Design, Analysis & Manufacturing of Go-Kart

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Abstract— The goal of paper is to design & analyse a racing vehicle (Go-Kart) to withstand all contours of tractions in all types of weather without damage. The objective for the team is to design and analyse a racing vehicle powered by a petrol engine. The vehicle must be safe, portable, easy to maintain. It should be able to negotiate with all the tractive forces without damage. Thus the frame design against failure becomes ethical. Typical capabilities on basis of which these vehicles are judged are maximum speed and acceleration. As weight is critical in a vehicle powered by a small engine, a balanced use of solid modelling and Finite Element Analysis (FEA) software is extremely useful in addition to conventional analysis. The following paper outlines the design and analysis of the Go-Kart chassis and other sub-systems design. It will cover the design constraints like material selection, initial design, and structural analysis. It will finally cover the results of the actual real world usage of the vehicle.
Key words: Design, Analysis

I. Introduction

We approached our design by considering all the possibilities and designed models in Solidworks and analysed them in Ansys. The Design method is iterative and the chassis was modified after analysis.

The purpose of this project was to design, manufacture, teamwork and communication skills of the team members to prepare them for working in industry. The team was divided into six subdivisions in order to design all the main aspects of the vehicle. The subgroups were: the transmission team, wheel and geometry team, frame design team, steering team, braking team and manufacturing team. The frame team focused on producing a chassis that is lighter. This team has focused on shortening the frame so that less material is used. The drive train team focused on designing a more efficient power train design. This was achieved by using CVT. The steering team focused on better maneuverability of the Kart so as to negotiate every turn. Also the team worked to achieve smaller turning radius and perfect rolling condition. The braking team focused on minimum braking distance and time; without losing control of the vehicle. Additionally team worked on complete and continuous locking of the brake disks when actuated

A. *Overall Vehicle Details*

Length	2210 mm
Width	1390 mm
Wheel base	1450 mm
Track width	970 mm
Ground Clearance	52 mm
Front Tyre Size	10’’ x 4.5-5 SM 68 TL
Rear Tyre Size	11’’ x 7.1-5 SM 68 TL
Max. Speed	68 Km/h
Max. Acceleration	1.26 m/s²
Stopping Distance	4.178 m
Kerb Weight	90 Kgs
FAW to RAW Ratio	0.3:1

Table 1:

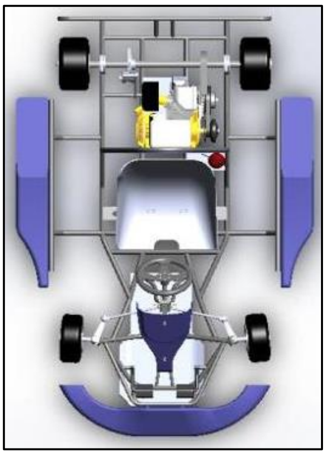

Fig. 1: The Enigma – Isometric View

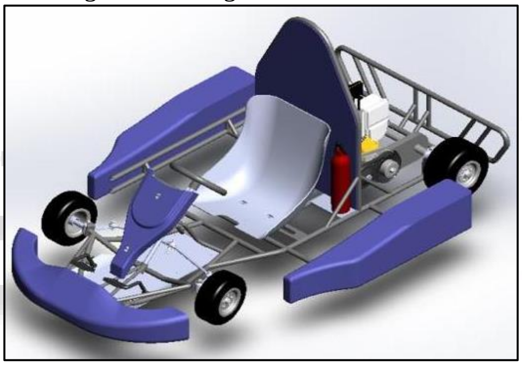

Fig. 2: The Enigma – Top View

II. Chassis Design

A. *Objective*

The chassis is the component in charge of supporting all other vehicle’s subsystems with the plus of taking care of the driver safety at all time. The chassis design need to be prepared for impacts created in any certain crash or rollover. It must be strong and durable taking always in account the weight distribution for a better performance.

B. *Material Selection*

According to the rulebook of the competition the chassis material must have at least 0.18% carbon content. The following materials which are commercially available and are currently being used for the chassis of a racing vehicle were shortlisted-AISI 1020 steel & AISI 4130 Steel. Both the materials were analysed with outside diameter 25 mm and wall thickness of 3 mm.

After reviewing each of these analyses it is evident that the best choice would be use AISI 4130 Chromalloy tubing with a 1 inch diameter and a 0.11811inch wall thickness.

