

# Design Analysis and Optimization of Go-Kart using Finite Element Analysis

Jawagar Shrehari J<sup>1</sup>, Raagul Srinivasan K A<sup>2</sup>B.E. Student, Department of Mechanical Engineering, Dr. N.G.P. Institute of Technology, Coimbatore, India<sup>1,2</sup>

**ABSTRACT:** The objective of this paper is to highlight the design report of the Go – Kart vehicle. The performance of a Go-kart depends a lot on the chassis design. Thus, this project takes a look at the investigation of chassis design, simulation and fabrication. We approached our design with a rough 2D sketch of the chassis and we created the virtual assembly of our go-kart using a CAD modeling software Solid works and the analysis was done using Ansys16 software. Based on the analysis the model was retested with boundary conditions under the practical parameters. So the design focuses on safety, serviceability, strength, ruggedness, standardization, cost, ergonomics and aesthetics

**KEYWORDS:** Go-kart, Chassis, Finite element analysis (FEA), Drive Train

## I. INTRODUCTION

The design objectives set out to be achieved were three simple goals applied to every component of the car: durable, light-weight, and high performance, to optimizing the design by avoiding over designing. In order to ensure that the design of the chassis achieve the standard level, detailed analysis was made through ANSYS16 software. The purpose of this simulation was to investigate the strength and the flexibility of the chassis. The simulations carried out with several altered parameters for various impact tests, which would also help in reducing the cost. With this we had a view of our kart. Our primary objective is to design a safe and functional vehicle based on rigid and torsion free frame, well mounted power train and with a bit of innovation are the key aspects in our design.

## II. RELATED WORK

In the design of today's vehicles there is a strong emphasis on reducing the vehicle weight and to lower the aerodynamic drag in order to leave a smaller ecological and economical footprint. This has shown to affect the crosswind sensitivity for ground vehicles, which is of importance for handling and safety [1]. Most engines have a single spark plug per cylinder, a notable exception being in aircraft where the complete ignition system is duplicated to improve reliability. The spark usually is provided by a battery and coil, although until the 1920s, a magneto often was used. For a continuously variable transmission (CVT) to produce an improvement in fuel economy, it must be efficient and have a wide span (range of ratios). There are two significant types of CVT-the Van Doorne belt system, and the Torotrak system-both of which have been the subject of much development work [2]. The steering system is a mechanism on a vehicle that serves to regulate the direction of the vehicle by means of deflecting the front wheels [3, 4]. It takes an astonishing amount of energy to decelerate a moving vehicle-in fact it takes the same amount of energy to decelerate from one speed to another as it would to accelerate between the two speeds-except that we can decelerate faster because most of the inertial forces are working for us rather than against us [5].

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

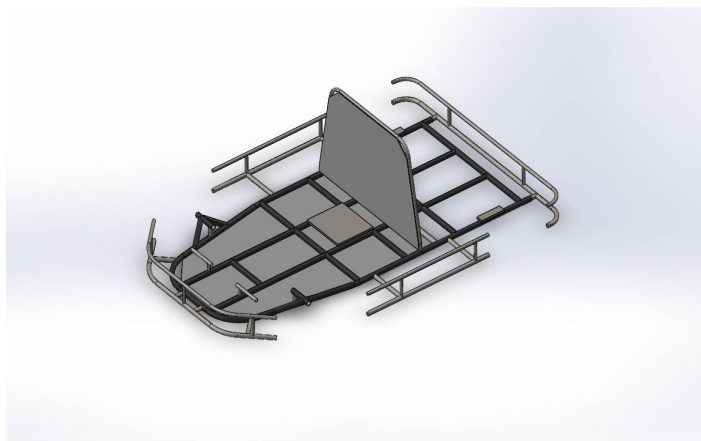
Vol. 5, Issue 11, November 2016

**TABLE I: Technical Specification**

Description	Specification	Description	Specification
Wheel Base	1260 mm	Engine &Transmission	8.4 Bhp @7500 rpm
OverallHeight	620 mm	Max.Torque	8.3 Nm @5500 rpm
Overall Length	1900 mm	Gear ratio(rear axle)	1:1:5
Overall width	1270 mm	Steering	Ackerman(pivot type)
Ground clearance	2.5" inches	Turning Radius	2.5M
Overallweight	160 Kg	Brake	Disc Type
Material	Seamless mild steel (asper analysis)	Tire size	11/7.10-5(rear) 10/4.50-5(front)

