FORMULA BHARAT 2021

DESIGN REPORT

Team ID	**:**	**96**
Team Name	**:**	**Team V-LOCITY**
College Name	**:**	**Visvesvaraya National Institute Of Technology**
City	**:**	**Nagpur, Maharashtra**

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INTRODUCTION

This design report is submitted by Team V-locity from VNIT, Nagpur. Visvesvaraya National Institute of Technology (VNIT) is located in Nagpur and is one of the 31 NITs in the country. The environment in the college coupled with interest of students in automobiles led to the creation of Team V-locity in 2009. Since then the team has been regularly participating in SAE events. The goal of 2021 for Team V-locity, VNIT Nagpur is to design and fabricate a vehicle which embodies innovation and elegance, while ensuring safety. The team was divided into departments, each responsible for the design and the manufacturing of the subassemblies of the vehicle, and the management team for the sponsorship and managerial activities. This report diagnoses the attributes of the vehicle subassemblies, highlighting the considerations, functions and processes behind their design.

Design goals:

The primary goal that was kept in mind while designing was to build a car for the weekend racing event which would have a simple yet innovative structure with optimum space utilization and using simple yet standard manufacturing methods for the manufacturing of the car.

The team having previously participated in SAE Supra 2019, comprehensively scrutinized the previous designs and addressed the grey areas. The suspension system was redesigned and optimized to increase the stiffness and also improve the overall drivability of the car. We have tried to achieve a low center of gravity as well as centralized weight distribution with the placement of components. The components were so placed that be conducive to the driver for ease of access to all the components for future customization and repair. Thus, we were able to come up with a design that would be simple yet efficient and conducive to the weekend racing event.

CHASSIS:

a) Material used-

1. AISI 1018 steel tubes

Reasons-

1. Light weight 2. Cheap 3. Good strength 4. Excellent weldability b) Type of welding-

Arc welding

1. Dimensions of members-

MATERIAL	DIMENSIONS (mm)	DENSITY (gm/cc)	YIELD STRENGTH	ELASTIC MODULUS	MASSS PER UNIT LENGTH
AISI 1018	o.d.=25.4				
	i.d=20.4				1.4 kg/m
	i.d=21.4				1.2 kg/m
	i.d.=22.2	7.8	400MPa	210GPa	1.0 kg/m

Rules from rulebook-

1. T2.3 Wheelbase- The car must have a wheelbase of at least 1525 mm (60 inches). The wheelbase is measured from the center of ground contact of the front and rear tires with the wheels pointed straight ahead.

2. Main Hoop- T3.11.4 In the side view of the vehicle, the portion of the Main Roll Hoop that lies above it attachment point to the Major Structure of the Frame must be within ten degrees (10°) of the vertical.

3. T3.11.6 In the front view of the vehicle, the vertical members of the Main Hoop must be at least 380 mm (15 inch) apart (inside dimension) at the location where the Main Hoop is attached to the Major Structure of the Frame.

4. Front Hoop- T3.12.4 The top-most surface of the Front Hoop must be no lower than the top of the steering wheel in any angular position.

5. T3.12.5 The Front Hoop must be no more than 250 mms (9.8 inches) forward of the steering wheel. This distance shall be measured horizontally, on the vehicle centerline, from the rear surface of the Front Hoop to the forward most surface of the steering wheel rim with the steering in the straight-ahead position.

6. T3.12.6 In side view, no part of the Front Hoop can be inclined at more than twenty degrees (20°) from the vertical.

7. MAIN HOOP BRACING- T3.13.2 The Main Hoop must be supported by two braces extending in the forward or rearward direction on both the left and right sides of the Main Hoop.

8. T3.13.4 The Main Hoop braces must be attached as near as possible to the top of the Main

Hoop but not more than 160 mm (6.3 in) below the top-most surface of the Main Hoop. The included angle formed by the Main Hoop and the Main Hoop braces must be at least thirty degrees (30°).

9. FRONT HOOP BRACING- T3.14.2 The Front Hoop must be supported by two braces extending in the forward direction on both the left and right sides of the Front Hoop.

Side Impact Structural member must connect the Main Hoop and the Front Hoop. With a 77kg (170 pound) driver seated in the normal driving position all of the member must be at a height between 300 mm (11.8 inches) and 350 mm (13.8 inches) COCKPIT- 21. T4.1 Cockpit Opening 22. T4.1.1 In order to ensure that the opening giving access to the cockpit is of adequate size, a template shown in Figure 8 will be inserted into the cockpit opening. It will be held horizontally and inserted vertically until it has passed below the top bar of the Side Impact Structure (or until it is 350 mm (13.8 inches) above the ground for monocoque cars). No fore and aft translation of the template will be permitted during insertion. e) Static and Dynamic Analysis of Roll Cage-

10. T3.14.3 The Front Hoop braces must be constructed such that they protect the driver's legs and should extend to the structure in front of the driver's feet.

