RNI: MAHENG/2016/68030 **VOLUME 6 (2021), ISSUE 6: 45-68** **ISSN: 2456-5342 (print)**

MGMUNIVERSITY
JAWAHARLAL NEHRU ENGINEERING COLLEGE

VISTA INTERNATIONAL JOURNAL ON ENERGY, ENVIRONMENT & ENGINEERING

VISTA

Designing and manufacturing a formula student car

Shubham Choutmal, Krushna Bhagwat* and Dr. Arvind Chel
Department of Mechanical Engineering, MGM's Jawaharlal Nehru Engineering College, MGM University Aurangabad, (M.S.), India

***Corresponding Author's Email ID : bhagwatkrish.48@gmail.com**
Mobile no: +91-9503761531

A B S T R A C T

This paper presents a detailed design procedure and its analysis in context of safety of the vehicle developed for Formula Bharat vehicle competition. This vehicle has an open wheel with rear-wheel-drive autocross race car with an engine of KTM duke 390. The vehicle developed has tubular frame chassis with independent suspension for the wheels. This paper intends to provide concise methodology for design and manufacturing with up gradation to accommodate innovation in design. The objective of design of 'Formula'type vehicle is to simulate real-world engineering and their related challenges. This paper narrates the design and analysis results of Chassis, Intake Manifold to streamline the air flow to the Engine, Rear and Front Uprights, Rocker Arm for suspension system, Rack and Pinion assembly of steering system. The adaptive design completed in several stages taking into account each and every aspect of design to get acquainted with given specifications.

Keywords : *Formula Bharat, KTM Engine, Suspension, Manifold Restrictor, Streamline flow*

1. Introduction:

Formula Bharat is a society which conducts different events for engineering students in order to give them the opportunity to explore their knowledge and learn things practically. There is requirement to manufacture a student formula race car [1]. After design and manufacturing of car, one test the race car before participating in the competition. This event for students is kept for learning team work, decision making, marketing skills etc. Generally, this event is divided into two different levels in which the first one is called as the virtual round in which all the participating teams are asked to give a presentation of 15 minutes on their design of the car followed by question-and-answer session for 10 minutes by the judges and accordingly, each team will be awarded the points. All the teams who make their way after clearing the first round are asked to build their car and to participate in the second round, which is generally of 4-5 days in which teams are asked to present their design reports and give business presentations to the judges. After that, all the student formula cars are inspected by the judges whether it abided by all the rules which are mentioned in the rule book or not. Only when designed car clear technical inspection, one gets permission to participate in different static and dynamic events like tilt test, noise test, egress test, autocross, skid pad, endurance, etc. But throughout the whole process, the major area of concern which is always to be kept in mind by the participants is the safety of the driver, and that can be achieved by performing a good number of iterations in the design as per the results obtained after the analysis of the formula student race car. So, this paper reports all about the developed Formula race car and the necessary tests which are to be performed on the race car. There is focus on optimizing the results in order to ensure and achieve the safety of the driver.The car components selections carried out for various car components are explained briefly about chassis, engine and transmission, suspension system, brakes, steering.

1.1. Chassis:

The main functions of a chassis in motor vehicles are as follows:

- To support the vehicle’s mechanical components and body
- To deal with static and dynamic loads, without undue deflection or distortion

The chassis will be constructed using normalized 4130 alloy steel, which is often referred to as ‘chromoly’. This material is stronger and more ductile that it’s lower costing alternative of Cold rolled steel. It also exhibits better welding properties leading to simpler manufacturing of the chassis as given in Table 1.

Table 1. Properties of AISI 4130

PROPERTIES	AISI 4130
Yield Strength	435 N/mm2
Ultimate Strength	670 N/mm2
Modulus of Elasticity	20447 N/mm2
Shear Modulus	7998 N/mm2
Poisson Ratio	0.29

1.2. Engine and Transmission:

When selecting an engine to be used in the Formula SAE competition, the most important thing to consider is the official Formula SAE rule book. The competition limits the displacement to 610 cc, putting us at the top of this limitation. Hence, we started exploring various options that are compared as narrated in Table 2.

1) KTM 390Duke 2) KTM 200Duke
3) HONDACBR600 4) YamahaR6

Table 2. Comparison of engine specifications

Specifications	KTM 390 Duke	KTM 200 Duke	HONDA CBR 600	Yamaha R6
Displacement (Cubic Centimeter, cc)	373.2	199.5	599	599
Max. Power (Brake Horse Power)	43	24.6	118	122
Max Torque (Nm)	35	19.2	66	64.7
No. of cylinders	1	1	4	4
Max Speed (Kmph)	169	138	252	262
Max Speed (rpm)	9500	10000	13500	14500

The engine chosen was based on the best power to weight ratio, availability and the restriction of using a maximum 610cc displacement engine for the FSAE competition. The KTM 390 Duke Engine is best suited for the above needs and is the most powerful 390cc engine producing 43bhp for a weight of just 36kgs.The engine has a cassette built 6 speed wet multi plate clutch gearbox with a primary reduction ratio of 2.67 [2].

The basic function of intake manifold is to get air from the carburetor or throttle body directed into the intake ports as shown in Fig.1. When designing the engine package for a Formula Student car, as well as other automotive applications, it is very important to design a quality intake system. The function of the intake manifold system is to deliver combustion air to the engine. Specifically, the primary design goal is to distribute the combustion mixture evenly to each intake port, as doing so improves the engine's ability to efficiently and effectively produce torque and power. [3]

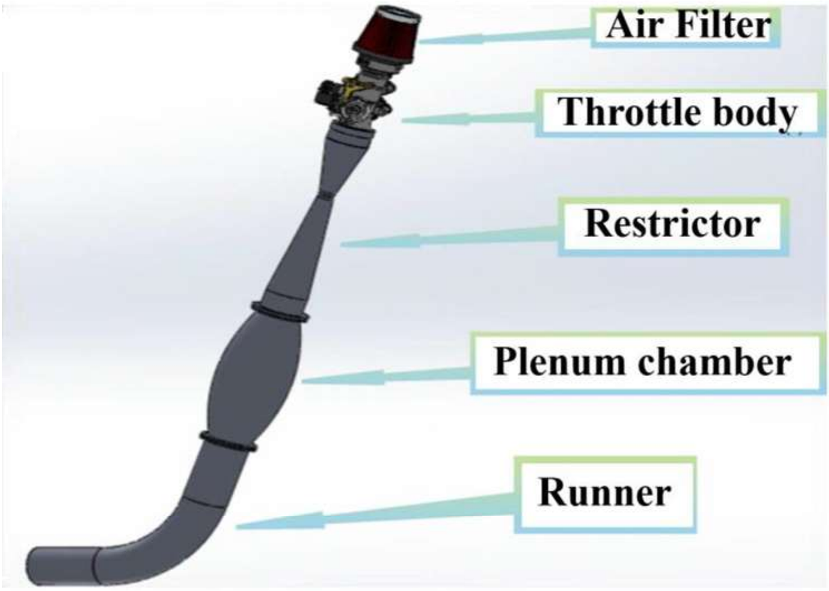

Fig.1. Components of intake manifold

1.3. Suspension system:

The suspension system of a vehicle refers to the group of mechanical components that connect the wheels to the frame or body as shown in Fig. 2. A great deal of engineering effort has gone into the design of suspension systems because of an unending effort to improve vehicle ride and handling along with passenger safety and comfort. Purpose of suspension system is as follows -

- To isolate the vehicle from disturbances so that the driver can keep control of the vehicle, without causing discomfort to passenger.
- System should minimize vertical motion, as well as pitch and roll movements, as the vehicle passes over an irregular road, performs turning maneuvers, and is accelerated or braked heavily.
- Apart from these basic operational aspects, the suspension should also provide a good level of comfort for the passengers, minimizing the movements and accelerations imposed on and perceived by them.
- The level of comfort is increasingly seen as one of the main contributing factors for purchase decision and satisfaction.
- The disturbances can be caused by irregularities on the road, or caused by loads inherent of the operation of the vehicle, such as acceleration, braking and turning, as well as aerodynamic loads.

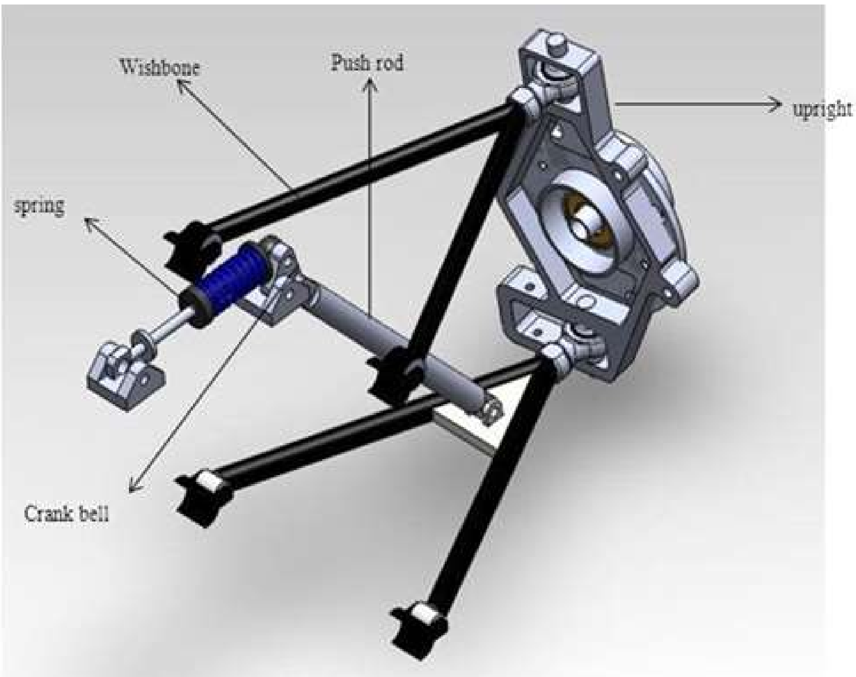

Fig.2. Components of the suspension system

1.4. Steering:

Primary function of steering system is to achieve angular motion of the front wheels to negotiate a turn. Steering system converts the rotary motion of the steering wheel into angular motion of the front wheels. Different components of steering system are shown in Fig.3. Main objectives of steering system are as follows-

- Steering system provides directional stability to vehicle.
- To obtain perfect rolling condition.
- To minimize the efforts required to turn the vehicle.
- To minimize the turning radius of vehicle.
- It minimizes tyre wear.
- To facilitate straight ahead recovery, after completing turn.

