Design and Manufacturing of Fsae Vehicle- Selection, Modification, Static and Dynamic Analysis of Fsae Vehicle

[1]Upendra S. Gupta, [2]Savan Parmar, [3]Prasanna Gupta, [4]Rahul Singh
[1,2,3,4]Dept. of ME, SVITS, Indore, India

Abstract
This paper provides in-detail description of the design considerations, static & dynamic analysis and mathematical data involved in the design of a FSAE Vehicle. 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 FSAE against failure.

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 FSAE Roll cage Vehicle. Roll cage is widely used in vehicle design to give a support for all components. It has been study heavily to improve the performance of the car and create a suitable one for different type of vehicle. In the race car event, roll cage is built to boost it until a limit with the intention of win over challenger. Roll cage of a vehicle is usually designed with purpose to hold the load from the components of vehicle and mass from driver and passenger. Roll cage need to satisfy a number of requirements whose aims partly conflict because of different operating conditions which are loaded and unloaded weight, acceleration and braking force, level or uneven road and straight running or cornering. Nowadays, most of the components of vehicle are in the stage of replacing with Steel and aluminum materials. This is due to the properties of material that can be designed freely to hold the load from any direction. Its light-weight property makes it possible to enlarge the performance of the car while maintaining low weight. In the Formula SAE industry already widely used of steel, aluminum and compositematerials and most of the part has been already replace with it. Thus, aluminum and steel material is highly important and it is worthy to study about it. 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 Creo software. Later the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of AnsysSoftware(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. This is aimed to design the frame of an FSAE vehicle 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

The roll cage is a tabular design, which consists of rectangular and circular cross section being attached to a roll cage. First the roll cage is modeled using Creo S/W with available dimensions. There was a significant advantage in creating a model with the help of Creo S/w that could be used for crash analyses. Then the model was meshed using ANSYS (14). In order to reduce the complexity of the model, discrete element approximations were used where possible. The CRASH ANALYSIS required the common model to be modified by adding masses representing the wheels, engine and driver.

A. Roll Cage Configuration & 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 Rollcage design,Since safety of driver is paramount to us, the roll cage is required to have adequate factor of safety even in worst case scenariosTo have greater torsional stiffness to ensure lesser deflection under dynamic loading and enhanced physical object [1].

Table 1: Material Properties of AISI 1018 (seamless)

Material	Yield strength	Outer diameter	Thickness	carbon percentage
AISI1018 (Seamless)	360 Mpa	1.”	2 mm	0.18%

B. Roll Cage Design

To begin the initial design of the frame, some design Guidelines were required to be set. They included intended Transmission, steering and suspension systems and their Placement, mounting of seat, design features and manufacturing methods. It is also required to keep a minimum clearance of 3 inches between the driver and the roll cage members.It is also necessary to keep weight of the roll cage as low as possible to achieve better acceleration. It is necessary to keep the center of gravity of the vehicle as low as possible to avoid toppling.[6]. Mounting heavier components such as engine, driver seat etc. directly on the chassis is one way of achieving low center of gravity. Also it is imperative to maintain the integrity of the structure. This is done by providing bends instead of welds which in turn reduces the cost [1].

C. Brake System

We are using disc brakes rather than drum brakes because they give more cooling air volume and radiating surface and less inertia. Disc brake inertia is lower than drum brakes (60% less). Stopping distance, Braking torque ,Stopping time and Generated heat of disc brakes is less than drum brakes. The brakes are composed of the ventilated disc of outer diameter of 220mm and inner diameter 160mm and 3mm thickness. Brakes caliper are of floating type of with double piston as these are more economical, lighter in weight and also require fewer parts than fixed caliper. The total weight of the vehicle along with 77 kg driver was estimated to be 320kg.
The weight distribution for the car was estimated to be approximately 40:60 from front to rear [1-2].

