A Research Study Of Roll Cage And Braking System For Go Kart

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Abstract-In this paper includes study of roll cage and braking System for Go kart. Go kart is small four wheeled vehicle without suspension or differential .It is a vehicle which is simple, light weight and compact and simple to operate. Roll cage or chassis specially design such way that light weight, strength durable AISI 1018 mild carbon steel for offers a good balance of mechanical properties like toughness and ductility.

Braking system is a most important part of go kart in system stopping the rotation of wheel form of rotation. to achieve best perform from brake system the brakes uses lock up rear, while minimum cost and light weight

Index Terms-Go-Kart, chassis, AISI 1018, Braking system, racing

INTRODUCTION

The first in America (U.S.A) manufacture kart in GO Kart manufacturing Co. Company in 1957 and in 1958 much colic was first produce engines for kart. Its first engine called mcculloch MC10 was an inspire form chain-saw two stroke engine and in 1960s, motor motorcycle engine also use for go Kart. In India in1999 karting is starts a few local level events eventually leading to the launch of national championship sports and organize. GO-Kart is a single seated four wheel vehicle build for racing and entertainment. some university and collage also encourage student activities on motor sports and organize. GO- Kart is a single seated four wheel vehicle of roll cage and braking system. Roll cage his design is made with the helps solid works software. A Framework of reinforcement protects a car passenger cabin in the events that it should roll on its roof .Roll cage is the structure consist of the various cross section tubes and tubing thickness to protect the driver .for chassis design requirement we use Discover 125cc engine and choose AISI 1018 having good strength with flexible in nature.

The brake system design includes the single 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[5] we have extensive design and carried out the regards separate parameter of disk ake system involve in GO kart

1 Roll cage of GO-KART

A **Roll cage** is a specially engineered and constructed frame built in (or sometimes around, in which case it is known as an (exo cage) the passenger compartment of a vehicle to protect its occupants from being injured or killed in an accident, particularly in the event of a rollover.

1.2DESIGN OBJECTIVES OF CHASSIS

1. Provide full protection of the driver, by obtaining required strength and torsional rigidity, while reducing weight through diligent tubing selection

2. Design for manufacturability, as well as cost reduction, to ensure both material and manufacturing costs are competitive with other Go-Karts.

3. Improve driver comfort by providing more lateral space in the driver compartment

4. Maintain ease of serviceability by ensure that chassis members do not interfering with other subsystems

1.3 Chassis construction:

The chassis of a Go-Kart consists of following components suitably mounted:

i. Engine
ii. Transmission system, consisting of the chain sprocket, rear axle.
iii. Road wheels.
iv. Steering system.
v. Brake.
vi. Fuel tank.

All the components listed above are mounted on the conventional construction, in which a separate frame is used and the frameless or unitary construction in which no separate frame is employed.

1.**3 Function of the Frame**:

1) To support the chassis components and body.
2) To withstand static and dynamic loads without undue deflection.
4.3 Loads on the frame:
1) Weight of the vehicle and the passenger, which causes vertical bending of the side members.
2) Vertical loads when the vehicle comes across a bump or hollow, which results in longitudinal torsion due to one wheel lifted (or lowered) with other wheels at the usual road level.
3) Loads due to road chamber, side wind and cornering force while taking a turn, which results in

lateral bending of side members.
4) Load due to wheel impact with road obstacles may cause that particular wheel to remain obstructed while the other wheel tends to move forward, distorting the frame to parallelogram shape.
5) Engine torque and braking torque tending to bend the side members in the vertical plane.
6) Sudden impact loads during a collision, which may result in a generable collapse.

1.4 Frame Design:
The frame is designed to meet the technical requirements of competition the objective of the chassis is to encapsulate all components of the kart, including a driver, efficiently and safely.
Principal aspects of the chassis focused on during the design and implementation included driver safety, drive train integration, and structural weight, and operator ergonomic. The number one priority in the chassis design was driver safety. By the competition rules and Finite Element Analysis (FEA), the design assured

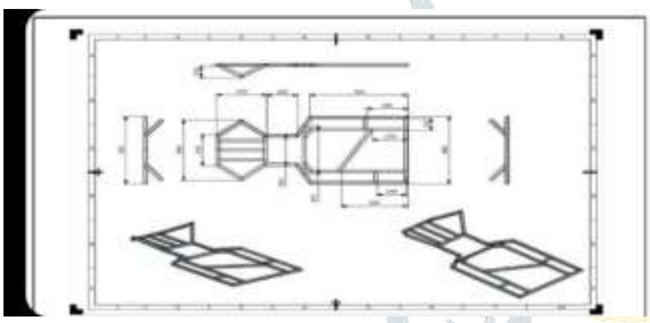

Fig a..CAD model of chassis

Table 1. Chassis Dimensions

PARAMETERS	VALUE
Vehicle length	1568mm
Vehicle width	1388mm
Roll cage material	AISI 1018

