Design & Manufacturing of All Terrain Vehicle (ATV)- Selection, Modification , Static & Dynamic Analysis of ATV Vehicle

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***Abstract*- This paper provides in-detail description of the design considerations, static & dynamic analysis and mathematical data involved in the design of a ATV Vehicle. The focus has been laid on the simplicity of design, high performance, easy maintenance and safety at very reasonable prices.The design and development comprises of material selection, chassis and frame design, cross section determination, determining strength requirements of roll cage, stress analysis and simulationsto test the ATV against failure .During the entire design process, consumer interest through innovative, inexpensive, and effective methods was always the primary goal. Most of the components have been chosen keeping in mind the easy availability and reliability.**

According to recognition of customer's need we are going to design a vehicle which is ergonomic, aerodynamic, highly engineered and easily manufactured. Hence it makes the vehicle more efficient. Our vehicle can navigate through almost all terrain, which ultimately is the objective behind the making of any all-terrain vehicles. We began the task of designing by conducting extensive research of each main component of the vehicle.

***Keywords:* Roll cage, material, finite element analysis, strength, Power train; Final-drive, Rack and Pinion, Suspension, Brakes,**

I. INTRODUCTION

The objective of the study is to design and develop the roll cage for All - Terrain Vehicle. Material for the roll cage is selected based on strength, cost and availability. The roll cage is designed to incorporate all the automotive sub-systems. A software model is prepared in Solid works software. Later the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of Ansys Software(14). 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. As weight is critical in a vehicle powered by a small engine, a balance must be found between the strength and weight of the design. To best optimize this balance the use of solid modeling and finite element analysis (FEA) software is extremely useful in addition to conventional analysis. There are many ATV's in the market, but they are not manufactured in India. These ATV's are assembled here. So we are giving a cost effective design of an All Terrain Vehicle Frame. Since the chassis is the main 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 hill climbing, pulling, acceleration and maneuverability on land as well as shallow waters. This is aimed to design the frame of an ATV which is of minimum possible weight and show that the design is safe, rugged and easy to maneuver. Design is done and carried out linear static analysis and Dynamic analysis for the frame.

II Design Methodologies

A. Roll cage Configuration, Design & Material

One of the key design decisions of our frame that greatly Increases the safety, reliability and performance in any automobile design is material selection. To ensure that the optimal material is chosen, extensive research was carried out and compared with materials from multiple categories. The Objectives of Roll cage design, Since safety of driver is paramount to us, the roll cage is required to have adequate factor of safety even in worst case scenarios To have greater torsional stiffness to ensure lesser deflection under dynamic loading and enhanced physical object.

Material	*Yield strength*	*Outer diameter*	*Thickness*	*carbon percentage*
ST 52.3 (with seam)	460 Mpa	*1.25"*	*2 mm*	*18.899%*

Table 1: Material Properties of ST52.3

B. Steering System

The quality of the steering system and geometry also dictates the performance of the ATV. We prefer rack and pinion steering over other steering systems due to Its low cost, Simple construction & Immediate response

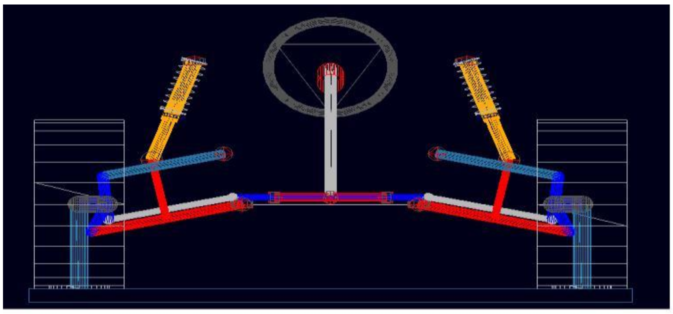

Fig1:Adams Steering System

Steering System	
Steering mechanism	Adams steering
inner angle	39.8°
outer angle	25.46 deg
Radius turning-	2.55 m
inner angle actual	37.33 deg
outer angle actual-	24.82 deg
steering Ratio-	18.53:1
Percentage Ackerman	87.23 per
Length of tie rod	48.259 "
rack	22"
Tie rods ends-	13.12"
steering Arm length	3.48"
king to king pin-	50.98"
Ackerman angle-	23.01 deg
Included Angle	9 deg

Table2: Technical Specification of steering System

C. Design criteria of Adams steering model

The objective of steering system is to provide max directional control of the vehicle and provide easy manoeuvrability of the vehicle in all type of terrains with appreciable safety and minimum effort. Typical target for a quad vehicle designer is to try and achieve the least turning radius so that the given feature aids while manoeuvring in narrow tracks, also important for such a vehicle for driver's effort is minimum. We researched and compared multiple steering systems. We need a steering system that would be easy to maintain, provide easy operation, excellent feedback, cost efficient and compatible to drivers ergonomics.

