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Design and Analysis of All Terrain Vehicle

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***Abstract**:The concept of the All-Terrain Vehicle is that has a capability to be driven on any kind of terrain (road). It is a type of vehicle which is accomplished of driving on and off paved or gravel surface. It is generally categorized by having bulky tires with profound, open treads and a stretchy suspension.This paper deals with the detailed description ofdesigning a roll cage by taking inputs from SAE BAJA rule book 2016, suspension and steering of an All-Terrain Vehicle (ATV). Our primary focus is to design, analyze a single-sitter fun to drive, multipurpose, safe, strong, and high performance off road vehicle that will take the harshness of rough roads with maximum safety and driver comfort.The design consideration consists of material selection, design of chassis and suspension, simulations to test the ATV against failure.*
***Keywords:**SAE-BAJA, All-Terrain Vehicle, Roll cage, Design, Finite Element Analysis, Suspension.*

I. Introduction

The objective of the study is to design and analyze the roll cage and suspension of ATV which can resist high end loads and gives safety as well as comfort to the driver. Material used for this roll cage is selected based on strength, carbon percentage,availability and cost. The roll cage is designed to integrate all the automotive sub-systems. A CAD model is prepared in Solid works software preceding it is tested against all modes of failure by conducting several simulations and stress analysis with the assistance of ANSYS Software. Based on the result acquired from these tests, the design is modified therefore. After successfully designing the roll cage, it is ready for fabrication. There are many ATV's in the marketplace, but they are not factory-made in India and are assembled here. A cost effective All-Terrain Vehicle is designed. Since the chassis is the main part of an automotive, it should be robust and light mass. Thus, the chassis design becomes very essential. Typical capabilities on basis of which these vehicles are arbitrated are hill mounting, dragging, speeding up and manoeuvrability on land as well as shallow waters. This is intended to design the frame of an ATV which is of least possible weight and show that the design is safe, rugged and easy to manoeuvre. Design is done and carried out linear static analysisThe centre of gravity and roll centre are maintained at a desired position in order to have a good handling capability of the vehicle.

The following part describes the detailed description of the design and analysis of chassis and suspension.

II. Design Methodologies

2.1 Design Of Roll Cage

The design and development process of the roll cage involves various parameters like- material selection, frame design and finite element analysis.

2.2 Material Selection

One of the key design decisions of our frame is the material selectionwhich primarily focuses on strength, safety, reliability and performance of vehicle. To confirm that the best material is chosen, wide research was carried out and compared with materials from severalclasses. The key classes for comparison were strength, mass, and price. The details of each step are given below.

Also as per the rule book restriction there should be at least 0.18% of carbon content in metal. The material assortment also depends on number of influences such as carbon content, material properties, availability and the most important restriction is the price. Initially, three materials are considered based on their availability in the market. They are AISI 1018, AISI 1040 and AISI 4130. By using**Pugh's concept of optimization**, [1].This is the scoring matrix which is a form of prioritization matrix.We have chosen AISI 1040 for the wishbones. The main criteria were to have better material strength and lesser weight along with bestprice of the material.

Comparison Of Materials:

The properties of the belowstated materials which were considered for wishbones are as follows,

Properties	Aisi 1018	Aisi 1040	Aisi 4130
Carbon Content (%)	0.18	0.40	0.30
Tensile Strength (Mpa)	440	620	560
Yield Strength (Mpa)	370	415	460
Hardness(Bhn)	126	201	217
Cost (Rs./Metre)	325	425	725

Pugh's Matrix

DescriptionCriteria	AISI 1018	AISI 1040	AISI 4130
Total Weight	**-2**	**0**	**+1**
Yield Strength	**-1**	**0**	**+1**
Tensile Strength	**-2**	**+2**	**0**
Cost	**+1**	**0**	**-2**
Elongation at break	**-2**	**+1**	**0**
Net Score	**-6**	**+3**	**0**

Hence, AISI 1040 is selected for wishbones because the net score is highest for AISI 1040.

