DESIGN OF A COMPACT GO KART VEHICLE

R. Santosh Kumar[1], R. Rohith[2], S. Srinivas[3]

[1,2,3]*Department of mechanical engineering, Panimalar Engineering college, Affiliated by Anna university, Chennai-600123, Tamil nadu, India.*

Abstract - *The design of the go kart is the amalgamation of the design for chassis, driver ergonomics, cost effectiveness and safety while avoiding the possibility of over design. Design software's along with manual calculations were utilized. The goal of our project is to design and fabricate a fully functional Go-Kart vehicle which is suitable for competitive racing. To achieve this, a bottom up approach of designing was utilized. Design process began with the chassis (foundation) and then was built up to the exterior parts.*

Key Words: Go kart, Design, Calculations.

1. INTRODUCTION

We approached our design by considering all the possible alternatives for the system and modeling them in CAD software like CATIA, SOLIDWORKS, CREO ect.., and subjected to analysis using ANSYS software for finite elemental analysis of the complete go kart vehicle. Based on the analyses result, the model was modified and retested and a final design was choosed. Our main objective is to fabricate a Kart which is comfort for the driver, to achieve this design and analysis was done for assumed height and weight of the driver. The position of various components was optimized based on the driver's comfort. The manufacturing approach we used was cost effective, ergonomics and ease of fabrication.

2. TECHNICAL SPECIFICATIONS OF THE VEHICLE:

2.1 Overall dimensions

WHEEL BASE	42.5"
REAR TRACK WIDTH	42"
TOTAL WEIGHT	125 Kg
GROUND CLEARANCE	2.5"

2.1.1 Chassis material

- Material used: AISI 1018
- Ultimate strength: 440 MPa
- Yield strength :370 MPa

2.1.2 Chemical composition

ELEMENT	%COMPOSITION
Carbon	0.14 - 0.20%
Phosphorous and Sulphur	< 0.050%
Manganese	0.6 - 0.9%

2.1.3 Engine

The engine used is Honda stunner, 125 cc

Engine technology	Air cooled, 4 stroke, SI engine
Length*width*Height(approx.)	300 mm* 200mm*250 mm
Bore x stroke	52.4 x 57.838 mm
Displacement (cc)	124.7
Net power output	11BPH@ 8000 rpm
Net torque	11 Nm @ 6500 rpm
Overall Reduction Ratios	1st gear: 31.645 2nd gear: 21.604 3rd gear: 14.577 4tthgear: 12.245 5th gear:10.62
Fuel tank capacity	10 liters
Oil capacity	0.5 L

2.1.5 Wheel Specification:

Tyre Size	Position	Section width(mm)	Overall Diameter(mm)
4.5 x 10-5	Front	114.3	254
7.1 x 11-5	Rear	180.34	279.5

2.2 BRAKING SYSTEM

Braking system of our vehicle was designed to achieve maximum braking efficiency and to provide greater vehicle safety

2.2.1 Brake pedal

Passenger cars generally use a pedal ratio between 4 to 6. A pedal ratio of 5:1 was selected.

2.2.2 Master cylinder

Analyzing the available ones, we chose the master cylinder of TVS APACHE RTR 160. The piston diameter of this MC is 19.05 mm.

2.2.3 Caliper and rotor selection

The caliper and the rotor are selected from KTM DUKE 390 with diameter of caliper being 32 mm and effective radius of rotor being 100 mm.

2.2.4 Brake system specifications

Brake type: Single disc brake

Brake fluid :**DOT 4**

Brake disc: Diameter **- 195 mm**

Thickness**- 5 mm**

Brake pad lining thickness: **4.5 mm**

Master cylinder diameter: **19.05 mm**

Caliper inside cylinder: **32 mm**

Assumptions:

Pedal ratio = 5: 1

Max force applied by the driver on the brake pedal= 400 N

Area of the Master cylinder piston (A_{MC}) = π x $(D_{MC})^2$ ÷ 4 = π x $(0.01905)^2$ ÷ 4 =2.85*10^-4 m^2

+ Area of the piston in the calipers (A_{CAL}) = π x $(D_{CAL})^2$ ÷ 4= π x $(0.032)^2$ ÷ 4= 8.0384*10^-4 m^2

Effective rotor radius = 100 mm

Co-efficient of friction between the brake pad and disc,

μ_B = 0.3 (wet)

= 0.7 (dry)

Brake fluid = DOT-4

Disc diameter = 195 mm

Disc material = Cast Iron

Losses due to compression were neglected.

2.2.5 Brake analysis

Force on the master cylinder (F_{MC}) = max force applied x pedal ratio = 400 x 5 = 2,000 N

Pressure developed in the system on applying the brakes, p = F_{MC} ÷ A_{MC} = 2,000 ÷ 2.85*10^-4 = 7.0177*10^6 N/m^2

The above pressure is same throughout the system.

Force on the calipers (F_{CAL}) = p x A_{CAL} = 7.0177*10^6 x 8.0384*10^-4 = 5,641.842 N

Brake Type	Single disc brake
Recommended fluid	Dot 4
Brake Disc	Diameter-195
Brake pad thickness	4.5mm
Master cylinder diameter	19.05mm
Caliper inside cylinder	32mm

Force on the rotor / disc (clamping force) = F_{CAL} x 2 = 5,643.842 x 2 = 11,287.637 N

Total Frictional force = clamping force x μ_B = 11,287.637 x 0.3 = 3,386.306 N(wet)

Total Frictional force = clamping force x μ_B = 11,287.637 x 0.7 = 7,901.341 N(dry)

2.2.6 Assuming wet surface

Torque on the rotor = Frictional force x Effective rotor radius = 3,386.306 x 0.1 = 338.6306 1N-m

Deceleration produced = Frictional force ÷ mass of the kart = 3,386.306 ÷ 190 = 17.8226 m/s^2

Stopping distance (from max speed to zero) = $(\text{Velocity})^2$ ÷ (2 x deceleration) = $(23.92)^2$ ÷ (2 x 21.1644) = 1**3.51 m**

2.2.7 Assuming dry surface

Torque on the rotor = Frictional force x Effective rotor radius = 7,901.341 x 0.1 = 790.13411N-m

Deceleration produced = frictional force ÷ mass of the kart = 7,901.341 ÷ 190 = 41.38 m/s^2

Stopping distance (from max speed to zero) = $(\text{Velocity})^2$ ÷ (2 x deceleration) = $(12.76)^2$ ÷ (2 x 49.38) = **6.88 m**

2.2.8 Thermal analysis

Kinetic energy of the vehicle at max speed = mV^2 ÷ 2 = (190 x 23.92 x 23.92) ÷ 2 = 54355.80J

Brake time = Stopping distance ÷ speed = 13.51÷ 23.92 = 0.564 s

Brake power = Kinetic energy ÷ Brake time = 54355.80 ÷ 0.564 = 96375.53 W

Rubbing area on one side of the disc=(π÷4) x [(pad outer diameter) ^{2} – (Pad inner diameter) ^{2}]