- Static mass front M_f =128N
- Static mass rear M_r=192N
- Relative centre of gravity height X=h/wb=0.145 where h is height of centre of gravity=9", wb is wheel base=62 "
- Front Dynamic axle load
- M_{fd}=(Mf +X*a*M)*g=148.416kg
- Rear Dynamic axle load Mrd=(M_f -X*a*M)*g =171.584kg (Where a is deceleration (0.44gunits)
- Braking force rear on each tyre

B.F_r=(W_r/2)* μ*g=370.312 N
- Braking force front on each tyre

B.F_r=(W_f/2)* μ*g=320.311 N

Calculating by maximum braking force
- Torque T=B.Fr*D/2=102.242N-m (where D= diameter of tyre=21.74")
- Disc effective radius r_e=(220+160)/4=95mm
- Clamp load C=T/(r_e*μf*n)= 1630.65N (where n =2,no. of friction faces)
- System pressure P=C/(2*A)= 1.201MPa (A=area of piston) area is twice as double piston caliper is used

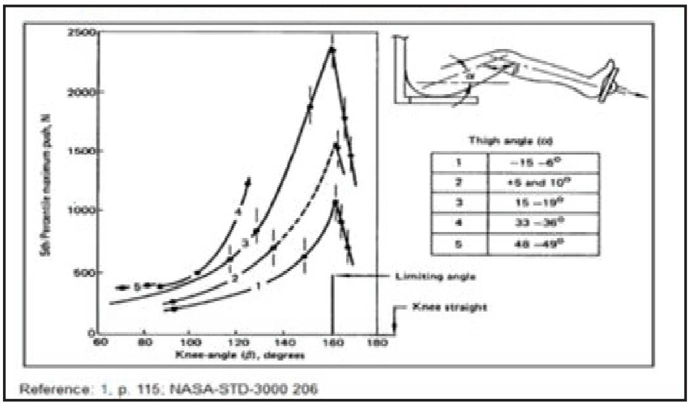

Fig. 2: From the Graph Given by NASA when the Driver is Innormal Condition the Pedal Force Applied by him is 150N

- Pedal ratio =P*A*/250= 4.054:1
- (Where A*=area of master cylinder)
- Average deceleration by considering response time, a_{ave} = v/((v/a)+0.3g) = 0.421g

(where v is maximum speed i.e. 105km/h or 29.166m/s)
- Stopping distance = v^2/(2*g* a_{ave}) =102.984m
- Stopping time = v/a*g = 6.75 sec.
- Power=20163.68W.[2].

D. Steering System(Design Selection Criteria)

We prefer rack and pinion steering over other steering systems due to:

1. Low Cost
2. Simple Construction & Working
3. Immediate Response

Steering Geometry, Steering Angle & Steering Ratio

- 2.25 turns lock to lock which implies that steering wheel can turn 1.125*360 = 405° on one side.
- By the deflection of 405° of steering to left.
- Length of steering arm d = 3.06"
- Left wheel steer angle actual = 36.55 °
- Right wheel steer angle actual = 25.99 °
- Steering Ratio = 405/31.27 = 12.95:1.

[2].

Turning Radius

- R = (a2 + I2*(Cot d)2)0.5
- Cot(d) = (Cot (in) + Cot (out))/2
- R= 2.61 M. [2].

Percentage Ackerman

- Outside ideal steer angle = tan^-1(wheel base/turning radius+track/2)=26.75°
- Inside ideal steer angle = tan^-1(wheel base/turning radius-track/2)=38.84°
- Actual ackerman turning angle =(inside steer angle – outside steer angle)

  =36.55-25.99 = 10.56°
- Ideal ackerman turning angle = inside ideal steer angle – outside ideal steer angle= (38.84-26.75) =12.09°
- % Ackerman = 100 *(Ackerman/ideal Ackerman)
- % Ackerman=87.34%. [1-2].