(i)Steering Geometry Steering Angle and Ratio

- 3.2 turns lock to lock which implies that steering wheel can turn 1.6*360=576 degree on one side.
- By the deflection of 576 degree of steering to left Length of steering arm d =8.841cm.
- cos (left wheel steer angle)=$(2d_2-x_2)/2d_2$=37.33
- Cos (right wheel steer angle)=24.82
- Left wheel steer angle = 37.33 degree
- Right wheel steer angle = 24.82 degree.
- Difference b/w left wheel steer angle & right wheel steer angle = 37.33-24.82 = 12.51degree.
- Steering ratio= 576/35.075= 18.53

(ii) Turning radius

- $R=(a^2+l^2\cot^2 d)^{(1/2)}$
- Cotd=(cot (in)+cot(out))/2=((39.80+25.46)/2)=32.63
- R=2.55 mt

(iii) Percentage Ackermann

- Outside ideal steer angle=(wheel base/turning radius+track/2)=39.80
- Inside ideal steer angle=(wheel base/turning radius-track/2)=25.46
- Ackerman =(inside steer angle – outside steer angle)
  = (37.33-24.82)= 12.51 degree
- Ideal Ackerman = inside ideal steer angle – outside ideal steer angle =(39.80-25.46) =14.34 deg
- % Ackerman = 100 *(ackerman/ideal ackerman)
- % Ackerman=87.23%

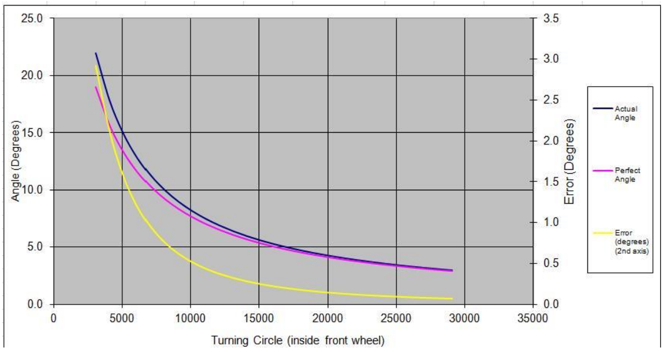

Fig2: Graph of Steering Geometry

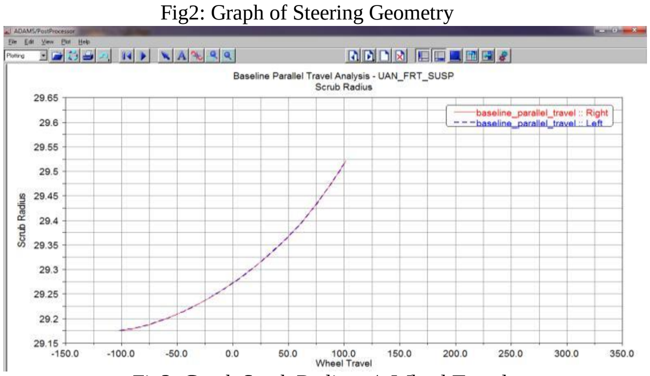

Fig3: Graph Scrub Radius v/s Wheel Travel

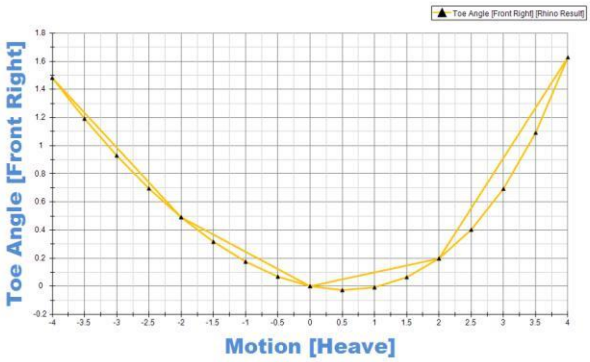

Fig4: Graph optimum Kinematic analysis of adams steering Model

D. Suspension System

The overall purpose of a 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 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. Double A Wishbone was selected for it's simple design and ability to provide a good travel.

Motion Ratio Front	0.86
Motion Ratio Gear	0.6
Front Spring Constant	12.86N/mm
Rear Spring Constant	39.6/mm
Camber	-1
Caster	3.6
Toe	0
Sprung mass	185
Unsprung Mass	90
Wheel Travel Front Left	6 "
Wheel Travel Front Right	6 "
Wheel Travel Rear Left	5 "
Wheel Travel Rear Right	5 "

Table3: Technical Specification of suspension system

E. Design Criteria of Suspension System

(i)Determination of spring rates

A frequency range of 100 to 125 (for the sprung mass) was used to obtain the testing range of2.197kg/mm and 3.793kg/mm for spring rates.125 mm was elected due to large chunk of wheel travel so that driver do not feel any discomfort.

$$f_1 = 1/2\pi \sqrt{\{(k_s*k_t)/(k_s*k_t)/m_s\}}$$

(ii)Spring Design Consideration

Helical Close Coiled springs were selected.

