IJEAST

INTERNATIONAL JOURNAL
OF ENGINEERING APPLIED SCIENCE
AND TECHNOLOGY

VOLUME : 1 ISSUE : 9 Print / Issue Publication Date: 09-Oct-2016

ISSN : 2455-2143

Indexed In

WWW.IJEAST.COM

editor@ijeast.com

International Journal of Engineering Applied Sciences and Technology, 2016
Vol. 1, Issue 9, ISSN No. 2455-2143, Pages 95-102
Published Online July – August 2016 in IJEAST (http://www.ijeast.com)

DESIGN REPORT OF A GO KART VEHICLE

Shaik Himam Saheb, RavSandeep Kumar Kona, Md. Hameed
Assistant professor, Department of Mechanical Engineering
Guru Nanak Institute of Technology,
Hyderabad

ABSTRACT - This paper concentrates on explaining the design and engineering aspects of making a Go Kart for student karting championship 2015. This report explains objectives, assumptions and calculations made in designing a Go Kart. The team's primary objective is to design a safe and functional vehicle based on rigid and torsion free frame. The design is chosen such that the Kart is easy to fabricate in every possible aspect

I. INTRODUCTION

We approached our design by considering all possible alternatives for a system and modeling them in CAD software SOLIDWORKS and subjected to analysis using ANSYS based on analysis result, the model was modified and retested and a final design was fixed. The design process of the vehicle is based on various engineering aspects depending upon

- **Safety and Ergonomics**
- **Market Availability**
- **Cost of the Components**
- **Safe Engineering Practices**

With this we had view of our Kart. We set up some parameters of our work and team has been divided into core groups.

- **Design**
- **Engine and Transmission**
- **Steering**
- **Brakes and wheels**
- **Business and Management**

II. LITERATURE REVIEW

Being a new team we required a clear idea of basic requirements, parameters and design of Go Kart. We made a detailed study on Go Kart and visited Runway9, Hyderabad a Famous Go Karting Spot. We gained more knowledge during our field study and our basic doubts on dgn were cleared.

III. DESIGN OF KART

The following design methodology was used during design:

- Requirements
- Design calculations and Analysis
- Considerations
- Testing
- Acceptance

IV. MATERIAL SELECTION

IS1161 is selected for the chassis because of the following reasons

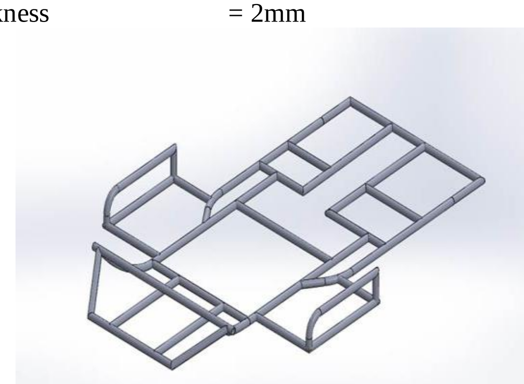
- Machinability
- Weld ability
- Availability

Properties Of IS1161 Seamless Tube

Properties	Metric
Ultimate tensile strength	415Mpa
Yield tensile strength	310Mpa
Poisson's ratio	0.3

