© 2016 IJEDR | Volume 4, Issue 2 | ISSN: 2321-9939

Design, Static and Dynamic analysis of an All-Terrain Vehicle Chassis and Suspension System

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***Abstract* - This paper provides in-detail description of the design and structural analysis of chassis and suspension system of a standard All-Terrain Vehicle. The design and development comprises of material selection, chassis and frame design, cross section determination, and determining strength requirements of roll cage, stress analysis, design of the entire double wishbone suspension system and simulations ton test the ATV against failure. The static and dynamic structural analysis is also done on the chassis for validating the design. Initially, a prototype design of the chassis was made as a 3-D CAD model using Solidworks CAD software. The designed ATV is an off-road vehicle powered by 305 cc, four strokes, 10 BHP engine Brigg Stratton engine and driven by manual transmission. Material selection was based on the basis of factors like weight, cost, availability and performance during the entire design process, consumer interest through innovative, inexpensive, and effective methods was always the primary goal. The manufacturing objective is to design a vehicle which is safety ergonomic, aerodynamic, highly engineered and customer satisfaction which can make it highly efficient. The proposed design of ATV can navigate all most all terrain which is the primary objective behind the design and fabrication of any all-terrain vehicles.**

***Index Terms* - Roll cage, material, finite element analysis, Front & Rear Suspension, Simulation of suspension system, LOTUS, ANSYS**

I. INTRODUCTION

The objective of the study is to design and analyze on static and dynamic failures of the chassis for All - Terrain Vehicle. Material selected for the chassis based on physical strength, cost and availability. The roll cage is designed accordingly to provide all the automotive sub-systems. A software model is prepared in Solid works software and for finite Element analysis the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of Ansys Software (14.0). Based on the result obtained from these tests the design is modified accordingly. After successfully designing the roll cage, it is ready for fabricated. The vehicle is required to have a combination frame and roll cage consisting of steel members. The ATV should run continuously for four hours in various terrains, especially loose and uneven roads with high bumps, deeper potholes and muddy terrain on the surface. The input from the road surface to the ATV is hard/soft and always varying its rattle space with body and suspension, longitudinal acceleration in forward motion and lateral acceleration when cornering. This property results reduced in steering stability, controlling and handling performance of the ATV by drivers. So we are giving a cost effective design of an All-Terrain Vehicle Frame and suspension system. Since the chassis is the integral part of an automotive, it should be strong and light weight. Thus, the chassis design becomes very important. Typical capabilities on basis of which these vehicles are judged are braking test, bumping, hill climbing, pulling, acceleration and maneuverability on land as well as shallow waters. The aim is to design a frame with ultimate strength to show that the design is safe, rugged and easy to maneuver. Design is done and carried out the linear static and dynamic failures of frame and suspension system.

II. DESIGN METHODOLOGIES

Roll cage Configuration, Design & Material

The roll cage plays a crucial role in providing the desired strength, endurance, safety and reliability to the vehicle. The roll cage is designed in such a way that the driver seat, engine, transmission system, suspension system, brake system, fuel system and steering mechanism can be mounted on it. The objectives considered were that the roll cage must be designed with high yield and tensile strength steel tubes as a triangulated space frame, number of welded joints should be very less in favor of bent joints, strength and weight ratios should be maintained at all times when vehicle is in dynamic mode, must provide maximum spaces for the moving parts, must be designed in such a way that provides maximum driving reliability and most importantly the driver's safety, must have ease of serviceability by ensuring that the roll cage members do not interfere with other subsystems and the roll cage members should maintain their integrity in order to protect the driver in the event of a rollover or any impact.