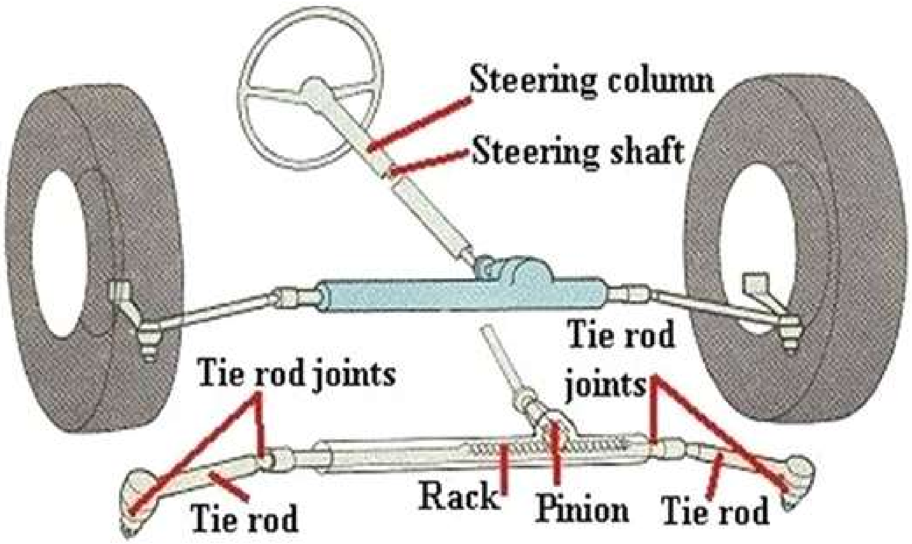

Fig.3. Steering system of the car

2. Design of Automobile:

The design of automobile involves design of all components like chassis, steering, suspension, brake system as explained briefly below.

2.1. Design of Chassis:

After material selection preparing CAD model of chassis was a next step. Based on past design knowledge, anthropometric data of tallest driver was taken and previous 3-D chassis model was modified as shown in Fig. 4. Solid works software tool was used for designing. SAE rules were taken care of while designing the system components.

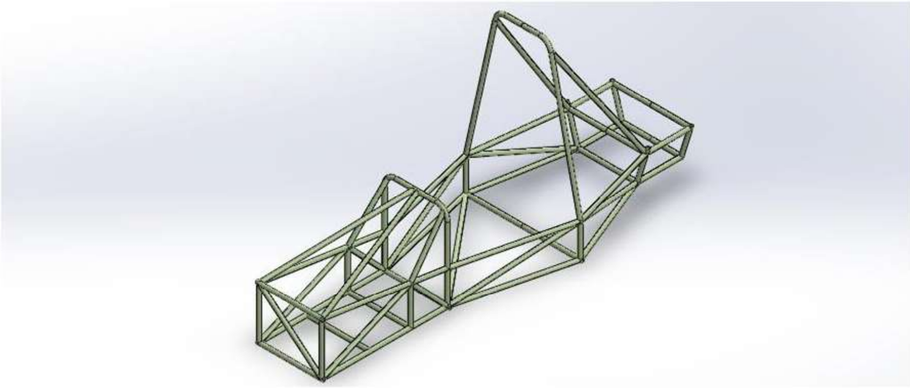

Fig.4. Developed CAD model of chassis

2.1.1. Lateral load transfer:

When cornering in a steady turn, load is transferred from the inside pairs of the wheels to the outside pair due to centrifugal force. This load transfer is called lateral load transfer.

$$\text{Lateral Load Transfer(Lb)} = \frac{\text{Lateral Acceleration} \times \text{Centre of Gravity} \times \text{Weight}}{\text{Track width}}$$

2.1.2. Lateral bending:

Lateral bending is due to the load transfer while cornering which is equal to centrifugal force and thus the force of 4774.3 N is acted on the side impact member in cockpit thus the equivalent stress is calculated (246 MPa) which is well under permissible limit. Thus, material will not start to yield during lateral bending.

2.1.3. Longitudinal load transfer:

Such load transfer occurs in a longitudinal plane under linear acceleration or deceleration

$$\text{Longitudinal Load Transfer(Lb)} = \frac{\text{LongitudinalAcceleration} \times \text{Centre of Gravity Weight}}{\text{Track width}}$$

2.1.4. Acceleration and brake test:

Due to inertia effect, acceleration forces tend to act in opposite direction to the motion of body. The mass of driver is assumed 70kg and drive train 50kg and acceleration of engine is This is less than the permissible stress 435MPa thus chassis is safe.

2.1.5. Torsional rigidity:

In order to design a car of maximum torsional stiffness the basis or generalized equation for torsion must be examined. Figure 5 below is a basic shaft constrained at one end and an applied torque T at the other, with Φ denoting the resultant twist of the shaft.

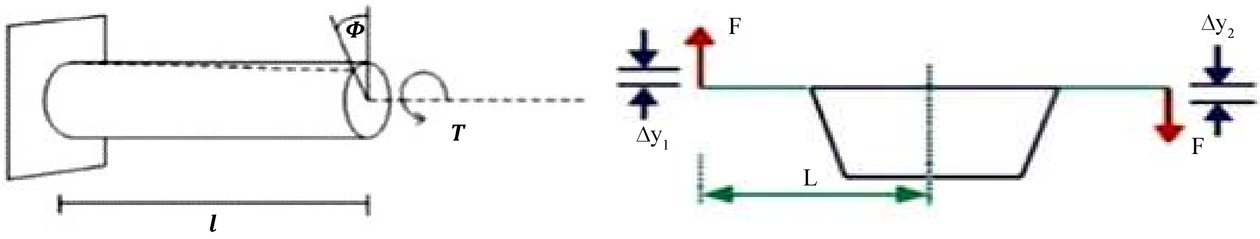

Fig.5. Bending load on the structure of chassis.

Torque (T) on the chassis can be determined as follows:

$\boldsymbol{T = \Phi JG/l}$

This equation can then be rearranged to express torsional stiffness,

Torsional rigidity, $\boldsymbol{T/\Phi = JG/l}$.

This expression displays that torsional stiffness in proportion to both the polar moment inertia and material shear modulus, whilst being inversely proportional to the length.The torsional rigidity can be calculated by finding the torque applied to the frame and dividing by the angular deflection:

$$K = \frac{T}{\phi}$$

$$K = \frac{F \times L}{\tan^{-1}\left[\frac{\Delta y1 + \Delta y2}{2L}\right]}$$

Where,

K=Torsional Stiffness

T=Torque

ϕ=Angular deformation

F=Shear Force

y1, y2= Translational displacement

Force Applied=1130N

y1=y2 =1.68mm=0.00168mm, L=20mm

Torsional rigidity, K = 482Nm/deg

2.2 Engine and transmission:

2.2.1 Design of Drive train:

The determination of final drive ratio is given below [3]:

Overall Gear Ratio

= (Gear Ratio × Primary Reduction Ratio × Final Drive Ratio × Differential Reduction Ratio)

Consider Vehicle’s top speed at 6th Gear Ratio as 110 Kmph= 68.35 mph

Maximum Speed (mph) = (Speed, rpm x Tyre Diameter, inch) ÷[Overall Gear Ratio × 336]

68.35 = [9000 × 21] ÷ [Overall Gear Ratio × 336]

Substituting the Overall Gear Ratio in the above equation, one get

68.35= [9000 × 21] ÷ [(0.8400 × 2.67 × Final Drive Ratio × 1) × 336]

Hence, Final Drive Ratio = 3.67:1

Table 3. Gear ratio with different gears

Gear	Gear Ratio	Overall Gear Ratio	Max. Speed (Kmph)
1st	2.66	26.12	34.65
2nd	1.85	18.19	49.75
3rd	1.42	13.92	65.02
4th	1.14	11.19	80.85
5th	0.95	9.37	96.60
6th	0.84	8.22	110

2.2.2 Final Gear Ratio:

A few quick hand calculations using the bikes maximum speed were conducted to determine the required final gear ratio. Research revealed the top speed of the standard KTM Duke 390 is 160km/h (44.44m/s) with 45/15 gear ratio. The bikes rear wheel measures 567mm in diameter and the car wheels are 540.2mm in diameter.

v = wr,

where: v = velocity (m/s), w = angular velocity (rad/s), and r = radius (m).