1.2 Frame Design

The entire chassis is designed by following the SAE BAJA Rule book 2016 [2]. Initial design is started by designing the integration of cockpit and steering box (To avoid welds) by consideringthe ergonomics of driver comfortseating,pedalling and steering rack. The rear part of chassis is designed by considering the positions of engine and transaxle.

It is also mandatory to keep a least clearance of 3 inches between the driver and the roll cagemembers [2]. We can achieve better acceleration by keeping the roll cage weight as low as possible. Centre of gravity should be kept as low as possible to avoid collapsing.We avoided welds thereby giving more significance to bends. A layout of the chassis within the given geometricalrestrictions is as shown in Fig.1

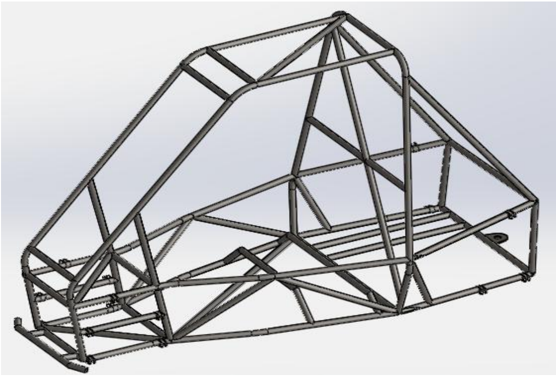


Fig 1. Roll Cage CAD Design

1.3 Finite Element Analysis

After confirming the frame along with its material and cross section, it is very important to test the rigidity and strength of the frame beneath severe environments. The frame should be able to withstand the impact, roll over conditions and provide maximum safety to the driver without suffering much deformation.

Loading Analysis –

To imprecise the loading that the vehicle will see anstudy of the impact loading seen in various types of accident was vital. To appropriately model the impact forces, the deceleration of the after impact needs to be found. To approximate the nastiest case situation that the vehicle will perceive, research into the forces the human body can endure was completed. It was found that human body will pass out at loads much higher than 7g. And the crash pulse scenario average set by industries is 0.15 to 0.3sec. We well-thought-out this to be around 2.5 sec. It is assumed that worst case crash will be seen once the vehicle runs into stationary object.

Load calculations:

The mass of the vehicle is 300 kg. The analysis is performed assuming the vehicle hits the static rigid wall at top speed of 60kmph. The collision is expected to be flawlessly plastic i.e. vehicle comes to rest after collision.

Initial velocity u=16.67m/s
Final velocity v=0.
Impact time as 0.18 s.
By applying Newton's 2nd law,
F = change in momentum/time
F= (m*(v-u))/t F= (300*(0-16.67))/0.18
F = 27783.3N

FEA of Roll cage

A CAD modelof the roll cage was designed in CATIA and was imported into ANSYS Mechanical in IGES format. ANSYS was used to do static structural analysis of the chassis. Automatic fine meshing is done for the entire roll cage. The below mentioned Impact tests were conducted on our chassis and the following results were obtained.

For AISI 1040 alloy steel-

Young's modulus	415GPa
Poisson's ratio	0.27-0.29 (say0.28)
For all the analysis the weight of the vehicle	300kgs

Main Objectives of FEA of Roll Cage-

The main objective is to have adequate factor of safety (FOS) even in worst case scenarios to ensure driver safety.

Static Analysis:-
1)Frontal Impact
2) Rear Impact
3)Side Impact
4)Roll over test

The mass of the vehicle is 300 kg. The impact test or crash test is performed assuming the vehicle hits the static rigid wall at top speed of 60 Kmph. The collision is assumed to be perfectly plastic i.e, vehicle comes to rest after collision.