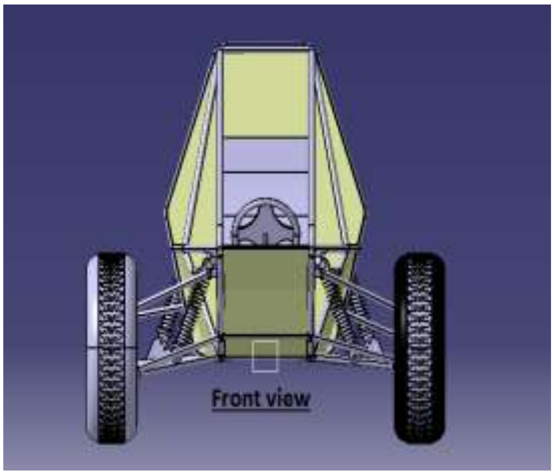

Fig.1.1: Front View of the Vehicle

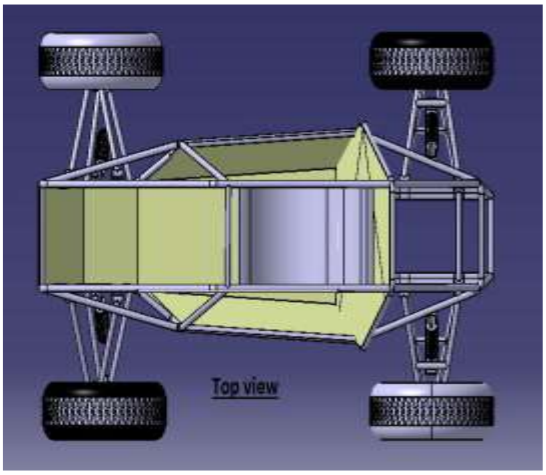

Fig.1.2: Top View of the Vehicle

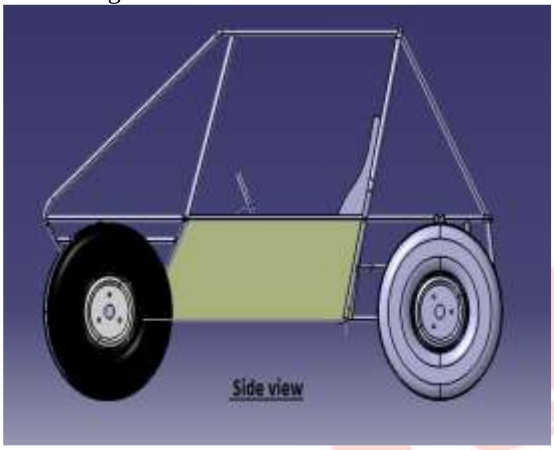

Fig.1.3: Side View of Vehicle

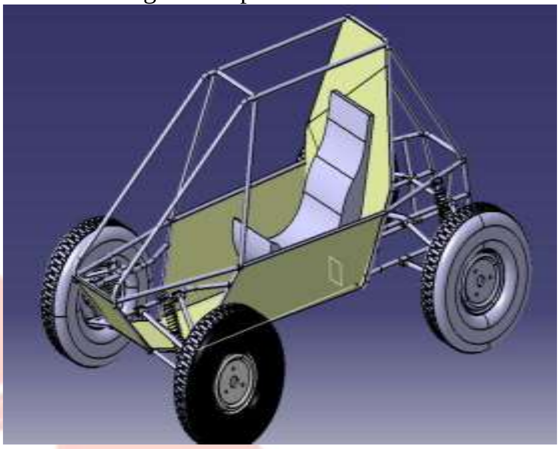

Fig.1.4: Isometric View of Baja Vehicle.

Material Selection

The material used for the required roll cage was circular steel tubing with an outside diameter of 25 mm (1 inch), wall thickness of 3.05 mm (0.120 inch) and a carbon content of at least 0.18 (Baja SAE et al, 2014). The research was conducted to choose the best possible material. The choice of material was limited to steel as per SAE rules. The material was selected on the basis of cost, availability, performance and weight of material. After thorough research, two best materials were found for the designing of the roll cage i.e.: Steel AISI 4130 Chromoly alloy and Steel AISI 1018. The reasons for using round tubing (seamless) were it is lighter than square tube as smaller gauge sizes can be used to handle the same stress as a wider square tube and a round tube always out performs the square tube. Table 1.2 shows Mechanical properties of Steel AISI 1018 tube.

Physical properties	Steel AISI 1018 Properties	Steel AISI 4130 Chromoly alloy
Density	0.284 lb./in	0.284 lb./in
Ultimate Tensile Strength	63,800 psi	97,200 psi
Yield Tensile Strength	53,700 psi	63100 psi
Modulus of Elasticity	29,000 ksi	29,700 ksi
Bulk modulus	20,300 ksi	20,300 ksi
Shear modulus	11,600 ksi	15,400 ksi
Poisson’s ratio	0.290	0.290
Elongation Break	15%	25.5%
Hardness brinell	126	197

Table 1.1: Mechanical properties of Steel AISI 1018 Tube & Steel AISI 4130 chromoly alloy

Design of Roll Cage

According to the constraint in the rulebook, the maximum speed of the vehicle is assumed to be 60 km/h or 16.66m/s. Calculations below were calculated in order to design the roll cage in best possible way.