- Material of wire = ASTM A228 Modulus of Rigidity (G) = 79.24 GPa
- Diameter of coil (D) = 91.59mm and 76.352mm
- Parameters selected for spring wire No. of Turns was now calculated from the following formula: **$K = (G*d^4)/8*n*D^3$**
- No of Active Turns for front spring = 8 and for rear = 8
- Equivalent air suspension system is used.

Spring	Wire Diameter(d)	Total Length
Front	10mm	368.3mm
Rear	10mm	508mm

Table4: Technical Specification of Helical closed coil springs

(ii)Bump Steer

The pivot point of the tie rod affects bump steer so that the steering is unaffected by bump steer. We geometrically kept the instantaneous centre of arm and tie rod coinciding with each other. The velocity ratio and motion ratio of tie rod ends and a arm ends were kept almost equal. The mounting point of tie rod at wheel below the central axis end tie rod end at wheel below the central axis and tie rod end at the chassis was kept above the central axis. This by iteration was found non-disturbing centre changes so satisfactory bump steer.

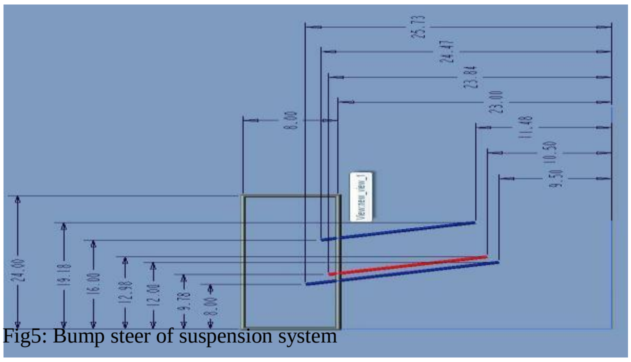

Fig5: Bump steer of suspension system

F. Driveline System

During the design the aim was to achieve a driveline which allows the maximum torque in the first gear while a top speed of 60 km/h in the top gear.In the choice of CVT or Manual gearbox we chose a manual gearbox according to the suggestions of driver as well as the research about its advantages.We followed the design methodology that the power available at the output shaft of the engine must be transmitted to the wheels at an appropriate ratio. We calculated the static friction force on the tyres due to road and calculated the first gear ratio to rotate the tyre from a stand still position.In the design we first assumed the tire size engine specifications and the loss factor. The data is as follows

- Wheel diameter = 24 inches
- Circumference = 1.914 m.
- Loss factor = 12%
- Engine power = 8.8 Hp at 3600 RPM
- Engine = 305 cc, 10 hp at 3800 rpm

Gear	Speed (km/h)	Engine Drive shaft (RPM)	Axle of driveline (RPM)
1	12	3400	104.49
2	21	3400	183.246
3	28	3400	287.95
4	43	3400	437.95
Reverse	7	3400	61.08

Table5: speed of drive shaft on different gears

Gear	Gear ratio
First	32.53
Second	18.55
Third	11.80
Fourth	7.76
Reverse	55.66

Table6: Final driveline data including differential

In order to reduce the vibrations produced by the engine and the driveline we are using anti vibration mountings of neoprene rubber.

G. *Design consideration of Brakes*

The purpose of the braking system is to increase the safety and maneuverability of the vehicle. In order to achieve maximum performance from the braking system, the brakes have been designed to lock up all four wheels at the same time. It is desired from a quad bike that it should have effective braking capability to negotiate rigid terrains. We are using disc brakes rather than drum brakes because

- More cooling air volume,
- Generated heat is less than drum brakes.
- Braking torque is less.
- The brakes are composed of the disc of outer diameter of 220mm and inner diameter 160mm and 3mm thickness.
- Brakes calliper are of floating type of with double piston as these are more economical, lighter in weight and also require fewer parts than fixed calliper.
- The total weight of the vehicle along with 60 kg driver was estimated to be 350kg.
- The weight distribution for the car was estimated to be approximately 40:60 from front to rear.
  Ü Static weight front W_f =140N

  Ü Static weight rear W_r =210N
- Static load distribution Ψ

ü Ψ_f =0.4 and Ψ_r =0.6

ü Relative centre of gravity height $X=h/w_b=0.3$

Where h is height of centre of gravity w_b is wheel base

Dynamic Condition:

- Front Dynamic axle load,$((1-\Psi_f)+X.a)M$=283.5kg
- Rear Dynamic axle load,$((1-\Psi_r)-X.a)M$=66.5kg (Where a is deceleration (0.7gunits)
- Ü Braking force rear on each tyre

  $B.F_r=(W_r/2)*a*g$=721.035 N
- Ü torque $T=B.F_r*R$=219.77N-m (where R= radius of tyre)
- ü Disc effective radius r_e=(220+160)/4=95mm
- ü Clamp load $C=T/(r_e*\mu f*n)$= 3855.614N (where n =2,no. of friction faces)
- ü System pressure P=C/A= 2.84MPa (A=area of piston)

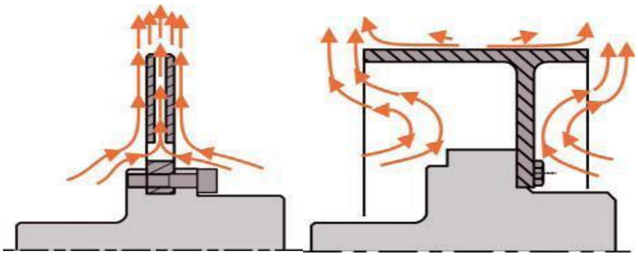

Fig6: Disc brake & Drum Brake

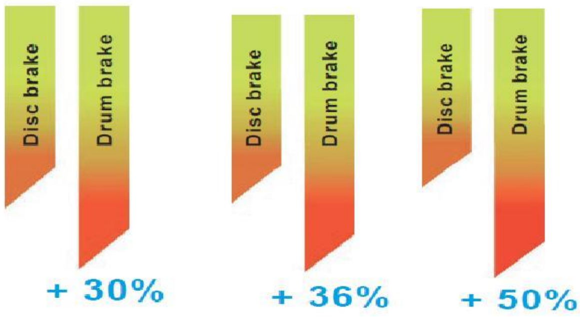

Fig7: Comparison of disc brake & drum brake with respect to stopping time ,Heat generation & Braking Torque.

From the graph given by NASA when the driver is in normal condition the pedal force applied by him is 250N.

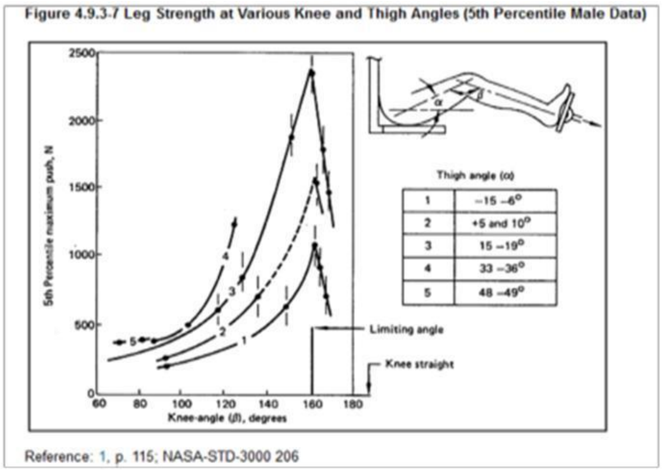

Fig8: graph given by NASA

Pedal ratio =$P*A^*/250$= 5.75:1 (where A^* =area of master ylinder)Average deceleration by considering response time, a_{ave} = v/((v/a)+0.3g) = 0.623

(where v is maximum speed i.e. 60km/h)

Stopping distance $=v^2/(2*g*a_{ave})$=22.73m Stopping time =v/a*g=2.727 sec. Power=17832.98W

H. *Driver Ergonomics*

For the purpose of driver comfort/ergonomics an assembly was imported to CATIA software. This assembly included the roll cage of the car, seat, steering system and the driver. The driver was placed in order to mimic the actual situation during the race.

Using CATIA:

- Proper pedal positioning is ensured for easy operation of pedals by driver.
- Posture of driver is examined by using human builders ergonomics module of Catia.
- The left hand reach envelop and right hand reach envelop as well as vision of the driver was examined by the Catia ergonomics analysis.
- Proper safety distance as mentioned in the rule book was also taken into consideration.
- We have used 95% male human builder for the above driver ergonomics analysis.