11. T3.14.4 The Front Hoop braces must be attached as near as possible to the top of the Front Hoop but not more than 50.8 mm (2 in) below the top-most surface of the Front Hoop.

12. OTHER SIDE TUBE RECQUIREMENT- T3.16 -Other Side Tube Requirements If there is a Roll Hoop brace or other frame tube alongside the driver, at the height of the neck of any of the team's drivers, a metal tube or piece of sheet metal must be firmly attached to the Frame to prevent the drivers' shoulders from passing under the roll hoop brace or frame tube, and his/her neck contacting this brace or tube.

13. Frontal Impact Structure- T3.18.1 The driver's feet and legs must be completely contained within the Major Structure of the Frame. While the driver's feet are touching the pedals, in side and front views no part of the driver's feet or legs can extend above or outside of the Major Structure of the Frame.

14. T3.18.2 Forward of the Front Bulkhead must be an energy-absorbing Impact Attenuator.

15. BULKHEAD- T3.19.2 Except as allowed by T3.19.3, The Front Bulkhead must be located forward of all non-crushable objects, e.g. batteries, master cylinders, hydraulic reservoirs.

16. T3.19.3 The Front Bulkhead must be located such that the soles of the driver's feet, when touching but not applying the pedals, are rearward of the bulkhead plane. (This plane is defined by the forward-most surface of the tubing.) Adjustable pedals must be in the forward most position.

BODYWORK

17. T3.24.1 Sharp edges on the forward-facing bodywork or other protruding components are prohibited.

18. T3.24.2 All forward facing edges on the bodywork that could impact people, e.g. the nose, must have forward facing radii of at least 38 mm (1.5 inches). This minimum radius must extend to at least forty-five degrees (45°) relative to the forward direction, along the top, sides and bottom of all affected edges.

19. T3.25.1 The Side Impact Structure for tube frame cars must be comprised of at least three (3) tubular members located on each side of the driver while seated in the normal driving position

20. T3.25.3 The locations Initial speed of the car = 60 Km/hr.=16.67 m/s

K.E. of the vehicle = .5* (300 Kgs) * sqr.(16.67 m/s)

Total crush energy = .5*(300Kgs)*sqr.(7 m/s) = 7350 J

Net energy after the attenuator has been crashed = 34333.33J

Velocity after crush **-** KE net = .5*m*sqr.(v)

Thus v= √ (2 KE net/m) = 15.129 m/s.

Assuming time taken for the car to come to rest after crush is .4 sec, deceleration = v/t = 4*g

{Standard value used while designing cars}

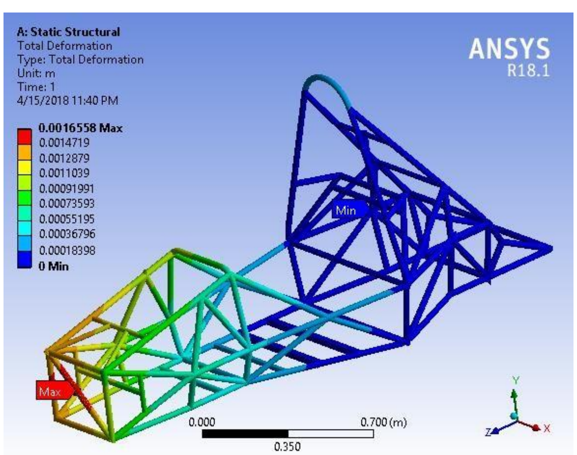
Assuming final mass of car = 300 kg, total force = 300*4*g = **11.77 kN**