= (π÷4) x [(0.195)^2 – (0.152) ^{2}]

[Width of the pad = 43 mm]

= 0.01172 m^2

Total rubbing area = 0.01172 x 2

= 0.02344 m^2

Heat flux (q) = brake power ÷ Total rubbing area

= 96375.53 ÷ 0.02344

= 4111584.12 W/ m^2

Rise in temperature of the disc while braking (T) = $\frac{0.527 \times q \times \sqrt{\text{Brake time}}}{\sqrt{[\text{Density} \times \text{sp.heat} \times \text{thermal conductivity}]}}$

= (0.527 x 4111584.126 x √0.306) ÷ √ (7250 x 500 x 58) ....[Cast Iron disc]

T = 112.22'C

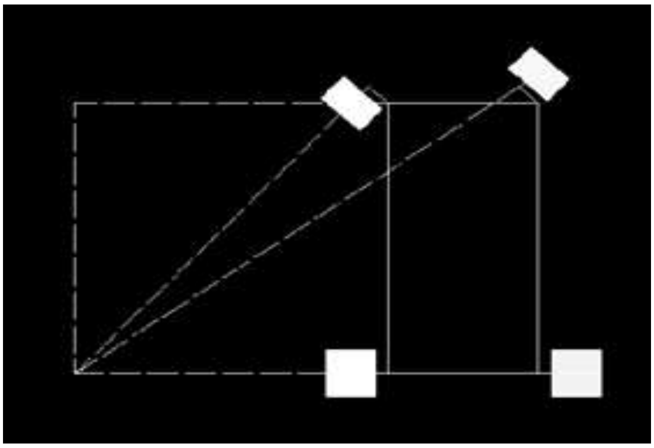
Max temperature produced in the disc during hard braking = 36.51 + 30 (ambient temperature) = **142.2'c**

3.1 STEERING SYSTEM

The steering used here is based on Ackermann's principle in order to avoid the need for tyres to slip sideways, when following the path around a curve.

3.1.1 Steering calculations

When a vehicle is cornering, each wheel describes a turning circle. The outer turning circle, or its radius, is the main subject of interest. The calculation is not precise because when a vehicle is cornering the perpendiculars through the centers of all wheels do not intersect at the curve centre point (Ackermann condition). In addition, while the vehicle is moving dynamic forces will arise that will affect the cornering maneuver. The values stated below were found out by using the graphical method.



NOTATIONS	VALUES
STEERING GEOMETRY	Ackermann
WHEEL BASE	1040mm
FRONT TRACK WIDTH	680mm
REAR TRACK WIDTH	1050mm
DISTANCE FROM FRAME TO WHEEL CENTER (FRONT WHEEL)	340mm
DISTANCE FROM FRAME TO WHEEL CENTER (REAR WHEEL)	320mm
INNER STEER ANGLE	27degree
OUTER STEER ANGLE	19.32degree
TURN RADIUS	2440.155mm
CASTER	1 degree
TOE IN AND TOE OUT	2 degrees
KING PIN INCLINATION	N/A
TIE ROD	Equal length of 340mm
STEERING RATIO	1:1
STEERING ARM LENGTH	460mm
STEERING ARM ANGLE	74.22 degree

PERCENTAGE OF ACKERMAN STEERING ACHIEVED:
Let us assume the inner turning angle to be 27 degrees, (A)

Now, by formula

Tan A = L/(R-D/2)

Where, L- Wheel base

D- Track width

R- Turning radius

Therefore,

Tan27= 1070/(R-680/2)

R=2440.115mm

Similarly

Tan B= L/(R+D/2)

B=18.92 degree

Steering arm angle = 4*A*tan(d/2l)

SAA=73.2degree

Now,

% Ackerman = 6*(A-B)/atan(1/(tanB-1)-B

=87.77%

Therefore, percentage of Ackerman achieved= 87.77percentage

3.1.2 Target performance of the kart

- STUNNER 125 engine was chosen as it offers the max power (11 BHP @8000RPM) under the 125CC segment.
- Direct drive was implemented (no jackshaft) so that the transmission efficiency ameliorates.
- When a choice was to be made between the two types of chains (R 1278) for the drive, #R 1278 was chosen as the load transfer capability of this chain was higher and also the minimum breaking load is 18200 N. This gave us a high FOS.
- A small experiment was done to determine the max force a driver can apply on the brake pedal. The seating position was simulated and in the place of the brake pedal, a weight machine was kept. The subject was able to apply a maximum force of 40 Kgf or 400 N.
- The target stopping distance during hard braking from max speed (135.6 Kmph) to zero was 6 to 13 m. To achieve this, a pedal ratio of 5:1 was considered. The calculated stopping distance was 6.8 m under dry conditions, which is assumed to be the track conditions.
- The effective rotor radius of the disc used is 0.1 m. This offered higher braking torque on the disc and thus higher braking force on the tires.
- A live rear axle provides better traction when compared to a dead rear axle. Hence a live rear axle was used.

4.1 TRANSMISSION SYSTEM

Engine: Honda STUNNER125 cc

Transmission:

Primary reduction = 3.35

Final reduction = 1.41

Secondary reduction:

First gear	3.076: 1
Second gear	1.944: 1
Third gear	1.473: 1
Fourth gear	1.19: 1
Fifth gear	1.038:1

Rear Axle Type: live axle

Wheels: (in inches)

Front = 4.5 x 10-5

Rear = 7.1 x 11-5

Diameter of the rear wheel (D) = 280 mm [circumference = 398.98 mm]

Gear Ratio (G): 1.41[14 T clutch sprocket and 20 T axle sprocket]

The calculations have been done for 8000 RPM crankshaft speed as max power is available only at that speed.

Assumed total weight of the kart:

- Driver = 65 Kg
- Chassis = 15 Kg
- Engine = 30 Kg
- Other stuff = 80 Kg

Total = 190 Kg

Max angular speed of the rear wheel (N):

Gear	Reduction ratio	Speed of the Rear wheel (RPM)
First gear	14.29	514.8
Second gear	9.182	871.62
Third gear	6.95	1151.07
Fourth gear	5.62	1423.48
Fifth gear	4.90	1632.6

Chain: R 1278 OR 08B-1(pitch, p = 12.7 mm)

Since the engine used has a five gear speed variation system, some of the calculations below are done for each gear.