Table 2: Technical Specification of Steering System

Technical specification of Steering System	
Steering angle	23.68°
Ackerman Angle	18.71°
Length of Tie Rod	38.15"
Rack	16"
Tie Rod Ends	11.075"
Steering Arm Length	3.18"
King to King pin	42"
C Factor- Rack Travel pre revolution of pinion	= 3.5/2.25 =1.55

E. Electronic

The two Polaris master switches, specified in the SUPRA SAEINDIA 2014 Rulebook, have been used in the car. Out of the two master switches, the primary master switch is located on the right hand side of driver, on the main hoop at shoulder height of driver. It is a rotary type, direct acting switch and it can be easily actuated from outside the car. The other master switch is located on the left hand side of driver in the cockpit, on the front dashboard unobstructed by the steering wheel. It is a push/pull type emergency switch which can be easily actuated by the driver in any emergency situation. Actuating any of the master switches will disable the power supply to the engine and stop the engine. Apart from the two master switches a brake pedal-over-travel-switch is also installed in the car to meet the emergency condition of brake system failure. In case the brake pedal over travels, the switch will be activated and will stop the engine from running. A 12V sealed lead acid battery is used to provide adequate starting capacity to actuate the starter motor. The battery is attached to the frame of car and is placed on the back side of car. Highly durable wires were used so that they can bear high current flowing thorough them and large amount of heat generated. A non-adhesive heat shrink is also used to provide strain relief, mechanical protection and water proofing to the wires. Wheel speed sensor is used to prevent the car from over speeding. Fuel level sensor and engine speed sensor are also used. Brake light and indicator lights have been used [6].

F. Engine

The source of propulsive power is of course the engine. In accordance to rule book and our comparative analysis we found the Royal Enfield lightening engine having piston displacement of 535CCthe best suited to us. It's a four stroke, spark ignition, air cooled single cylinder engine generating maximum power of 26bhp at 5400rpm and maximum torque of 38NM @ 4000 rpm. The design methodology which we followed for a racing vehicle was based on the study that max performance in longitudinal acceleration of vehicle is determined either by engine power or traction limits on the drive wheels. At low speeds tire traction may be the limiting factor but since we are designing for speeds engine power will mainly account for the limits. Since maximum attainable power according to power performance function Pe = Hence we took 10 % range around this maximum power to determine our engines working range and calculated the angular velocity range (ω_1 and ω_2) as shown in graph plotted between Power vs. ω. [1-6].

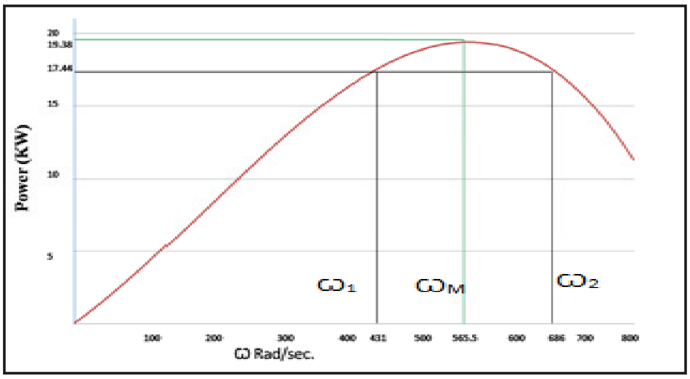

Fig. 3: Graph Plotted Between Power Vs. ω

G. Transmission

The design methodology for determining the gear ratios was based on engines working range in terms of rpm and gear box stability condition. This determined engines working range was swept in all the gears for best performance. After calculating gear ratios we found TATA Nano transmission best suited to us [6].

- The gear speed equation used is

$\omega_e = (ni*nd*Vx)/R_w$

Where, ωe=angular velocity of engine, ni =respective gear ratio, nd = differential transmission ratio, R_w = effective tire radius, V_x = forward velocity of engine

ω_e=565.8 rad/s,

ω_{max}=4.2*n4*52.78/0.276

Where, ω_{max} =686 rad/s, ω_{min} =431rad/s, ω_m=558.5rad/s

So n_4= 0.88.76

Similarly other gear ratios have been calculated.

- For stable gear box a constant relative gear ratio at a constant vehicle speed was considered such that:

$Cg = n_{i-1}/n_i$

1st=3.7 ; 2nd=2.17 ; 3rd=1.28 ; 4th=0.75.