Rearranging and finding the angular velocity of the bikes rear wheel, Wras follows:

$\therefore W_r = v_r / r_r = 44.44 / (0.567 / 2) = 156.77$ rad/s

Then the front sprocket angular velocity, WF is found with standard gear ratio 45/15 as follows:

$\therefore W_F = (45 / 15) \times 156.77 = 470.31$rad/s

The desired angular velocity of the car wheels, WC is determined as follows:

$\therefore W_C = v_C / r_C = 36.11 / (0.54 / 2) = 133.69$ rad/s

Now the required final gear ratio will as function of number of teeth can be written as follows:

$\therefore W_F / W_C = t_C / t_F$

Hence, $t_C / t_F = 470.31 / 133.69 = 3.52$

This allowed the front and rear sprocket teeth combination to determine the gear ratio. With an off the shelf selection of 13, 14 or 15 tooth front sprockets then a suitable rear sprocket size was required to obtain the gear ratio. A 53-tooth rear sprocket with a 15-tooth front gives a gear ratio of 3.53, so this gear ratio was chosen as given in Table 4.

Next, the top speeds in each gear at the maximum design rpm (9000rpm) were determined using the service data (KTM Duke 390 2014). The following gear ratios were used in the calculations:

Primary Reduction Ratio = 80:30 = 2.66:1

With the secondary reduction ratios evaluated the option of running a range of front sprockets to change the top speeds could be achieved depending on the track layout.

Table 4. Final Gear Ratio

Gearing (53/15)	Gear Ratio	Overall Gearing	RPM	Maximum Speed (kmph)	Acceleration (m/s2)
1st	2.66	25.12	9000	36.47	9.90
2nd	1.85	17.50	9000	52.37	6.89
3rd	1.42	13.39	9000	68.44	5.27
4th	1.14	10.76	9000	85.11	4.24
5th	0.95	9.01	9000	101.68	3.55
6th	0.84	7.91	9000	115.79	3.12

The thickness of sprocket teeth is taken in between 8-12mm as per the availability of chain.

2.2.3 Design of Intake manifold :

The geometric design of the intake system affects the volumetric efficiency of the engine, and thus directly affects the performance of the vehicle. For the restrictor, the design of convergent-divergent nozzle is considered. Considering packing within vehicle, both the end diameters are constrained 43.5mm inside diameter of throttle body and 20mm of the restrictor as shown in Fig. 6, with ANSYS FLUENT code, numbers of iterations of steady flow analysis are carried out. Inlet pressure is taken equal to 1atm and outlet pressure has several values ranging from 0.1 to 0.8 atm.

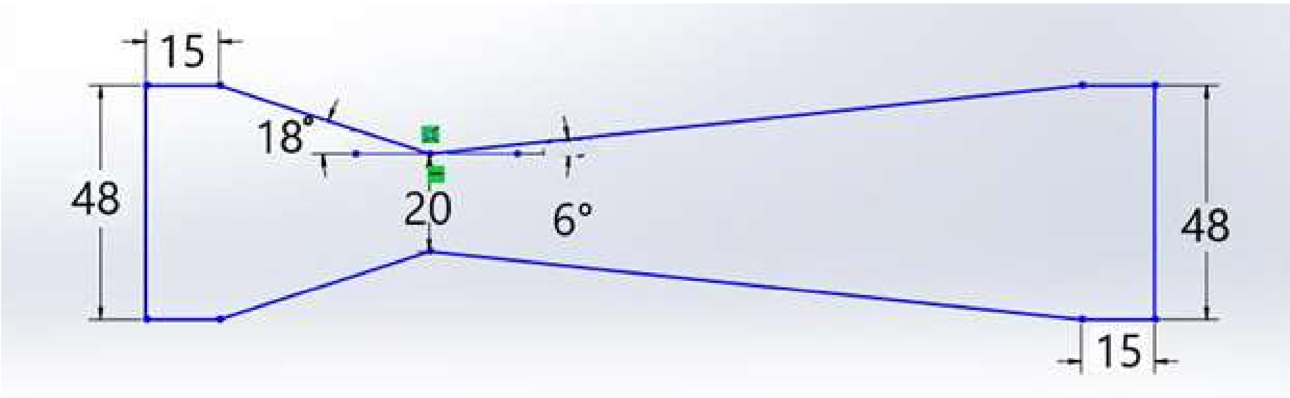

Fig.6. Developed CAD model of restrictor at Intake Manifold

Selections of the appropriate boundary conditions are very important in the case of solving boundary value problems, otherwise leads to numerical instability of the system. Considering the above factors in mind, the mass flow rate for choking condition Is calculated as shown below: -

Mass flow rate : $m = rVA$

For Ideal compressible gas :

$$\dot{m} = \frac{Ap_t}{\sqrt{T_t}}\sqrt{\frac{\gamma}{R}}M\left(1+\frac{\gamma-1}{2}M^2\right)^{-\frac{\gamma+1}{2(\gamma-1)}}$$

Where,

A = Area (ft2)

r = Density

V = Velocity

R = Gas constant
M = Mach no.
T = Room temperature (Rankine)
p = Pressure (psf)
γ = Specific Heat Ratio.
At = Throat area and Mach number, M = 1 (for minimum throat area)

Mass flow rate through restrictor (m),

$$\dot{m} = 0.532 \frac{Ap_t}{\sqrt{T_t}}$$

Let, Area at throat= A and Pressure at throat = Pt

$$\dot{m} = 0.532 \frac{Ap_t}{\sqrt{T_t}}$$

Hence, Mass flow rate through restrictor, m˙ is 0.0742 Kg/s.

The diameter of the spherical plenum was changed in order to vary plenum volume 1 time to 2 times engine displacement. Hence,the plenum volume [4] is taken as twice the engine displacement volume (373.2cc). Hence, the plenum volume is 746.4cc.

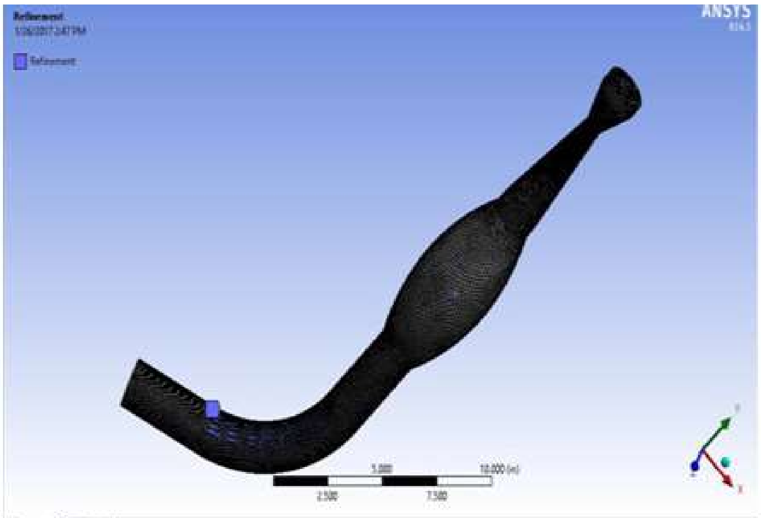

Fig. 7. Meshing of intake manifold

The formula for optimum intake runner length [4] is -

L = (EVCD*0.25*V*2)/ (rpm*RV)

Runner length 'L' is determined as 339.42mm.

Where,
EVCD = Effective valve close duration
RV = Reflective value
V = pressure wave speed

2.3 Design of Suspension system:

2.3.1 Front uprights:

For the front arrangement, the sizing of the upright was defined by the desired scrub radius and kingpin inclination the final design arrived at a scrub radius of 33mm with a static kingpin inclination (without camber incorporated) of 5°.

2.3.2 Rear uprights:

The rear upright was a bit simpler to size as the scrub radius and Kingpin inclinations were not deemed as important in the rear as wheels aren't steered. The main consideration here was the packaging because the upright has to accommodate both the pickup points for the suspension arms but also the pickup point for the toe link, a rod that attaches to the chassis and upright alongside the suspension arm mounts in order to provide toe angle adjustment.

2.3.3 Roll Centre:

Although it wasn't possible to specify an exact location of the roll Centre as in Fig. 8 and maintain it for each iteration as this would defeat the purpose of trying different geometries, it was possible to reinforce a common trend throughout all iterations regarding the placement of the roll Centre.