C. Analysis

Each impact test is a worst case scenario that could potentially occur to the vehicle. There are three tests:

1) Front Impact Test
2) Rear Impact Test
3) Side Impact Test

D. Front Impact Test

Generally in the case of pure elastic collision in frontal impact the linear velocity remains at 64Kmph according to ENCAP (The European new car assessment program).

Hence the value of force is calculated by mass moment equation that is- f=p×Δt

Where ΔT is the duration of time, generally the collision takes place for a very short duration of time. We assumed this time as ΔT=1.01 seconds. And the gross weight of vehicle is Estimated some around (M-150Kg), hence the moment of the vehicle at 64 Kmph or 17.8 m/s that is-

– P=M×V
– P=150×17.8
– P=2670 kgms

And the fontal impact force, i.e.

– F=P×ΔT
– *F*=2670×1.10
– F=2937 N

Now the calculated force were placed on the frontal part of frame by keeping the rear part fix on Hyper-mesh the resulting along with the images as:

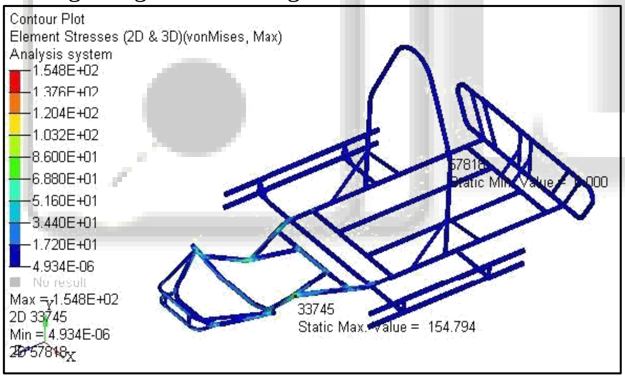

Fig. 3: Maximum Stress during Front Impact

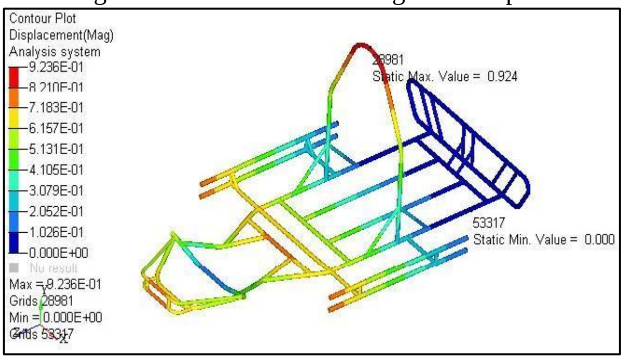

Fig. 4: Maximum Deformation during Front Impact

E. Side Impact Test

For the side impact the velocity of the vehicle is taken as 48kmph or 13m/s according to ENCAP standard& then the force is calculated. In the case of collision by side impact, the value of the impact force generated is calculated in the same way as in front impact.

F. The Side Impact Force

F=2194.5 N

Hence the calculated force was placed on one side of the model of frame while keeping another side fixed and the stresses were simulated. The images are shown below

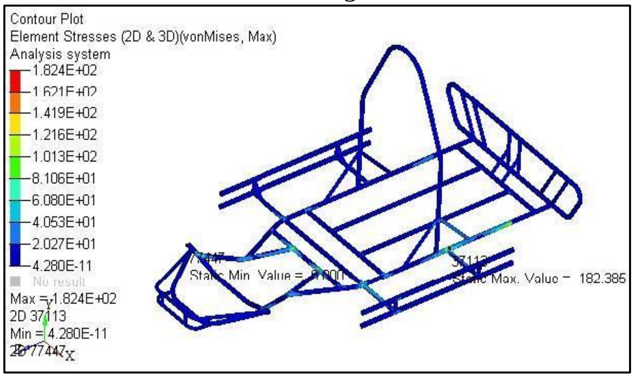

Fig. 5: Maximum Stress during Side Impact

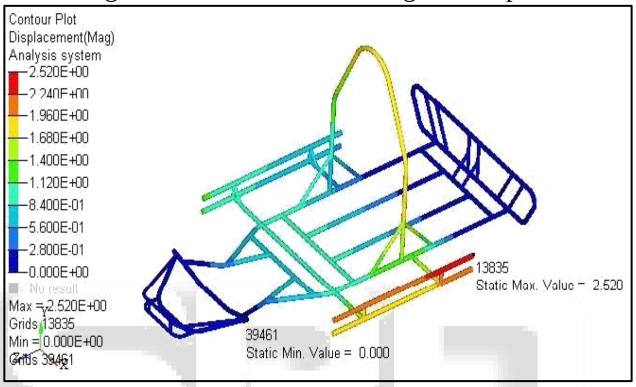

Fig. 6: Maximum Deformation during Side Impact

G. Rear Impact Test

The rear impact force is also calculated in the same way as remaining two. In the case of velocity of collision were taken 50kmph or 13.8m/s by the calculation & also as according to the ENCAP standards & the force is calculated.