### III. OPTIMIZATION OF CHASSIS

#### A. MATERIAL SELECTION



**Fig 1.** 3D View of Chassis

The frame that supports everything right from steering to the rear power trains also supporting the necessary body works are the chassis. The chassis was mainly designed through the Solid works software. The chassis is a double cradle type and is suspension less. The material AISI 1012 Seamless Mild Steel with 0.10% of carbon is used in the frame design because of its good weld ability relatively soft and strengthens as well as good manufacturability. A good

## International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

strength material is important in a roll cage because the roll cage needs to absorb as much energy as possible to prevent the roll cage material from fracturing at the time of high impact. Mild Steel has been chosen for the chassis because it has structural properties that provide a low weight to strength ratio. The frame was designed using 1.25 inch diameter tube with a thicker wall of 3 mm is used instead of 1.5 inch diameter tube with a thinner wall for manufacturability purposes. Although the thinner wall, 1.25 inch diameter tube would be slightly lighter than the thicker wall, 1 inch diameter tube, it would have been more material and more difficult to weld. Then it is also assured by analysis in solid works software. The various Physical properties and chemical properties of the material are listed in Table II and Table III as follows.

**Table II: Material Properties**

S.No.	Properties	Values
01.	Tensile strength, Ultimate	450 Mpa
02.	Tensile strength, Yield	380 Mpa
03.	Bulk modulus	200 Gpa
04.	Shear modulus	80 Gpa
05.	Modulus of elasticity	200 Gpa
06.	Poisson's ratio	0.29
07.	Elongation at break	16%

**Table III: Chemical Composition**

S. No.	ELEMENTS	VALUE
01	Carbon,C	0.10%
02	Manganese,Mn	0.47%
03	Silicon,Si	0.025%
04	Sulphur,S	0.008%
05	Phosphorus,P	0.014%
06	Nickel, Ni	0.014%

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## B. DEFORMATION TEST OF THE CHASSIS

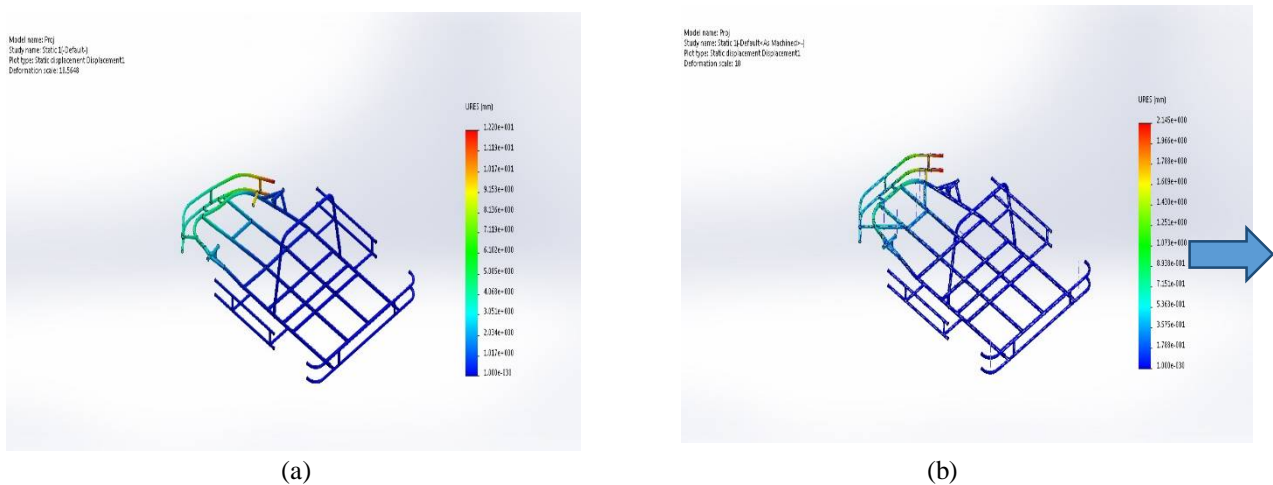


Fig 2:(a) Failed Chassis (b) Rectified Chassis

## C. FRONTAL IMPACT ANALYSIS

Generally in the case of pure elastic collision in frontal impact the linear velocity remains at 60Kmph according to ENCAP (The European new car assessment program). Hence the value of force is calculated by mass moment equation that is  $F = P \times \Delta T$