4.2 Max Speed of the kart:

$V_{(max)4} = (\pi \times D \times N) \div 60$

$= (\pi \times 0.28 \times 1632.6) \div 60$

$= 23.92 m/s$

$\mathbf{V_{(max)4} = 86.122\ Kmph}$

Similarly, for the other gears,

First gear	7.54 m/s
Second gear	12.77 m/s
Third gear	16.87 m/s
Fourth gear	20.86 m/s
Fifth gear	23.92m/s

4.3 Max Acceleration:

$a_{(max)1} = [g \times HP] \div [V \times weight]$

$= [9.81 \times 7673.5] / [7.54 \times 190 \times 9.81]$

$\mathbf{a_{(max)1} = 5.35}$

First gear	5.35 m/s^2
Second gear	3.162 m/s^2
Third gear	2.384m/s^2
Fourth gear	1.93 m/s^2
Fifth gear	1.688m/s

4.4 Transmission:

Gear ratio = 1.41

Clutch sprocket (Z1) = 14

Axle sprocket (Z2) = 20

Engine torque (T1) = 11 N-m at 6500 RPM

Torque at the axle (T2) = ?

Engine sprocket speed (N1) =2301.96

Axle sprocket speed (N2)=1632.6

First gear	514.8
Second gear	871.62
Third gear	1151.02
Fourth gear	1423.48
Fifth gear	1632.6

4.5 Pitch circle diameter of the sprockets:

Engine sprocket: 14T

D_1 = p x cosec (180÷14)

= 12.7 x cosec (12.85)

D_1 = 57.07mm

Axle (driven) sprocket: 20T

D_2 = p x cosec (180÷20)

= 12.7 x cosec (9)

D_2 =81.18mm

D_2< D (280 mm) The axle sprocket doesn't hit the track during operation.

4.6 Chain Drive Calculations:

R 1278 Chain specs:

Pitch (p) = 12.7 mm

Max Roller diameter (D_r) = 8.51 mm

Min width between the inner plates (W) = 8.00 mm

Max pin body diameter (D_p) = 4.45 mm

Max plate depth (G_{pl}) = 11.70 mm

Transverse pitch (P_t) = -NIL-

Max overall over joint = 20.5 mm

Bearing area = 0.50 cm^2

Weight per metre = 7 N

Min breaking load = 18200 N

4.6.1 Chain velocity:

$V_{C(max)} = \pi \times D_1 \times N1 \div 60$

$= \pi \times 0.069 \times 2301.96 \div 60$

$V_{C(max)}$ = 8.31 m/s

First gear	726.27 RPM
Second gear	1288.9 RPM
Third gear	1623.0 RPM
Fourth gear	2007.106 RPM
Fifth gear	2301.96 RPM

4.6.2 Chain Length:

L=Kxp
K = Constant

$K = [(Z1+Z2) \div 2] + [(2 \times C) \div p] + \{ [(Z2 - Z1) \div (2\pi)]^2 \times [p \div C] \}$

Z1 = 14

Z2 = 20

P = 12.7

C = 30p – 4 = (30 x 12.7) – 4 = 377 mm …C = minimum distance between sprocket centers.

$K = [(14 + 20) \div 2] + [(2 \times 377) \div 12.7] + \{[(20 - 14) \div (2\pi)]^2 \times [12.7 \div 377]\}$

K = 76.40

L = K x p = 76.37 x 12.7 = 970.28 mm

L ≈ 970 mm

4.6.3 Factor of Safety of the chain drive:

$$\textbf{FOS} = \frac{\text{Breaking load}}{\text{Total load on the driving side}}$$

= W_B ÷ W ……A

For R1278 , W_B = 18200 N

W = Tangential force (F_T) + Centrifugal force (F_C) + Sagging tension (F_S) ……B

$F_T = \frac{\text{power to be transmitted}}{\text{Speed of the chain}}$

$= \frac{7673.5}{7}$

= 1096.21 N

$F_C = m \times v^2 = 0.7 \times (8.31)^2 = 48.33N$

$F_S = k \times m \times g \times C$ ….. [k = constant = 3, C = minimum centre distance between sprockets = 0.377 m]

= 3 x 0.7 x 9.81 x 0.377

= 7.849 N

B W = 1096.21+ 48.33 + 7.849

W = 1152.38N

A **FOS** = $\frac{18200}{1152.38}$

= **15.79**

4.7 Transmission efficiency:

Efficiency = 1 – (power lost ÷ input power)

Input power = 7673.5 W

Power lost = (Drag force + Rolling resistance) x Velocity

Drag force (F_d) = C_d x A x V^2 x density of air ÷ 2 [A = Frontal area] = 0.7 x 0.57 x $(23.92)^2$ x 1.16 ÷ 2

$\mathbf{F_d = 132.41\ N}$

Rolling resistance (R_r) = μ x W x number of wheels = (0.03) x (190 x 9.81) x 4 [for Asphalt, μ = 0.03]

$\mathbf{R_r = 223.668\ N}$

Power lost =(132.41+223.68) x 14.07 = 5010 **W**

Efficiency = 1 –(5010÷ 7673.5)

= 0.3471 = **34.71%**

4.8 Wheel torque

T_w = Overall Gear ratio x transmission efficiency x engine torque

= Overall gear ratio x 0.3471 x 11

Gear	Overall gear ratio	Wheel torque (N-m)
First gear	15.54	59.33
Second gear	9.18	35.05
Third gear	6.95	26.535
Fourth gear	5.62	21.45
Fifth gear	4.90	18.70

4.9 Tractive effort:

TE = T_w ÷ (Radius of the rear wheel, R)

First gear	423.78 N
Second gear	250.35 N
Third gear	189.53 N
Fourth gear	153.21N
Fifth gear	133.57 N

Checking the above obtained values…

Max tractive effort that can be applied = $TE_{max} = \mu_t$ x (Weight of the kart, W) …For Asphalt, μ_t = 0.7
$\mathbf{TE_{max}}$ = 0.7 x (160 x 9.81) = **1304.73 N**

Since the tractive effort produced in 1st, 2nd, 3rd and 4th, 5th gears is lesser than the max applicable tractive effort, slip won't occur. There is a LESS chance for wheel slip.

4.10 Gradability:

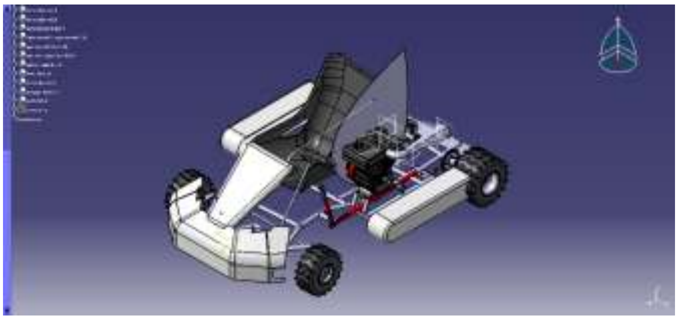
Tractive effort = W x sin θ

W = m x g = (160 x 9.81) = 1570 N

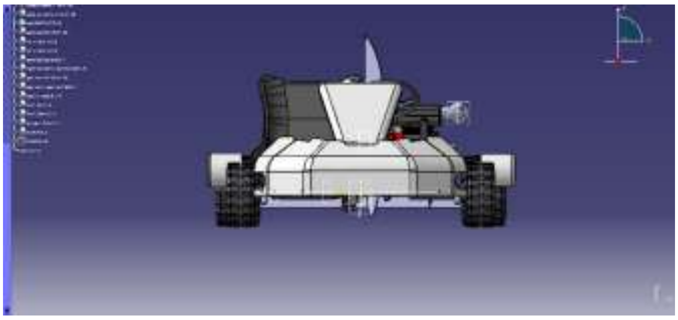
First gear	13.12 deg
Second gear	7.718 deg
Third gear	5.831 deg
Fourth gear	4.703 deg
Fifth gear	4.105 deg