Frontal Impact	8g
Max. Deformation	5.496 mm
Max. Stress	151.25N/mm2
Factor of Safety	2.74 (Design id safe)

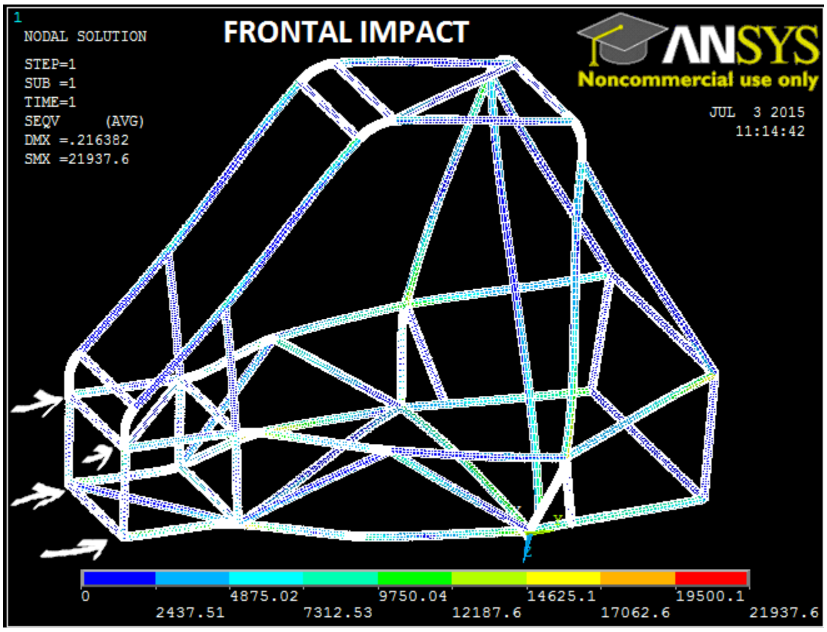


Fig 2. Frontal Impact Vonmises stress

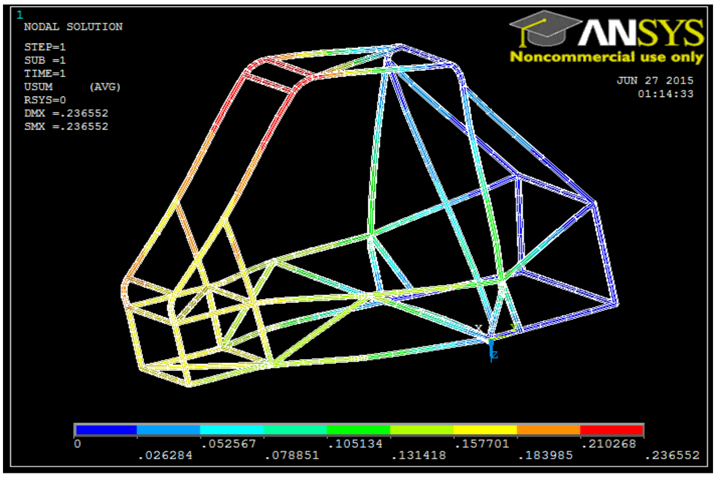


Fig 3. Frontal Impact Deformation

Rear Impact	8g
Max. Deformation	7.95 mm
Max. Stress	207.157 N/mm^2
Factor of Safety	2.12 (Design is safe)