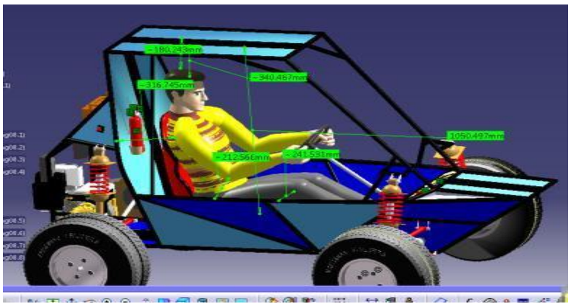

Fig9: Safe Distance of Vehicle

Fig10: Vision of Vehicle

Fig11: Right hand reach of Vehicle

Fig12: Left hand reach of Vehicle

III Results & Discussion of Static & Dynamic Analysis of ATV Vehicle

The Finite Element Analysis (FEA)of the vehicle was done using ANSYS. The stress analysis was done under worst case scenarios and maximum forces were applied in the analysis. Adequate factor of safety were ensured for all the components under these worst case conditions.

The FEA of Rollcage and suspension components was done using ANSYS Workbench 14. The analysis for rollcage included front impact, rear impact, side impact, rollover, front bump, rear bump and torsion. For all the analysis the weight of the vehicle is taken tobe350kg.

Technical Parameter	*Front impact*	*Side Impact*	*Rear Impact*
Velocity (Km/h)	*50*	*40*	*40*
Time of impact (s)	*0.2*	*0.3*	*0.2*
Force (kN)	*24.305*	*12.96*	*19.44*
In terms of G's	*7.09*	*3.78*	*5.66*

Table 7: Analysis of Roll Cage by using ANSYS

Technical Parameter	*Front impact*	*Side Impact*	*Rear Impact*	*Front Bump*	*Torsion*	*Roll over*
Max. Eqvt. Stress (Mpa)	*159.46*	*189.66*	*132.96*	*353.25*	*242.73*	*319.53*
Max. Deform at Member (mm)	*1.06*	*8.57*	*1.13*	*13.01*	*9.6*	*15.9*
Factor of safety	*3.01*	*2.53*	*3.6*	*1.04*	*1.97*	*1.5*