5. 3D VIEWS OF THE KART

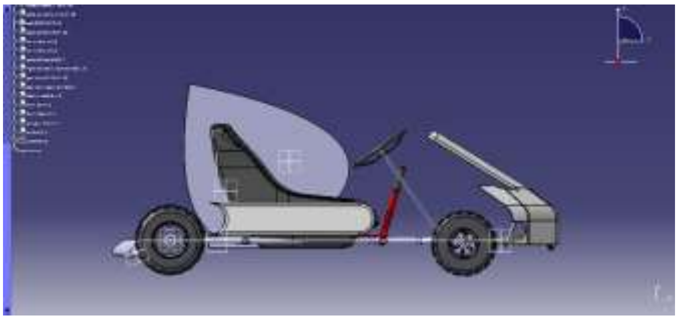
ISOMETRIC VIEW:



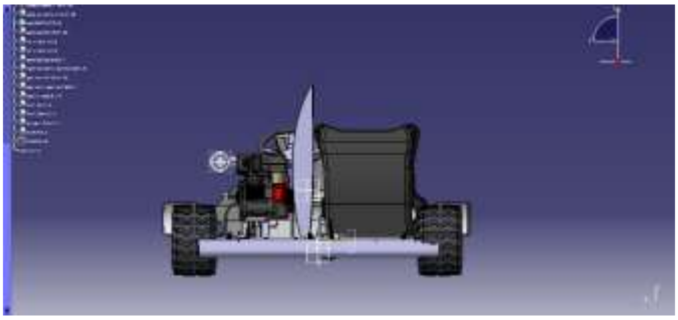
FRONT VIEW:



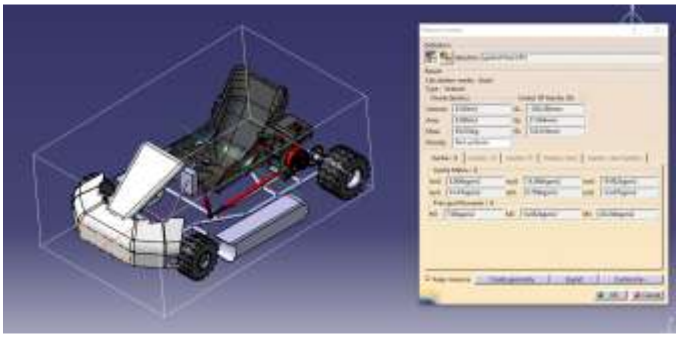
SIDE VIEW:



REARVIEW:



5.1Calculation of center of gravity using CATIA



The center of gravity is somewhat in the engine side and between the seat and engine.

5.2 DESIGN PROCESS :(design methodology)

- A hollow rod was chosen instead of solid rod for the purpose of cost reduction and better power to weight ratio. Hence the outer and inner diameter of the rod is 26 mm and 20 mm.
- The priority was given to design a stable structure with an aerodynamic arrangement in the front part of the chassis.
- A non-symmetric rear end was designed to facilitate the engine support, driver comfort and also provide a slight rear lift during turns.
- Cross bars were used to provide uniform stress distribution under running conditions.
- Chassis was designed with only x-axis and y-axis. The usage of z-axis was avoided to prevent complications and to simplify the analysis process.

5.3 ERGONOMICS

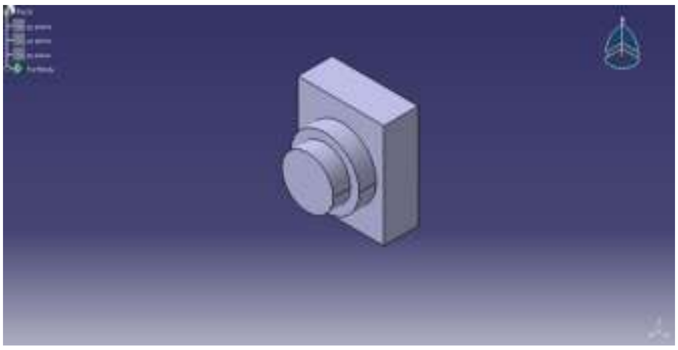
Driver ergonomics played a major role in designing of our vehicle chassis. The cockpit has been designed to allow considerable comfort of the driver. Large leg space and enough room for movement inside the cockpit are some salient points.

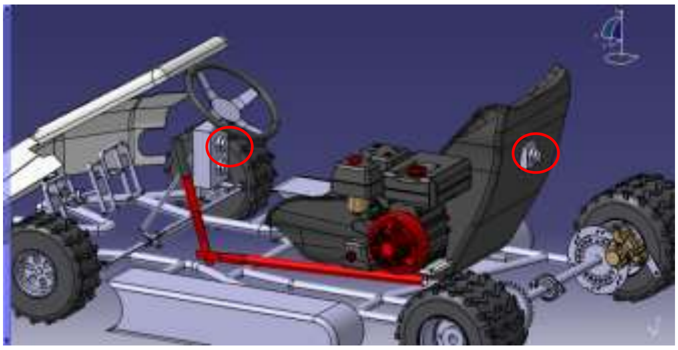
The approach adopted for driver ergonomics was to question our driver on his requirements and using him as our base for measurements, calculation and designing of our chassis. The output has been successful design of cockpit that is safe and comfortable with driver in. The chassis has been designed to enhance the driver's visibility.

All the essential controls in vehicle have been placed such a way that it can be accessed with ease. The accelerator, brake pedals are positioned such that the driver shall stretch his legs for a long time without any stress.

We have placed the kill switch near the centre of the vehicle for easy accessand the other one in the side of the seat.

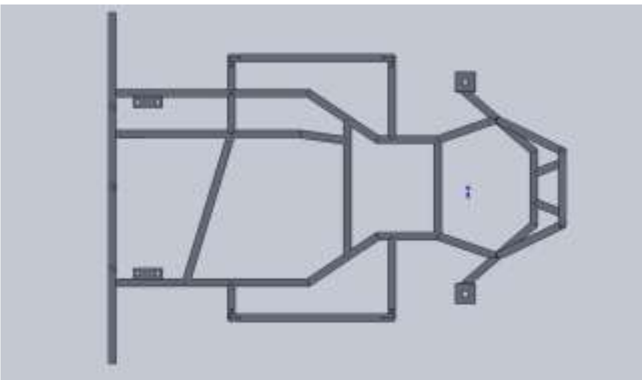
• Wide and spacious cockpit.
• Egress time: 5 seconds.
• All controls within the reach envelop of the driver's hand.
• The seat has been designed to withstand any kind of sudden motion.