Where  $\Delta T$  is the duration of time, generally the collision takes place for a very short duration of time. We assumed this time as  $\Delta T = 1.01$  seconds. And the gross weight of the vehicle is Estimated some around ( $M=180$  KG), hence the momentum of the vehicle at 60 Km/h or 16.667 m/s that is-

$$P = m \times v$$

$$P = 180 \times 16.667$$

$$P = 3000 \text{ kgm/s}$$

And the frontal impact force i.e.-

$$F = P \times \Delta T$$

Where,  $P = m \times v$

$$P = 180 \times 20$$

$$P = 3600 \text{ kgm/s}$$

And the rear impact force-

$$F = P \times \Delta T$$

$$F = 3000 \times 1.10$$

$$F = 3960 \text{ N}$$

Now the calculated force were placed on the frontal part of frame by keeping the rear part fix on Ansys16 and the result along with the image as

## D. SIDE IMPACT ANALYSIS

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

A side impact analysis is very necessary and the boundary conditions are taken relating to the real life situation that a go kart will experience. 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. For the side impact the velocity of vehicle is taken 72kmph or 20m/s according to ENCAP Standard and then the force is calculated i.e.-  $F = P \times \Delta T$

Where,  $P = m \times v$

$$P = 180 \times 20$$

$$P = 3600 \text{ kgm/s}$$

The side impact force –

$$F = 3600 \times 1.10$$

$$F = 3960 \text{ N.}$$

Hence the calculated force were placed on one side of the modal of frame while keeping another side fixed and the stresses were simulated the image is shown as-

## E. REAR IMPACT ANALYSIS

The rear impact force is also calculated in the same way as remaining two. In this case the velocity of collision was taken 72kmph or 20 m/s by the calculations and also as according to the ENCAP standards.

The calculations are as-

Where,  $P = m \times v$

$$P = 180 \times 20$$

$$P = 3600 \text{ kgm/s}$$

And the rear impact force-

$$F = P \times \Delta T$$

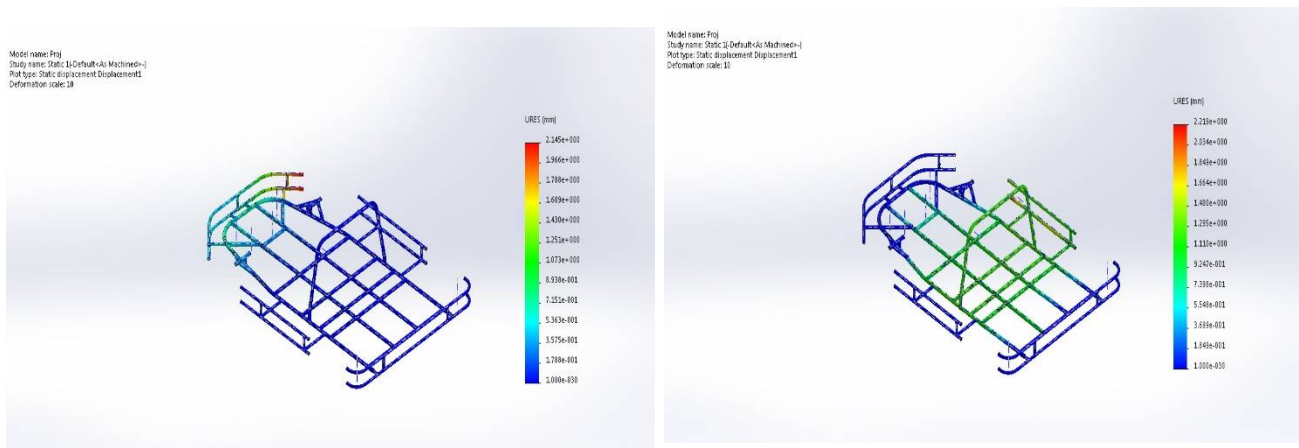
$$F = 3000 \times 1.10$$

$$F = 3960 \text{ N}$$

Hence the calculated value of the rear impact force was placed on the rear part of the frame while keeping the frontal part fixed.

## IV. SIMULATION & RESULTS

### A. RESULTS



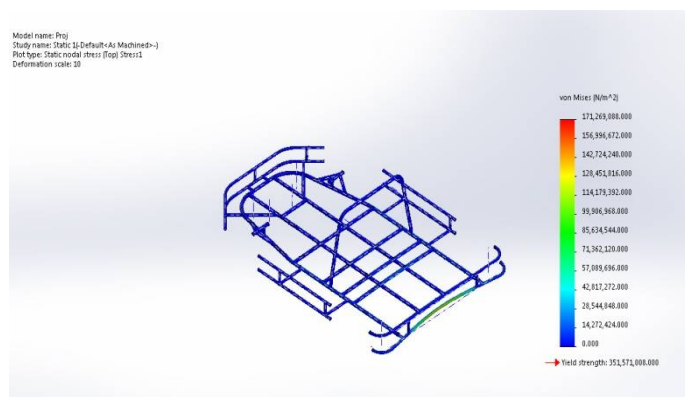
(a)