- Acceleration capacity at different speeds is calculated using acceleration capacity equation-

$A_x= (P_M*\eta)/mV_x$

Where , η=overall driveline efficiency

The required time to reach the desired speed was evaluated using traction force equation which was further integrated. The traction eqn used is Te = (Rw* Fx)/η* ni*nd ,Where, Te = engine torque η = overall driveline efficiency [1-2].

H. Suspension

We have used independent double wishbone type suspensions. In this A-arms are non parallel and actuation of spring is done with the push rod mechanism. These type of suspension enables us to place the spring inboard of vehicle body, thus providing relatively smooth air flow from sideways of vehicle body and also lowers the unsprung mass [5,7].

Spring & Damper Selection:-For Spring and Damper system we have used the Monoshock suspensions which are easily available and economical.

Table 3: Static Vehicle Parameters

Parameter	Front	Rear
Wheelbase	1575 mm	
Track Width	1168 mm	
Vehicle'sC.G Height	254 mm	
Weight Distribution	43:57	
Unsprung Mass	58 Kg	
Wheelbase	44mm above	57mm above
Static Camber	-1.5°	-1°
Caster Angle	2°	
King Pin Inclination	3°	

I. Ride Rate Calculations

Given:-g=9.81 m/s2,Mass on front wheels =152kg, Mass on rear wheels =197kg, Natural Frequency = 163cpm(Front), 147cpm (Rer),Unit Conversion:-1N=0.2248lbs

- **Wheel rate(lbs/in) = (Wheel Frequency(CPM) / 187.8)2 x Sprung Weight(lbs)**

For Front

=(163/187.8)^{2}x 152/2 x 9.81x0.2248

=23710N/m

For Rear

=(147/187.8)^2 x 197/2 x 9.81x0.2248

=24070N/m

- **Motion Ratio Sag Values**

Sag = W (N) / k (N/m)

Sag Front = (152 x9.81) / 45000 x 2

=17.5mm

Sag Rear= (197 x9.81) / 45000 x 2

=21mm

Motion Ratio = Minimum Wheel Travel / Sag

Motion Ratio Front = 25.4 / 17.6 = 1/ 0.7

Motion Ratio Rear = 25.4 / 21 = 1/ 0.82

- **Coil Rate:-**

Coil Rate = Wheel Rate / Motion Ratio^2

Coil Rate Front = 23710/ 0.7^2

= 48387 N/m

Coil Rate Rear = 24070/ 0.82^2

=35800N/m [2-3,5].

Table 4: Static Vehicle Parameters

Technical Parameter	Front impact	Side Impact	Rear Impact	Roll over
Max. Eqvt. Stress (Mpa)	416.75	371.74	431.9	412.8
Max. Deform at Member (mm)	1.06	1.94	2.27	7.53
Factor of safety	1.08	1.21	1.05	1.09

J. Tires
Dimension = 185/60
Rim = 13"

III. Finite Element Analysis
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 Roll cage components was done using ANSYS Workbench 14. The analysis for roll cage 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 to be 325Kg

IV. Results & Discussion

A. Front Impact Test
Front Impact Test was carried out assuming a vehicles having 325 kg mass and travelling with velocity of 110km/h colliding Head on with a stationary wall. The impact force was calculated using the kinetic energy transfer theory.
Impact energy = (1/4) x M x (velocity)^2
Work done= force x displacement [3].

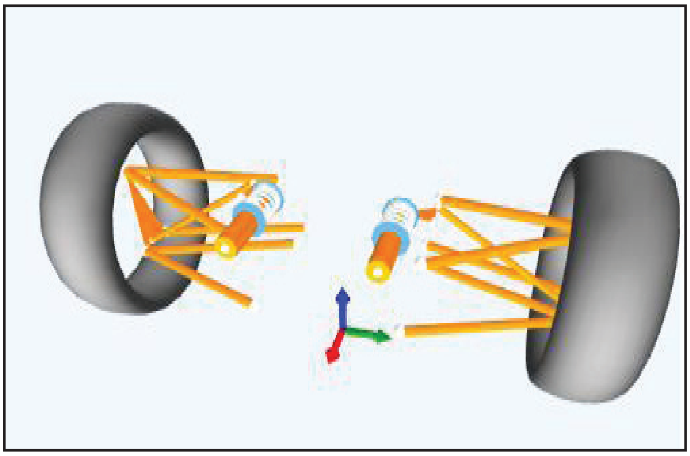

Table 4: Analysis of Roll Cage by using ANSYS(14)