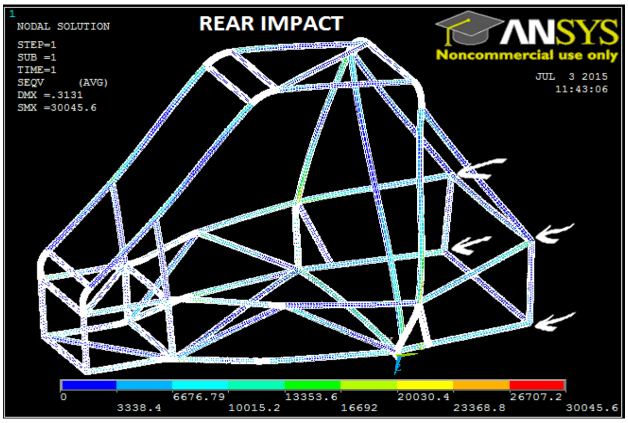


Fig 4. Rear ImpactVonmises Stress

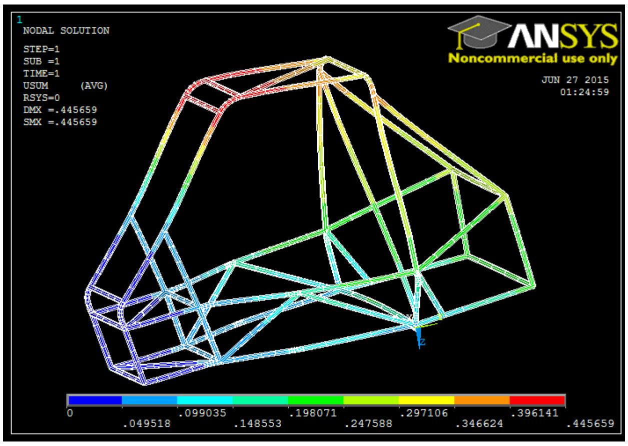


Fig 5. Rear Impact Deformation

Side Impact	3g
Max. Deformation	2.65176 mm
Max. Stress	161.309 N/mm2
Factor of Safety	2.57 (>2 Design is safe)

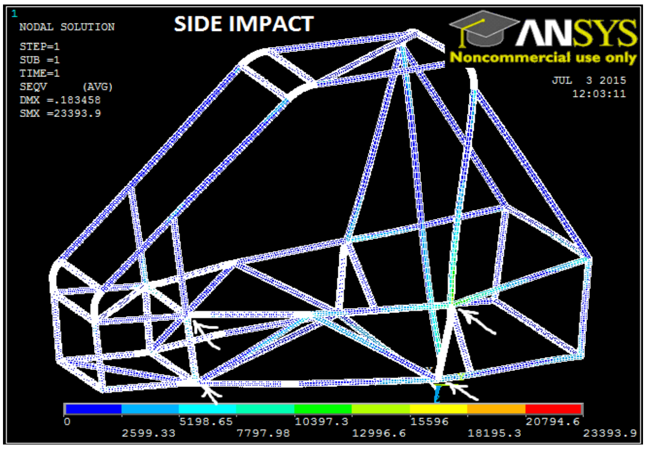


Fig 6. Side ImpactVonmises Stress

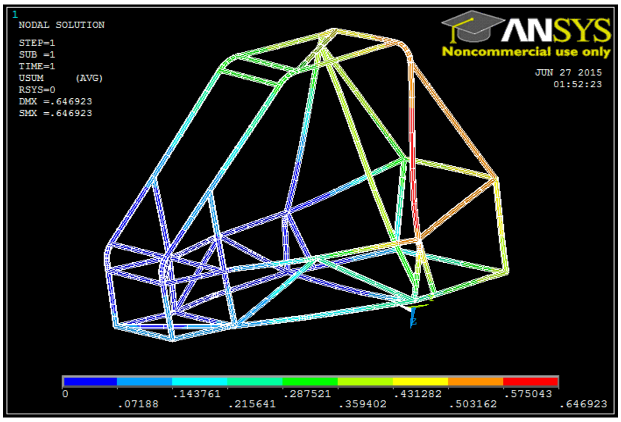


Fig 7. Side Impact Deformation

Roll over	3g
Max. Deformation	4.659 mm
Max. Stress	167.91767N/mm2
Factor of Safety	2.47 (>2 Design is safe)