Let Wnet = Net work done, f = Force and d = Distance travelled

Now,

$$W_{net} = \tfrac{1}{2} mv^2 \text{ final} - \tfrac{1}{2} mv^2 \text{ initial} \quad (1)$$

$$W_{net} = - \tfrac{1}{2} mv^2 \text{ initial} \quad (2)$$

$$\text{But, } W_{net} = \text{Impact force} \times d \quad (3)$$

It was considered that for static analysis, the vehicle comes at rest within 0.1 seconds after impact (Sania and Karan *et al*, 2013). Therefore, for a vehicle which moves at 16.66 m/s, the travel of the vehicle after impact is 1.66 m (Sania and Karan *et al*, 2013). From equations (1), (2) and (3)

$$\text{Impact force} = \tfrac{1}{2}\, mv^2 \text{ initial} \times 1/d \qquad (4)$$
$$\text{Impact force} = \tfrac{1}{2} \times 235 \times (16.66)2 \times 1/1.66$$
$$\text{Impact force} = 19{,}632.852 \text{ N}$$
Therefore, Impact force by speed limit = 19,633 N

The Baja vehicle will have a maximum of 7.9 G's of force during impact, G = Mass of the vehicle × Gravitational force acting on the vehicle (Sania and Karan *et al*, 2013).
$F = m \times a = 235 \times 7.9 \times 9.81 = 18{,}212.265$ N
Impact force by acceleration limit = 18,212 N
The above calculated values are practically comparable.

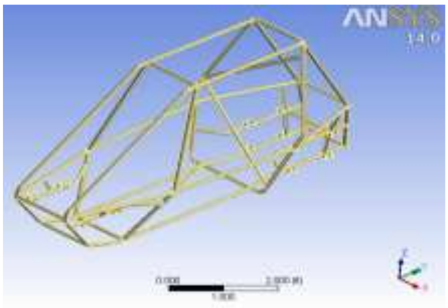
Fig. 1.5: Isometric View of Roll Cage

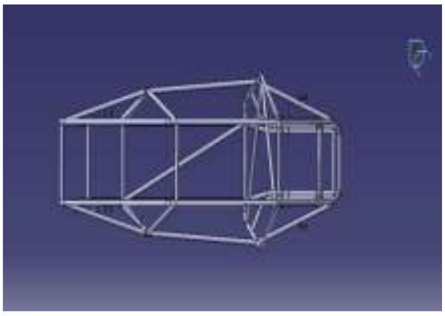
Fig.1.6: Top View of Roll Cage


Fig.1.7: Rear View of Roll Cage

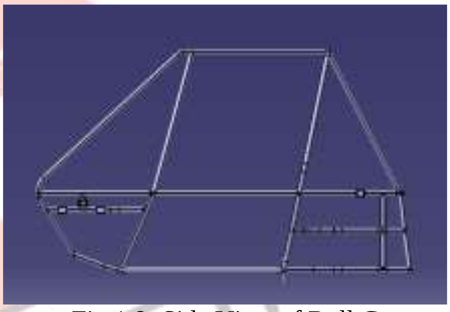
Fig.1.8: Side View of Roll Cage

Triangular in structure
Chassis have been supported with all possible triangular structure so that forces acting on members can be distributed uniformly throughout the members. Shocker has been mounted passing through the center line of triangle.

Finite Element Analysis
After finalizing the frame along with its material and cross section, it is very essential to test the rigidity and strength of the frame under severe conditions. The frame should be able to withstand the impact, torsion, roll over conditions and provide utmost safety to the driver without undergoing much deformation. The solution of a general continuum by the finite element method always follows an orderly step by step process.

Step 1: Discretization of structural domain
Step 2: Selection of a proper interpolation model
Step 3: Derivations of element stiffness matrices (Characteristic matrices) and load vectors.
Step4: Assemblage of element equations to obtain the overall equilibrium equation.
Step 5: Solution of system equations to find nodal values of the displacements (field variable)
Step 6: Computation of element strains &stresses from the known model displacements

III. FRONT IMPACT ANALYSIS

Deceleration of 10 G's was assumed for the loading which is equivalent to a static force of 26,698 N (equivalent to6000 lbf) load on the vehicle, assuming the weight of the vehicle is 270.16 Kg (600 lbs.).Load applied: 26698N/m2 on front corner
Constraints: ALL DOF's=0 on Rear corner points
Note: Here we applied load of 10G. The research found that the human body will pass out at loads much higher than 9 times the force of gravity or 9 G's. A value of 10kG's was set as the goal point for an extreme worst case collision.

