1.768
-20.00	0.9298	0.5996	4.1464	6.0188	1.762	1.762
-10.00	0.4688	0.2945	4.5734	6.5035	1.756	1.756
0.00	0.0000	0.0000	5.0001	7.0000	1.749	1.749
10.00	-0.4766	-0.2837	5.4264	7.5082	1.741	1.741
20.00	-0.9613	-0.5566	5.8526	8.0281	1.732	1.732
30.00	-1.4544	-0.8188	6.2787	8.5598	1.723	1.723
40.00	-1.9564	-1.0705	6.7048	9.1034	1.713	1.713
50.00	-2.4678	-1.3119	7.1311	9.6593	1.702	1.702
60.00	-2.9894	-1.5432	7.5575	10.2278	1.691	1.691
70.00	-3.5216	-1.7645	7.9843	10.8093	1.680	1.680
80.00	-4.0653	-1.9761	8.4115	11.4044	1.669	1.669
90.00	-4.6212	-2.1781	8.8393	12.0135	1.657	1.657
100.00	-5.1902	-2.3708	9.2677	12.6373	1.645	1.645
110.00	-5.7731	-2.5541	9.6968	13.2763	1.633	1.633
120.00	-6.3710	-2.7284	10.1269	13.9314	1.620	1.620
130.00	-6.9849	-2.8937	10.5580	14.6034	1.608	1.608
140.00	-7.6158	-3.0500	10.9903	15.2929	1.596	1.596
150.00	-8.2649	-3.1975	11.4240	16.0009	1.584	1.584

Table 03. Result table for 1st iteration

The above table is obtained from the simulation which is interpolation of the graph shown above. It contains different parameter like camber angle, toe angle, castor angle. KPI angle, damper ratio and spring ratio against wheel travel. Here we have to consider the table which contains camber angle and wheel travel. After averaging the values for camber angle change is **0.04829 deg/mm of wheel travel.**

5.1.1.2 Iteration 6:-



Graph02. Wheel travel (mm) vs camber gain (deg) (6th Iteration)

This is also the graph obtained from Lotus Shark simulation software for 6th iteration. This is the graph we obtained after modifying the co-ordinates of suspension geometry from 1st iteration by changing lengths and angles of control arms i.e. IBJ and OBJ.

FRONT SUSPENSION - BUMP TRAVEL
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-80.00	0.5103	4.4819	0.2669	6.4640	2.370	2.370
-60.00	0.4174	3.0090	1.4671	6.4909	2.357	2.357
-40.00	0.3005	1.8005	2.6546	6.6005	2.340	2.340
-20.00	0.1613	0.8087	3.8316	6.7743	2.319	2.319
0.00	0.0000	0.0000	4.9998	7.0000	2.294	2.294
20.00	-0.1843	-0.6503	6.1604	7.2686	2.268	2.268
40.00	-0.3932	-1.1609	7.3145	7.5737	2.238	2.238
60.00	-0.6289	-1.5460	8.4629	7.9105	2.207	2.207
80.00	-0.8938	-1.8168	9.6063	8.2754	2.174	2.174
100.00	-1.1907	-1.9822	10.7453	8.6659	2.139	2.139
120.00	-1.5230	-2.0492	11.8802	9.0801	2.102	2.102
140.00	-1.8941	-2.0234	13.0115	9.5169	2.064	2.064
160.00	-2.3080	-1.9095	14.1392	9.9757	2.025	2.025