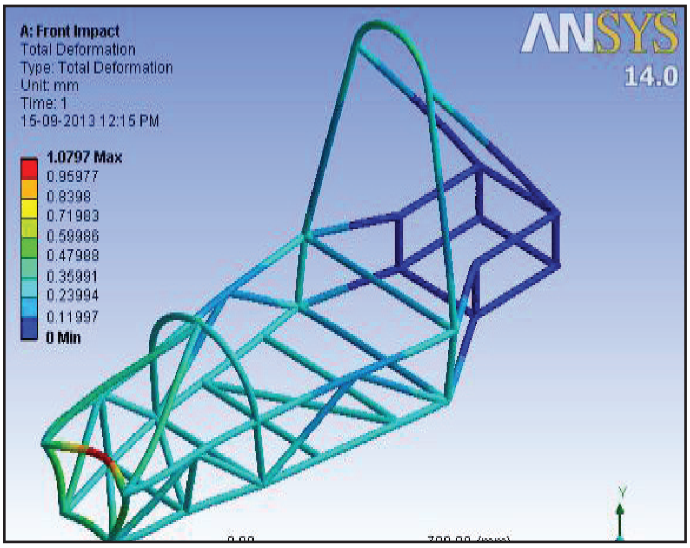

Fig. 5: Finite Element Analysis of Roll Cage for Front Impact Test by using ANSYS (14)

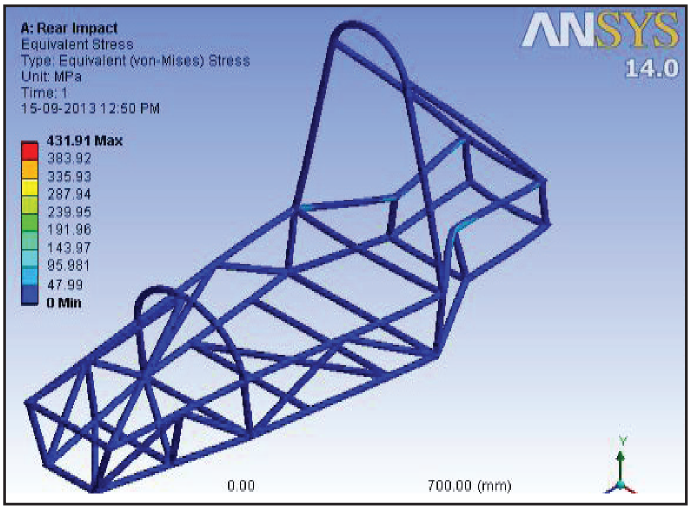

Fig. 6: Finite Element Analysis of Roll cage for Side Impact Test by using ANSYS (14)

B. Arms & Clamps
The A-arms and the clamps are the main parts that connect the unsprung mass with the roll cage. Hence they are an important areato look and check for any possibility of failure. FEA of both was done to ensure the same. 3g bump forces were taken while analysingboth .

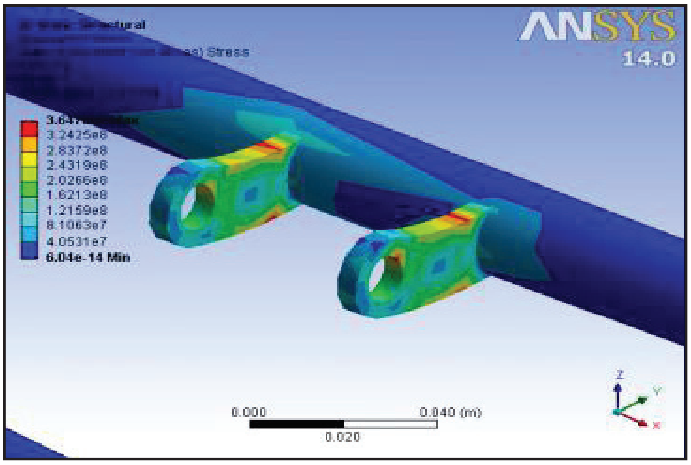

Fig. 7: Finite Element Analysis of Roll Over by Using ANSYS (14)