Table 8: Impact Assessment Data by using Ansys Software12

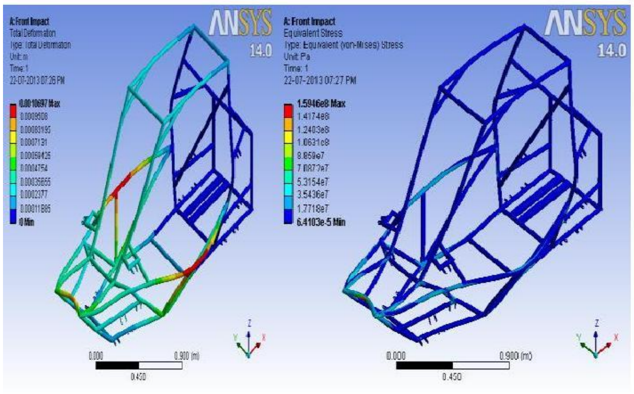

Fig13: Finite element analysis of Front Impact

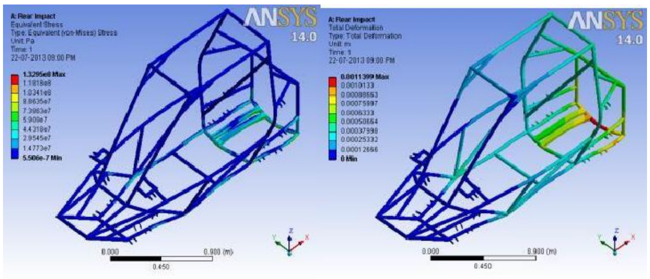

Fig14: Finite element analysis of Rear Impact

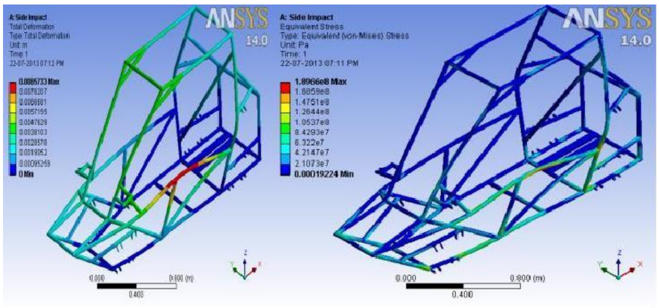

Fig15: Finite element analysis of Side Impact

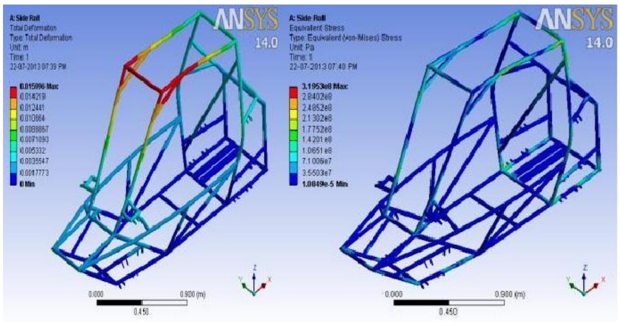

Fig16: Finite element analysis of Roll over

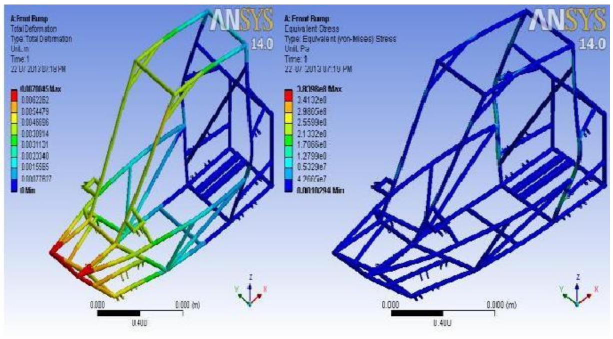

Fig17: Finite element analysis of Front Bump

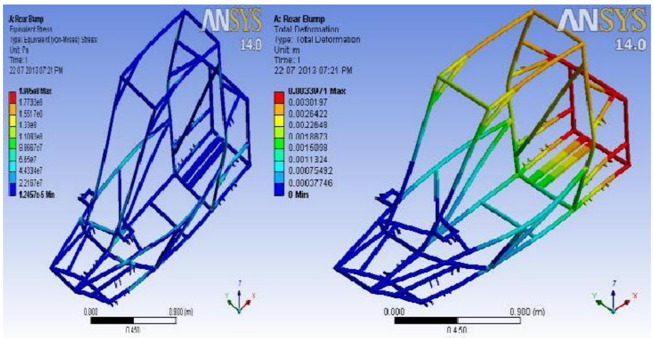

Fig18: Finite element analysis of Rear Bump

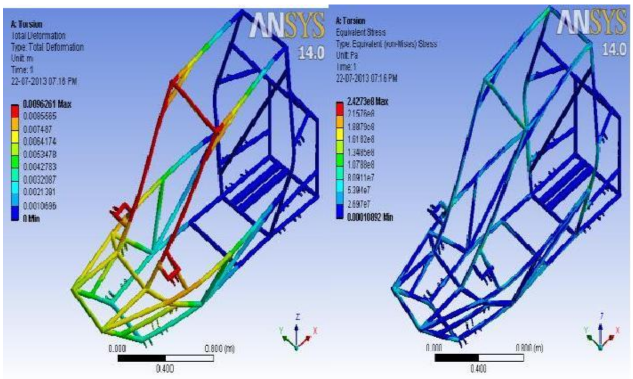

Fig19: Finite element analysis of Torsion

(i)Analysis of suspension pickup points & Clamps:

The Suspension pickup points and the clamps are the main parts that connect the unsprung mass with the roll cage. Hence in the rough terrains they are an important area to look and check for any possibility of failure. FEA of both was done to ensure the same. 3g bump forces were taken while analysing both.

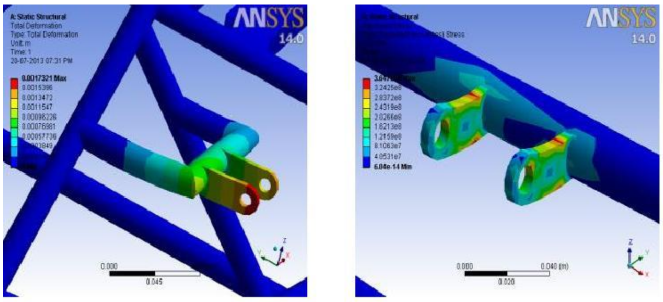

Fig20: Finite element analysis of suspension pickup points & Clamps

(ii)Analysis of knuckle

The Knuckle undergoes various direct, shear and thrust forces during the plying of vehicle. And since we have manufactured a custom design, it becomes more necessary to ensure its safety. FEA of the knuckle was done taking in account the brake calliper clamp load of 3929.8 N, bearing force of 1000 N and 3g bump forces during the run.

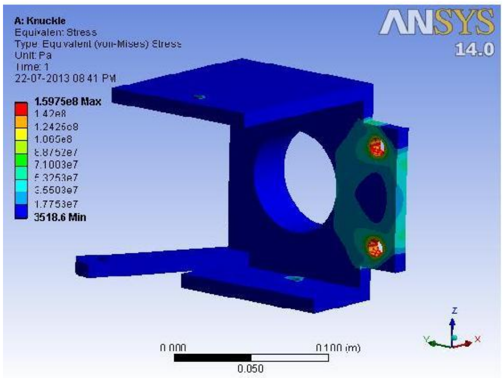

Fig21: Finite element analysis of knuckle

(ii)Dynamic Analysis on MSC Adams

The Dynamic Analysis of the suspension system of the vehicle was done on MSc Adams. Separate Analysis for Front and Rear Suspension Systems were done.The Spring Constant for the front suspension was Kept 2.197 kg/mm. The initial Caster Angle was kept +3.6 deg. The initial Camber was kept -1.0 degree. A Dynamic Analysis where this Front Suspension system undergoes a bump of 100mm was conducted and the Resulting Graphs. Various suspension parameters were