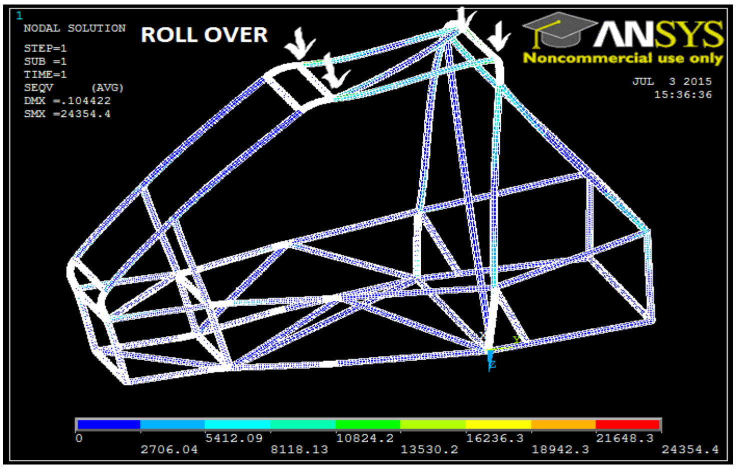


Fig 8. Roll Over ImpactVonmises Stress

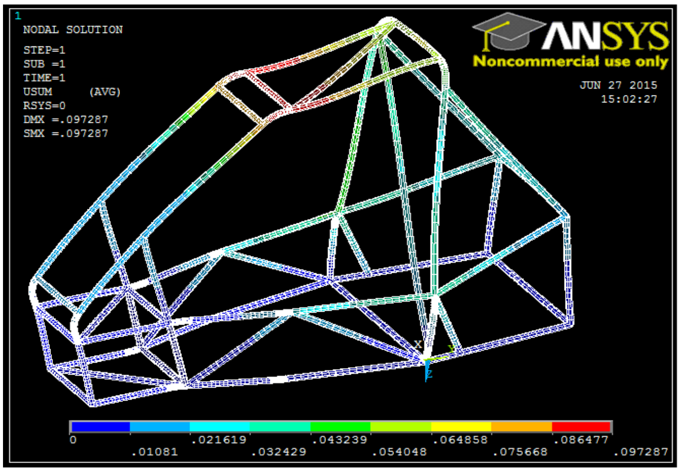


Fig 9. Roll Over Deformation

1.4 Suspension Design

The aim of Suspension is to maintain the ground contact of tires throughout the ride. The roll centre is maintained near to the centre of gravity which improvises the stability and handling capability of the Vehicle.

The Centre of Gravity was tried to keep in middle of the vehicle & closest to the ground for optimum stability.

Objective

1. To have Comfort, safety and manoeuvrability for our vehicle.
2. To have better riding and vehicle handling.
3. Protect the vehicle from damage and wear from force of impactduring landing after jumping.
4. Maintaining correct wheel alignment

Design Methodology –

The overall purpose of a suspension system is to absorb impacts from coarse irregularities such as bumps and distributethat force with least amount of discomfort to the driver. We completed this objective by doing extensive research on the front and rear suspension arm's geometry to help reduce as much body roll as possible. Proper camber and caster angles were provided to the front wheels. The shocks will be set to provide the proper dampening and spring coefficients to provide a smooth and well performing ride.

Estimated weight of the vehicle	230 kg
Driver with accessories	70 kg
Overall weight of the vehicle	300 kg
Un-sprung mass	52 kg
Sprung mass (with driver)	248 kg

Basic Calculation in Spring Design

Front lower wishbone length	= 403.225mm
Damper mounting	= 221.773mm
Motion ratio	= 218.403/403.225
Natural frequency	= 1.2Hz

According to this motion ratio, natural frequency and taking 47.17% sprung mass for front, spring rate is calculated as

Spring Constant	= 6.6N/mm
Suspension travel	= 2.5inch
Length of shock absorbers	= 12inch

Similarly for rear taking 52.83% sprung mass, the spring rate is calculated as

Motion ratio	= 0.97
Natural frequency	= 1.5 Hz
Spring Constant	= 11.54N/mm
Travel	= 4 inch

Design Of Front And Rear Suspension System

For the front, we are using unequal A-shaped Control Arm Double Wishbone System. This was selected based on calculations for Roll Centre, Camber Angle, Caster Angle, King-pin Inclination, Scrub Radius, etc. The design was tested under static analytical conditions and found to be safe. The dynamic calculations were stimulated and analysed in LOTUS. Graphs plotted justified design considerations.

On the rear side, we have used A-shaped control arm for providing high stability, at the same time to minimize the yaw motion without affecting the travel.

Suspension arm was made of AISI 1040 steel pipe of OD 1 inch with 3 mm wall thickness. In front we have used ball joints of Maruti 800 and in rear we have used bushes of 1 inch diameter and 2 inch length with the aim of minimizing the rear-yaw motion.