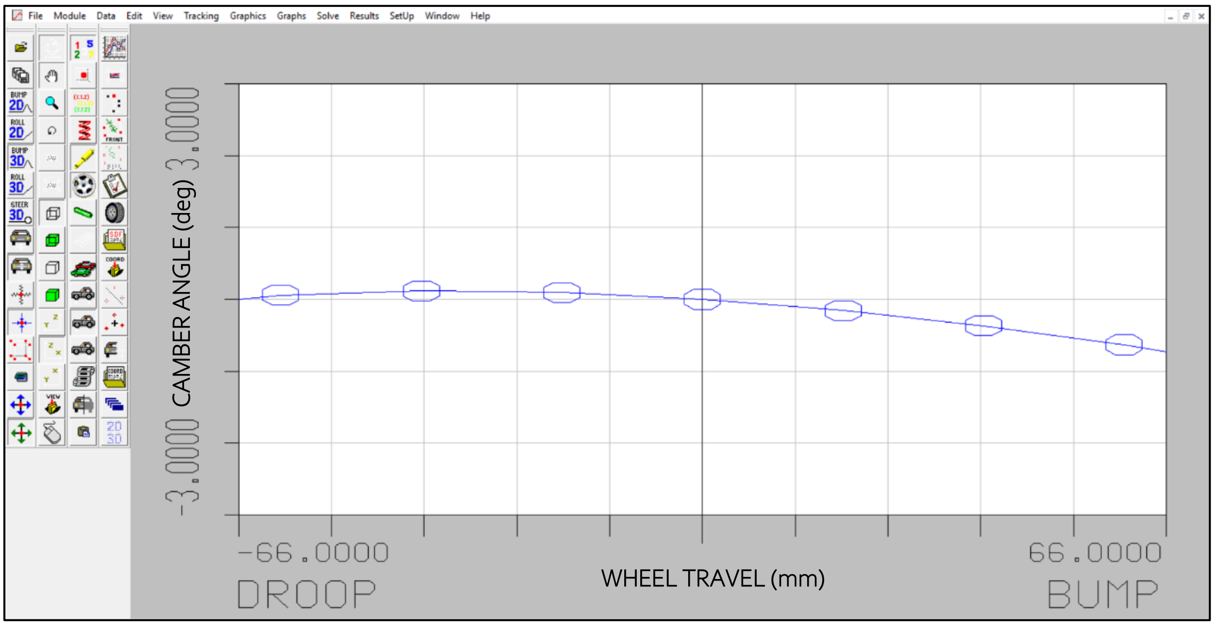
INCREMENTAL SUSPENSION PARAMETER VALUES

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)
-80.00	27.29	0.00	285.64	365.64	-41.88	12.44	34.23	34.23
-60.00	29.58	0.00	270.29	330.29	-29.17	8.88	25.77	25.77
-40.00	31.87	0.00	256.65	296.65	-18.06	5.66	17.26	17.26
-20.00	34.21	0.00	244.39	264.39	-8.38	2.71	8.67	8.67
0.00	36.62	0.00	233.27	233.27	0.00	0.00	0.00	0.00
20.00	39.14	0.00	223.10	203.10	7.20	-2.51	-8.77	-8.77
40.00	41.80	0.00	213.74	173.74	13.29	-4.83	-17.64	-17.64
60.00	44.65	0.00	205.07	145.07	18.36	-6.99	-26.64	-26.64
80.00	47.72	0.00	197.00	117.00	22.45	-9.00	-35.77	-35.77
100.00	51.06	0.00	189.46	89.46	25.63	-10.86	-45.05	-45.05
120.00	54.74	0.00	182.39	62.39	27.92	-12.57	-54.48	-54.48
140.00	58.83	0.00	175.73	35.73	29.36	-14.14	-64.08	-64.08
160.00	63.43	0.00	169.45	9.45	29.98	-15.57	-73.86	-73.86

Table 04. Result table for 6th iteration

The above table is obtained from the simulation which is interpolation of the graph shown above. It contains different parameter like camber angle, toe angle, castor angle. KPI angle, damper ratio and spring ratio against wheel travel. Here we have to consider the table which contains camber angle and wheel travel. After averaging the values for camber angle change is **0.01174 deg/mm of wheel travel.**

5.1.1.3 Iteration 9:-



Graph03. Wheel travel (mm) vs camber gain (deg) (Final Iteration)

This is also the graph obtained from Lotus Shark simulation software for final (9th) iteration. This is the graph we obtained after modifying the co-ordinates of suspension geometry from 8th iteration.

FRONT SUSPENSION - BUMP TRAVEL
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-100.00	1.7825	7.1703	0.2237	5.2183	2.698	2.704
-80.00	1.2579	5.2952	1.1704	5.6386	2.627	2.633
-60.00	0.8389	3.6834	2.1223	6.0172	2.561	2.566
-40.00	0.5004	2.2860	3.0783	6.3658	2.499	2.503
-20.00	0.2250	1.0671	4.0379	6.6918	2.438	2.443
0.00	0.0000	0.0000	5.0003	7.0001	2.379	2.384
20.00	-0.1841	-0.9357	5.9655	7.2940	2.322	2.326
40.00	-0.3344	-1.7559	6.9330	7.5756	2.265	2.268
60.00	-0.4568	-2.4731	7.9028	7.8461	2.208	2.211
80.00	-0.5557	-3.0975	8.8748	8.1062	2.151	2.154
100.00	-0.6348	-3.6370	9.8490	8.3563	2.094	2.097
120.00	-0.6970	-4.0985	10.8254	8.5961	2.036	2.039
140.00	-0.7447	-4.4875	11.8041	8.8251	1.978	1.981
160.00	-0.7798	-4.8084	12.7850	9.0427	1.919	1.922