V. DIMENSIONAL SPECIFICATIONS

Round tube of dimension = 25.4mm OD
Thickness = 2mm



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VI. JUSTIFICATION

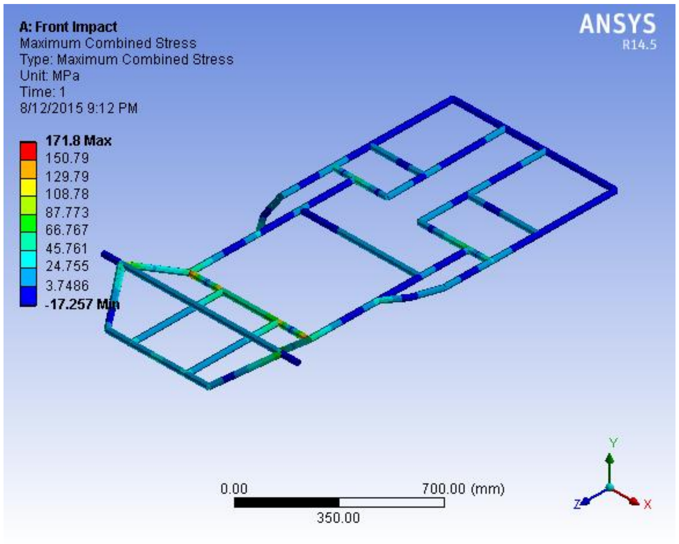
Round hollow tubes are light in weight

VII. FABRICATION PROCESSES

Lathe Work, Cutting, Drilling, Milling, Shaping , Grinding, Polishing, Finishing, Welding

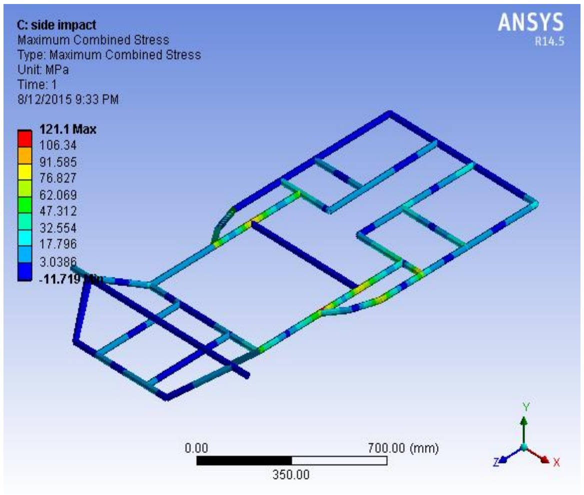
Front Impact Analysis

Using the gross weight of the vehicle is 160kg
The impact Force was calculated based on G-load of 5



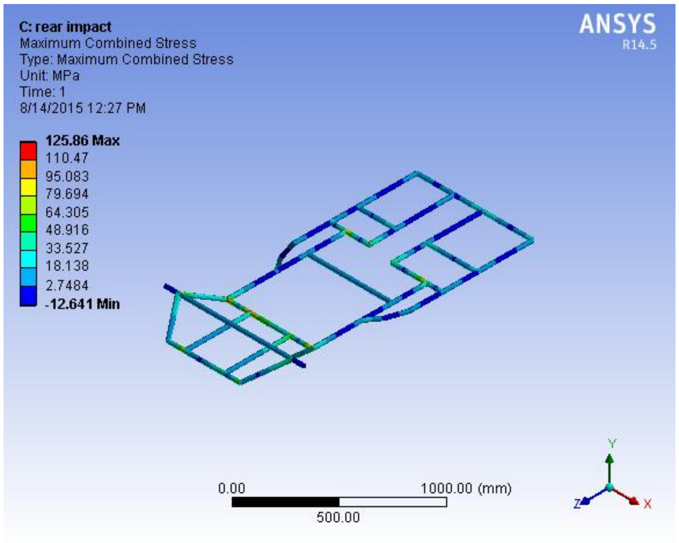
Load applied: 5000 N
Maximum stress induced: 171.8 Mpa
Displacement: 1.9mm
Factor of safety: 2.2

Side Impact Analysis



Load applied: 3500N
Maximum stress induced: 121.1Mpa
Displacement: 1.14mm
Factor of safety: 2.5

Rear Impact Analysis



Load applied: 2500N
Maximum stress: 125.86Mpa
Displacement: 6mm
Factor of safety: 2.4

Wheel Hub Analysis

The following are the considerations took for the design of wheel hub for 25.4 mm shaft:
Material: Mild Steel
Braking torque: 975N-m