(b)

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016



(c)

**Fig 3:** (a) Frontal Impact Analysis, (b) Side Impact Analysis, (c) Rear Impact Analysis

**Table IV: Load & Constraints**

	Front Impact	Side Impact	Rear Impact
<b>Constraints</b>	Rear Bumper Fixed	Opposite Bumper Fixed	Front Bumper fixed
<b>Load Given</b>	3960N	3960N	3960N
<b>Max. Deflection</b>	2.145mm	2.249mm	2.170mm

**Table V: Analysis Details**

Content	Description
Study Name	Static 1
Analysis type	Static
Mesh type	Shell mesh

## V. STEERING SYSTEM

### A. OBJECTIVE

The steering system is designed to withstand the stress of safely maneuvering the vehicle through any type of possible condition at the time of driving. The purpose of the steering system is to provide directional control of the vehicle with minimum input. The main goal for steering is to have steering radius of 4m or less and to have 100% Ackerman steering.

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## B. DESIGN

Simplicity and safety were the main design specifications for the vehicle's steering system. While designing the steering system the constraints that we possessed were center alignment of steering system.

Table VI: Steering Specifications

Description	Specification
Inner turning angle	30 deg
Outer turning angle	22 deg
Turning radius	2.5m
Castor angle	12 deg
Camber angle	0 deg
King pin inclination	7 deg
Tie rod length	0.36m
Steer wheel diameter	11 inch

The formulae's used for steering calculation are:

$$R = \frac{d}{2} + L \cos \left( \frac{A+B}{2} \right)$$

$$\% \text{ Ackerman} = \frac{A-B}{\tan^{-1} \left( \frac{1}{\tan(B)-1} \right) - B}$$

Where,

**R** is the turning radius,

**L** is the length of the cart,

**A** is the angle of the inside angle of the wheel

**B** is the angle of the outside wheel

**d** is the width of the car.

To determine the Ackerman percentage, equation (2) was used. Given that, 100% Ackerman angle is desired, A at 30-degrees and B at 22-degree was the best option.

This gave a turning radius of 3.2m.

Inner and outer turning angle is calculated by the formulae-

Outer angle-

$$\tan A = L / (R - d/2)$$

Inside angle-

$$\tan B = L / (R + d/2)$$

Caster angle is the most important factor governing how the kart will handle. It will make the kart more stable in rough condition and the kart's straight line stability will also be improved. King pin inclination is used to making a steering tend to return to the straight ahead or centre position. If kingpin is incline at 12 degree, it gives self centering effect and leads to less steering effort.

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

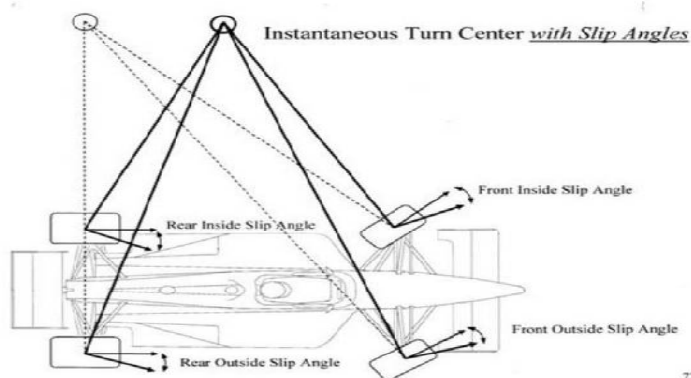


Fig 4:Steering Geometry

Therefore our **kart does not skid** because the inside front wheel is angled just a little more than the outside front wheel.

Table VII: Steering Design Considerations

CONSIDERATION	PRIORITY	REASON
Simple design	Essential	Minimize weight to maximize power to weight ratio of car
Low steering ratio	Essential	Quick steering Response
Ackerman geometry	High	To avoid skidding
Minimize bump steer	Desired	To conserve Momentum

## VI. DRIVE-TRAIN

### A. OBJECTIVE

The drive-train is a very important part of the racing cars, taking into consideration that all of the car's power is transferred through the drive-train system to the ground. The challenge is to harness the engine's 3.5 basic horsepower (BHP) and distribute it to the ground in the most efficient way. The drive-train needs to be able to operate in the lowest and highest gear ratios while performing in all of the different aspects of the competition. The goal of the drive train is to transfer power from the engine of the vehicle to the wheels. The power transferred must be able to move the vehicle. Acceleration is also an important characteristic controlled by the drive train. There are several different methods of power transmission that have been used in cars. And also works on infinite no. of gear ratios according to the speed of engine.