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)
-100.00	-0.37	0.00	423.90	523.90	-67.31	24.76	39.54	39.46
-80.00	1.90	0.00	388.19	468.19	-49.53	19.40	32.02	31.96
-60.00	3.98	0.00	357.08	417.08	-34.20	14.24	24.31	24.27
-40.00	5.93	0.00	329.37	369.37	-20.98	9.30	16.41	16.38
-20.00	7.79	0.00	304.26	324.26	-9.65	4.55	8.30	8.29
0.00	9.61	0.00	281.13	281.13	0.00	0.00	0.00	0.00
20.00	11.42	0.00	259.54	239.54	8.10	-4.37	-8.51	-8.49
40.00	13.24	0.00	239.15	199.15	14.77	-8.56	-17.23	-17.20
60.00	15.11	0.00	219.67	159.67	20.08	-12.58	-26.18	-26.13
80.00	17.05	0.00	200.87	120.87	24.12	-16.45	-35.36	-35.30
100.00	19.10	0.00	182.53	82.53	26.93	-20.18	-44.78	-44.71
120.00	21.30	0.00	164.47	44.47	28.54	-23.76	-54.46	-54.38
140.00	23.67	0.00	146.51	6.51	28.98	-27.21	-64.43	-64.33
160.00	26.27	0.00	128.46	-31.54	28.26	-30.54	-74.69	-74.58

Table 05. Result table for final (9th) iteration.

The above table is obtained from the simulation which is interpolation of the graph shown above. It contains different parameter like camber angle, toe angle, castor angle. KPI angle, damper ratio and spring ratio against wheel travel. Here we have to consider the table which contains camber angle and wheel travel. After averaging the values for camber angle change is **0.00985 deg/mm of wheel travel.**

Ideal value for camber change per millimeter of wheel travel is zero deg/mm. But practically this is not possible to achieve so **0.00985deg/mm** of wheel travel is closet we can achieve without compromising other parameters of vehicle.

5.1.2 CAE Results:

5.1.1.1 Front Knuckle:-

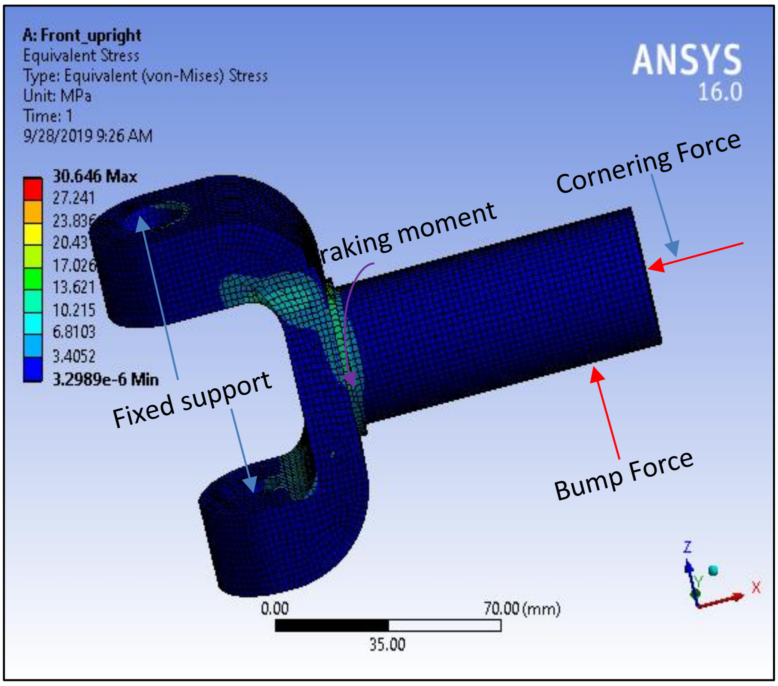


Fig 09: CAE analysis of Front upright

Type of analysis	No of node	No of element	Max Stress
Static Structural	25,758	22,759	30.646(MPa)
Transient structural	16,758	18,859	75.25(MPa)

Table 06. Meshing criteria and result of CAE analysis of front knuckle

This is the stress Contour plot obtained from the analysis of front knuckle in Ansys R16 CAE software. Maximum generated stress in component is 75.25MPa

$$\text{FOS} = \frac{Allowable\ stress\ or\ Yield\ stress}{Maximum\ stress}$$

$$= \frac{545MPa}{75.25MPa}$$

= 7.24

The red part indicates the maximum stress generated in the component.