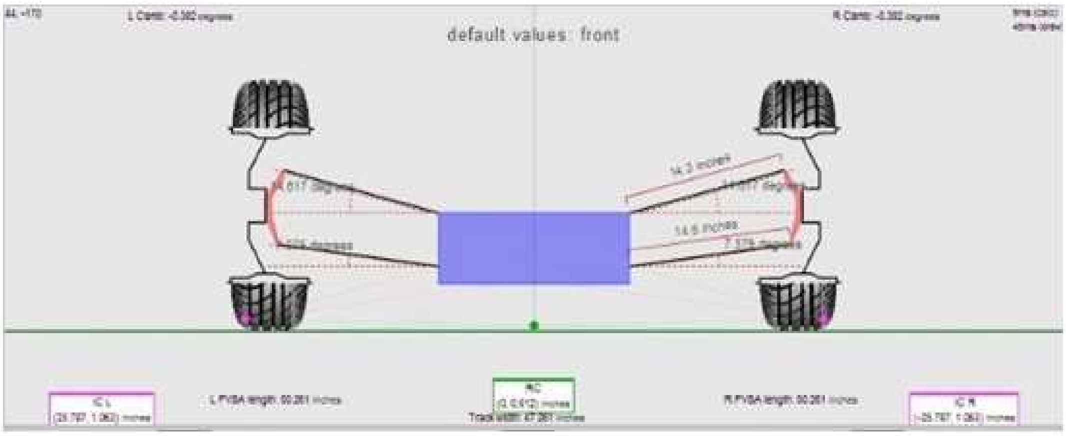

Fig.8. Front roll Centre denoted by green dot

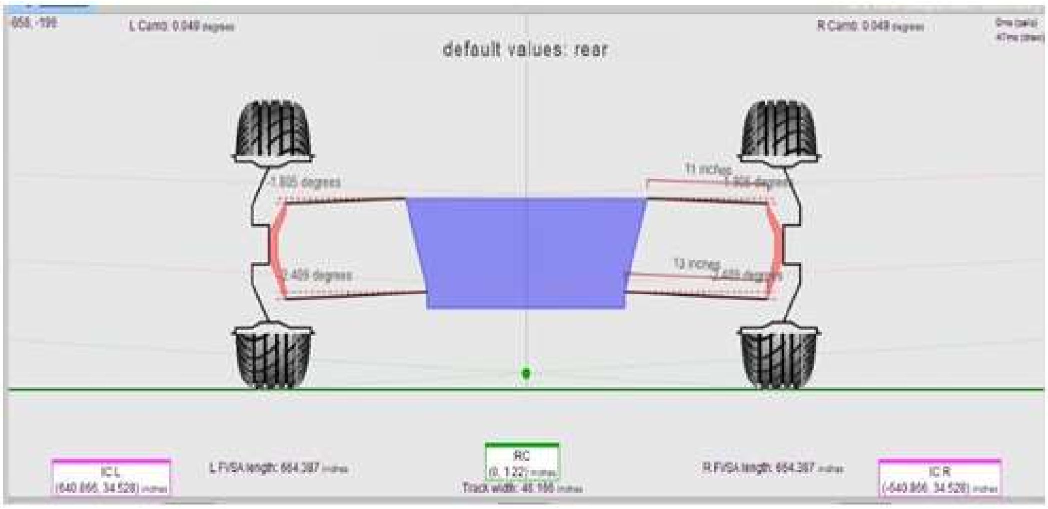

Fig.9. Rear roll Centre.

2.3.4 Relative camber of the front and rear:

As a vehicle enters a corner lateral load transfer will compress the outboard front spring inducing a lot of roll at the front end of the vehicle[5].

Mass of vehicle without driver=240Kg
Mass of vehicle with drive, M=310Kg
Mass on front wheels, M_F=108.5kg
Mass on rear wheel, M_R=201.5kg
Un-sprung mass of vehicle, M_{us}=61.233kg
Front un-sprung mass of the vehicle, M_{unF}=28.779kg
Rear un-sprung mass of the vehicle, M_{unR}=32.454kg
Sprung mass of the vehicle, M_s=248.76kg
Front sprung mass of the vehicle, M_{SF}=79.72kg
Rear sprung mass of the vehicle, M_{SR}=169.04kg
Natural Frequencies:
Front Natural FrequencyN_F=2.5Hz
Rear Natural FrequencyN_R=2.3Hz
Front Motion ratio = 1.428
Rear motion ratio = 1.25
Front track width = 47.468” = 1.21 m
Rear track width = 46.153” = 1.17m
Wheelbase = 64” = 1.62m
Front spring rateK_(sF),

$$K_{sF} = 4\pi^2 \times {N_F}^2 \times \frac{M_{SF}}{2} \times (Motion\ Ratio)^2$$

$$K_{sF} = 4\pi^2 \times (2.5)^2 \times (79.721/2) \times (1.4285)^2$$

$$K_{sF} = 20.055 N/mm$$

Rear spring rate K_{sR} can be determined as follows:

$$K_{sR} = 4\pi^2 \times {N_R}^2 \times \frac{M_{SR}}{2} \times (Motion\ Ratio)^2$$

$$K_{sR} = 4\pi^2 \times (2.3)^2 \times (169.046/2) \times (1.25)^2$$

$$K_{sR} = 27.581 N/mm$$

2.3.5 Wheel rate : The wheel rate can be determined as follows [5]:

$$\text{Wheel Rate, Kw}\left(\frac{\text{N}}{\text{mm}}\right) = \frac{SpringRate(N/mm)}{(MotionRatio)^2}$$

Front WheelRate, KwF= 20.055/ (1.4285)2
Front WheelRate, KwF=9.83N/mm
Rear Wheel Rate, KwR = 27.581/ (1.25)2
Rear Wheel Rate, KwR =17.65N/mm

2.3.6 Roll rate : The roll rate can be determined as follows [5]:

$$\text{Roll Rate, } K_{\phi s} = \frac{\pi \times T2 \times {K_w}^2}{180 \times 2 \times K_w}$$

Front role rate = $124.98 Nm$/degree angle

Rear role rate = $210.85 Nm$/degree angle

2.3.7 Roll moment at l_g lateral acceleration:

$M1g = (\text{h} - Rc) \times Ms \times g$

Front,

$M1g = (\text{h} - Rc) \times Ms \times g$

$M1gF = (\text{h} - RcF) \times MsF \times g$

$M1gF = (0.2084 - 0.018) \times 79.721 \times 9.81 = 148.904 Nm$

Rear,

$M1gR = (\text{h} - RcR) \times MsR \times g$

$M1gR = (0.208 - 0.037) \times 169.046 \times 9.81 = 283.576 Nm$

2.3.8 Spring load : The spring load for Front (FSF) canbe determined as follows [6]:

$FSF = FF1 \times MotionRatio$

FSF = 532.1925 × 1.428

FSF = 759.97N

Spring load for Rear, $FSR = FR1 \times MotionRatio$

FSR = 988.35 × 1.25

FSR = 1232.44N

2.3.9 Spring deflection:

Spring Deflection on Front =FSF/ KSF = 759.97/ 20.055

Spring Deflection on Front = 37.89mm

Spring Deflection on Rear =FSR/ KSR = 1235.446/ 27.581

Spring Deflection on Rear =44.79mm

2.4 Design of Brakes –

The vehicle has two independent hydraulic systems and it is actuated by a single brake pedal. The pedal directly actuates the master cylinder. Here no cables are used for this purpose. All rigid brake pipes are mounted securely along the roll cage. Flexible fluid lines were used for better mounting.

Pedal Ratio = A/B = 6

Force applied by Driver (F1) = 40*9.81 = 392.4 N (max.)

Force transferred to the Brake Actuator (F2) = F1*A/B= 2354.4 N (Max)

Velocity of vehicle (v) = 40 Km/hr = 11.10 m/s (Assumed optimum velocity)

Stopping time (t) = v*m/F2 = 1.46 sec

Deceleration (a) = (v-u)/t = 7.602 m/s^2

Stopping Distance (S) ={ (v^2) – (u^2)}/2a = 8.1037 m

Work Done in Braking = $mv^2/2S$ = 2356.62 J

Radius of tyre (R_w) = 115.1 mm

Disc Radius (R_b) = 105 mm

Weight distribution of Vehicle on Static Condition=35% on Front axle and 65% on rare axle.

2.4.1 The Brake Pedal:

Exists to multiply, the force exerted by the Driver's Foot from elementary statics, The force output of the Brake Pedal Assembly ({L2/L1}=6:1 and Fd=40Kg).

Fbp = Fd x {L2/L1}= 40 x 9.81 x 6= 2354.4 N

Where,

Fbp = The Force output of brake pedal assembly.

2.4.2 The Master Cylinder:

The hydraulic pressure generated by Master Cylinder:

D = 19.05 mm = 19.05 x 10^{-3} m

Pmc = Fbp/Amc = (2354.4)/{(3.14)(19.05*10^{-3})2} = 2.066 x 106 N/m^2.

Where,

P_{mc} - The hydraulic pressure generated by The Master Cylinder.

A_{mc} - The effective area of the master cylinder piston.

2.4.3 Brake Fluid, brake Pipes and hoses:

Assuming no losses along the length of the brake lines, the pressure transmitted to the caliper will be equal to:

Pcal = Pmc

Where,

Pcal= 2.066 x 106 N/m2

Pcal = The hydraulic pressure transmitted to the caliper.

2.4.4 Caliper:

The one-sided linear mechanical force generated by the caliper will be equal to:

Fcal = Pcalx Acal

Fcal = 2.066 x 106 x 3.14 x (26.44)2 x 10-6

Fcal=4535.058 N.

Where,

Fcal = The one sided linear mechanical force.

The clamp force will be equal to, twice the linear mechanicalforce as follows:

Fcalmp = Fcal x 2

Fcalmp= 9070.1164 N.

Fcalmp = The clamp force generated by the caliper.

2.4.5 The Brake Pads:

The Clamping force causes Friction which acts normal to this force and tangential to the place of the rotor, the Friction Force is given by:

Ffriction = Fclamp x ubp

ubp = co-efficient of friction between brake pad and rotor.

2.4.6 The rotor:

The torque is related to the brake pad frictional force as Follows:

Torque = FFrictional x Reffective

Where,

Tr - The torque generated by rotor.

Reffective- The effective radius of the rotor measured from the rotor center to the center of pressure of the caliper pressure.

The torque will be constant throughout the assembly.

Tt= Tw = Tr=285.708 Nm.

Where,

Tw - Torque found in tyre wheel.

2.4.7 The Tyre:

The force reacted between the tyre and the ground (rt = 0.165)

Ftyre = Tt/Rt = 1731.56 N.

The total braking Force generated is defined as the SUM of the Frictional Forces at the Four tyres which is given as follows[7]:

Ftotal = Ftyre left front+ Ftyre right front+ Ftyre left rear+ Ftyreright rear.