1.5 Material use and its composition:

The roll cage for GO-KART used material is considered depending upon the various factors such as maximum load capacity, absorption force capacity, strength, rigidity. The material selected for the roll cage building is AISI 1018. AISI 1018 is a mild/low carbon steel AISI 1018 carbon steel is a free machining grade that is the most commonly available grade around the world. Although its mechanical properties are not very unique, it still can be easily formed, machined, welded and fabricated.

1.5.1Chemical composition:

Table no 2
AISI 1018 Chemical composition:

COMPOSITION	AISI 1018
Iron (Fe)	98.81 to 99.26%
Manganese (Mn)	0.60to 0.90%
Carbon ©	0.15 to 0.20%
Sulfur (S)	0 to 0.050%
Phosphorus (P)	0 to 0.040%

1.5.2Physical properties

Table no 3
AISI 1018 physical properties

PROPERTIES	AISI 1018
Density	7.87 g/cm^3
Elastic (Young's, Tensile) Modulus	210 *GPa*
Elongation at Break at 50 mm	15%
Poisson's Ratio	0.29
Tensile Strength: Ultimate (UTS)	430 to 470 *MPa*
Bulk Modulus	159 GPa
Yield strength	264 MPa
Thermal conductivity	51.9 W/m-k

1.6 Structural Rigidity

Overall frame structural rigidity is important to enhance the capabilities of a 4-wheeler vehicle. To measure the overall frame rigidity, tensional rigidity analysis was conducted through CAD. The objective of the tensional rigidity analysis was to manipulate the chassis design within the CAD software to increase the amount of torque per degree of chassis deflection. By theoretically increasing this value, the actual vehicle could have the ability to be more torsion-ally rigid, making it able to withstand more intensive without failure. Which is equivalent to the gross weight is calculated i.e. Gross weight = 120kg and the equivalent force that is – $F = M \times g$ $F = 120 \times 9.81$ $F = 1177.2$ N The calculated force is placed on one of the corner of the frame while other three corners were kept fixed by constraining. Hence according to the result obtained, the frame would be torsion-ally rigid

We have selected the material as AISI 1018 for chassis pipes. Yield Strength = 370 MPa12

Taking, factor of safety =2

Σ permissible = 230 MPa.

Tensile strength = 440Mpa

The point loads of 98.06N, 490N, 14N, 637N, 1470N and 117N are acting on beam at points C, D, E, F, G and H due to the steering, fuel tank, chassis, engine, driver and battery, respectively as shown below. BMD for chassis

RA and RB are the reactions at point A and point B respectively. ΣFY =0

RA + RB = 550N

Moment at point A:

ΣMA =0

10 × 100+20 × 200+50 × 535+140 × 680+300 × 750+30 × 850-1070 × RB =0 RB = 352.757N

Therefore, RA = 197.243N Now,

For bending moment diagram,

Bending moment at point A (MA) = 197.243 × 0 = 0

Bending moment at point C (MC) = 197.243 × 100 = 19724.3N-mm = 19.7243N-m

Bending moment at point D (MD) = 197.243 × 200-10 × (200-100) = 38448.6N-mm = 38.4486 N-m13

Bending moment at point E (ME) =197.243 × 535-10 × (535-100)-20 × (535 -200) = 94475N-mm = 94.475N-m

Bending moment at point F (MF) =197.243 × 680-10 × (680-100)-20 × (680-200)-50 × (680-535)

= 111475.2N-mm = 111.4752N-m

Bending moment at point G (MG) =197.243 × 750-10 × (750-100)-20 × (750-200)-50 × (750-535)- 140 × (750-680) = 109882.3N-mm = 109.8823N-m

Bending moment at point H (MH) =197.243 × 850-10 × (850-100)-20 × (850-200)-50 × (850-535)-140 × (850-680)-300 × (850-750) = 77606.6N-mm= 77.6066N-m

Bending moment at point B (MB) = 197.243 × 1070-10 × (1070-100)-20 × (1070-200)-50 × (1070-535)-140 × (1070-680)-300 × (1070-750)-30 × (1070-850) = 0

From above, the maximum bending moment (Mmax) is at point F. Therefore,

Mmax = 111.4752N-m

We take, fixed outer diameter of pipe (D) = 25.4mm Y= D/2 = 12.7mm

Moment of inertia (I) = (π/64) × (D4-d4) = (π/64) × ((25.4)4-d4) We know that,

Mmax / I = σ permissible / Y 111475.2 / I = 230 / 12.7

I = 6155.3697mm4

6155.3697 = (π/64) × ((25.4)4-d4)

From these we get, d = 23.2226mm

t = (D-d)/2 =1.0887mm

Therefore, for pipe of Outer Diameter 25.4 mm, 1.0887mm thickness is required in order to prevent bending of chassis due to point loads. But by taking into consideration the constraint given in rulebook and results of analysis of chassis in ANSYS16.0, We have selected cross section of 1.8 mm thickness.