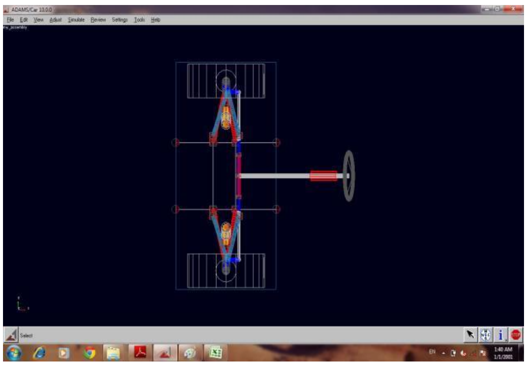

Fig22: dynamic analysis of suspension systen

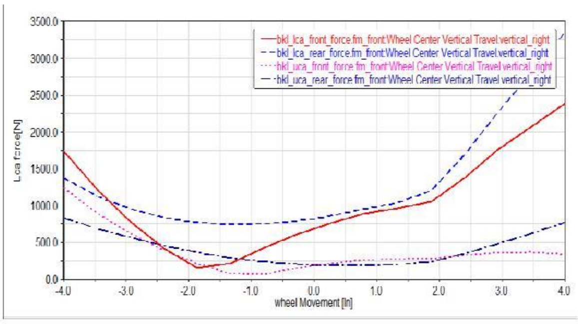

Fig23: graph of lca Force v/s wheel Movement

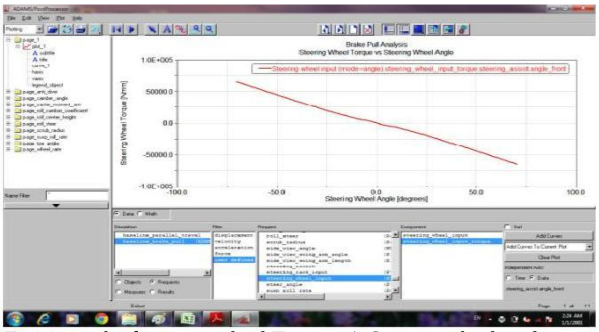

Fig24: graph of steering wheel Torque v/s Steering wheel angle

(iii)Dynamic Analysis on MSC Adams for front suspension system

The Spring Constant for Rear suspension was selected as 2.197kg/mm.The initial Camber Angle was kept -1 deg. The Kingpin Angle was kept +9 deg. The Rear suspension system then undergoes a dynamic test where it encounters a bump of 100mm

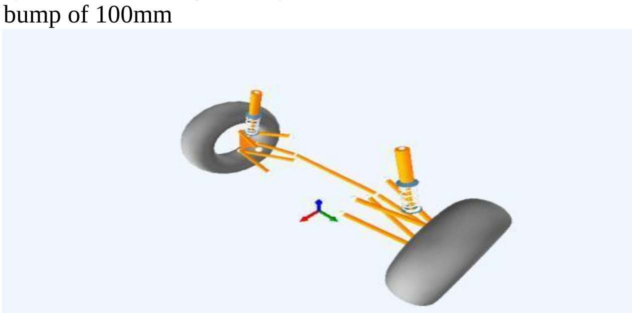

Fig25:Front Suspension System

(iv)Dynamic Analysis on MSC Adams for rear suspension

The Spring Constant for Rear suspension was selected as 3.793kg/mm. The initial parameters were kept Camber Angle 0 deg, Kingpin Angle 0 deg and caster Angle 0 deg. The Rear suspension system then undergoes a dynamic test where it encounters a bump of 100mm.

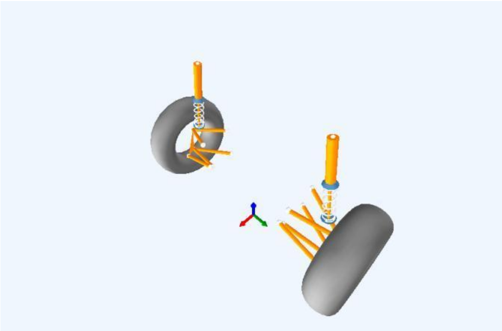

Fig26: Rear Suspension System

(iv)Dynamic Analysis on MSC Adams for safety factor, travel, deflection & Motion Ratio

For travel of 8 inch simple kinematic and dynamic analysis is used Travel for front spring= 5.16 inch,Travel for rear spring=3 inch

(v)Dynamic Analysis on MSC Adams for A Arms

For A arms loads are calculated using Adams and four bar linkage method and analysis is performed in ansys software (14) to optimise its hard points and motion ratio.