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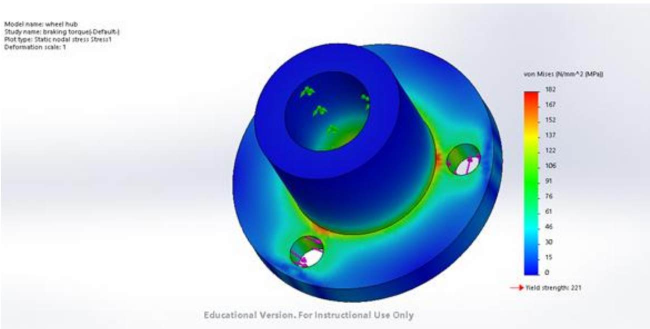
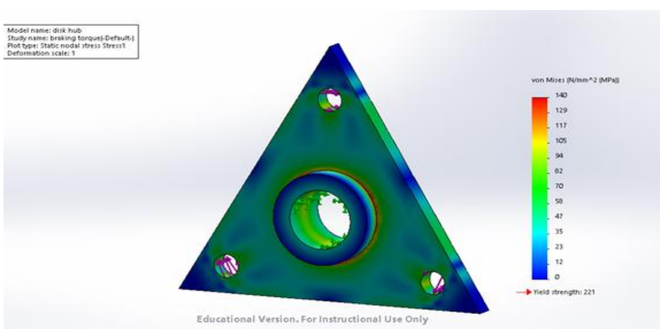
Load: 1500N



Maximum stress : 182Mpa
Yield strength: 221Mpa
Factor of safety: 1.21

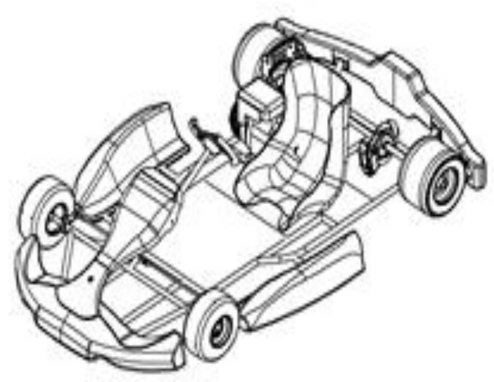
Disc Hub Analysis

The following are the considerations took for the design of wheel hub for 25.4 mm shaft:
Material: Mild Steel
Braking torque: 975N-m
Load: 1500N