Calculations for Torsional Stiffness:

Torsional Stiffness = G * J / 14

L Where,

G -Modulus of Rigidity

J - Polar moment of Inertia L- Length

Modulus of rigidity for given material is 80GPa

For D = 25.4mm, t = 1.8mm

The polar moment of Inertia (J) = (π/32) × (D4-d4)

= (π/32) × ((25.4)4-(21.8/4)

= 18690.3525mm4

Length (L) = 1070mm

Torsional stiffness = T/θ = (G× J)/L

= ((80×103) × 18690.3525)/1070

= 1.3974×106N-mm/rad

= 1397.4N-m/rad

Similarly, we have calculated the torsional stiffness values for different cross sections that are easily available in market. Therefore, we get following result

Calculations for Bending Stiffness:

Bending Stiffness = E * I Where,

E–Young’s Modulus I – Moment of Inertia

Young’s modulus of given material (E)=210GPa

Moment of inertia (I) = (π/64) ×(D4-d4)

= (π/64) × ((25.4)4-(21.8/4)

= 9345.1763 mm4

Bending Stiffness = (210 × 103) × 9345.1763

= 1.9625× 109N-mm2

= 1962.5N-m2

Therefore, the torsional stiffness and bending stiffness for a given cross section are 1397.4Nm/rad and 1962.5N-m2 respectively.

2 Braking system of GO-kart

Brake system is very essential for the vehicle to reduce the speed or to stop the vehicle. A GO-Kart is a very small motor vehicle with four wheels, used for racing. The brakes are required for stop the vehicle within the smallest possible distance or to slow down the vehicle when we needed. Without the brakes we cannot control the vehicle speed so it is the most important system in automobiles.

2.1 Classification of braking system

- According to the purpose –

1.Primary or service brake
2.Secondary brakes

- According to the construction
  1 Drum brake
  2. Disk brake

- According to the method of power
  1. Mechanical brakes
  2 .Hydraulic brakes
  3. Air brakes
  4. Vacuum brakes
  5. Power assisted hydraulic brakes

6 .Magnetic brakes
7 .Electrical brakes

- .By method of application:
  1. Service or foot brakes
  2 .Parking or hand brakes

- According to the operation:
  1. Manual
  2. Servo
  3.Power operation

- According to the method of Braking contact

1.Internal Expanding Brake
2.External Contracting Brakes

- According applying Brake force:
  1. Single Acting Brakes.

2.Double Acting Brakes.

2.2Selection of brake :

We are using disc brake for rear wheel considering the respective advantages, availability, and their limitations. The following reasons support the selection of disc brakes for the front and rear wheels.16 Disk brake contributes for reduction in overall weight of the vehicle. More braking torque needs to be generated by the Rear brake even after weight transfer, because the single brake has to manage the braking torque requirement of the entire rear driveshaft The brake rotor is the component where the energy transfer takes place, while braking, the kinetic energy is transferred to heat energy by the rotor thus the factor of safety and thermal stability of the brake rotor is important. Hence structural and thermal analysis of the brake rotor is analysis using ANSYS (CFD). The driver ergonomics has to be considered while designing, this project aims to provide minimum pedal effort by fixing the optimal pedal ratio. The structural analyses of the brake pedal and brake rotor hub are done using ANSYS.

2.3 purpose of brake

- To decrease the speed of a vehicle using kinetic friction and keep it from rolling when stopped using staticfriction.18

- To design a braking system which take least time to bring the vehicle to stop.