Ftotal= 6926.27 N.

Where,

Ftotal = The total braking force reacted between the vehicle and the ground.

2.4.8 Deceleration of a vehicle in motion:

The deceleration of the Vehicle will be equal to;

Av = Ftotal/M

Where,

M - Mass of the vehicle.

2.4.9 Kinematics relationships of vehicle experiencing deceleration:

For a vehicle experiencing a Linear deceleration, the theoretical stopping distance of vehicle in motion can be calculated as follows: (When speed of vehicle is 40 km/hr)

SDv = Vv2/av x 2

Where, SDv - The stopping distance of the vehicle.

2.5 Design of steering system:

2.5.1 Rack selection:

Team has decided to manufacture Rack and pinion inhouse.

Design value

- Module (1m) = 2 mm
- Addendum (1m) = 2.011 mm
- Dedendum (1.25m) = 2.314 mm
- Pitch (π m) = 6.28 mm
- Pressure angle = 20°
- Tooth thickness (1.5708m) = 3.14 mm

Rack -

Length = 17”

No of teeth = 25

Cross section(Circular) = DIA 22 mm

Pinion -

PCD = 30 mm

Number of teeth = 15

Angular thickness = 7.5 °

Length = 50 mm

The material selected was AISI 8630. The new rack had a larger pinion reducing the lock angle from mid position of the steering wheel by 70^0. It showed positive results in FE analysis giving factor of safety of 2 under dynamic load, clearly being safe even in accidental conditions.

Thus, the majority of the steering components and setup from the previous vehicle remains the same. However, the goal of the steering modifications this time is to make the steering compact to achieve strict ergonomic targets set forth by the team and more importantly, to better the Anti-Ackerman geometry[8].

Factors fixed before performing iterations:

- Wheel track (w) = 47.5 inches
- Wheel base(l) = 64 inches
- Rack length = 17 inches
- Rack travel =4.21 inches
- Steering Wheel lock to center = 176 degrees

Iterations were performed by varying length of tie-rod and steering arm and rack offset from axle in AutoCAD. Based on these iterations the following values were obtained:

- Steering arm=90 mm
- Tie rod= 345.66 mm
- Turning radius unto CG= 2.356 m
- Turning radius unto centre of front axle=2.805m
- Initial Angle of Steering Arm = 30 degrees
- Inner Wheel(δ_i)= 29.36 degrees
- Outer Wheel(δ_o)= 44 degrees

2.5.2 Bump steer:

Bump steer is when the front wheel moves up and down. The front wheels should maintain a particular direction. It’s most important for the wheels to have minimal bump when negotiating turns[9]. There are certain elements of construction of the front-end components that will make this happen.

2.5.3 Steering ratio:

The steering ratio is the number of degrees you have to turn the steering wheel, for the wheels to turn a number of degrees. Steering ratio isconsidered 6:1 [10]

Wheel turn= 29.36°*6=176.16°

Now in how many turns 176°will achieve as PCD of pinion is30mm therefore,

No. of turns= 176*2/360=0.977turns≈1 turn

Rack travel = 1*2*3.14*30/2=94.2mm

2.5.4 Steering effort in static condition:

Maximum Normal Reaction at Front Wheels = 925.68 N

Torque at steering arm = μ x Normal reaction x Scrub

= 0.7 x (925.68 / 2) x 0.097 = 31.65 Nm

Force perpendicular to steering arm = 274.464 N

Force along the rack = 2 x (274.464/cos (20.9449)) = 587.764 N

Torque at pinion = Force along rack x Radius of pinion = 4.702 Nm

Radius of steering wheel = 150 mm

Effort at steering wheel = Torque at pinion / Radius of steering wheel= 31.347 N

2.5.5 Steering effort considering dynamic weight distribution at full braking:

Maximum Normal Reaction at Front Wheels = 2403.45 N

Torque at steering arm = μ x Normal reaction x Scrub

= 0.6 x (2403.45 / 2) x 0.0977 = 70.446 Nm

Force perpendicular to steering arm = 640.416 N

Force along the rack = 2 x (640.416 / cos(20.9449)) = 1371.449 N

Torque at pinion = Force along rack x Radius of pinion = 10.971 Nm

Effort at steering wheel = Torque at pinion / Radius of steering wheel = 73.143 N

3 Results and discussion:

3.1 Chassis:

3.1.1 Lateral load transfer:

When cornering in a steady turn, load is transferred from the inside pairs of the wheels to the outside pair due to centrifugal force. This load transfer is called lateral load transfer. Results of Lateral load applied on the chassis are shown in Fig.10. The chassis designed under 970.2 N loading condition has factor of safety is 2.47 more than acceptable limit 2.

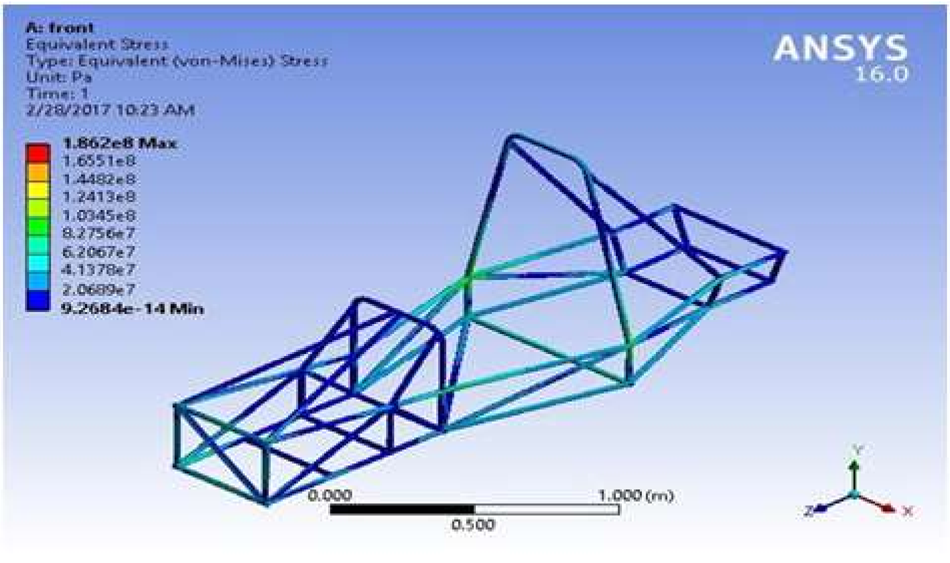

Fig.10. Stress analysis for lateral load

3.1.2 Lateral bending:

Lateral bending is due to the load transfer while cornering which is equal to centrifugal force and thus the force of 4774.3 N is acted on the side impact member in cockpit thus the equivalent stress is calculated (246 MPa) which is well under permissible limit. Thus, material will not start to yield during lateral bending. The chassis designed under 970.2 N loading condition has lateral bending with factor of safety is 3.23 more than acceptable limit 2.

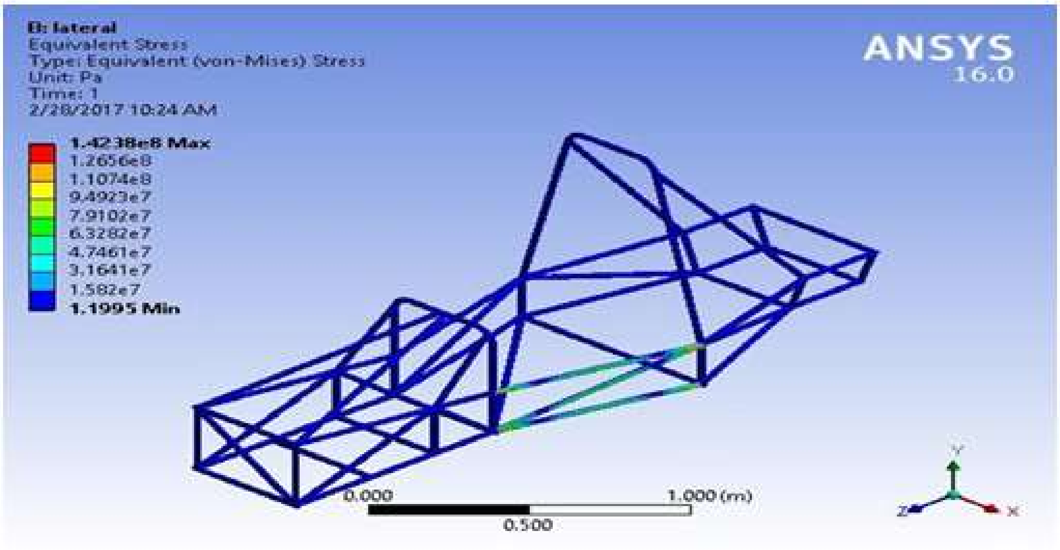

Fig.11. Bending analysis for lateral load

3.1.3 Acceleration and brake test:

Equivalent stress is calculated for this dynamic test which comes as 221 MPa. This is less than the permissible stress 435MPa, thus chassis is safe.