For Optimum testing of design taking,

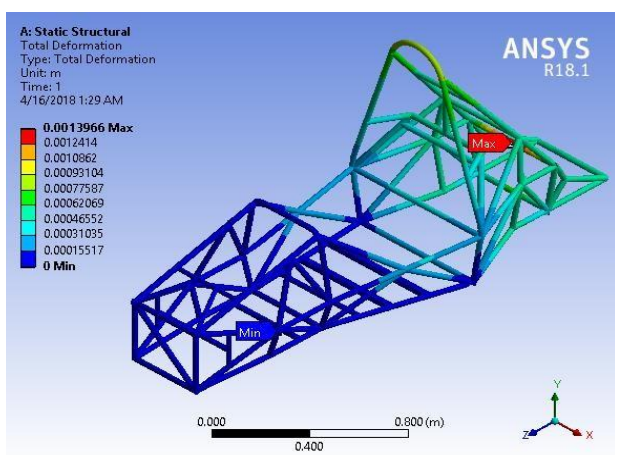
Factor of Safety = 1.7

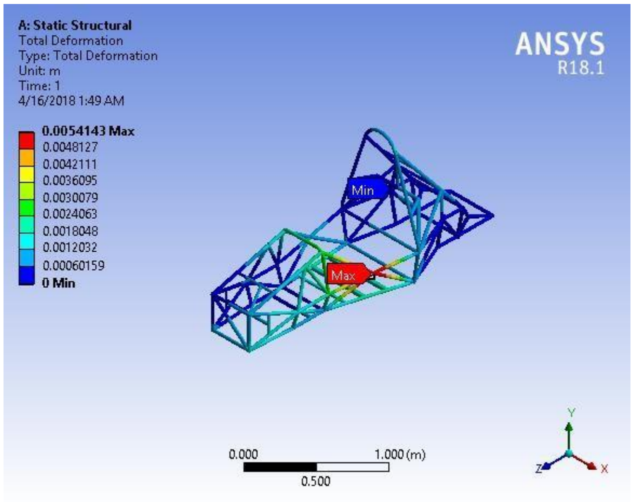
Force F = 20000 N

1.Front Impact :



2. Rear Impact:





3. Side Impact:

<u>SUSPENSION system:</u>

Type of Suspension - Pushrod Suspension

Pushrod suspension gives more stiffness to our vehicle than previous type of suspension (McPherson strut) used in our last vehicle.

The suspension was designed to reduce the sprung mass, give stable characteristics such as camber change, roll centre and to minimize bump steer. For this, the last year's suspension was analysed. We were able to save weight in knuckle, adapter plate and wishbone arms. The push-rod system is used, which allows us to change spring constant if necessary and also solves the problem of buckling which occurs in the case of simple spring system. The toes and camber of vehicles is kept to zero. Kingpin angle of 5 degrees is provided by structure of upright.

The team followed the following process steps:

a) Wheel selection
b) Tire selection
c) Wheelbase and track selection
d) Ground clearance optimization
e) Roll center optimization
f) Steering consideration
g) Braking considerations
h) All forces considerations
i) Packaging

Short long arm suspension geometry was used which helped us to minimize camber change. It also helped us in optimizing roll center height and keeping it almost at the same location with respect to car. Roll centre height was calculated considering, under-steer, non- toppling condition and reduction in scrub, using following formula. R*t = F*(roll center-center of mass) R= weight transfer due to roll

Track width rear= 1200mm

Track width front= 1300mm

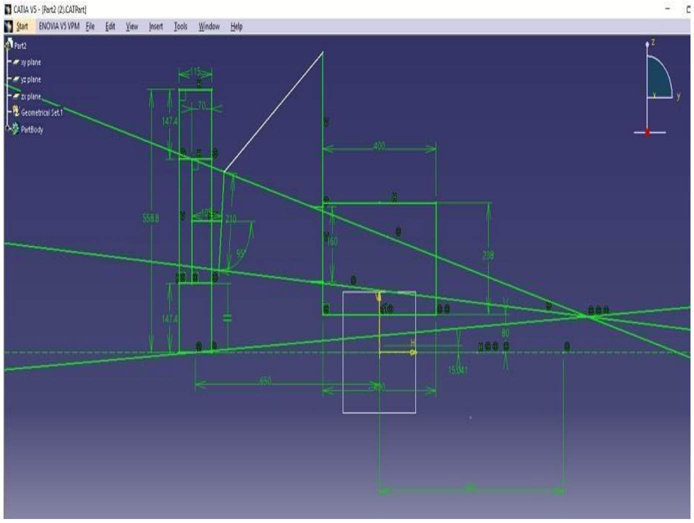
F_c = Centrifugal Force= 0.7mg

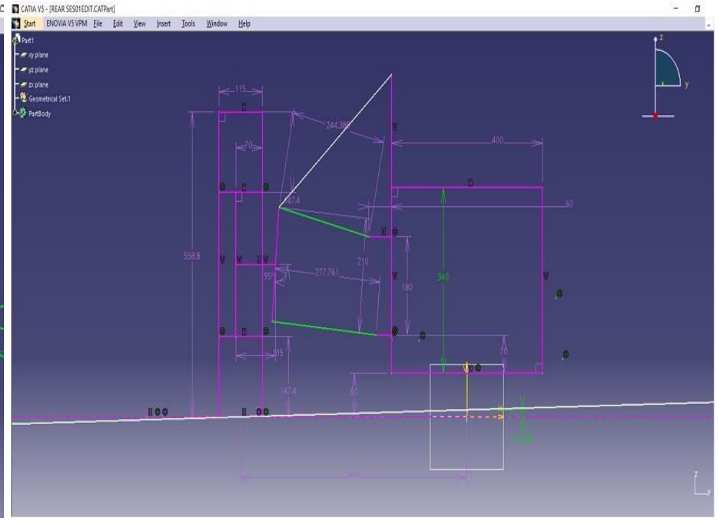
Also change in roll center height is kept small so that roll properties of the vehicle do not change drastically under rolling condition. According to these roll center calculations, four bar chains were sketched for front and rear suspension. Force analysis of the four-bar chain was carried out by applying maximum reaction force on the knuckle, F= 1428N for front & F= 1668N for rear.