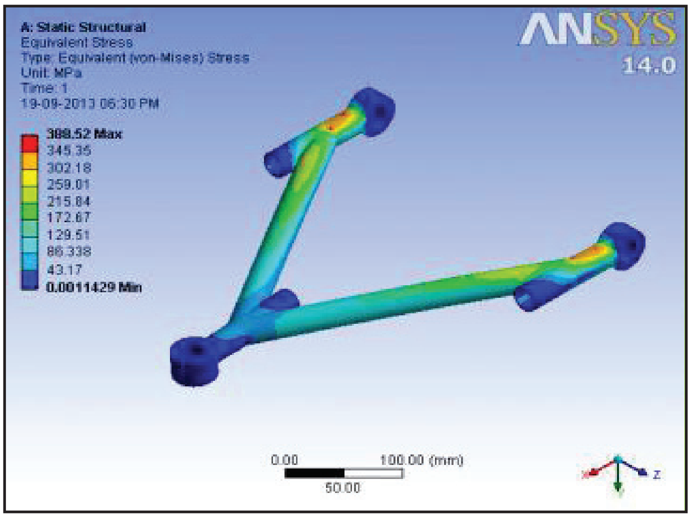

Fig. 8: Finite Element Analysis of Front Bump by using ANSYS (14)

C. 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 caliper clamp load of 1630.65 N, bearing force of 1000 N and 3g bump forces during the run.

1. Computational Fluid Dynamics Analysis

The CFD analysis of the outer body was done using ANSYS Fluent. Around 500 iterations were carried out to get the accurate results. For all the analysis the speed of the vehicle is taken to be 110 km/h. The design was optimized to produce least drag force and optimum downward lift during cornering.

2. Driver's safety

Because of the high speeds that are achieved in Formula One drivers have to be well equipped to handle the cars. The seatbelts keep the driver at his place all the time. We have 5 point system seat belt control as per our rule book. We have the helmet (SFI 31.1/2005) for the protection of the driver's head. Car is fitted with a fire extinguishing system. Car must have at least two mirrors mounted so that the driver has visibility to the rear and both sides of the car. As per our rule book driver should view at least 200° in fully seated condition [6].

Lift Forces - Direction Vector (0 1 0)

Zone	Forces (n) Pressure	Viscous	Total	Coefficients Pressure	Viscous	Total
car	-70.743378	0.19191073	-70.551467	-0.51266093	0.0013907328	-0.5112702
Net	-70.743378	0.19191073	-70.551467	-0.51266093	0.0013907328	-0.5112702

Drag Forces - Direction Vector (0 0 -1)

Zone	Forces (n) Pressure	Viscous	Total	Coefficients Pressure	Viscous	Total
car	93.650894	2.7437894	96.394684	0.67866641	0.019883609	0.69855002
Net	93.650894	2.7437894	96.394684	0.67866641	0.019883609	0.69855002