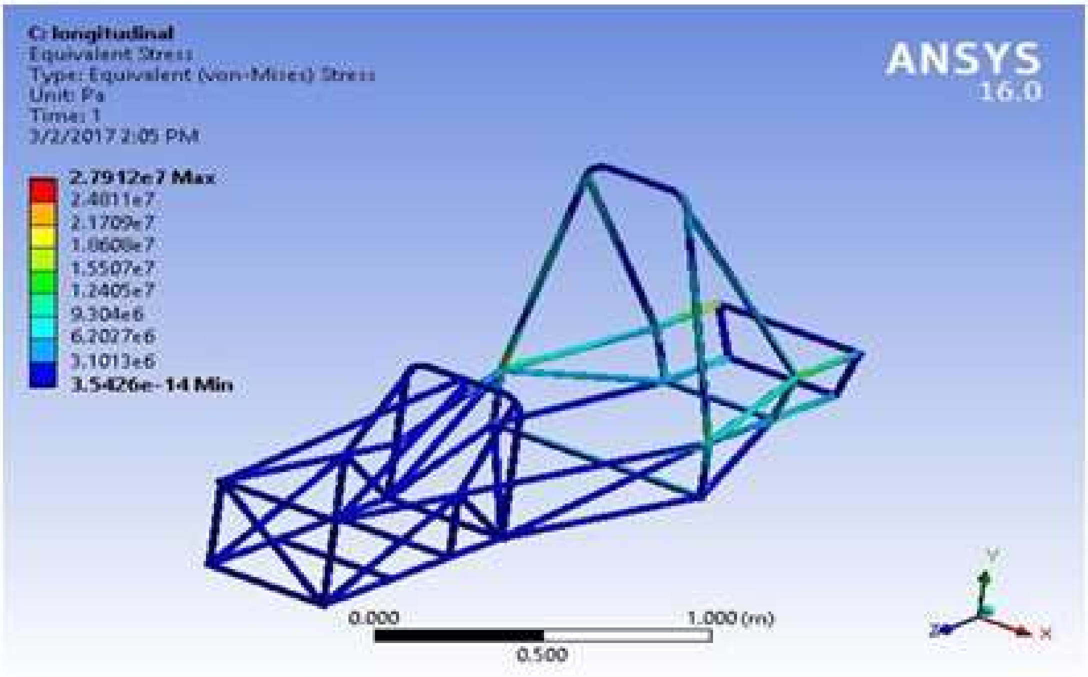

Fig. 12. Stress analysis in acceleration and brake test.

3.1.4 Torsional rigidity:

A best possible chassis would be one that has high stiffness; with low weight and cost. If there is significant twisting, the chassis will vibrate, complicating the system of the vehicle and sacrificing the handling performance.

Torsional rigidity= K=482 Nm/deg

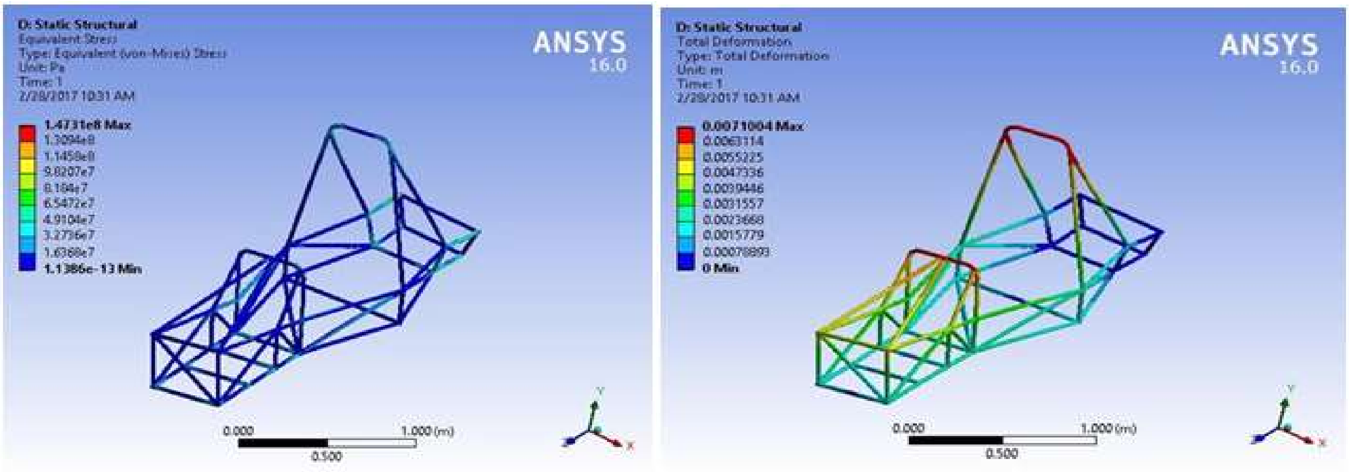

Fig.13. Torsional rigidity analysis.

3.2 Engine and transmission:

3.2.1 Sprocket:

Teeth on Engine sprocket (T1) = 15

Teethon Rear Sprocket (T2) = 52

Chain Pitch = 0.625 inch (15.875mm)

Outside diameter, OD1 = 169.9360mm

Outside diameter, OD2 = 357.6950mm

The thickness of sprocket teeth is taken in between 8-12mm as per the availability of chain.

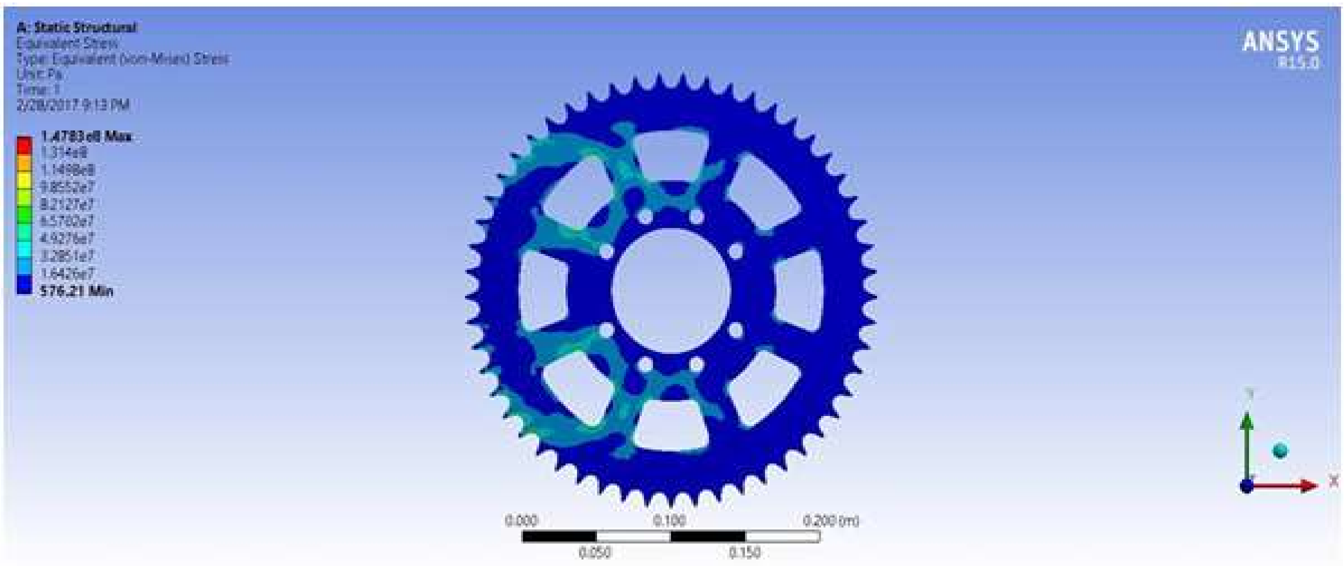

Fig.14. Structural analysis of sprocket

No deformation and accumulation of stress at one point is observed so the design is safe.

3.2.2 Intake manifold:

The primary goal in the design of an intake manifold system is to improve the engine's ability to efficiently and effectively produce torque and power through the even distribution of the combustion mixture to intake. In order to estimate the flow distribution of the system, as well as visualize and analyze the flow to find areas of recirculation and separation that increase pressure drop, simulations were done using CFD.