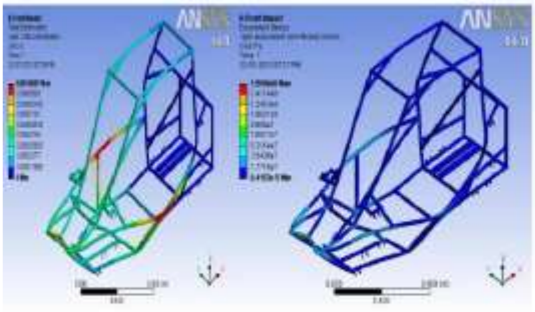

Fig1.9: Finite element analysis of Front Impact

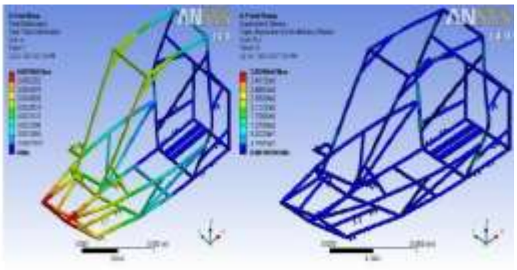

Fig 2.0: Finite element analysis of Front Bump

Front Bump Analysis

The next step in the analysis was to analyze the stresses on the shock mounts caused by a 8G load on the shock mounts. The loading was applied to the 2 shock mounts in the horizontal shock hoop in the front of the vehicle. Loading f=2000N is applied on shock mounts Constraints: All DOF's=0 at rear wheels and opposite front wheels.

Rear Impact Analysis

In this analysis a load of 8G was applied on rear corners by keeping front corners Constraint. Load applied 14000N/m^2 on rear corners Boundary conditions: All DOF's =0 on Front corner points.

Rear Bump Analysis

The next step in analysis was to analyze the stresses on the shock mounts caused by a 4G load on rear shock mounts. The loading was applied to the 2 shock mounts in the horizontal shock hoop in the rear of the vehicle. Loading F=2500N in applied on rear shock mounts. Here for loading we consider weight of driver and vehicle.

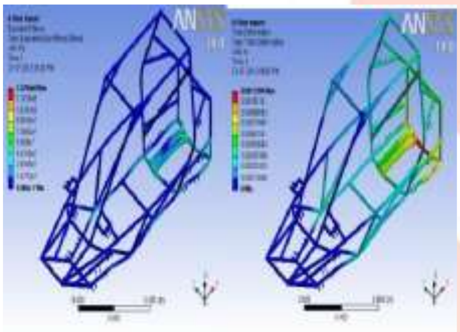

Fig 2.1: Finite element analysis of Rear Impact

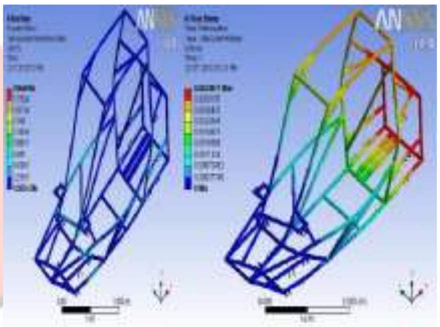

Fig 2.2: Finite element analysis of Rear Bump

Side Impact Analysis

Side impact occurs mostly when a Baja vehicle collides other side ways. In side impact a load of 4G is applied on side impact members by constraining base and opposite side. Load applied on side members 14000N/m2 Constraints: Assuming vehicle at static Opposite side impact members ALL DOF's =0.

Roll Over Analysis

Roll over mainly occurs at time of Cornering .RHO and FBM are subjected to loads. A load of 2G is applied on RHO and FBM junction. Loading F=7000 N is applied on top front points. Boundary conditions: ALL DOF's =0 on all key points of bottom members.