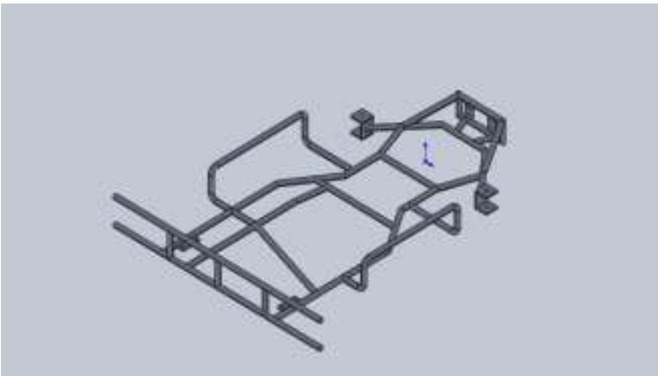


5.4 DIFFERENT VIEWS OF THE CHASSIS:

A. : TOP VIEW:



B. ISOMETRIC VIEW:



C. PROTOTYPE:

FLOOR PLANNING:

5.5 PROPERTIES OF MATERIAL AISI 1018:

CHEMICAL COMPOSITION

- Carbon - 0.20% (max)
- Phosphorous, sulphur - 0.050% (max)
- Manganese – 0.6-0.9%

MECHANICAL PROPERTIES

- Ultimate strength : 440 MPa
- Yield strength : 370 MPa
- The material has good weldability, machinability and cost effective.

5.6 WEIGHT CALCULATION OF CHASSIS:

Total length of the rods used in design=6 m
Density of the chassis material = 7.87gm/cm^3
Outer diameter of the rod = 26 mm
Inner diameter of the rod = 20 mm
Volume of the rod = $[\pi/4] \times [((26 \times 10^{-3})^2 - (20 \times 10^{-3})^2) \times 6]$

Volume of the rod = 1.3×10^{-3} m^3
Weight of the chassis = $1.3 \times 10^{-3} \times 7.87 \times 1000$
Weight of the chassis =10.23 Kg (11kg approx.)

6. CHASSIS ANALYSIS

This analysis was done in order to check the safety of the chassis design. It was completed by conducting dynamic, static, torsional and modal analysis over the chassis. Deceleration after the impact was assumed to be zero during Impact tests. Driver safety was ensured even at the worst case. The following tests were conducted

- Front impact
- Rear impact
- Side impact
- Static analysis
- Modal analysis
- Torsional analysis

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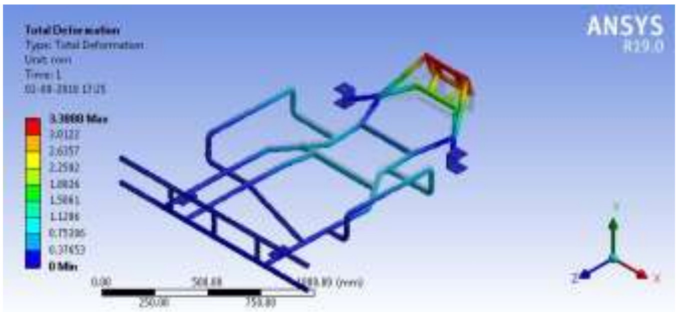
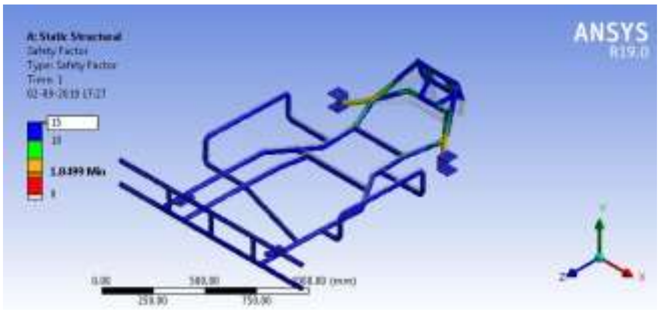
6.1 DYNAMIC ANALYSIS

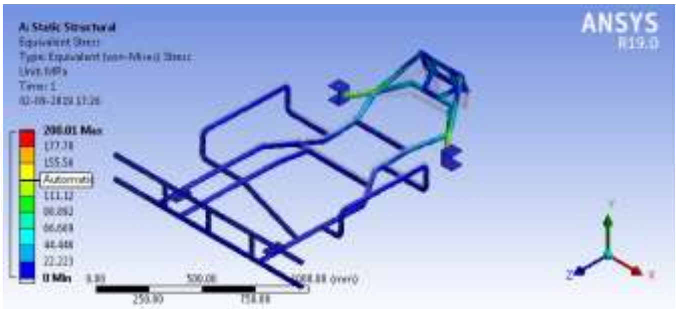
6.1.1 FRONT IMPACT

The first analysis to be completed was that of a front collision with a stationary object. In this case a deceleration of 10836 N was the assumed loading.For the front impact test, the front nodes are applied with the load calculated. The rear is completely constrained allowing displacement to occur only in direction of the load applied. The maximum stress after front impact was 200Mpa. But the yield stress of the material is 370 Mpa. So, our design proved to be safe.





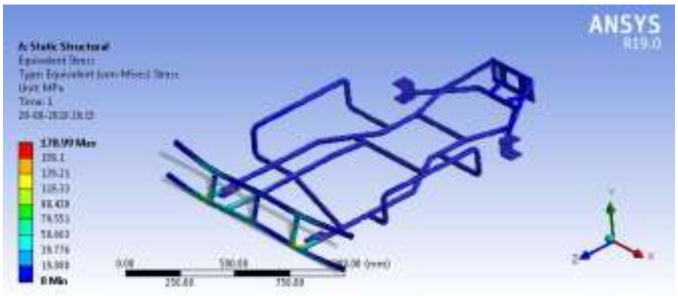
The factor of safety calculated from the front impact is 1.8499



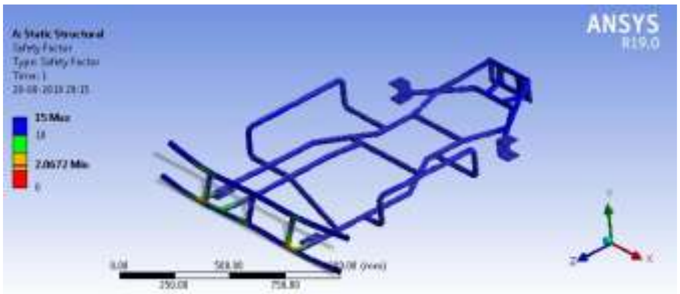
6.1.2 REAR IMPACT

Next rear impact analysis was done while assuming 7225.1N as the impact force.