Using this force analysis, forces in the pull-rods were calculated. Using this pull-rod force and spring stiffness, k=45000 N/m (Bajaj avenger), bell cranks were designed (Arm ratio: 6:8 (for front suspension) and 6.5:10 (for rear suspension)) . Knuckle was designed with mountings for wishbones, tie rod, and brake caliper & brake adaptor plate. We also took care of packaging because it is a part which contributes to most of air drag by suspension components and weight. It was then analyzed with following forces-

1. Reaction force due to weight of car: Front= 941.4N ,   Rear= 1412.6 N
2. Braking torque= 100 N.m
3. Force due to acceleration= 507.1N

Wishbones were designed to keep the stresses under maximum stress value= 157.4 MPa, considering factor of safety, welding and stress reversal factor.





Front Suspension Calculations

X= Force through upper wishbone

Y= Force through pushrod

Z= Force through lower wishbone

F_C = Centrifugal Force = 1318.46 N

R = Reaction Force by ground = 941.4 N

Bell Crank Ratio - Pushrod : Spring = 6:8

Applying Force and Moment Balance,

$X*\sin(13.4) + Y*\sin(40) + Z*\sin(4.32) = 941.4$ N

$-X*\cos(13.4) + Y*\cos(40) - Z*\cos(4.32) = 1318.46$ N

$X*\cos(13.4)*(0.411) + Z*\cos(4.32)*(0.147) - Y*\cos(40)*(0.384) + Y*\sin(40)*(0.044) = 0$

Solving these equations, we obtain

X = 500.2 N

Y = 1366.68 N

Z = -759.89 N

Therefore, Force transmitted through spring = 1366.68 * (6/8) N= 1025.01 N

Deflection required in the spring for that force = 5 cm

Therefore Spring Constant (k) = 1025.01/0.05= 20502 N/m

We have used a spring having spring constant 45000N/m

So, Factor of Safety = 45000/ 20502= 2.19

Rear Suspension Calculations

X= Force through upper wishbone

Y= Force through pushrod

Z= Force through lower wishbone

F_C = Centrifugal Force = 1977.69 N

R = Reaction Force by ground = 1412.64 N

Bell Crank Ratio - Pushrod : Spring = 6.5:10

Applying Force and Moment Balance,

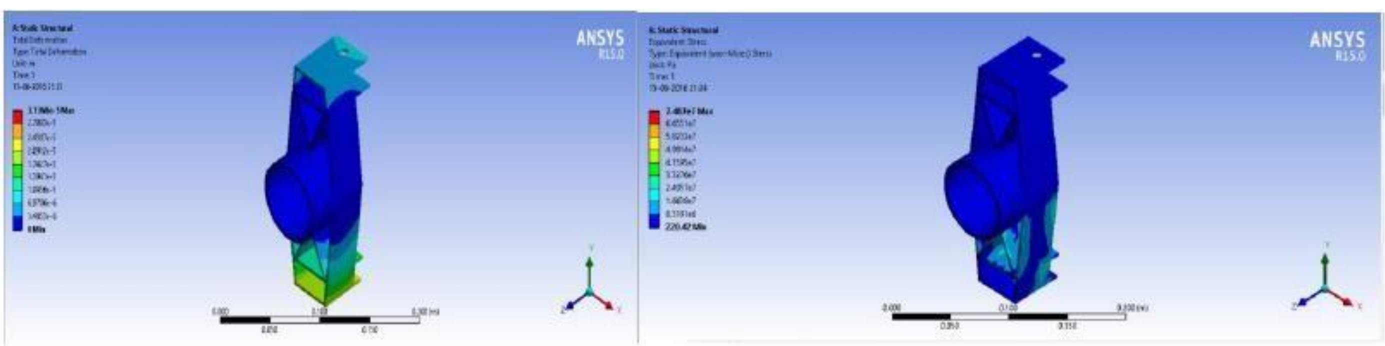
X*sin(12.765) + Y*sin(47.6) + Z*sin(5.122) = -1977.69 N
X*cos(12.765) + Y*cos(47.6) - Z*cos(5.122) = 1412.64 N
X*cos(12.765)*(0.384) + X*sin(12.765)*(0.044) + Z*cos(5.12)*(0.147) + Z*sin(5.122)*(0.025) - Y*cos(47.6)*(0.384) + Y*sin(47.6)*(0.044) = 0
Solving these equations, we obtain
X = 2100.53 N, Y = 1639.03 N , Z = -2932.81 N
Therefore, Force transmitted through spring = 1639.03 * (6.5/10) N = 1065.369 N
Deflection required in the spring for that force = 5 cm
Therefore, Spring Constant (k) = 1065.369/0.05 = 21307 N/m
We have used a spring having spring constant 45000N/m
So, Factor of Safety = 45000/ 21307 = 2.11