And the rear impact force-

F=2277 N

Hence the calculated value of the rear impact force was placed on the rear part of the frame while keeping the frontal part fixed. The analysis is shown as

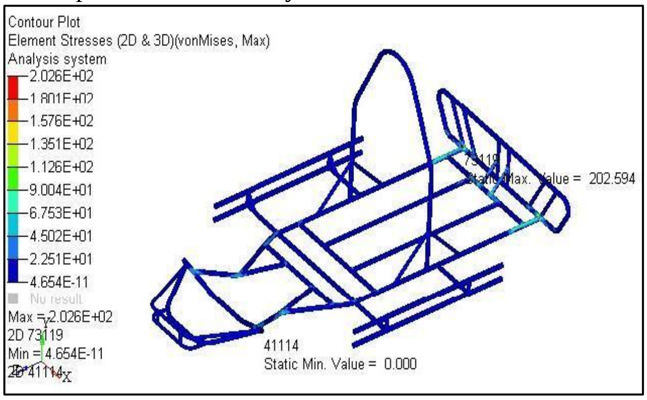

Fig. 7: Maximum Stress during Rear Impact

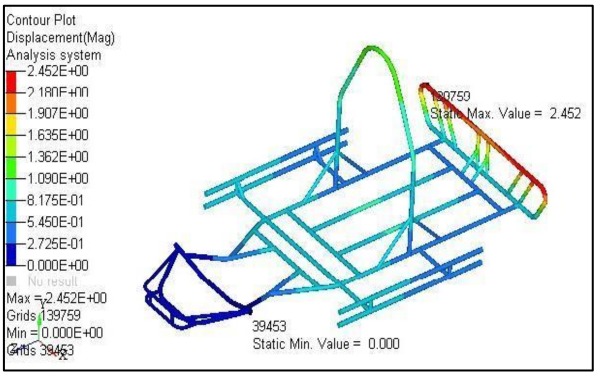

Fig. 8: Maximum Deformation during Rear Impact

Hence the conclusion of the safety analysis with result is tabulated as

FRACTORS	FRONT IMPACT TEST	REAR IMPACT TEST	SIDE IMPACT TEST
Impact Force	2937	2277	2194.5
Stress Generated	154.794	202.594	182.395
Total deformation	0.924	2.452	2.520
F.O.S	2.97	2.27	2.51

Table 2:

III. STEERING

A. Objective

The steering subsystem is responsible for the control of the vehicle. In the design process of this process of this subsystem the goal is to achieve a turning radius and steering stability. The speed of response and the driver´s input are also prime factors for the design of the steering system.

During steering, the wheel movement must be positive & exact no wheel should slide on the road.

B. Design

- The steering system works without a rack and pinion.
- The pitman arm is connected to 2 tie rods which work in to support the steering system to transmit the load applied by driver or transmit the driver's effort.
- The front wheels configuration has a -2/0° camber angle and a 3.25/0 ° caster angle.

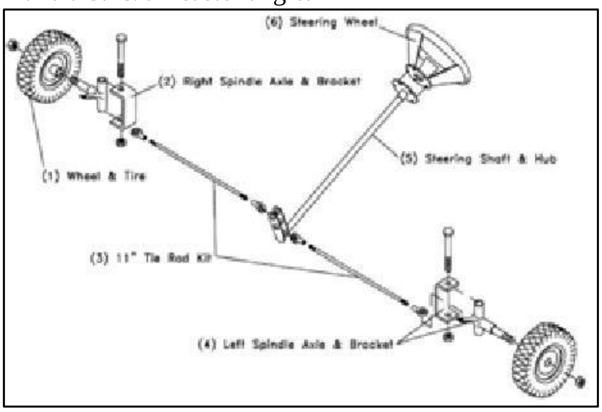

Fig. 9: Schematic Representation

The caster tends to drive the wheels forward, which makes it easier to maintain the car in a straight direction, also the inclination of the knuckle helps to reduce the turning radius to 2.10m