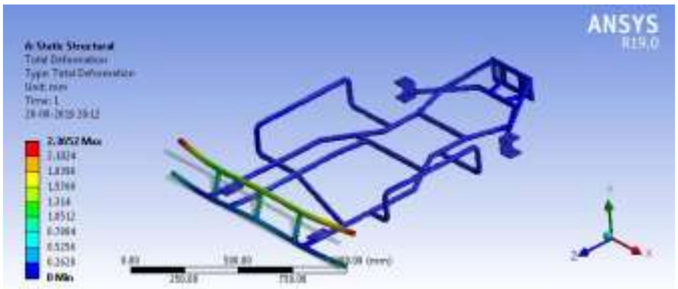
It is same as the front impact but the rear portion was constrained. The maximum stress after rear impact was 178.99Mpa. But the yield stress of the material is 370Mpa. So, our design proved to be safe.



The factor of safety for the given load is calculated as 2.03



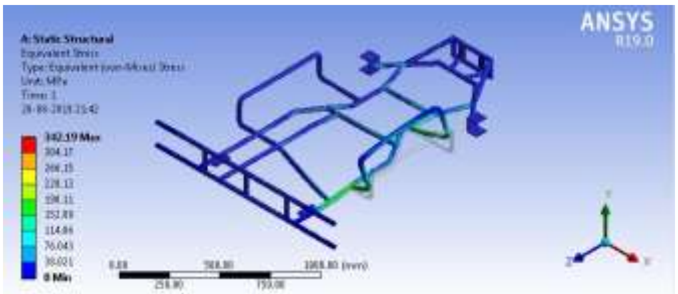
The total deformation for the given load is 2.365



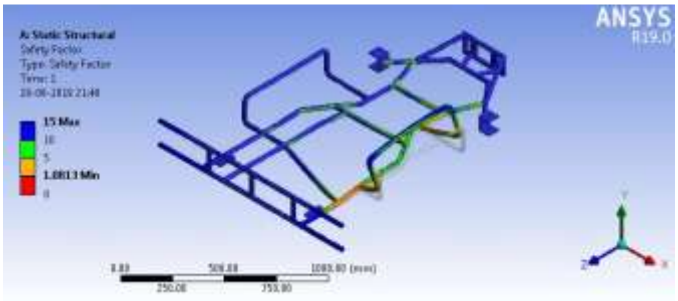
6.1.3 SIDE IMPACT

The next step in the analysis was to analyse a side impact with a 7225.1N load.

As a side impact is most likely to occur with the vehicle being hit by another kart vehicle it was assumed that neither vehicle would be a fixed object. The maximum stress after side impact was 342.19Mpa. But the yield stress of the material is 370MPa



The factor of safety is calculated as 1.08 for the given load



6.2 STATIC ANALYSIS

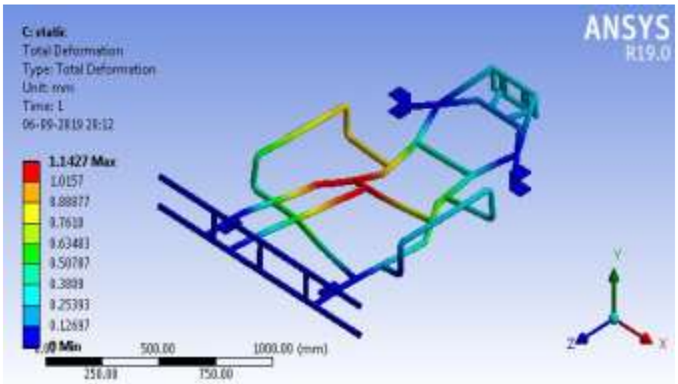
Total load acted=95+30=125kg=125*9.81=1226.25 N

Where, weight of the driver assumed as 95 kg at the max conditions and the weight of the engine is assumed as 30 kg.

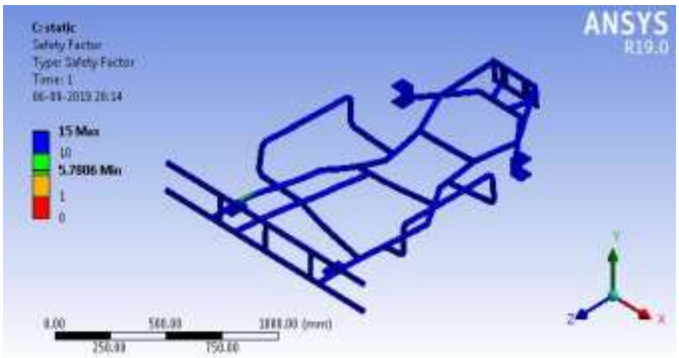
The weight is assumed to be acted on the specific regions such as the driver seating and the engine mounting.

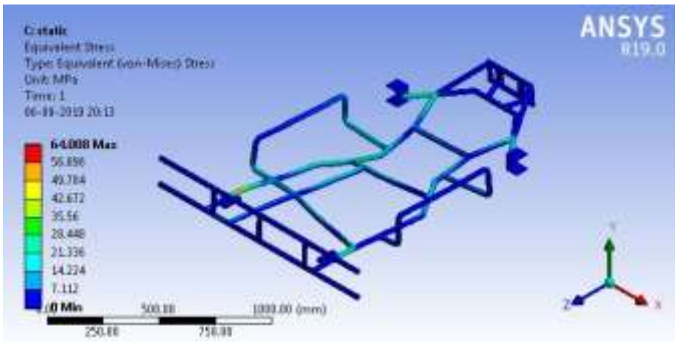
Both the deformation and the stress analysis are done for static analysis.

The weights of the other sub systems are negligible compared to driver's weight and the weight of the engine.



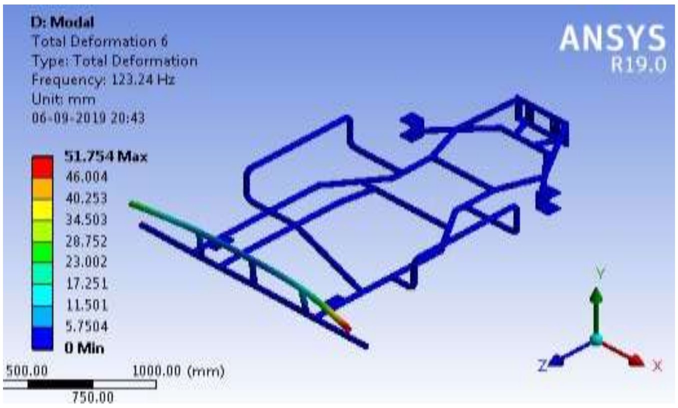
The maximum displacement during static analysis is 1.142. and fos=5.7