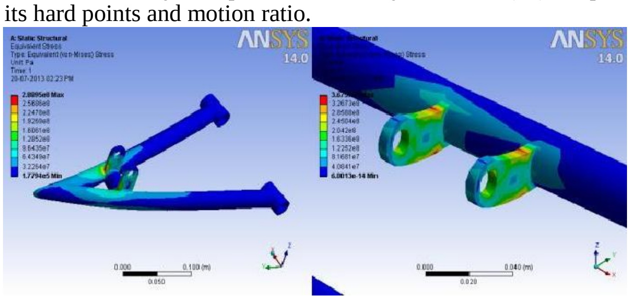

Fig27: Dynamic Analysis of A Arms

(vi)Technical Specification of ATV Vehicle

Engine Drive	Manual Four Speed
Suspension	Double Wishbone
Wheels Brakes	Disc Brakes
Steering ratio	18.5. : 1
Maximum Rack Travel	3”
Wheels	24” x 8” = 12”
Speed	58 Km/ H
Acceleration	
Stopping Distance	22.73 M
Deceleration	0.7 gm
Gradeability	-
Emission	-
Ground Clearance	12”
Turning Radius	2.5 M
Fuel consumption	10 m/ l

IV 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 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. The power-train used in the design offers easy operation and maintenance. 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 take frontal impacts of up to 159.46 Mpa and side impacts of up to 189.66 Mpa. This clearly reaffirms the vehicle’s ability to withstand extreme conditions.

V REFERENCES

[1] John C. Dixon; Suspension analysis and computation geometry; ISBN: 978-0-470-51021-6; October 2009

[2] Thomas D. Gillespie; Fundamental of Vehicle Dynamics; ISBN: 978-1-56091-199-9; February 1992.

[3] Zhanwang, Y., and Zongyu, C., “Dynamic Response Analysis of Minicar Changan Star 6350”, Proceedings of 2nd MSC worldwide automotive conference, MSC, 2000 http://www.mscsoftware.com/support/library/conf/auto00/p02200.pdf

[4] Kim, H. S., Hwang, Y. S., Yoon, H. S., Dynamic Stress Analysis of a Bus Systems”, Proceedings of 2nd MSC worldwide automotive conference, MSC, 2000. http://www.mscsoftware.com/support/library/conf/auto00/p03200.pdf

[5] Fermer, M., McInally, G., Sandin, G., “Fatigue Life Analysis of Volvo S80 Bi-fuel”, Proceedings of 1st MSC worldwide automotive conference, MSC, 1999 http://www.mscsoftware.com/support/library/conf/auto99/p00499.pdf

[6] Johansson, I., and Gustavsson, M., “FE-based Vehicle Analysis of Heavy Trucks Part I” Proceedings of 2nd MSC worldwide automotive conference, MSC, 2000 www.mscsoftware.com/support/library/conf/auto00/p01200.pdf

[7] Oijer, F., “FE-based Vehicle Analysis of Heavy Trucks Part II”, Proceedings of 2nd MSC worldwide automotive conference, MSC, 2000www.mscsoftware.com/support/library/conf/auto00/p01100.pdf

[8] Parnell, T., White, C., and Day, S., “Finite Element Simulation of 1800 Rollover for Heavy Truck Vehicles”, ASCE Engineering mechanics conference, Baltimore, 1999. http://citeseer.ist.psu.edu/407030.html

[9] Chiba, S., Aoyama K., Yanabu, K., Tachibana, H., Matsuda, K., Uchikura, M., “ Fatigue Strength Prediction of Truck Cab by CAE” , Journal of Mitsubishi Motors TechnicalReview, Vol.15, 2003, pp. 54-60. http://sciencelinks.jp/j-east/article/200310/000020031003A0250701.php

[10] Lee, D. C., Choi, H. S., Han, C. S., “Design of Automotive Body Structure Using Multicriteria Optimization” , Journal of Structural and Multidisciplinary Optimization, Vol. 32, 2006, pp. 161-167. http://www.springerlink.com/content/y70812k267632r47/

[11] Jin-yi-min, “Analysis and Evaluation of Minivan Body Structure” , Proceedings of 2nd MSC worldwide automotive conference, MSC, 2000. http://www.mscsoftware.com/support/library/conf/auto00/p00500.pdf

[12] Lee, J. N., Nikravesh, P. E., “ Steady State Analysis of Multibody Systems with Reference to Vehicle Dynamics” , Journal of Nonlinear Dynamics, Vol. 5, 1994, pp. 181- 192. http://www.springerlink.com/content/jwu5842568731t84/