5.1.1.2 Rear Knuckle:-

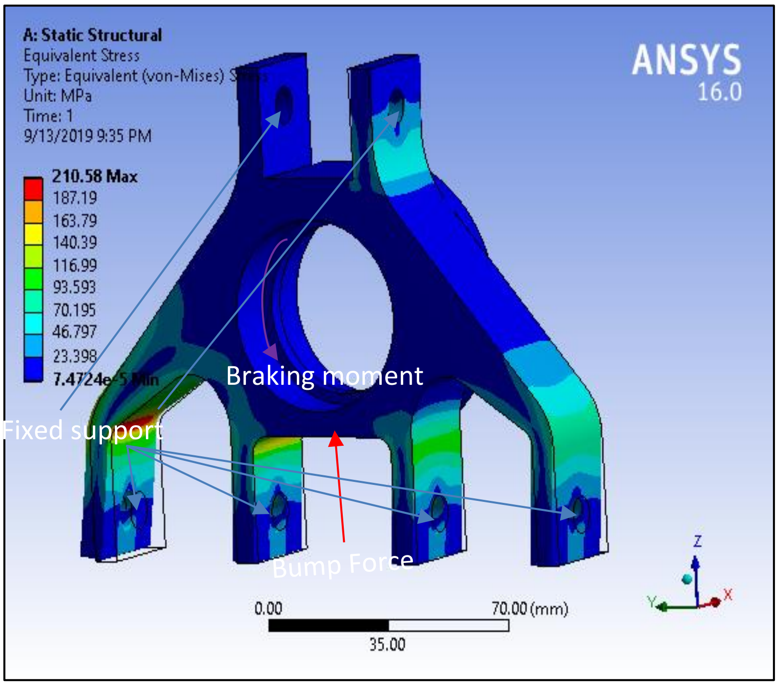


Fig 10: CAE analysis of Rear upright

Type of analysis	No of node	No of element	Max Stress
Static Structural	20,478	19,785	210.58(MPa)
Transient structural	14,458	12,745	214.47(MPa)

Table 07. Meshing criteria and result of CAE analysis of rear knuckle.

This is the stress Contour plot obtained from the analysis of front knuckle in Ansys R16 CAE software. Maximum generated stress in component is 214.47MPa

$$\text{FOS} = \frac{Allowable\ stress\ or\ Yield\ stress}{Maximum\ stress}$$

$$= \frac{545MPa}{214.47MPa}$$

= 2.541

The red part indicates the maximum stress generated in the component.

The maximum generated stress in rear upright is more than that of front upright because the weight distribution of vehicle is so that the 60 percent of the weight of whole vehicle in the rear side.

5.1.1.2 Lower control arm (A-arm):-

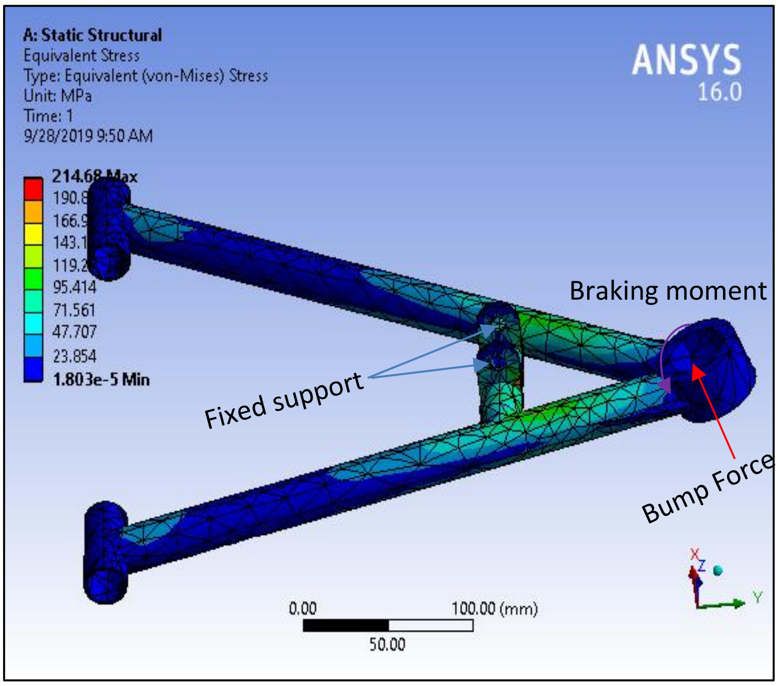


Fig 11: CAE analysis of lower control arm

Type of analysis	No of node	No of element	Max Stress
Static Structural	5,689	4,859	214.68(MPa)
Transient structural	3,458	2,485	290.14(MPa)

Table 08. Meshing criteria and result of CAE analysis of lower A-arm.