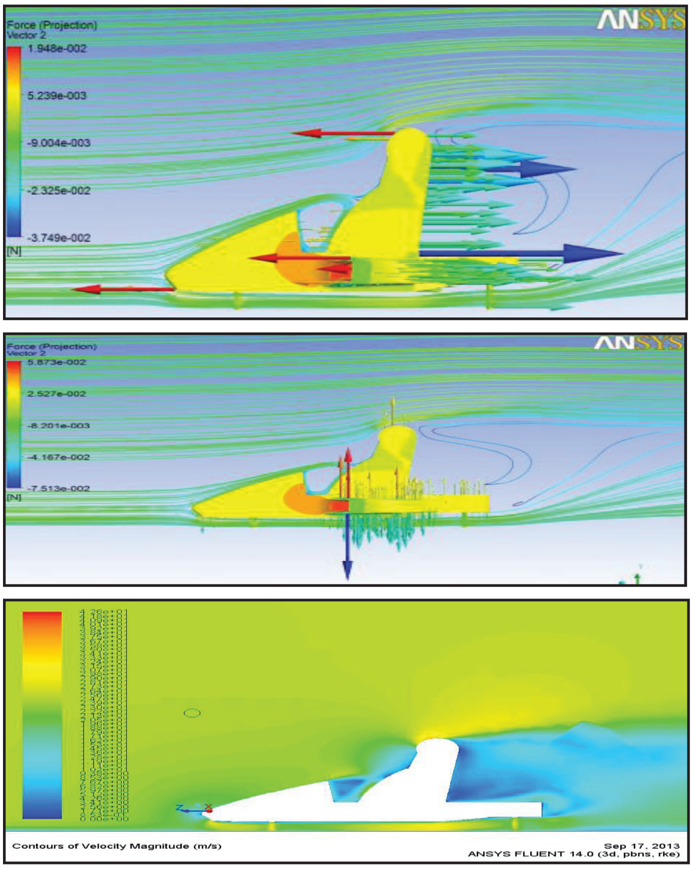

Fig. 9: CFD Analysis of Roll Outer Body r by Using ANSYS Fluent (14)

IV Conclusion

In the project is presented a detailed analysis of the crash behavior of the front and side impact rollcage structure that was designed to equip the formula FSAE car. The simulations performed by the finite element methods reveal some very important Facts. The design of the roll cage is improve driver safety by optimizing the roll cage by add and variation of mass in roll cage using ANSYS software. The design and position of the links between the energy absorbing structure and the car frame structure are found to be essential to determine the crash worthiness of the vehicle in case of front impact and side impact 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 Creo and ANSYS software packages to model, simulate, and assist in the analysis of the completed vehicle. Further, software analysis shows us that the vehicle can take frontal impacts of up to 416.75 Mpa ,Rear Impact of up to 431.9 Mpa,side impacts of up to 371.74 Mpa& Roll over 412.8 Mpa. This clearly reaffirms the vehicle's ability to withstand extreme conditions

Reference

[1] Design/Build of A Formula SAE Vehicle (L.Y Chan, M. Doecke, H. Lalwani, H.W Lau, T. Lau, C.C Lee, C.C Low.)
[2] Numerical And Experimental Analysis Of Formula SAE Chassis, With Recommendations For Future Design Iterations (The University Of Queensland)
[3] Torsional Chassis Stiffness And Crashworthiness Analysis Of The University Of Leeds (2000 Formula SAE / Student Racing Car)
[4] Design Of The Impact Attenuator For A Formula Student Racing Car: Numerical Simulation Of The Impact Crash Test (Mechanical Engineering Department, Politecnico Di Torino, Corso DucaDegli Abruzzi24, 10129 Torino, Italy)
[5] Introduction To Formula SAE Suspension And Frame Design Edmund F. Gaffney Iii And Anthony R. Salinas
[6] Formula SAE Rules, SAE International, USA, 2010.
[7] W. Jianyu, L. Yutao, H. Xiangdong,"Optimization of doublewishbone independent suspension for FSAE racing car", Machinery Design & Manufacture, Vol. 10, pp. 120-2, October 2011.

Upendra S. Gupta received his B.E degree in mechanical engineering from S.G.S.I.T.S, Indore, India, in 2005, the M.E (Gold Medalist). Degree in Design& Thermal Engg. from IET DAVV, Indore, India in 2010. He worked as a teaching assistant, lecturer and assistant professor with Department of Mechanical engineering, S.V.I.T.S Indore since 2005. Before teaching he worked in Gajra Bevel Gears corporation as an engineer for 1 year from march-98 to october99 and at TapasyaEngg. (STPSSarni) till 2002. He is currently at Reader's position in S.V.I.T.S, Indore. His research interest include thermal engineering design, applications of vortex tube, CFD, Nano-technology, power transmission and has successfully published his research in these topics. He has received Gold medal from the governor of MP In DAVV DikshantShamoroh 2013for his research work.