### B. ENGINE:

Briggs and Stratton 550 Series runs cooler and cleaner, delivering more power, smoother operation, longer engine life and improved fuel economy. Excellent performer for mid-duty applications. An all-round great performer, the 550



## International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

series line of engines utilize unique overhead valve (OHV) technology to provide solid performance, cleaner operation and longer life.

- Dura bore cast iron cylinder sleeve withstands wear and abuse to provide improved oil control and extended life.
- Gear driven duraLube splash lubrication system continuously supply oil to all internal parts magneto electric ignition system delivers quick and dependable start with no maintenance required.
- Maxi- clean automotive type pleated paper air filters provide superior protection for longer engine life.
- Super Lo-Tome muffler reduces noise levels and improves toner quality.
- Overhead valve design (OHV) runs cooler and cleaner to deliver more power, longer life and improved fuel economy.
- Dual Ball Bearings helps reduce engine and part wear for extended life.

### C. ENGINE SPECIFICATIONS

Power,  $P = 2.6 \text{ KW}$

Where,  $P = 2 \pi NT/60$

Max Torque =  $7.52 \text{ Nm @ } 3300\text{rpm}$

Gross torque =  $3.5\text{Hp} * 5252 \text{ (const.)} / 3060\text{rpm}$

Where, Gross Torque =  $5.5 \text{ ft-lbs @ } 3060\text{rpm}$

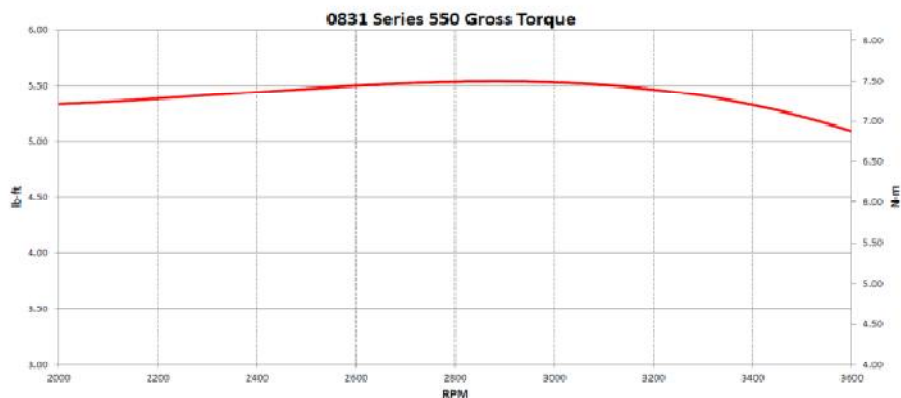


Fig 5: Engine Torque Curve

### D. CVT TRANSMISSION SYSTEM:

#### 6.4.1 Chain - Sprocket: No.1 (B/w Motor and CVT)

No of teeth in motor sprocket: 17 teeth

No of teeth in Pinion: 13 teeth

Ratio: 0.76

#### 6.4.2 Chain - Sprocket: No.2 (B/w CVT and Rear shaft)

No of teeth in Pinion and CVT sprocket: 13

The CVT has a centrifugal clutch that engages at 1400 rpm.

#### 6.4.3 Calculations

1. Motor RPM: 1320 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM = 1737 RPM

## International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

CVT Ratio at 1737 RPM: 22.11

Power shaft rpm = 1737/22.11 = 78.56 RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 78.56\*3.14\*0.4128\*60\*10<sup>-3</sup> = 6.1 Km/h

2. Motor RPM: 1680 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM = 2210.52 RPM

CVT Ratio at 2210.52 RPM: 16.71

Power shaft rpm = 2210.52/16.71 = 132.28 RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 132.28\*3.14\*0.4128\*60\*10<sup>-3</sup> = 10.29 Km/h

3. Motor RPM: 2100 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM=2763RPM

CVT Ratio at 2763 RPM: 10.80

Power shaft rpm = 2763/10.80 = 255.833 RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 255.833\*3.14\*0.4128\*60\*10<sup>-3</sup> = 19.9 Km/h

4. Motor RPM: 2280 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM=3000RPM

CVT Ratio at 3000 RPM: 8.997

Power shaft rpm = 3000/8.997 = 333.44RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 333.44\*3.14\*0.4128\*60\*10<sup>-3</sup> = 25.93 Km/h