Wheel Assembly:

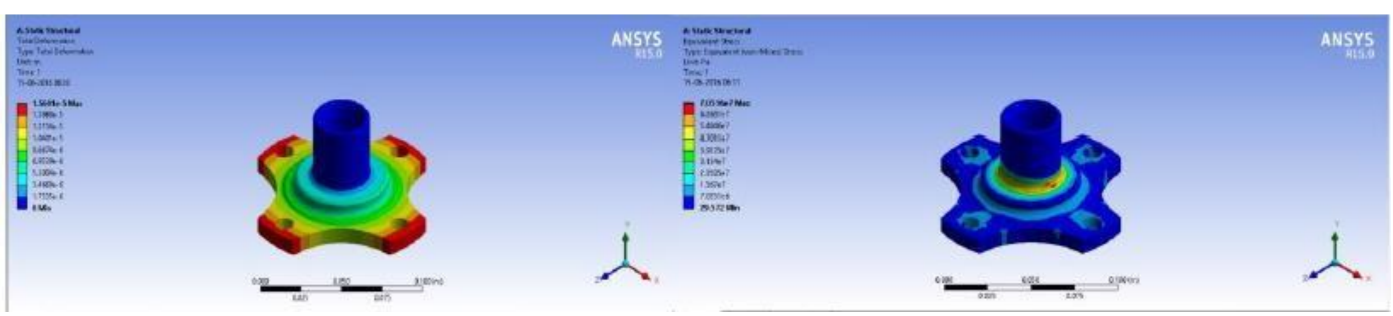
WHEELS AND TYRES

Tire selection is always a critical issue in race car. This year we are keeping same tire and alloy wheels used in former car. Reasons are, 145 width tires are sufficiently wider to give traction for a car having 37 bhp engine. For such engine cars large width tires reduce the fuel economy of the car along with increase in frictional and rolling resistance. Large tires and alloy wheels increases the car weight largely due to the reinforcement in rims. Also, considering rain chances on race track, we are keeping treaded tires instead

of slick tires. Further low width and treaded tires are ideal for front wheels for better steering maneuverability. Treads reduce the tire slip angle and hence Ackermann geometry closely holds the real steering geometry.



Upright Analysis



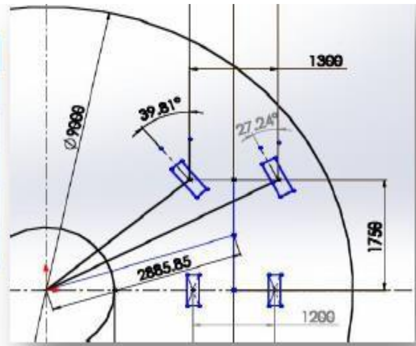
Hub Analysis

STEERING system:

Rack and Pinion steering system has been chosen over other steering systems for its compactness and ease of adjustment in Ackermann. The minimum width of the autocross track has been specified to be 3.5m and the outside diameter of the sharpest turn is 9m. Using these values, the calculations on SOLIDWORKS gave the angles of the front wheels to produce no scrub to be 39.81° for inner wheel and 27.24° for outer wheel. (fig)

Specifications	
Wheelbase	1750mm
Front Track Width	1300mm
Rear Track Width	1200mm
Average Steering Angle	33.52°
Ackermann %	135%
Steering Arm length	70mm
Turning Radius	2.88m
Centre to Lock	402.3°

Rack Specifications	
Rack Length	355.6mm
Rack Travel	104.77mm
Height of Rack	200mm (from ground)
C-factor	0.145
Steering Ratio	12:01



Iteration (Ackermann% & Steering Arm Length)	Inner Wheel Lock Angles	Outer Wheel Lock Angles	Difference from ideal inner Angle	Difference from ideal outer angle	Average Steer Angle
Ack-125% SAL 70mm	39.81°	27.95°	0°	-0.71°	33.88°
Ack-125% SAL 80mm	39.4°	27.7°	0.41°	-0.46°	33.55°
Ack-130% SAL 70mm	39.5°	27.46°	0.31°	-0.22°	33.48°
Ack-130% SAL 80mm	39.5°	27.38°	0.31°	-0.14°	33.44°
Ack-135% SAL 70mm	39.81°	27.22°	0°	0.02°	33.52°
Ack-135% SAL 80mm	39.81°	27.13°	0°	0.11°	33.47°
Ack-135% SAL 80mm	39.9°	27.16°	-0.09°	0.08°	33.53°

➢ From the above iterations, simulated on CATIA, at 135% Ackermann and for steering arm length of 70mm we get the best possible steering lock angles for both the wheels to keep scrub at minimum.