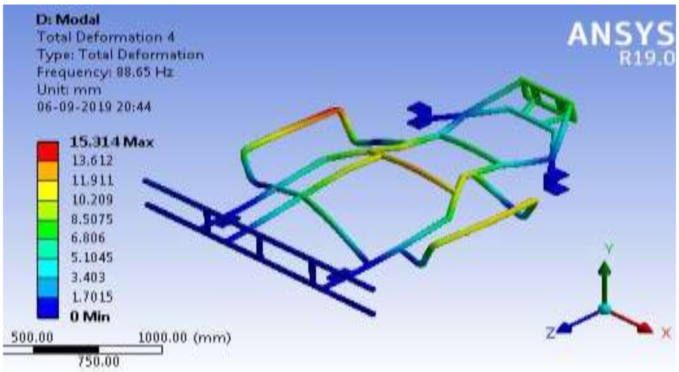




6.3 MODAL / VIBRATION ANALYSIS:

Next for the vibrational analysis, the vibrational frequency applied to the frame is assumed to be is 133.3HZ calculated for 8000 rpm engine(stunner).





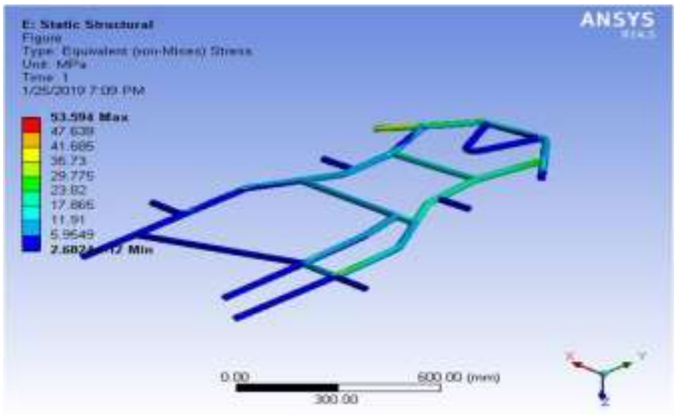
The maximum displacement during vibrational analysis is 51.5mm

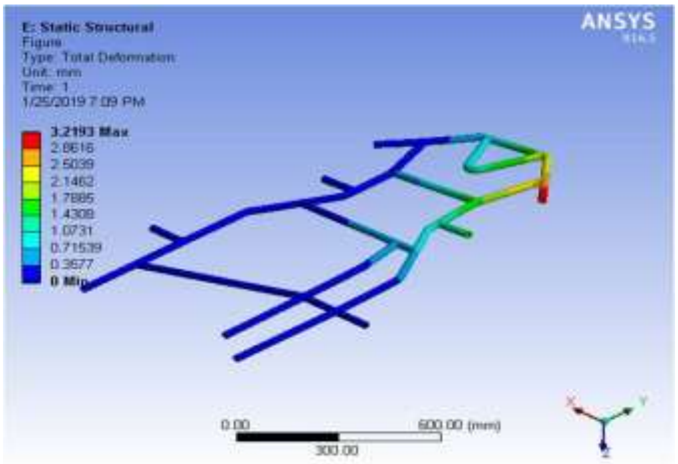
6.4 TORSION ANALYSIS

In torsional analysis, among four wheels three wheels are fixed and a bumping load is acted on one wheel.

The load acted on the single wheel =9.81(chassis weight+ driver weight+ engine weight)

9.81*(30+70+15)=1275.3 N

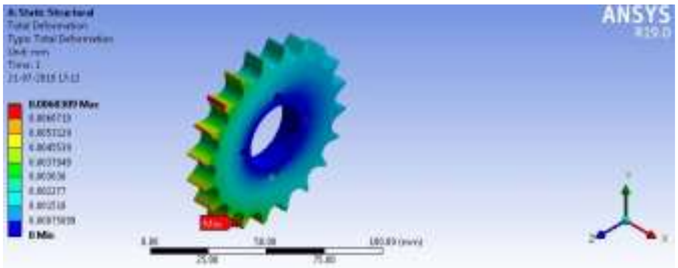




The maximum deflection of the chassis during torsional analysis is 3.219mm.

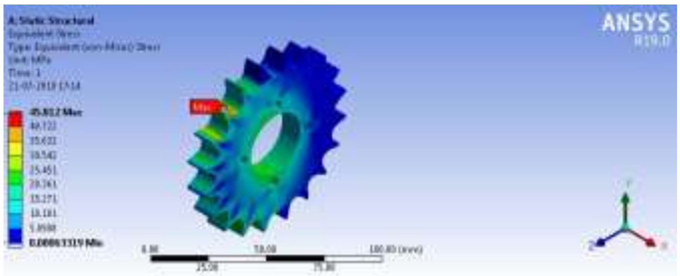
FOS=3.4

6.5 STATIC ANALYSIS OF SPROCKET

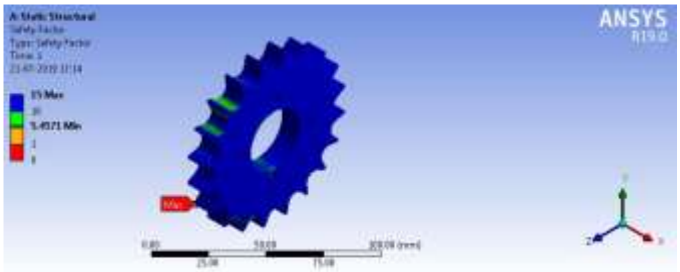


Here the inside is kept fixed. The moment of 157.29N-m at first gear engagement is applied to the teeth that meshes with the chain.

A maximum deformation of 0.0068mm is obtained.

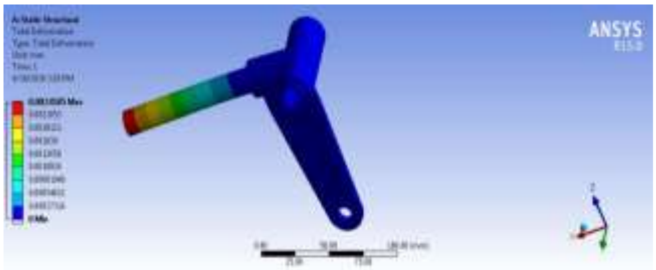


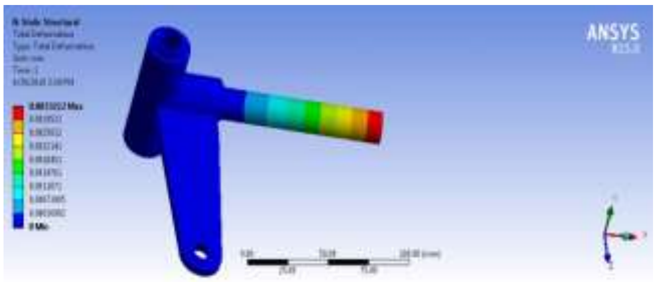
A maximum stress of 45.812MPa is obtained.