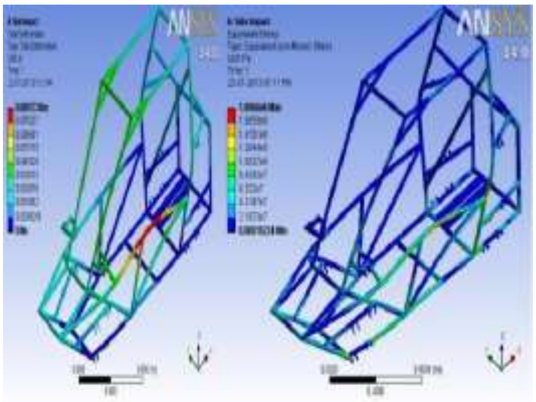

Fig 2.3: Finite element analysis of Side Impact

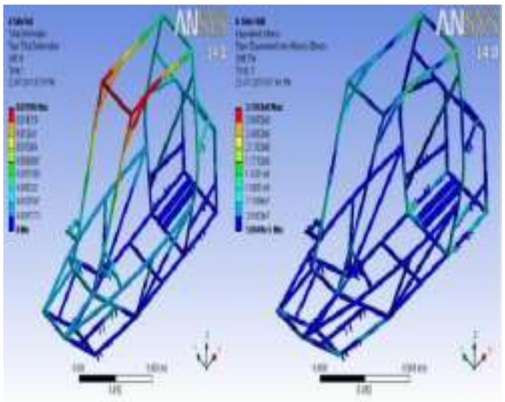

Fig 2.4: Finite element analysis of Roll over

Overall Analysis Result

Particulars	Front Impact	Rear impact	Side impact	Roll Over	Torsional rigidity	Front Bump test	Rear Bump Test
Total applied Force (N)	8G	8G	4G	2G	2G	1G	1G
Maximum total Deformation(mm)	6	13	18	7	2	3	4
Max. Combined stress (Mpa)	214	290	325	178	105	43	150
Factor of Safety	2.03	1.5	1.33	2.44	4.14	10.11	2.9

Table 1.2: Analysis Result Table

Suspension System Design

Suspension is a compromise between conflicting requirements. The suspension imparted to the vehicle was designed to provide maximum traction during cornering, stability in straight, to minimize the shock transferred to the roll cage and to provide enough ground clearance. Double A-arm suspension of unequal length was chosen to meet the above stated requirements. This design takes up a relatively large amount of space, but provides the most optimized wheel control, limiting tire scrub which can wear out tires quickly, and providing the maximum cornering grip. The front and rear suspension were simulated in optimum software. It also ensured the design was safe and compact.

Design Methodology & Objective

Designing a suspension which will influence significantly on comfort, safety and maneuverability contributing to vehicle road holding/ handling and braking for good active safety and driving pleasure. Protect the vehicle from damage and wear from force of impact with obstacles with maintaining correct wheel alignment. The overall purpose of suspension system is to absorb impacts from coarse irregularities such as bumps and distribute that force with least amount of discomfort to the driver. We completed this objective by doing extensive research on front suspension arm’s geometry to help reduce as much body roll as possible. Proper camber and caster angles were provided to front wheels. An independent rear suspension will be achieved with semi trailing arm links (with control links). The shocks will be set to provide the proper dampening and spring coefficient to provide a smooth and well performing ride. This whole analysis was done on LOTUS SUSPENSION software

Vehicle suspension specification	Values
Lateral Track Width (Front/Rear)	1517/1594.8 (mm)
Wheelbase	1710 mm
Ground Clearance	295.3 mm
Vehicle Weight	235 kg (518.086 lbs.)

Table 1.3: Vehicle Suspension Specifications

Front Suspension System

For our front suspension we have chosen a double arm wishbone type suspension. It provides spacious mounting position, load bearing capacity besides better camber recovery. Front unequal non parallel double wishbone suspension. The tire needs to gain negative camber in rolling situation, keeping the tire flat on the ground. Fox float R shocks feature an infinitely adjustable air spring, velocity-sensitive damping control, external rebound damping adjustment and ultra-light weight of 2 to 2.25lbs depending on size.

Rear Suspension System

An independent suspension system was chosen to be semi trailing link with upper & lower control arms keeping into consideration the rear loading and impact effects. The trailing link along with the upper and lower control arms helps in checking camber changes to be better. Since the motion of the semi trailing link is in the same plane as that of tires which allows proper motion of the shock absorber mounted on it. FLOAT R EVOL shocks feature a main air chamber with an infinite adjustable air spring, velocity sensitive damping control, additional air volume chamber (EVOL) for bottom-out adjustment, external rebound adjustment, and an ultra-light weight of 4 to 4.5lbs depending on size.