Calculation for Springs

Analytical method is used in spring rate calculation and for that we had to take some parameters given in table

Calculation for spring rate:

We found that spring rate is depends upon motion ratio and wheel rate in the following way

$$K_{spring} = (motion\ ratio)^2 \times K_{wheel}$$

$$f = \frac{1}{2\pi}\sqrt{\frac{K_{wheel}}{Sprung\ Mass}}$$

Front lower wishbone length=403.225mm
Damper mount= 221.773mm

Motion ratio $= \frac{\text{Damper mount} * \cos\alpha}{\text{Front lower wishbone length}}$
$= 0.541$

Frequency (f) =1.2Hz

According to this motion ratio, natural frequency and taking 47.17% of sprung mass for front, spring rate is calculated as

K *spring* = 6.6 N/mm
Suspension travel= 2.5inch
Length of shock absorbers= 12inch
Similarly for rear taking 52.83% sprung mass, the spring rate is calculated as
Natural frequency (*f*)= 1.5 Hz
Motion ratio= 0.97
K *spring* = 11.54 N/mm

Fig 10.Front & Rear Wish-bones

Alternative approach

We know that spring rate is calculated as:-

K *spring* $= \frac{Gd^4}{8nD^3}$

Where,
G - Modulus of rigidity or shear modulus of springmaterial
d - Wire diameter
n - Number of active coils
D - Mean coil diameter

After considering all the above calculated data the suspension was designed and implemented with the following specifications and dimensions.

Specifications	FRONT	REAR
Roll Centre (Static)	152.4 mm	390.5 mm
Static Camber	2 degree	NA
Static Caster	10 degree	NA
King pin Inclination	10 degree	10 degree
Scrub Radius	26.5 mm	19.05 mm

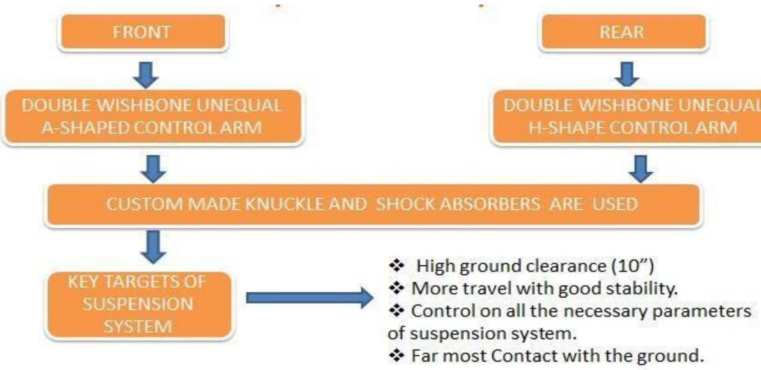
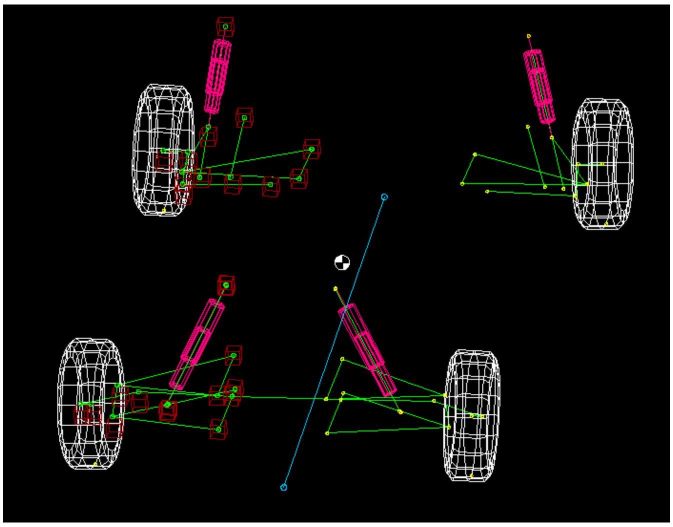