5. Motor RPM: 2520 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM= 3315.78RPM

CVT Ratio at 3315.78 RPM: 6.584

Power shaft rpm = 3315.78/6.584 = 503.61 RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 503.61\*3.14\*0.4128\*60\*10<sup>-3</sup> = 39.17 Km/h

6. Motor RPM: 3600 rpm

Wheel diameter: 0.4128m (16 inch)

Primary Reduction = 0.76

CVT input RPM= 4736.84 RPM

CVT Ratio at 4736.84 RPM: 6.33

Power shaft rpm = 4736.84/6.33 = 748.31 RPM

Speed (Km/h) = Power shaft rpm \*  $\pi$  \* Wheel Dia \* 60 /1000  
= 748.31\*3.14\*0.4128\*60\*10<sup>-3</sup> = 58.2 Km/h

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## VII. BRAKING SYSTEMS

### A. OBJECTIVE

The purpose of the brakes is to stop the car safely and effectively. In order to achieve maximum performance from the braking system, the brakes have been designed to lock up rear wheels, while minimizing the cost and weight.

### B. DESIGN

The brake system design includes the double disc at the rear axle to stop the vehicle. It is mounted in the one third part position of the axle with opposing the position of drive train sprocket hence also enables the good balancing requirement. Master cylinder is used at the front near the brake pedal providing the occupant to easily accessible space. The braking effort is 60% on the front. A proper master cylinder bore size was found by doing brake calculations based on the mass, center of gravity, master cylinder volume, weight transfer when the brakes are applied. The construction of the brake is rear hydraulic split. It is actuated hydraulically and the caliper that is used is floating type. The brake calculations are as follows.

**TABLE VIII: BRAKE DESIGN CONSIDERATIONS**

Considerations	Priority	Reason
Simplicity	High	Overall Goal of the Vehicle
Performance	High	Capable of Stopping Vehicle
Lightweight	High	Prevent air bubble within the brake lines
Reliability	Essential	Lightweight parts to minimize total weight
Ergonomics	Essential	Optimal pedal assembly fitment to suit every driver

When the brakes are applied the weight is transferred from the rear to the front. There will be a reaction force and inertia force acting on the wheels due to the weight of the vehicle. So there will be an overturning couple acting at the center. By resolving the horizontal and vertical components acting at the wheels and the center of the vehicle (In our case the brakes are applied at the rear only). So the retardation of the vehicle is calculated.

$$\alpha = g \left[ \frac{\mu(b-x)}{b+\mu h} \right]$$

Where b is the Wheelbase

The brake efficiency is considered to be 100% theoretically. In practical it is 50% brake efficiency is given by the foot and 30% hand brake. For calculating the retardation of the vehicle we have taken the car weight as 150kg from the Solid works modeled design and wheel base of 115cm

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

The center of gravity of the vehicle is at

$x = 0.573$  m - from front of the rear axle

$h = 0.3$  m - above the ground level.

Coefficient of friction,  $\mu = 0.6$  (in limiting case)

Retardation of vehicle,  $\alpha = 9.3$  m/s<sup>2</sup> (from the above weight transfer formula)

Condition: For velocity of vehicle,  $v = 60$  km/h = 16.67 m/s – The stopping distance and the stopping time calculation is as follows

Stopping Distance – formula taken from the laws of motion

$$S = \frac{v^2}{2\alpha} = 14.28 \text{ m}$$

Stopping Time - formula taken from the laws of motion =  $u/a$

$$t = 1.7 \text{ s}$$

From Newton's law of motion, we take

Braking force required for the vehicle,  $F = m \cdot \alpha$

Therefore,  $F = 150 \times 9.3 = 1395$  N

And the required Braking torque is calculated using,

Brake Torque = Brake Force  $\times$  Effective radius of rotor  $\{ (D + d)/4 \}$  for dry disc

B.T. = 64.7 Nm

Force applied by the driver on the pedal is taken as 100 N (app.) and the pedal ratio is 4:1. The area of the piston of the master cylinder is 1.

Therefore,

Brake pressure = pedal ratio  $\times$  pedal force / area of the cylinder piston

B.P = 5.5 Mpa

Appropriate Rotor(Disc), Master Cylinder and Caliper are Chosen nearer the calculated values.