This is the stress Contour plot obtained from the analysis of front knuckle in Ansys R16 CAE software. Maximum generated stress in component is 290.14MPa

$$FOS = \frac{Allowable\ stress\ or\ Yield\ stress}{Maximum\ stress}$$

$$= \frac{665MPa}{290.14MPa}$$

$$\mathbf{= 2.291}$$

The red part indicates the maximum stress generated in the component.

5.1.1.2 Lower control arm (H-arm):-

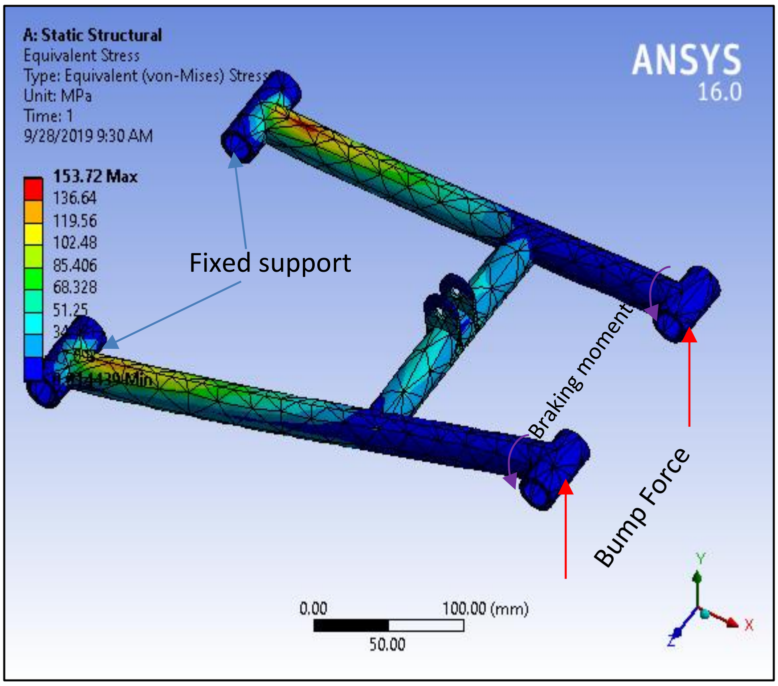


Fig 12: CAE analysis of H-Arm

Type of analysis	No of node	No of element	Max Stress
Static Structural	6,789	5,789	153.72(MPa)
Transient structural	4,789	4,201	210.52 (MPa)

Table 08. Meshing criteria and result of CAE analysis of lower H-arm.

This is the stress Contour plot obtained from the analysis of front knuckle in Ansys R16 CAE software. Maximum generated stress in component is 210.52MPa

$$\text{FOS} = \frac{Allowable\ stress\ or\ Yield\ stress}{Maximum\ stress}$$

$$= \frac{665MPa}{210.52MPa}$$

= 3.15

The red part indicates the maximum stress generated in the component.

The above CAE analysis result shows that all the components we are going to manufacture i.e. front knuckle, rear knuckle, A-arms and H-arms are **safe** with **minimum FOS 7.24**, **2.541**, **2.291** and **3.15.**

5.2 CURRENT PHASE:

Fig 13: Fixtures for front suspension assembly

The above figure shows how we manufactured our project using different kind of fixtures. In above picture we have use a threaded stud for restricting other degrees of freedom of the component.

Fig 14: Complete suspension assembly

The above picture shows our complete suspension assembly in whole vehicle.

CHAPTER 6

6.1 CONCLUSION:

From the design, analysis and simulation of our suspension system, it can be seen that we have minimize camber gain per inch of wheel travel till **0.0098deg per mm of wheel travel.** Also the use of double wishbone system at front and its modification H-arm with camber link at rear helps in compact packing due to its lesser requirement of components. By doing so we get ample of space in accommodating transmission and steering components with ease. Also due to the compactness of wishbone type suspension system we able to adjust the roll center so close to center of gravity and by doing so we able to minimize the rolling of body while turning and braking. Also by arranging the wishbone parallel to each other (in front suspension) we able to locate the instantaneous center very further away (nearly close to infinity) and hence able to minimize the camber gain as much as possible.

From this we conclude that the suspension system is so designed that it least affects other on the performance parameters of the All-Terrain vehicle.

6.2 REFERENCES

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