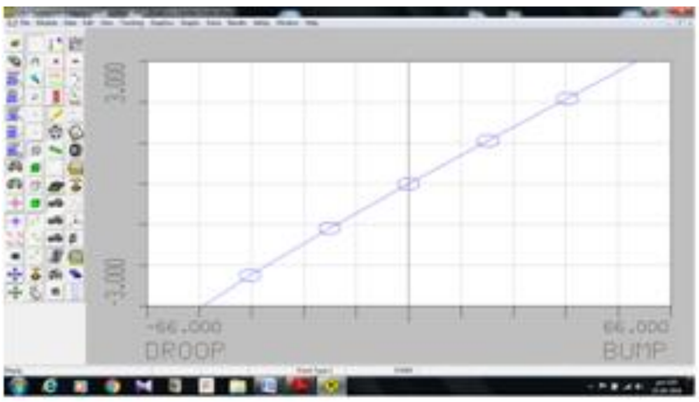
Fig 11. Suspension Design Methodology



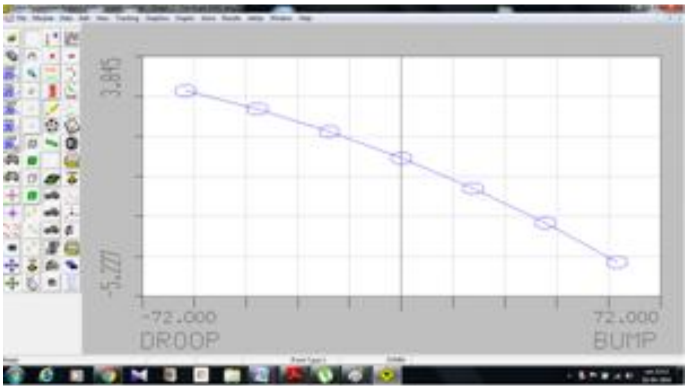
Suspension Design in Lotus Shark

III. Results

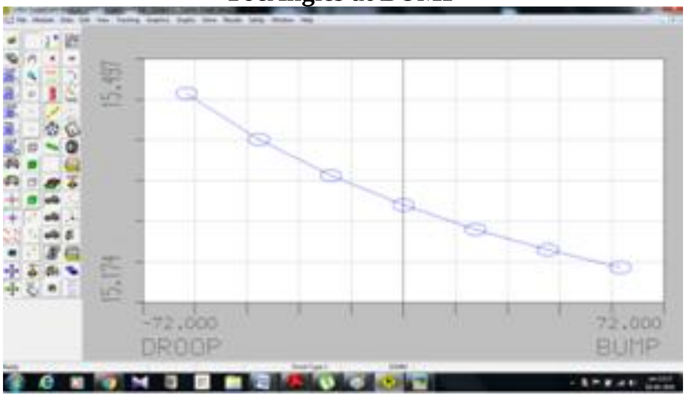
3.1 Graphical Results Of Suspension Geometry