**Table IX: Brake Technical Specifications**

Description	Specifications
Disk Outer Diameter	16 cm
Disk Inner Diameter	2.54 cm
Thickness of Disk	0.5 cm
Brake Pedal Force	10 N
Pedal Ratio	4 : 1
Coefficient of Friction	0.60
Brake line pressure	5.5 MPa
Brake Torque	64.7 Nm
Stopping time	2.8 sec
Stopping distance	14.28 m

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## **C. TYRESANDSELECTIONPARAMETERS**

TYRE DIMENSIONS used in the vehicle

**11/7.10-5 (rear)**

**10/4.50-5 (front)**

The type of tire that we are going to use is a radial tire (more properly, a radial-ply tire) is a particular design of vehicular tire (in British English, *tire*). In this design, the cord plies are arranged at 90 degrees to the direction of travel, or radially (from the center of the tire).

## **D. CONSTRUCTION PARAMETERS**

With only radial cords, a radial tire would not be sufficiently rigid at the contact with the ground. To add further stiffness, the entire tire is surrounded by additional belts oriented closer to the direction of travel, but usually at some "spiral" angle. These belts can be made of steel (hence the term steel-belted radial), polyester, or Aramid fibers such as Twaron or Kevlar. In this way, low radial tires separate the tire carcass into two separate systems: The radial cords in the sidewall allow it to act like a spring, giving flexibility and ride comfort. The rigid steel belts reinforce the tread region, giving high mileage and performance. Each system can then be individually optimized for best performance. Radial tires have different characteristics of springiness from those of bias-ply tires, and a different degree of slip while steering. The steel wires in radial tires become magnetic with use, and as they rotate, an alternating magnetic field is created. It is quite measurable with an EMF meter close to the wheel well when the wheel is rotating and is rich in harmonics up to several hundred hertz.

## **VIII. SUPPORT COMPONENTS AND PARAMETERS**

### **A. SEAT AND SITTING ARRANGEMENT**

The seat in this kart is also designed to be very light it is very simple made of plastic material and is attached to the chassis by four points only and can be adjusted in angle of back rest according to the requirement of the drivers comfort the back side angle of the seat is at 17 degrees which is the good position of the drivers body rest according to the ergonomics point of view and is kept almost parallel to the fire wall .the seat implemented in our go kart provides a good combination of weight reduction and ergonomics.

### **B. ASTHETICS**

Aesthetically, the design is improved by the use of more rounded corners than the straight. The unique use of rounded corners allows for a more pleasing look to the vehicle's body as well as a reduced number of welded joints. The use of continuous bended pipes also reduced the no of joints the lack of sharp edges on the chassis allows for the design of more streamlined body panels which not only look smoother, but may also have a positive effect on the overall aerodynamic drag forces.

### **C. GROUND CLEARANCE**

Standard ground clearance is given for our vehicle such that it prevents any part of vehicle touching the ground (except wheels). Also the ground clearance is taken from the lowest point under the vehicle.

**Ground clearance of the vehicle: 75 mm (2.9 inches)**

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## D. JACK POINT

The vehicle contains two jack points situated at the front and rear end of the chassis which has 10 inches of steel tube painted with red color

## E. BALLY PAN (FLOOR TRAY)

The bally pan is placed just behind the brake and accelerator pedals as per the rulebook to prevent any foreign particles from entering into the driver's area as well as in entire vehicle.