➢ The turning radius of the centre of the car comes out to be 2.88m which is within the limits presented by the earlier model.

➢ The steering arm is mounted 77.4mm from the bottom of the knuckle to reduce bump steer and the analysis is done on CATIA

BRAKES:

Brakes Calculations-
Assumptions made

1) Car weight = 400kg
2) Wheel base = 1700mm
3) Force distribution (front to rear) = 6:4
4) Stopping distance = 5m
5) Speed of car = 40km/hr
6) Coefficient of friction between tyre and road = 0.8
7) Tyre radius = 0.25m
8) Disc radius = 10cm
9) Coefficient of friction between pads and disc = 0.4
10) Calipers piston diameter = 1 inch
11) Master cylinder piston diameter = 18mm
12) Pedal ratio = 39:11
13) Front static load = 1600N
14) Rear static load = 2400N
15) Height of CG = 0.3m

Procedure of calculation

W=weight, m=mass, b=distance of C.G. from front tyre, c=distance of C.G. from rear tyre, L=distance between tyres.

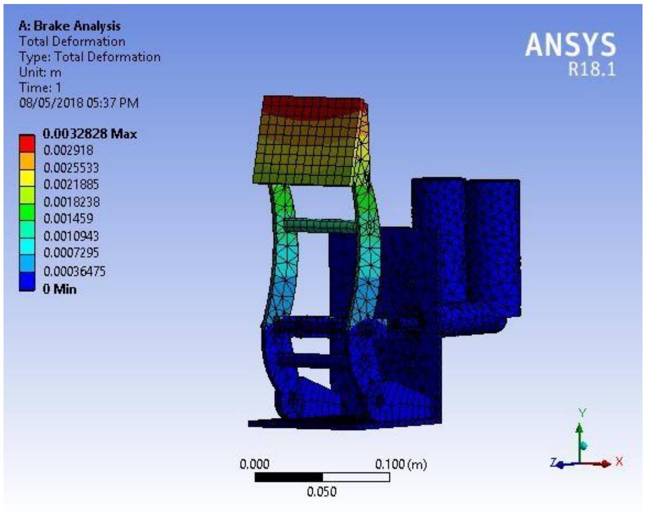
For stopping distance 5m and for initial velocity 40km/hr

v2=u2+2ds

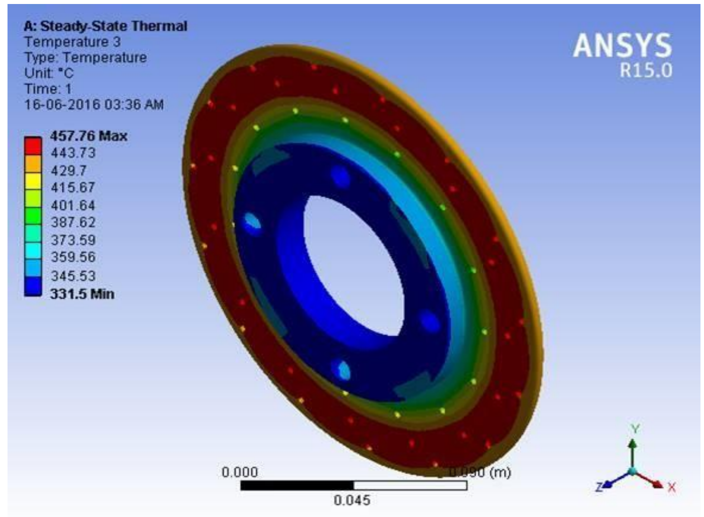
Where, v=final velocity, u= initial velocity, d=deceleration, s=
stopping distance d=u2/2s
d=11.122/2*5 =12.36 m/s2
deceleration ratio = 1.26g
1) Dynamic loads = W=m((c/L)+(Ax/g)(h/L)
Front dynamic weight =1600+ [(1.26*0.3*400*9.81)/1.7]= 2472.5 N
Rear dynamic weight =2400-(1.261*0.3*400*9.81)/1.7)=1527.5N
2) F=Wf*g*μ
Total front dynamic force = 2472.5*0.8 = 1978N
Total rear dynamic force = 1527.5*0.8 = 1222N
3) Front braking force on single tyre = Ff= 1978/2 = 989N
Rear braking force on single tyre = Fr= 1222/2 = 611N
4) T= F*r
Braking torque for single front tyre =989*0.25= 247.25Nm
Braking torque for single rear tyre = 611*0.25 =152.75Nm
5) Frictional force of single disc of front wheel=Fdf= 247.25/0.10=2472.5N
Frictional force of single disc of rear wheel = Fdr= 152.75/0.10=1527.5N
6) Clamping force on single disc of front wheel = Fclamp=2472.5/0.4=6181.25N
Clamping force on single disc of rear wheel=1527.5/0.4 = 3818.75N
7) Force given by M.C on one side caliper for front wheel =
Fcalfront=6181.25/2= 3090.625N
Force given by M.C on one side caliper for rear wheel = Fcalrear=3818.75/2= 1909.375N
8) Pcal=(Fcal/Acal)
Pressure for the front braking circuit=3090.625/0.5*10-3= 6.18MPa
Pressure for the rear braking circuit =1909.375/0.5*10-3=3.81MPa
9) Pcal=Pmc
Force of circuit = (Pmc*3.14*d2)/4
Force required for front braking circuit=6.18×106*3.14*0.0092/4 =393.15N
Force required for the rear braking circuit=3.81×106*3.14*0.0092/4 =242.4N
Total braking force for the Braking =393.15+242.4=635.55N
As the paddle ratio is 4:1 so,
10) Paddle effort taken by driver =635.55/4 =160N
11) Stopping time: We have v2=u2+2ds & v=u+dt
Rearranging the terms we get t=2s/u= (2*5)/11.12=0.899=0.9sec