1) Calculation
- Wheel base(b)=1.45 m
- Track Width (a)=0.970m
- Tie Rod Length (c)=0.77m
- Inner wheel maximum turning angle (θ): $= 38^0$
- Outer wheel maximum turning angle ($\emptyset$): $= 38^0$
- Steering ratio= 1: 1

2) *For* 38 degree Turning radius
- For Inner Front Wheel Radius Rif = 2.4551m
- For Outer Front Wheel Rof =2.2551 m
- For Inner Rear Wheel: RiR = 1.9559m
- For Outer Rear Wheel: RoR =1.7559 m
- Steering Wheel torque = 48.55Nm.
- Axial load = 24.27N
- Steering Safe Load: ML =532 N
- Where ML is maximum load.

Steering Calculations

3) $centrifugal\ force$=1321.693 N
- slenderness ratio=(LK) =34.54
- Hence,>30, critical load acts on it.
- SR<100, Considering short column

Hence, by Euler's rule-
- Pcr=4358.26 N

4) Safe compression force

$=PcrFOS$

=4358.26/1.5

=2905.50 N

Applied force

=2×48×9.81

=941.76 N

Hence, centrifugal force< tie rod buckling.

Applied force < safe compression force.

Hence, design is safe.

IV. BRAKE SYSTEM

A. Objective

Brake system is designed such that rear wheels will lock when brakes will come into play at full efficiency.

B. Design

For braking system we have used single hydraulic system working with Maruti 800 master cylinders and single pedal. Two ports of cylinder are blocked and one port actuates the brake calipers and one is connected to hydraulic switch for brake light indication.

Braking system has a TVS® caliper with rotor.

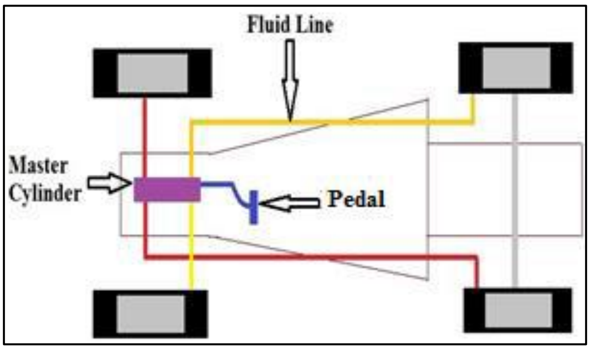

Fig. 10: Braking System

C. Calculations

- Rotor diameter = 0.21m
- Outer radius Ro = 0.12 m
- Inner radius Ri = 0.095 m θ=0.9773 $rads$

The results of the calculations:

- Pedal force is 26.76 kg
- Stopping distance=19.84m
- Stopping time=2.20 sec
- Similarly, for V=11.11m/s (40kmph)
- Deceleration rate=7.00m/s2
- Pedal force=26,76kg
- Stopping distance=8.813m
- Stopping time=1.58sec

If 50 kg applied,
50=FT 9.81×4
FT =1962 N
Therefore, stopping distance will be 4.178m

V. Drive Train

A. Objective

The objective of the drive train is to provide more than the enough torque to the wheels from the engine. The calculations were made in order to select the proper components. And also must provide the speed range of 4km/hr. to 65 km/hr. and to provide enough acceleration for racing.

B. Design

The main component of the drive train is the Briggs & Stratton engine which gives 7.45Nm (5.5 lb-ft) of torque at 2600 rpm and 3.5hp at 3600 rpm.

C. Specification of Power Train

1) Engine

- Cubic Capacity - 127cc
- Maximum Torque – 7.45Nm 2600@ rpm
- Power – 3.5Hp

2) CVT

We are using a continuously variable transmission of Comet which provides the following speed ratios.

- Speed Ratios:

a) Minimum Ratio- : 2.7: 1
b) Maximum Ratio: 0.9: 1

D. Resistance Calculation

1) Rolling Resistance

Rr=kr×m×g
=0.015×150×9.81
=20.0725 N

2) Air Resistance

Ra=12×ρ×A×cd×$(V/3.6)^2$
=12×1.202×1×0.5$(68.8/3.6)^2$
=109.75 N

3) Gradient Resistance

RG=mg sinθ (Whenθ=0)
=0 N

4) Total Tractive Resistance

Rt=Ra+Rr+RG
=109.75+20.072+0
=129.82 N

5) Brake power

BP=R_t×v/$n_{t\times3600}$
=129.82×68.800.95×3600
=2.6115 KW

6) Torque

P=2πNT/60
=2.6115×602π×4984.7
T=5.0Nm
IBJ center distance:
C = 0.75 m

E. Design of Chain & Sprocket

1) Firstly selecting the sprocket with diameter

Dp1 = 60 mm (As per standard selection chart of chain sprocket from design data book)
The torque transmitted from the CVT is enough to overcome the required torque.
Therefore selecting the chain drive with ratio 1:1 .
Dp1 = Dp2 = 60mm
And the torque transmitted is
T = 20.115 N/m
Therefore,
T = F × r
Therefore,
F = 20.115 ×103 ÷30
F = 670.5 N