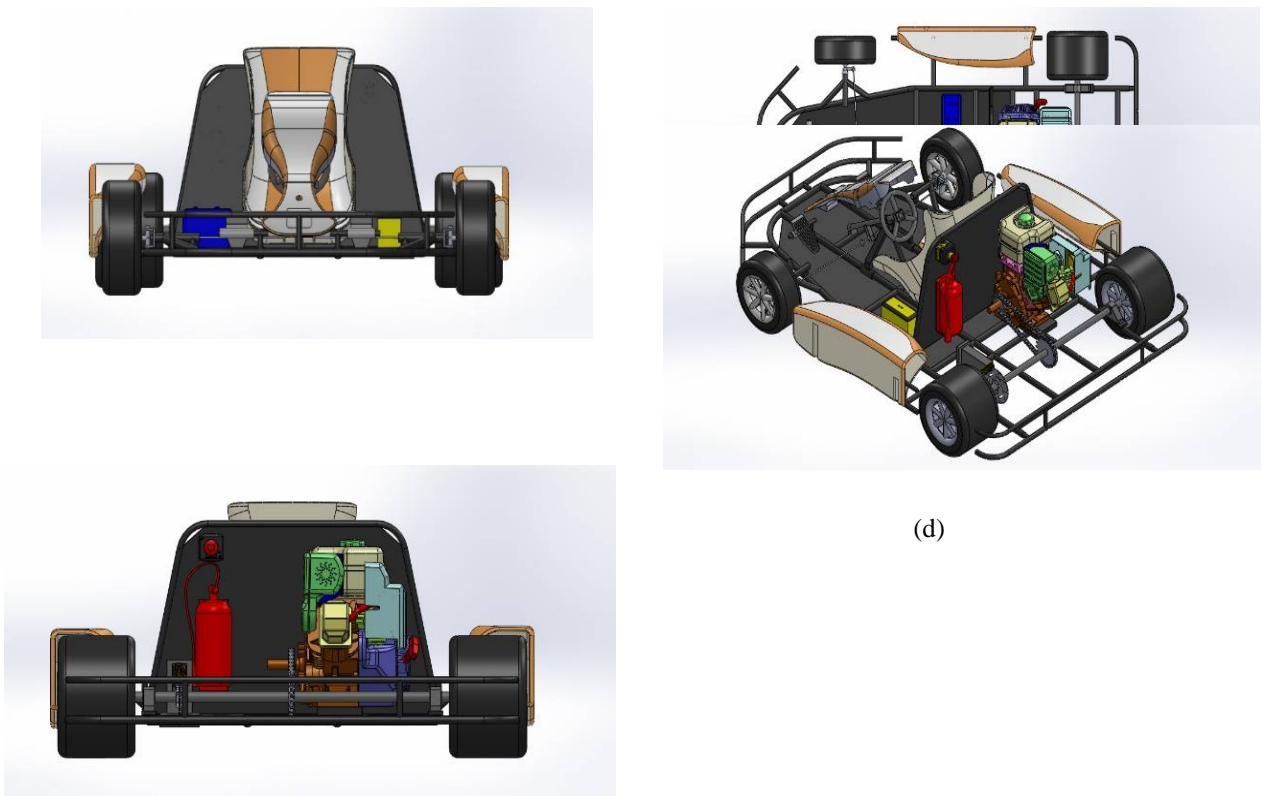
## F. ERGONOMICS

Ergonomically the kart is build up with proper dimensions. The driver seat is well cushioned and 3 inches away from the firewall. Alignment of the driver seat/driver sitting direction is parallel to vehicle's longitudinal axis. The driver is well sophisticated to drive a kart without any disturbance.

## G. KILL SWITCH

Two Kill switches are provided in our vehicle as a safety to the in case of emergency. One in the reach of the driver and the other for the reach of a Third person.

## IX. FINAL 3D VIEW



**Fig 6:** (a)Front View(b) Top View(c) Rear View(d) Orthogonal View

# International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 11, November 2016

## REFERENCES

- [1] Barnard, R. H., Road vehicle aerodynamic design - an introduction, *MechAero Publishing*, 2001.
- [2] Richard Stone and Jeffrey K. Ball, "Automotive Engineering Fundamentals", *Society of Automotive Engineers*, ISBN 0-7680-0987-1
- [3] L. Chu, et al., "Coordinated Control of Electronic Stability Program and Active Front Steering," *Procedia Environmental Sciences*, vol. 12, Part B, pp. 1379-1386, 2012.
- [4] Z. Gao, et al., "Dynamic Modeling and Steering Performance Analysis of Active Front Steering System," *Procedia Engineering*, vol. 15, pp. 1030-1035, 2011.
- [5] Carroll Smith, "TUNE TO WIN", *Aero Publishers, INC. 1978*, ISBN 0-87938-071-3
- [6] Warren J. Rowley "Introduction to Race Car Engineering", Book-1, 3<sup>rd</sup> Edition. ISBN-9780-9734-3200-8
- [7] R.S.Khurumi, J.K.Gupta., "A Textbook Of Machine Design", *EURASIA PUBLISHING HOUSE.(PVT).LTD* Revised edition 2014, ISBN-978-81-219-2537-2.
- [8] N.Kurumi, R.S.Kurumi., "STRENGTH OF MATERIALS (mechanics of solids)", *S.CHAND & COMPANY PVT.LTD.* Revised edition 2013, ISBN-81-219-2822-2
- [9] Ibrahim Zeid "Mastering CAD/CAM", *McGraw Hill Education(India) Pvt Limited.* Revised edition 2013, ISBN-9780-0712-3933-2
- [10] S.S.Rattan, "THEORY OF MACHINES" Fourth Edition., *Hill Education(India) Pvt Limited*, Revised edition 2014, ISBN-978-93 5134-347-9
- [11] PSG College Of Technology "DESIGN DATA Data book of Engineers", *Kalaikathir Achchagam, Coimbatore*, ISBN-978-81-927355 04.
- [12] Richard Stone and Jeffrey K. Ball, "Chassis Design; Principles and Analysis" *Society of Automotive Engineers*, ISBN 978-0-7680 0826-5676, pp.