- To ensures safety of the driver. Selection of Brakes: We had used a Hydraulic Disc Brake considering the following advantages, availability, and their limitations. For selection of best braking system in go-kart you have to kept some points in your mind:

a) Hydraulic system
b) Disc brake - Appache RTR rear200mm
c) Master cylinder- Appache RTR rear maste rcylinder
d) Brake lines- Appache RTR front
e) Caliper- Appache front double piston caliper

2.4Componant Specification:

- Brake Disc 150 mm (diameter) of Yamahafascino125
- Master cylinder 2.01*10-4 m2 (area) of TVS apache
- Brake fluid Dot-3
- Caliper 26 mm (diameter) of Bajaj pulsar

2.5 calculation for braking

- Pedal force applied by driver 2799.24 N
- Pedal lever ratio 6:1
- Force on caliper 6399.62 N
- Braking torque 295.6624 N/m
- Time taken to stop the vehicle 1.7s
- Stopping distance 1.19m

2.6 Procedure and Calculations: Stopping distance = V*V/ 2*μ*g
Braking force = mass of vehicle* deceleration
Deceleration = V/ time taken
Energy dissipated = [M*{V2(initial)- V2(final)}]/2

Speed (km/hr)	Stopping Distance(m)	Braking time (seconds)
40	6.29	1.13
50	9.83	1.42
60	14.16	1.69
70	19.27	1.98
80	25.17	2.56

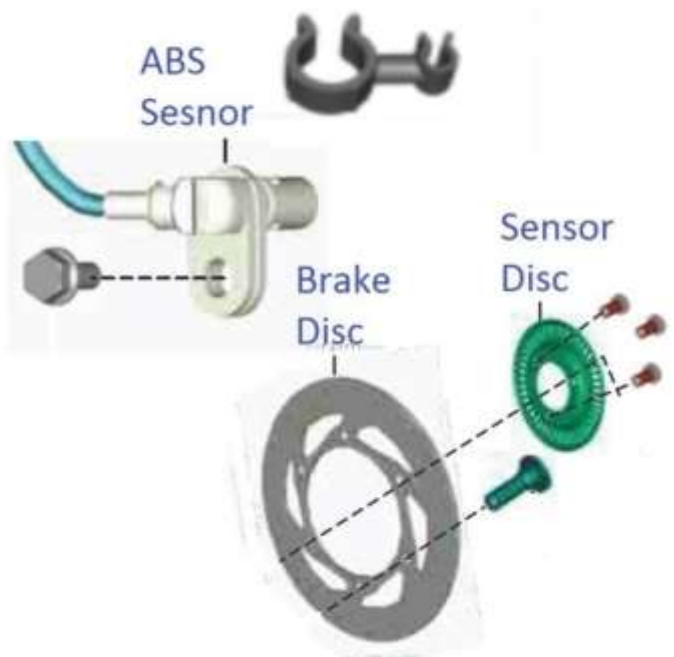

Fig. disk brake twin disk

2.7Design calculation

- Every racing vehicle requires good brakes to have control over the speed and stop the vehicle at any point of time within shortest time period and stopping distance.
- 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.
- To achieve best braking parameters in our Go Kart all the static and dynamic loads were calculated at the rear and front wheels

2.7.1 Parameters:

Mass of vehicle = 150 kg
Maximum speed of vehicle (s) = 80kmph (22.22m/s)
Tire radius (rtire):= 139.70mm
Height of C.G from ground surface (h) = 200mm
Calculation of stopping distance and time: u=initial velocity in m/s;
v=final velocity in m/s;
a=deceleration rate inm/s2;
s=stopping distance;
t=braking time;
v=0;
u=80 kmph (22.22m/s)
Assuming deceleration a=1 x g=9.81m/s2
(i.e. a=µxg; µ=the coefficient of friction between road and tyre Assuming µ=1 for asphalt and slick tires)
Using Newton's 3rd Kinematic equation, v2=u2+2as s=25.17m
Using Newton's 1st Kinematic equation
v = u + at t=2.56sec The stopping distances and braking time while stopping the kart to rest from various speeds are listed in following table

Table no 1

2.7.2 Dynamic weight

Dynamic weight transfer is the amount of change of vertical loads of the tires due to longitudinal acceleration imposed on Centre of Gravity of the car. Total static mass of vehicle including driver =150 kg (Static weight distribution is assumed to be 40%front and 60% rear)
Height of C.G from the ground surface (h) =200mm Wheelbase (b) =1070mm (Fs)F =Static weight on front wheels = 150 x 9.81 x 0.4 = 588.6N
(Fs)R =Static weight on rear wheels = 150 x 9.81 x 0.6 = 882.9N
Fd = Force due to dynamic weight transfer
Fd =275.05 N

Total dynamic weight on Front axle (Fd)F=(Fs)F+FD = 588.6 + 275.05 = 863.65N Total dynamic weight on Rear axle (Fd)R=(Fs)R-FD=882.9 - 275.05=607.85N

Master cylinder & caliper selection:

As only one disc is used on rear rigid driving axle weight on only rear wheels is considered. Maximum driving torque without slipping condition is:
Tf = (Fd)R× µ× r tire = 607.85 x 1 x 0.1397

=84.9N-m

Single piston caliper is suitable for required braking torque. Available single piston caliper in market is rear Brake Caliper of Bajaj Pulsar 220. The diameter of that Caliper is 32mm.