A factor of safety of 5.45 is obtained when the load is 157.29N-m (first gear is engaged)

6.6 STATIC ANALYSIS OF STUB AXLE

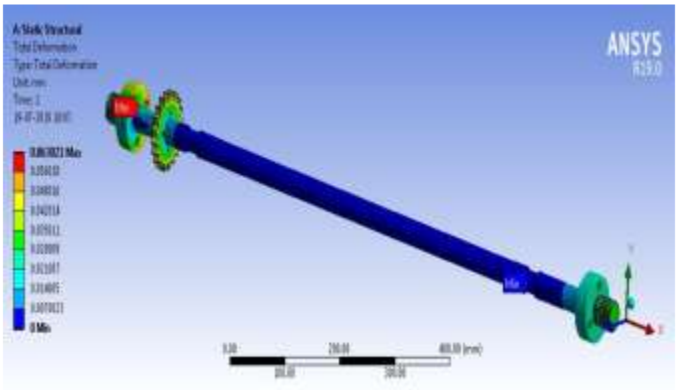




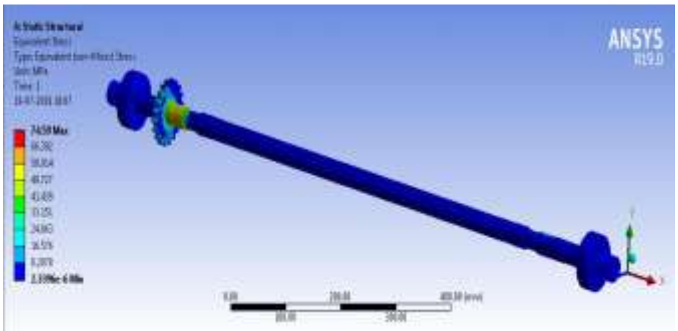
Here the axle part is fixed and the part attached to the wheel is given a load about 15 kN . here this load is due to the wobling of tires.we get quite a small amount of deformation of about 2.5*10^3mm in the left axle.

6.7 AXLE ANALYSIS:

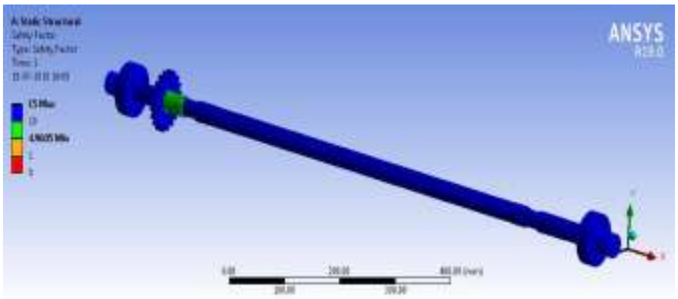
In the axle analysis,the live axle of the kart is mounted to the rear wheels. The loads and moment acting on the axle's end and sprocket is calculated and the results are obtained. The moment acting on the sprocket is calculated as 157.5N-m (Torque transferred when first gear is engaged). Also the load of the kart is evenly distributed to each wheel hub positions and its calculated as 520N on each wheels.



The deformation is calculated as 0.063021mm.



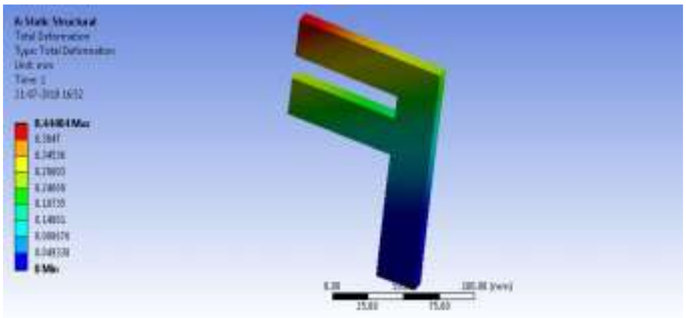
The maximum deformation is calculated as 74.59MPa



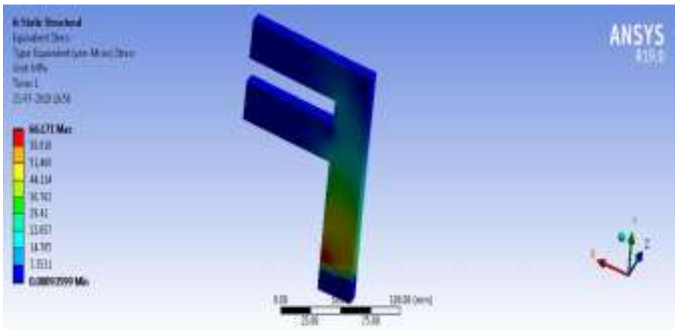
The factor of saftey is calculated as 4.905.

6.8 BRAKE PEDAL ANALYSIS:

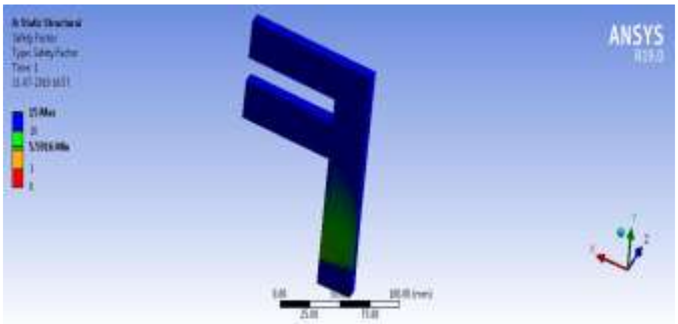
In the brake pedal analysis,The force given by the driver to the brake pedal is calculated and applied on the brake pad. By the standards the pedal force is about 400N.Its recommended that no more than this force is required to stop a passengers car.since a pressure of 200N applied othe brake and the result is observed.



A deformation of 0.4404mm maximum is obtained.



A maximum stress of 66.17MPa is obtained.



A maximum factor of saftery of 5.5 is obtained for a load of 400N.

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The design process is not a single-handed effort and it is our team, whom we wanted to thank for standing with us under all circumstances. We would also like to express our gratitude towards the Mechanical department and on the whole towards the college for their support and also BFKCT for providing such a wonderful opportunity to learn and grow.

Special thanks to our Head of the Department **Dr. L.Karthikeyan** and to the team mentor **Mr.S.Dhanasekhar** for their guidance and support.

7. CONCLUSIONS

The finite element analysis method is used to evaluate the system and create and modify the best vehicle design to achieve its goal. The main goal was to simplify the overall design to make it more light weight without sacrificing performance and durability also make the driver comfortable. The result is lighter, faster and more angle vehicle that improves go kart design.

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BIOGRAPHIES:

R. Santosh kumar
Department of Mechanical Engineering.

R. Rohith
Department of Mechanical Engineering.

S. Srinivas
Department of Mechanical Engineering.