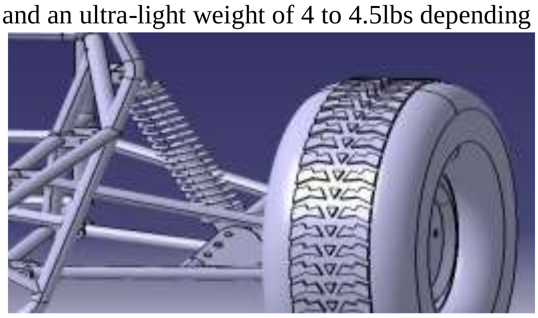

Fig 2.5: Front Suspension System

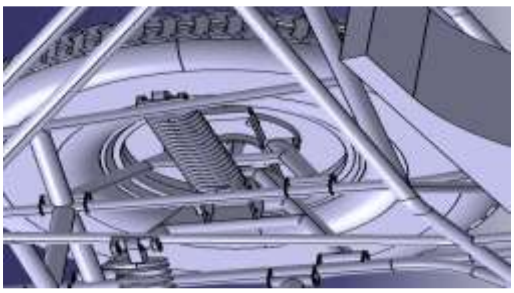

Fig 2.6: Rear Suspension System

Material Selecton

Tubing material: The suspension control arm are constructed of circular steel tubing. Factor such as strength, weight and cost were considered when choosing the control arm tubing material. Table 1.4:
Summary of material properties compares the different aspects of some of the materials considered:

Material	Carbon Content(%)	Yield Strength(Mpa)	Tensile Strength(Mpa)	Elastic modulus(Gpa)	Density(*1000 kg/m3)
DIN2391ST52	30%	436	670	190-210	7.7-8.03
Steel 1020 CD	20%	390	470	190-210	7.7-8.03
Steel 1018 CD	18%	370	440	190-210	7.7-8.03

DIN2391ST52 has higher carbon content than the other two alloys; therefore, it has better mechanical properties. DIN2391ST52 was again chosen for the tabs materials due to its superior properties. It was decided to use a minimum thickness of 0.08 inches steel plate for all the tabs in the suspension system.

IV. DESIGN AND ANALYSIS OF SUSPENSION ARMS AND UPRIGHT:

In order to withstand number of forces acting on suspension system which includes, wheel hub, stud/knuckle/upright and suspension arms had been designed with different design sequence depending up on compatibility in vehicle and were analyzed in ANSYS software. Most of the designs were completed in Soildworks and CATIA , CAD software.

Lower Wishbone A-Arm:

After of design changes, we came to finalize lower wishbone in the shape of A/V. Reason behind this is, It is the most effective structure to distribute stresses acting over the members. As we know maximum forces will be acting on lower arm.

Specification of Lower A-arm:

Pipe cross section 2 mm thickness	1 inch OD & 2 mm thickness
Arm length	16 inches
Distance between Arm members	11 inches

Total forces acting on lower arm have been discussed earlier, considering all those forces lower arm was analyzed FEA static structure with umber of cross section tube and final best results.

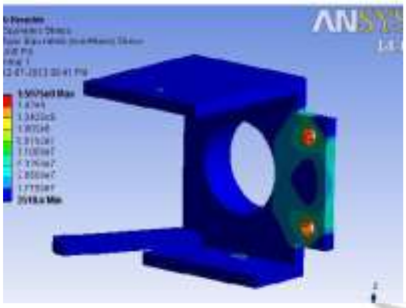
Fig 2.7: Analysis of Upright


Fig 2.8: Lower A-arm CAD Model

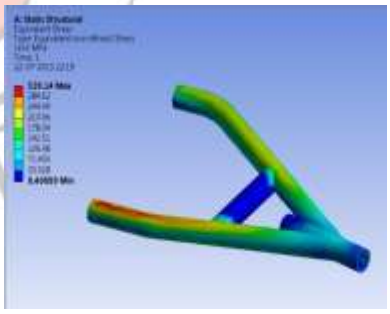
Fig 2.9: Analysis of A-arm

Upper wishbone Arm/U arm:

Wishbone suspension system is provide with an upper control system so that forces acting on lower arm will be distributed to upper arm too and also vehicle will be more stable dynamically as well as in static condition. Basically, forces that act upper wishbone are lateral force, breaking force and vehicle weight acting downward. Keeping in mind all these forces upper arm was designed to withstand all forces acting on it statically as well as dynamically. Upper wishbone provides better control over camber changes as well as caster arrangement. Just like lower arm, upper arm has also been checked for different size of tubes and best result was found for AISI 4130 chromoly having cross section, OD1inch and 2mm thickness. Shape of upper arm has been kept in shape of U, reason behind is that shocker is being mounted on lower arm which will pass through upper arm so enough space to accommodate, another reason is that U shape will cover largest part of the chassis which will increase the stability of vehicle.