2) Selecting 08A (ANSI - 40)

(As per standard selection chart of chain from design data book)
Pitch =12.70 mm , D1 = 7.95 mm
B1 = 7.85 mm ,pt = 14.85 mm
As the breaking load (min) = 13800 N
Therefore,
Our force transmitted does not exceed minimum breaking load.

3) Average Velocity

V = πDN/60 × 10^3
= (π ×60 ×4984.7)/60 × 10^3
Vavg = 15.6598 m/s

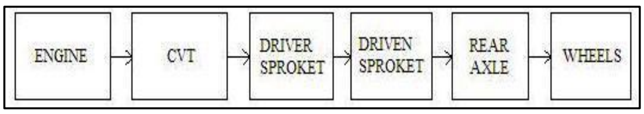

Fig. 11: Drive train

F. Performance

1) Force

F = T × iR

Where,

T = CVT output torque

i = velocity ratio

R = radius of tyre

Therefore,

F = 20.115 ×1.050.1314

F = 160.74 N

2) Acceleration

a = forcemass

= 160.74150

a = 1.07 m/*s*2

Gear Ratio	Torque	Force	Revolution	Angular Velocity	Top speed
	i/p torque *gear ratio	Torque R	N= 60P/2πT	ω= 2πN/60	ω*R
2.7	20.115	153.08	1661.5	173.99	22.86
0.9	6.705	51.027	4984.7	521.41	68.60

Table 3:

VI. Electric System

The electric system contains the break light; and emergency stop kill switches.

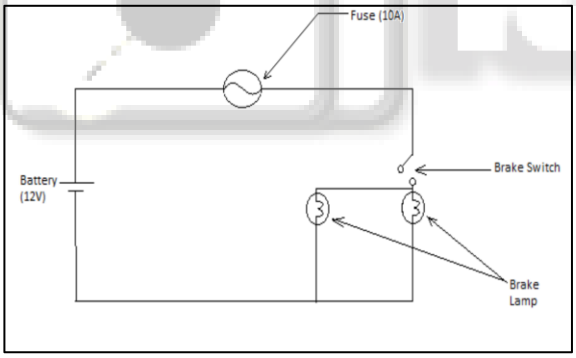

Fig. 12: Circuit Diagram

- There are two kill switches in the vehicle one over the wheel reach of the driver, and the second one outside of the car at the top right side of the rear body panels.
- This second location is easily accessible to team members and competition judges in case of emergency.

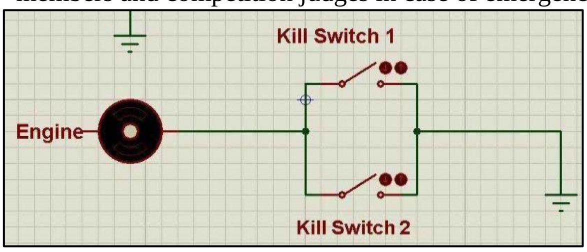

Fig. 13: Kill Switch

- The kill switches works by closing the circuit and killing energy to the engine causing immediate stop. However, pressing the kill switches does not affect the other electrical parts mounted on the vehicle.
- We are using the lead wire fuse box for preventive measures.

VII. Conclusion

The team's goal was to produce design that meets the SAE criteria for safety durability and maintainability as well as provides features that would have mass market appeal to the general Go-Kart enthusiast, comfort and aesthetics. Design decisions were made with each of these parameters in mind.

The team relied on individual member's knowledge and experience with Go-Kart vehicles as a tool for developing many of the initial subassembly designs for the Kart.

Where applicable, selection of components for each subassembly of the Kart was based on engineering knowledge gained through undergraduate level coursework. Reliance upon "engineering intuition" governed the selection of the remaining components. Computational design and analysis software were used to verify that each part of a subassembly design met or exceeded its stated objective. Use of these design tools also allowed the team to address and rectify conflicts between interfacing subassemblies before fabrication, saving both time and cost.

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