Braking Mechanism:

1) The braking system has two separate hydraulic circuits arranged in H-type arrangement to ensure braking in case any one of the circuit fails.
2) The brake rotors are drilled in order to make sure that the water or any debris from brake pads is removed. Thus, leaving the braking efficiency unaltered.
3) The Pedal ratio is taken as 39:11. This pedal ratio actuates sufficient force on the brake disc in order to bring it at rest. And thus proves useful in severe braking condition.



1. Brake pedal analysis



2. Temperature Distribution in brake disc

ENGINE SELECTION:

The engine selected for the car is of bike KTM Duke 390. Specifications: Max Torque= 35Nm at 7250rpm, 373.2cc, Single Cylinder 4 stroke liquid cooled engine (Max power 43HP at 9000rpm). Design Considerations: A reliable, safe and efficient drivetrain is desired. A good acceleration is desired with max velocity to be around 105 kmph. Engine is being used as a semi stressed member so that the vibrations are dissipated into the chassis to increase roll cage stiffness.

ELECTRONIC SYSTEM:

Before installing the engine into the car, all redundant ECU units were removed from it. Following are the important Electrical Units which are used in the car:

1. Wire harness/connectors
2. Dashboard panel
3. Kill switches
4. Fuses
5. Brake light bulb
6. Battery
7. Starter button
8. Brake Fuel Indicator

INTAKE AND EXHAUST:

▪ INTAKE SYSTEM

A venturi is designed such that minimum c/s diameter of the air passage is 20 mm. Appropriate flow analysis is done to find out the diverging and the converging angles of the venturi which are 6 and 12 degrees respectively. Intake manifold was added to optimize the efficiency and performance of the engine.

▪ EXHAUST SYSTEM

Exhaust system was designed according to rulebook. It was designed with objective of reducing noise level while keeping back pressure low. It was ensured that any component of exhaust did not touch any other components of car.

❖ ENGINE SUBASSEMBLY

▪ FUEL SYSTEM

Fuel tank is designed according to the space available and simplicity in manufacturing. Minimum possible volume is selected taking it to be 4 liters. Placement of Fuel tank is done considering the lowering of CG. Considering the rulebook, proper safety arrangements including belly pan, catch tank, fuel level indicator, filler neck height are used.

▪ COOLING SYSTEM

For engine cooling purpose single radiator with a larger surface area and the proper piping is used to increase the cooling capacity in order to cater to the demands of an increased load on the engine. The radiator is mounted sideways inside the ducts of the body so as to get maximum expose to the airflow. Also one fan is provided at the back of radiator to insure proper cooling.

DRIVETRAIN:

TYPE OF TRANSMISSION USED AND ITS REASON

Chain Drive is used for transmission, since it is readily feasible for the engine selected. Following calculations were done to find out sprocket dimensions and chain length:

1. **Sprocket dimensions** (to find secondary ratio):

Let N be the secondary ratio.

Equating> (initial torque on wheels considering static friction) = (initial torque on engine)

μmgR = T* (prim ratio)*(1st gear ratio)*N*(efficiency) ....(a)

Values taken: μ=1 (dry road), m=320kg (car+driver), Rwheel=13in=0.325m, Prim ratio=8/3, 1st gear ratio=8/3, Efficiency=0.8 (assumption).