Dynamic analysis of suspension on LOTUS software:

The suspension dynamics describes the orientation of the tire as a function of wheel travel and steering angle. The motions of the tire are highly dependent on the type of suspension. The various suspension systems can be designed on LOTUS/ ADAMS suspension software. The type of suspension system was selected by measuring the track width and chassis coordinates, steering angle, caster and camber angle, wheel rates, roll stiffness, king pin angle and tire scrub. The roll center position and instant center was found on LOTUS software. The characteristic curves of caster angle, camber angle toe, toe angle was drawn on different bumping analysis.

Determination of Roll center:

Determination of roll center plays a very important role in deciding the wishbone lengths, tie rod length and the geometry of wishbones. Roll center and ICR is determined because it is expected that all the three elements- upper wishbone, lower wishbone and tie rod should follow the same arc of rotation during suspension travel. This also means that all the three elements should be displaced about the same center point called the ICR. Initially, wishbone lengths are determined based on track width and chassis mounting. These two factors- track width and chassis mounting points are limiting factors for wishbone lengths. Later, the position of the tire and the end points of upper arm and lower arm are located.

The vehicle center line is drawn. The end points of wishbones are joined together to visualize the actual position of the wishbones in steady condition. When the lines of upper and lower wishbones are extended, they intersect at a certain point known as Instantaneous Center (ICR). A line is extended from ICR to a point at which tire is in contact with the ground. The point at which this line intersects the vehicle center line is called the Roll Center.

Now, extend a line from ICR point to the steering arm. This gives exact tie rod length in order to avoid pulling and pushing of the wheels when in suspension.

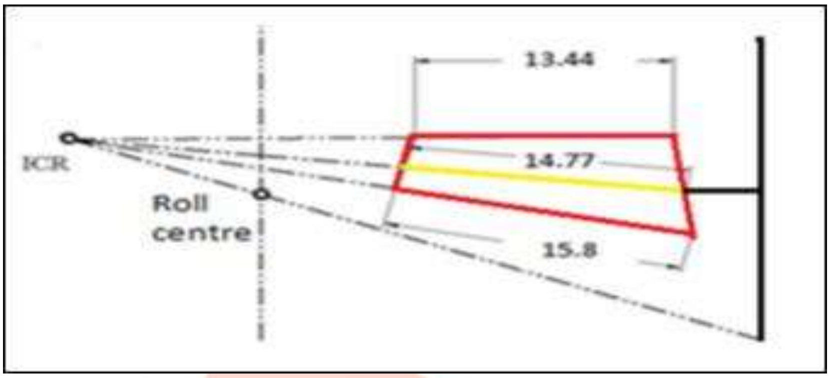

Fig 3.0 Determination of Roll Center

Design of Spring

A spring is an elastic object used to store mechanical energy. Springs are usually made out of spring steel. The force exerts in a spring is proportional to its length of compression and elongation. The spring constant of a spring is the change in the force it exerts, divided by the change in deformation of the spring. Spring is used in order to absorb shocks and for providing springing action for better comfort of the passenger.

V. SIMULATION OF SUSPENSION SYSTEM

Lotus Engineering Software has been developed by automotive engineers, using them on many power train and vehicle projects at Lotus over the past 15 years. It offers simulation tools which enable the user to generate models very quickly, using a mixture of embedded design criteria and well-structured interface functionality.

VI. SUSPENSION SYSTEM IN LOTUS

Lotus simulation software has been used to simulate the suspension geometry of double wishbone suspension system. Various co-ordinates of the entire system are given as input and the virtual model is built. It looks like as shown.