As there is only one disc used for braking the rigid driving axle, Master Cylinder (MC) is preferred over Tandem Master Cylinder (TMC) to avoid blocking extra outlet port.

2.7.3 Design of Braking circuit:

We found the force applied by foot of driver by using spring balance to be 150N on an average. We have assumed a pedal ratio of 6 :1 to achieve an optimum pedal travel and minimum driver effort

Braking circuit:

Pedal force applied by driver (Fa)	= 150N
Brake Pedal Ratio	= 6 : 1
Force applied to master Cylinder (F1)	= 900N
Diameter of Master cylinder piston (D1)	= 12mm
Area of master cylinder (A1)	= 11.31×10-5 m2
Brake fluid Pressure (P) = F1 / A1	= 79.57bar
Diameter of caliper piston(D2)	=32mm
Brake caliper piston area (A2)	= 8.042×10-4 m2

According to Pascal's Law, Assuming pressure is transmitted without any loss to the Caliper Force Transmitted to Brake caliper (Fc)=PxA2 =6399.02N
Total force transmitted to brake pad(Fp)=2×Fc =12798.038N Braking Torque (Tb) = Fp × µpad& disc× reffective
=12798.038×0.33×(0.07)

Where reffective = radius at which pad apply force on disc

Tb = 295.63 N-m

Hence braking torque is greater than the required frictional torque So wheels will get locked. The brake pedal has an adjustment of 2 pedal ratios as 4.5:1 and 3:1.
Hence, according to driver feedback, pedal ratio can be adjusted, depending upon the track conditions and dynamic events.

2.7.4 Rotor Specifications:

We are going to manufacture disc of our own design. Specifications of disc are:
Material = EN8
The reason behind selecting EN8 is to obtain high strength, specific heat, better heat transfer and
good thermal conductivity of the disc.
Outer dia = 150 mm
Effective radii = 75 mm
Thickness =4 mm
PCD= 80mm (3 bolts of dia 8)
Brake rotor and Hub
Design of Rotor Hub:

Material:EN8
For EN8 σyt=323.23 MPa
Torque to be transmitted (Mt)= Torque due to traction
= μ x m x g x 0.6 x radius of tire
= 123.3411 N-m
=123341.1 N-mm
Fixing the hub length as 30mm,
Shaft diameter = 25mm24
Hub diameter (dh) = 41mm
Thickness of flange (t) = 6mm
Force on hub=Mt / radius of hub
=123341.1/20.5
=6016.64N
Shear stress=F / A
=6016.64/ (π x dh x t)
=6016.64/(3.14 x 41 x 6)

Brake Pedal

2.7.5 Pedal Calculation:

The force applied by foot of driver while braking is assumed to be 150N. The value is found
out by taking a number of readings of force actually applied by our driver on spring balance and
taking the average.
Force applied by foot of a driver on pedal (N) = Average reading in spring balance x 9.81 Now
the pedal ratio is assumed to be 5.5 : 1
Force on master cylinder = Pedal ratio x Force on pedal by foot of driver
= 5.5 x 150
= 825 N
This force generates a pressure of 41.65 bar and a braking torque of 154.77Nm in the
respective hydraulic circuit to lock the wheels. Required force on pedal for generating 123.34
Nm torque to lock the wheels is 120N. So, from above calculations application of braking force
is in comfort of driver.
Selecting FOS = 2 D3 = 4329.638
D = 16.253 mm
Calculation by Torsion for finite life Sse =0.577 x Se
S se = 99.643
D
3 = 16 x Mt / (π x Sse / FOS)
= 12606.804
D = 23.2007
Hence by using design for fluctuating load, the diameter obtained is 23.2007mm

Conclusion :

Result concluded that the 150cc Engine For GO-Kart was finally design and analysis we found Some factors About It AISI 1018 are one suitable material for Go- Kart chassis material also suitable for large scale production. we found the various deformation result by analysis of using CATIA and Auto-CAD Software.The design of braking system ensure driver safety precaution calculation found the stress on disk brake less than yield stress of 210GPa thus the safety on brake hub is ensured. The paper gives overall information about Roll cage and braking system.

Acknowledgment:

The team expresses its sincere gratitude to **Mr. Hemant P. Gawade** Professor (Mechanical) for their continuous support and encouragement. We are also thankful to our faculty advisor **Mr. Rupesh B. Fonde** (R&D) for the valuable advice and supervision.

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