Camber Angles at BUMP



ToeAngles at BUMP

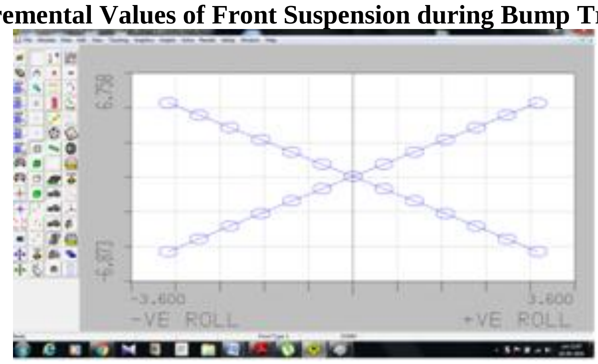


CasterAngles at BUMP

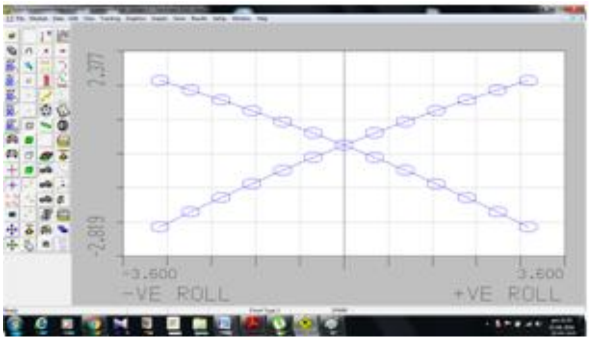
INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-60.00	-3.5177	2.5540	15.4506	12.9552
-40.00	-2.2539	1.8563	15.3897	11.8766
-20.00	-1.0936	1.0005	15.3417	10.9462
0.00	0.0000	0.0000	15.3028	10.1247
20.00	1.0547	-1.1464	15.2705	9.3859
40.00	2.0939	-2.4510	15.2432	8.7115
60.00	3.1402	-3.9369	15.2199	8.0878

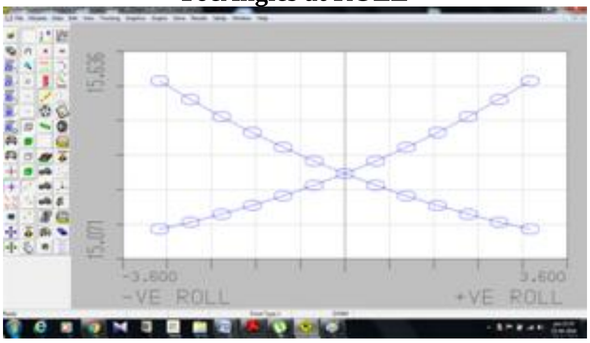
Incremental Values of Front Suspension during Bump Travel



Camber Angles at ROLL



ToeAngles at ROLL



CasterAngles at ROLL

INCREMENTAL GEOMETRY VALUES

ROLL ANGLE (deg)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-3.00	4.8103	-2.0841	15.1518	5.8901
-2.50	4.0122	-1.7030	15.1709	6.5823
-2.00	3.2132	-1.3361	15.1923	7.2795
-1.50	2.4130	-0.9828	15.2161	7.9820
-1.00	1.6110	-0.6426	15.2424	8.6901
-0.50	0.8068	-0.3151	15.2713	9.4042
0.00	0.0000	0.0000	15.3028	10.1247
0.50	-0.8100	0.3030	15.3371	10.8519
1.00	-1.6236	0.5940	15.3743	11.5864
1.50	-2.4414	0.8733	15.4145	12.3286
2.00	-3.2640	1.1409	15.4579	13.0791
2.50	-4.0919	1.3967	15.5047	13.8386
3.00	-4.9259	1.6407	15.5550	14.6077

Incremental Values of Front Suspension During Rolling

1. From the following graphs and table we can conclude that since there is no much variation in toe angle bump steer condition is avoided.
2. Our spring constant (k) values are in optimum range , so handling and riding experience on our vehicle will be very comfortable.
3. King pin inclination is optimum, so the return ability of wheels to straight position is better.

3.2 Cae Analysis Result

S.No.	Name of the Test	FOS	Max stress. N/mm^2	Max. Displacement (mm)
1.	Front impact (8G)	2.744	**151.25**	**5.496**
2.	Rear impact (8G)	2.123	**207.157**	**7.95**
3.	Roll over (3G)	2.471	**167.917**	**4.659**
4.	Side impact (3G)	2.573	**161.309**	**2.651**

IV. Conclusion

The objective of designing and analyzinga single-passenger off-road race vehicle with high safety and low production costs seems to be accomplished. The design is first conceptualized based on personal experiences and intuition. Engineering principles and design processes are then used to verify and create a vehicle with optimal performance, safety, manufacturability, and ergonomics. The design process included using Solid Works, CATIA and ANSYS software packages to model, simulate, and assist in the analysis of the completed vehicle. After initial testing it will be seen that our design should improve the design and durability of all the systems on the car and make any necessary changes up until the leaves for the competition. Multiple unique design features provide easy adjustability that give the owner more control over the vehicle. Further, software analysis shows us that the vehicle can withstand in extreme off road conditions.

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