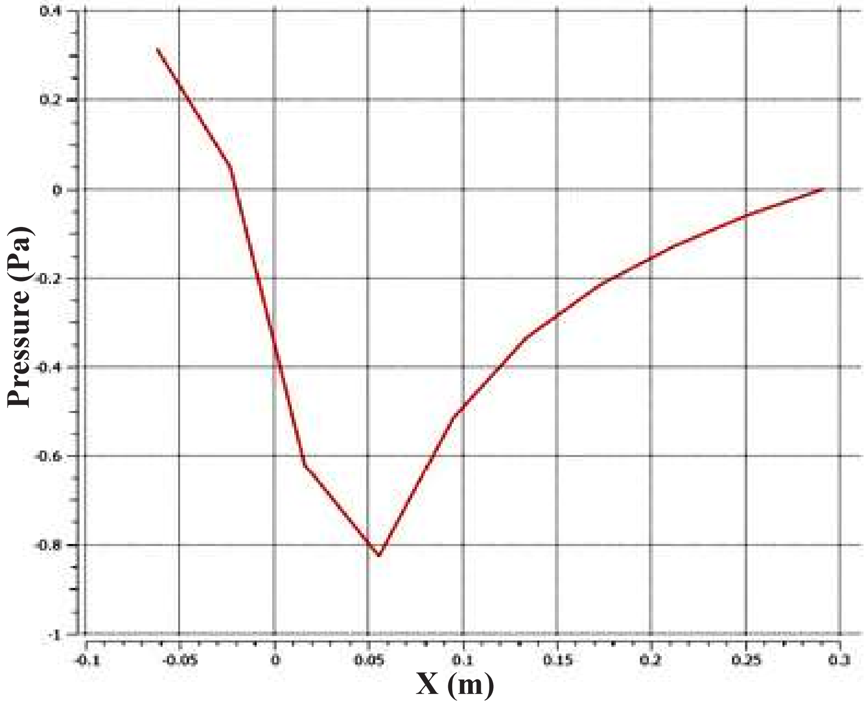

Fig. 15. Pressure contours diagrams

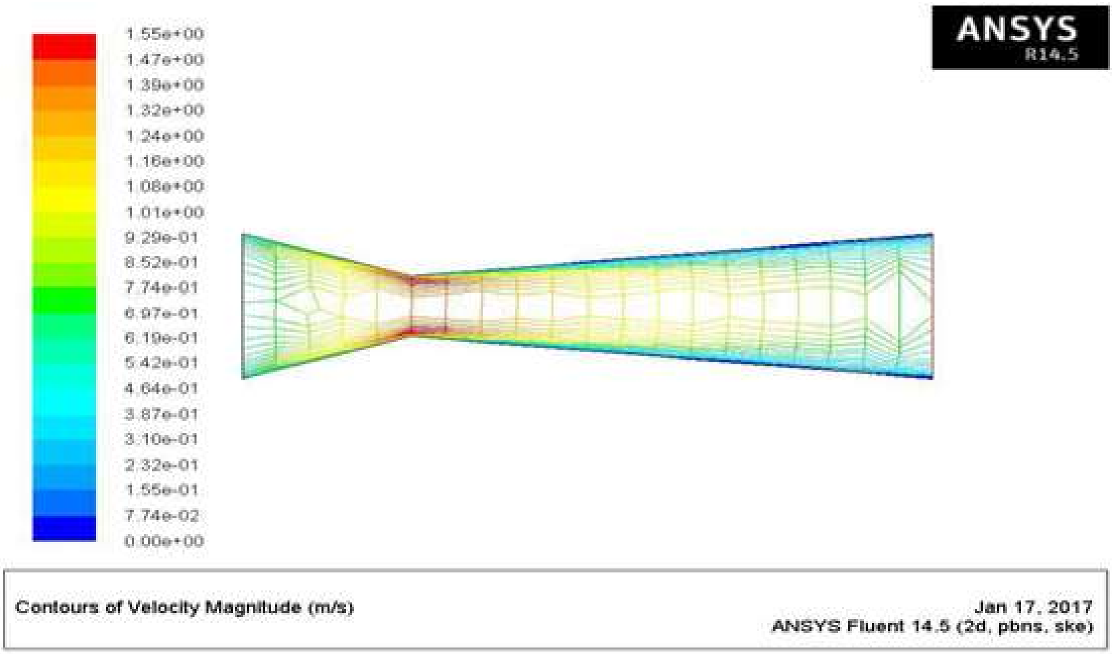

Fig.16. CFD analysis for velocity contours

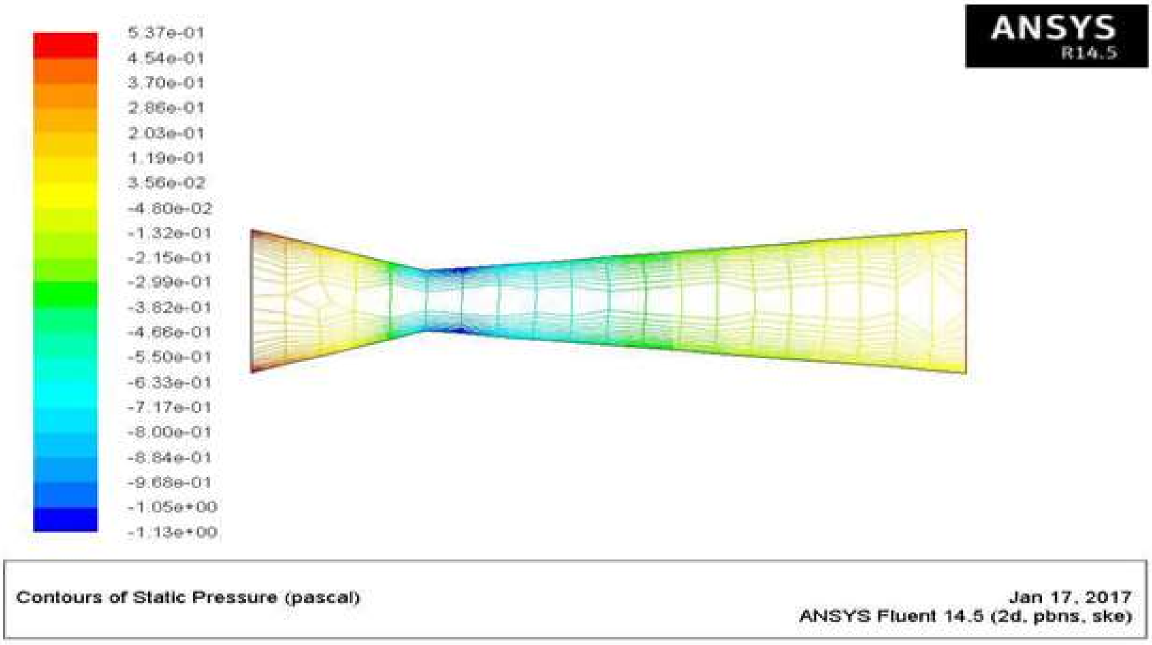

Fig.17. CFD analysis for pressure contours.

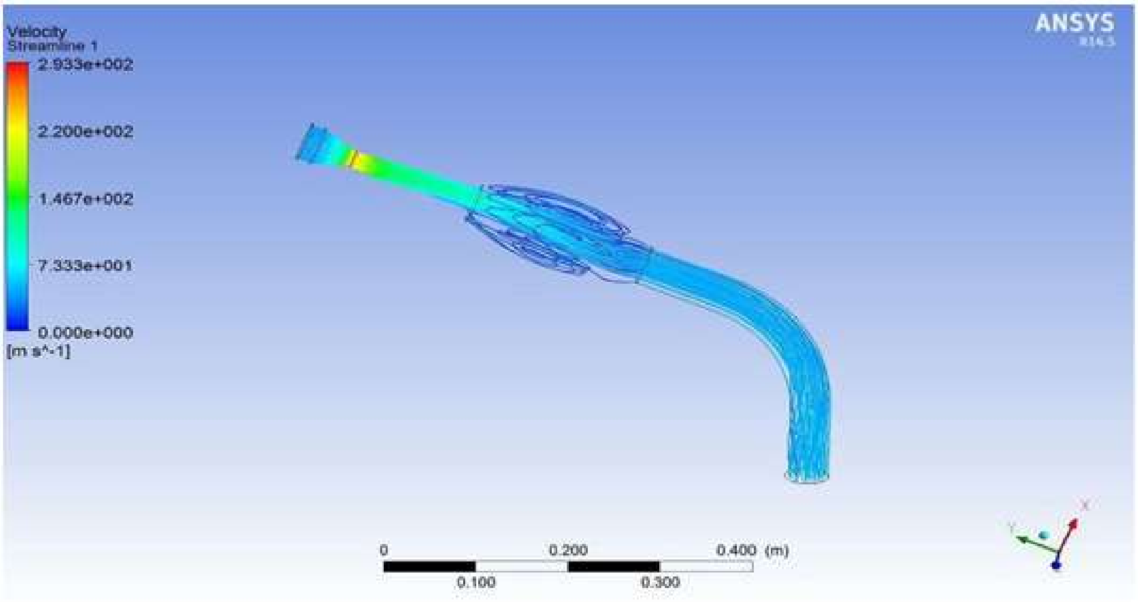

Fig.18. Velocity streamline diagram

Thus, the overall effect was to maximize the power offered by the restricted engine. The factors were evaluatedto optimize horsepower at 6000 to 7000 rpm.

1) Mass flow rate through restrictor is 0.0742 Kg/s.

2) Plenum volume - 746.4cc.

3) Runner length is 339.4252mm.

The optimal results can offset the effect of restrictor, and improve the restricted engine's ability effectively inthe high-speed range (5000 to 7000 rpm).

3.3 Suspension system:

3.3.1 Upright:

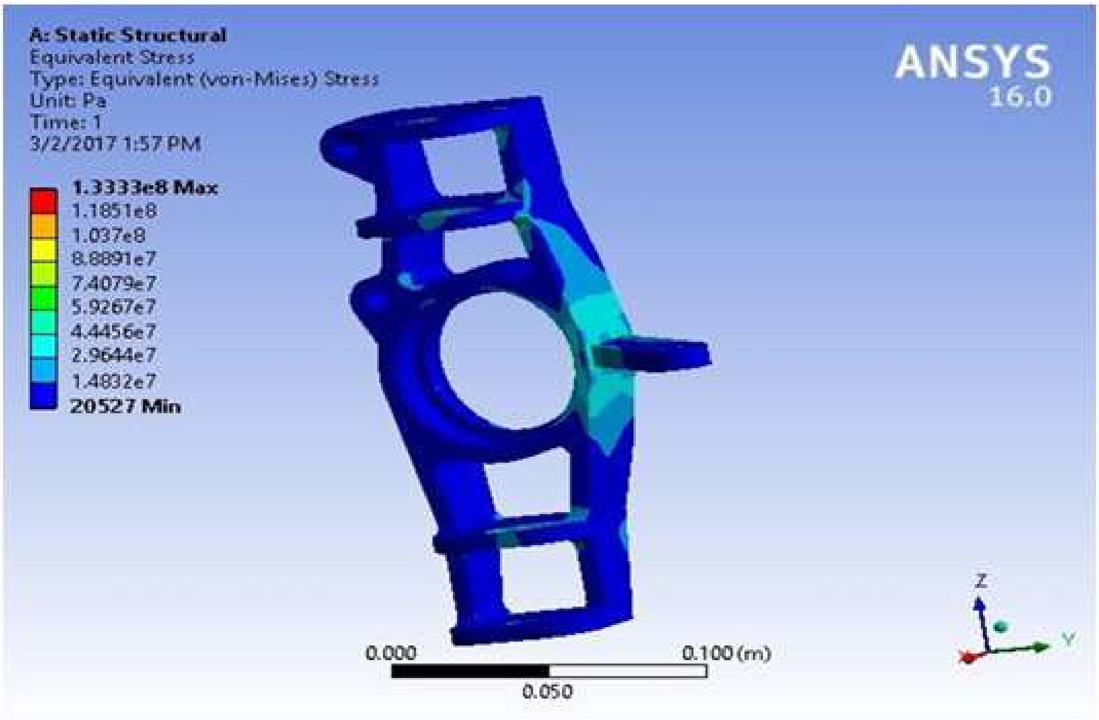

Fig.19. Upright stress analysis.