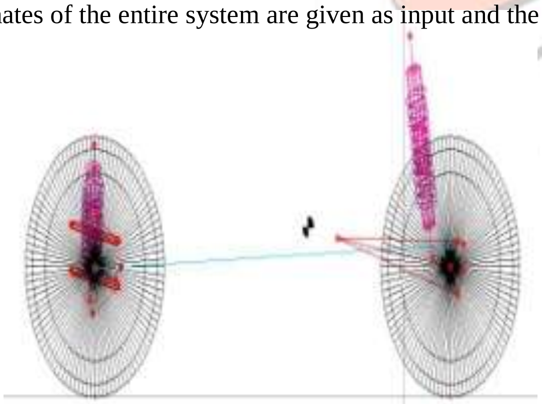

Fig 3.1: Suspension Geometry in Lotus

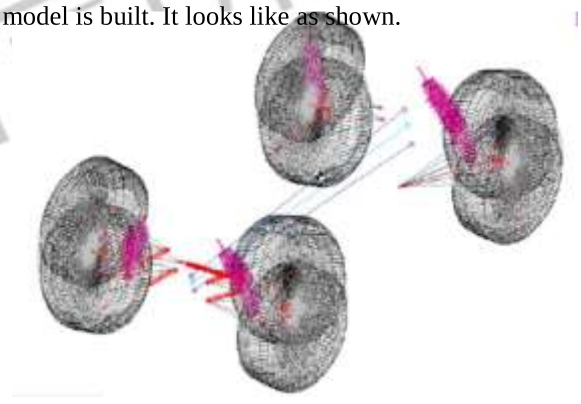

Fig 3.2: Camber Change in Bump

Plot of Camber Angle Vs Roll Angle

From the below graph of Camber Angle vs. Roll Angle, it is clear that, as the camber of the tire varies in bump and droop then roll angle also varies. The camber angle varies from -40 to +40 with roll angle.

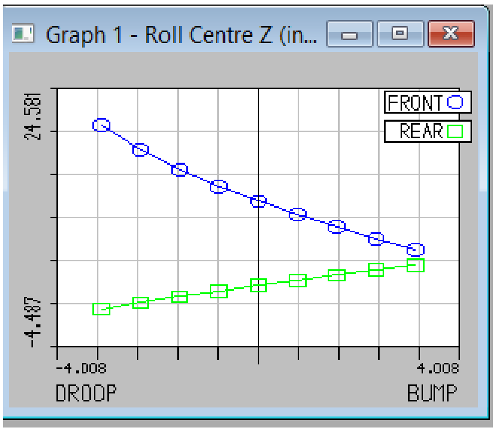

Fig 3.3: Camber change during bump

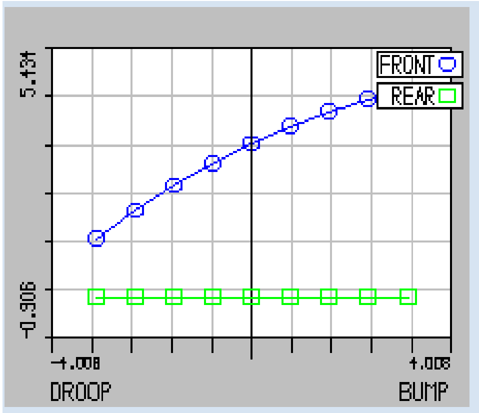

Fig 3.4 :camber change during bump

Suspension parameters	Values
Suspension Travel in Jounce	140.5 mm
Suspension Travel in Drop	65.8 mm
Front Roll Center height	310.5 mm
Rear Roll Center Height	320.5 mm
Camber Angle	20º
Camber Angle	10º
Damper Travel	158.4 mm
Spring Rate	13.8
Spring Rate(Front/Rear)	266.7 mm
Spring Wire Diameter(Front/ Rear)	8 mm
Number of Turns of Spring(Front/Rear)	16
Toe change in Travel	Minimal
Toe In (Degree)	0
Weight Distribution Bias (Front/Rear)	45/55%

Table 1.5: Calculated Suspension Parameters

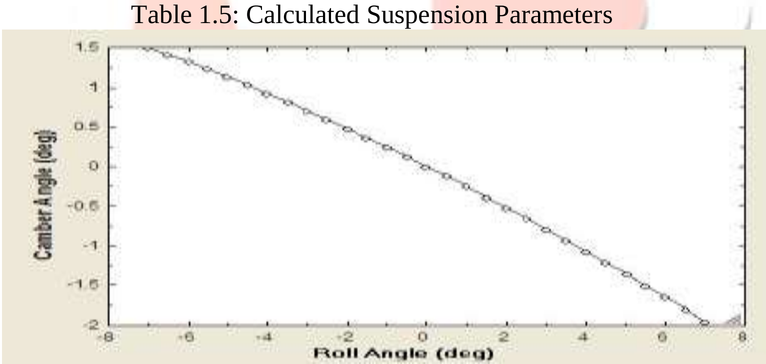

Fig 3.5: camber angle vs. Roll angle Graph

VII. CONCLUSION

The objective of designing a 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 14.0 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 vehicle. The Roll-cage Design by analyzing the failures on static structural analysis on ANSYS & the suspension system designed by considering all the input parameters on LOTUS SUSPENSION SYSTEM can be further modified for decreasing the weight and cost. Transverse leaf spring can be used to reduce the distribution of sprung weight on the suspension assembly. Pneumatic suspensions can be added in the future for better performance.

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