T is calculated from foll:-

T=P1+P2*(2πN/60)+P3*(2πN/60)2

where, P1=(Pmax)/ K=35.18, P2=(Pmax)/ K2=0.035 , P3=(Pmax)/K3 =3.55*10-5 and K=(2 πNp)/60

Here Pmax=35kW, Np=9500 rpm, Putting above values, T=44Nm Equation (a) becomes =>

1019.2=250*N N=4 Sprocket diam=3.5*4=14mm Pitch comes out to be p=15.87mm

2. Chain Length

Assuming sprocket to sprocket dist=25cm=250mm,

No of links= Ln=2*(250/15.87)+(15+60)/2+(60-15)/2π+(15.87/250)=76.23≈76

Chain length= Ln*p =76*15.87= 1209mm ≈120cm

Chain sprocket and chain are selected that of KTM Duke 390 since the dimensions of them match with the required dimensions as calculated.

▪ Gear Transmission: 6 speed gearbox of KTM Duke 390

▪ Differential: Open differential gearset is used.

▪ Clutch System: Hydraulically actuated clutch is used.

▪ Axle: Consists of C V Joints Rzeppa.- 2 fixed ,2 plunging

Ergonomics:

• Cockpit layout has been constructed considering the driver comfort.

• Prototype roll cage was built to understand drivers comfort position, seat arrangement and steering wheel alignment. Driver position have been made as low as possible to lower the CG of the vehicle.

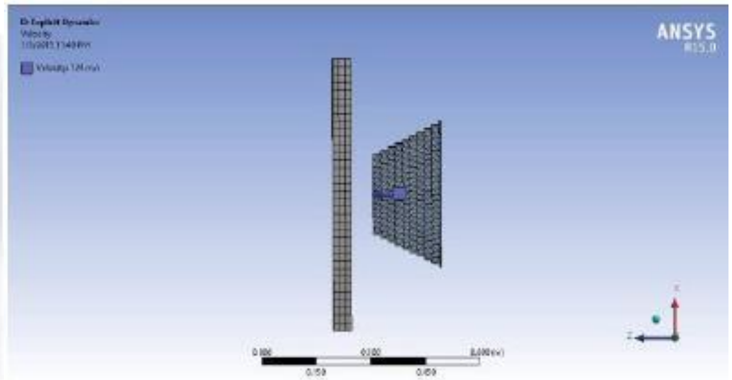
Safety:

IMPACT ATTENUATOR DATA - Design Selection

design	cost	weight	Reliability	safety	feasibility
Weighing factor	0.35	0.25	0.2	0.3	0.3
foam	6	8	7	8	8
honeycomb	8	8	9	8	8

From the above matrix clearly, Honeycomb is the better choice for us. Also, its light weight and low price make the material appealing in that it has the ability to be created relatively cheap and also easily replaced.

1. Honeycomb Pyramid Solid Model



2. Ansys test setup

Calculations:

Vehicle of mass 300kg is impacted with velocity of 7m/s.

So, its K.E. will be (M+m) $*v^2/2$,

where m is the mass of Impact Attenuator and M is the mass of vehicle,

K.E. = $0.5* (300+0.954) * (7)^2 = 7373.373$J

Impact attenuator is expected to absorb minimum this amount of energy.

Finite Element Analysis

Analysis was performed in the Ansys Explicit Dynamics. Initial testing on the models often failed or was unable to finish due to the complexity of the honeycomb pyramid structure. Our Team created a replacement part for the honeycomb that could be used in the finite element testing. The component had the mechanical properties of the honeycomb yet was modelled as a solid block. This solid block was used for testing in Ansys.

Mechanical properties of Honeycomb replacement solid block:

Test Setup in Ansys: The Impact Attenuator was allowed to impact on the rigid wall at velocity of 124m/s. v= (K.E. x 2/ m)^0.5 ,m=mass of impact attenuator The impact scenario is such that the vehicle collides with the wall with IA in the front and Impact attenuator is expected to absorb all the K.E. But the connectors are not possible in Ansys explicit dynamics. So the Impact Attenuator is given the whole K.E. which gives the above calculated velocity for test.

Property Value Units:

Elastic Modulus 71,000 N/mm^2 **Poisson's Ratio .3**

Shear Modulus 440 N/mm^2 **Density 8.32 x 10-5 g/mm^3**

Tensile Strength 290 N/mm^2 **Compressive Strength 4.6 N/mm^2**

Yield Strength 268 N/mm^2

Test Results: Impact Attenuator was found to absorb 7458J of energy which complies with the rules