The Fig.19 shows the accumulation of the stresses in upright. There is no concentration of stresses at any single point and the maximum value doesn't cross the threshold value, design is safe.

3.3.2 Rocker:

There is no concentration of stresses at any single point and the maximum value doesn't cross the threshold value, design is safe.

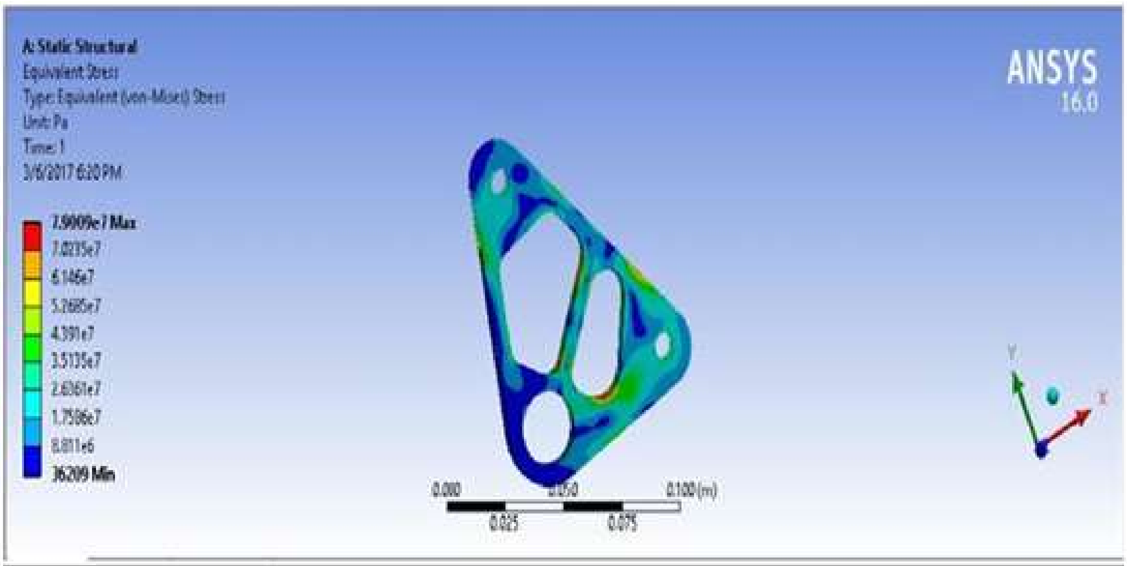

Fig.20. rocker stress analysis.

Different geometrical values of the suspension system are as given in the Table 5.

Table 5. Final values of suspension geometries

Parameters	Front	Rear
Wheelbase	1625.6mm	
Track width	1220.724mm	1172.6164mm
Camber	-0.734o	0.049o
Roll centre height	8.99mm	30.988mm
Scrub radius	33mm	38mm
Kingpin inclination	5o	0o

3.4 Brake system:

The objective of the braking system is to stop the car safely and efficiently and to lock all four wheels at same time and to obtain minimum stopping distance. Hydraulic disc brakes are used on all Four Wheels. Diagonal Split system is used for the safety of driver. In case of failure of one brake line system second brake line system will operate efficiently on other two diagonal tyres to maintain straight line stability of the vehicle. For driver ergonomics Floor mounted brake pedals, lower CG of pedal assembly is selected.

- The Force output of brake pedal assembly, Fbp= 2354.4 N
- The hydraulic pressure generated by The Master Cylinder = 2.066 x 106 N/M2.
- The hydraulic pressure transmitted to the caliper = 2.066 x 106 N/m2
- The one-sided linear mechanical force = 4535.06 N
- The clamp force generated by the caliper = 9070.11 N
- co-efficient of friction between brake pad and rotor = 3174.54 N
- The torque generated by rotor = 285.71 N.m
- The total braking force reacted between the vehicle and the ground= 6926.27 N
- Acceleration = 2.012 g.
- The stopping distance of the vehicle = 2.612 m.

3.5 Steering system:

Iterations were performed by varying length of tie-rod and steering arm and rack offset from axle in AutoCAD. Based on these iterations the following values were obtained:

- Wheel track (w) = 47.5”
- Wheel base(l) = 64”
- Rack length = 17”
- Rack travel = 4.21”
- Steering arm = 90 mm
- Tie rod= 345.66 mm
- Turning radius unto CG= 2.356 m
- Turning radius unto Centre of front axle=2.805m
- Initial Angle of Steering Arm = 30 degrees

Steer Angles:

- Inner Wheel (δ_i) = 29.36 degrees
- Outer Wheel (δ_o) = 44 degrees

4. Safety and ergonomics:

For Safety compliances driver accessible kill switches has mounted. One has designed cockpit ergonomically as per rulebook. The vehicle also consists of the stock OEM electronics of KTM DUKE 390 the basic requirements as mentioned in the rule book Master kill-switch, Cockpit kill-switch, brake over travel kill-switch and brake lights are incorporated as well.

5. Conclusion:

Considering all specified constraints in the design rule book, the vehicle which has been designed for racing capability will definitely give its best performance on the track. The basic advantage of this vehicle is that it is combination of optimum design, low cost. As the mechanisms which are going to be used in this vehicle are simple ones hence the production, use and maintenance of the vehicle is easy and quick.

- Design of the chassis of previous model has been studied and modification as per new rule book is done. Thus, improving the chassis also by reducing the weight.
- Designof an air intake system for the FORMULA BHARAT race car is done. The purpose of compensating the pressure losses because of the restrictor of 20 mm according to the SAE rulebook is achieved and ultimately the power loss of the engine is reduced.
- Uprights and rocker have been modified and enhanced for the better performance. Also, cost is reduced.
- For driver ergonomics Floor mounted brake pedals was selected, lower CG of pedal assembly.
- The goal of the steering modifications this time is to make the steering compact to achieve strict ergonomic targets set forth by the team and more importantly, to better the Anti-Ackerman geometry.

6. Acknowledgement:

We take this opportunity to express deep sense of gratitude to all automobile expert faculty members. We also wish to convey our sincere thanks for his consistent involvement, valuable guidance, scholarly encouragement, selflessness, time investment and thought-provoking discussions which inspired us to make this report fruitful. We are grateful to all faculty members of Dept. of Mechanical Engineering especially Prof. (Dr.)Nitin Phafat, Incharge IIRC and Prof. (Dr.) M.S. Kadam, Head of Mechanical Engineering Department for their motivation for this national level activity "Formula Bharat". In high esteem, we also express our word of thanks to honorable Principal Prof. (Dr.) H.H. Shinde and Management for providing us motivation for participation and extend the excellent infrastructure facilities at Incubation Centre of Dept. of Mechanical Engineering.

7. References:

[1] SAE 2016, Formula SAE Rules,www.fsaeonline.com/content/2016_FSAE_Rules.pdf accessed on February 2018.

[2] KTM 390 duke, 2018, Engine Owner's Manual, www.ktmshop.se/documents/18_3213782_en_OM.pdf accessed on February 2018.

[3] John Baechtel, 2012, Performance Automotive Engine Math:Fast Engine Math for Street and Race Applications (Sa Design-Pro), CarTech Inc. Publisher, Oct 2012ISBN: 1934709476.

[4] M.A. Ceviz , M. Akin, 2010, Design of a new SI engine intake manifold with variable length plenum, Department of Mechanical Engineering, Faculty of Engineering, University of Atatürk, Erzurum 25240, Turkey, 21 March 2010.

[5] Gaffney III E. F., Salinas A. R., 2016, Introduction to Formula SAE Suspension and spring design, University of Missouri–Rolla, 971584

[6] Gillespie T.D., 1992, Fundamentals of Vehicle Dynamics, Society of Automotive Engineers.

[7] Mehmood A., Khan A. A., and Khan A. A., 2014, Vibration Analysis of Damping Suspension Using Car Models, International Journal of Innovation and Scientific Research, 9(2): 202-211.

[8] Milliken, Douglas L., and William F. Milliken, 1995, Race Car Vehicle Dynamics. Warrendale:Society of Automotive Engineers (SAE).

[9] Pacejka, H.B., 2012, Tyre and Vehicle Dynamics, 3rded., ButterworthHeinemann,Oxford.

[10] V. Arvind, 2013, Optimizing the Turning Radius of a Vehicle Using Symmetric Four-Wheel Steering System, International Journal of Scientific and Engineering Research, 4(12): 2177-2184.