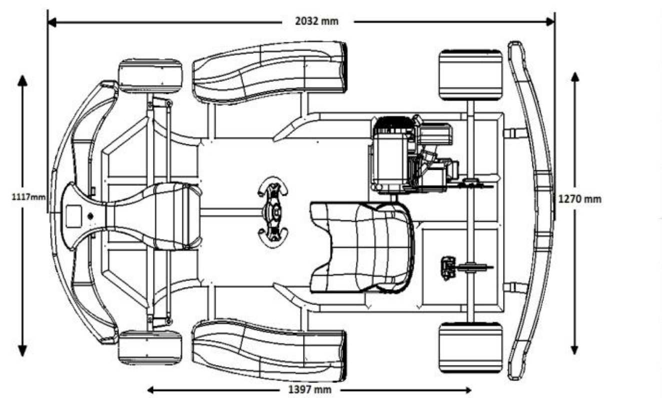


Maximum stress: 140Mpa
Yield strength: 221Mpa
Factor of safety: 1.57

VIII. VEHICLE DRAWING AND DIMENSIONING

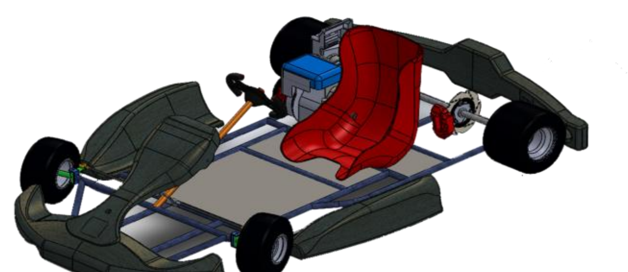


Isometric view of the vehicle

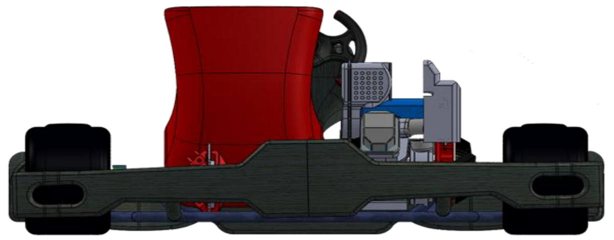


Top view of vehicle

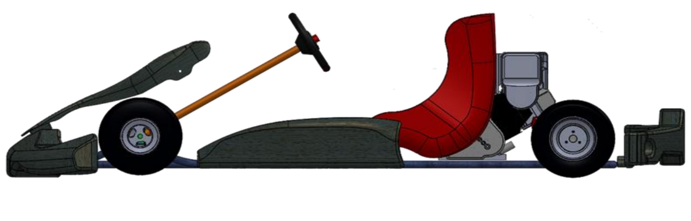
IX. DIFFERENT VIEWS OF THE VEHICLE



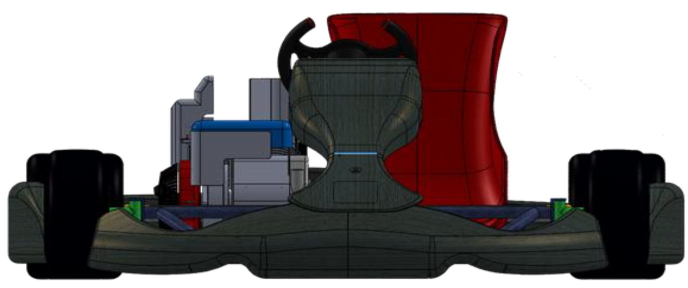
Isometric View of the vehicle



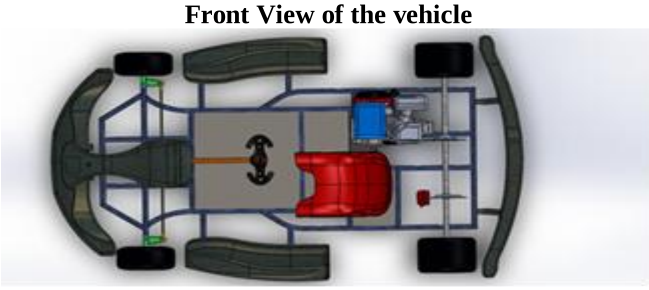
Rear View of the vehicle



Side View of the vehicle



Front View of the vehicle



Top View of the vehicle

X. DESIGN PARAMETERS

CHASSIS	Seamless tube	**IS 1161**
WHEELBASE	1066.8 mm	
OVERALL LENGTH OF VEHICLE	1778 mm	
TRACK WIDTH	914.4 mm 1066.8 mm	Front Rear
TYPE OF ENGINE	3.5HP 127cc engine	
STEERING	Mechanical linkage	
WHEELS AND TYRES	10*4.5*5 11*7.1*5	front rear
BRAKES	Hydraulic disc brake	Rear
TRANSMISSION	Centrifugal clutch	
MASS OF THE VEHICLE	87.8kg	approx.
GROUND CLEARANCE	2 Inch	from bottom most part

XI. STEERING SYSTEM

Mechanical arrangement is planned to be used this type of steering system was selected because of its simple working mechanism and a steering ratio of 1:1 so to simple we have used mechanical type linkage.

Our steering geometry is having 99% Ackerman and also gives **60degree** lock to lock turn of steering wheel which is

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very suitable for the race track as it allows quick turns with a small input and being more precise at the same time. We also attain a perspective turning radius of 2.37meter.

COMPONENTS	**DIMENSIONS**
Tie Rod	14 inch x 0.5 inch ϕ
King-Pin	3 inch x 1.5 inch x 11mm ϕ
Bracket	3.5 inch x 2.5 inch x 0.5 inch
Pit-man arm	1 inch x 2 inch x 10 mm ϕ
Bolt	10 mm ϕ
Steering shaft	20 inch x 1 inch ϕ
Steering wheel	10 inch ϕ

According to the Ackermann geometry the front tyres will rotate about the mean point as a result the entire force will act on the outer front tyre on a corner. Thus the cornering traction will be primarily governed by the outer tyre.

We have chosen the mechanical linkage because it is cheap, light in weight and easy to manufacture.

XII. CONSIDERATIONS FOR STEERING SELECTION

- Amount of steering wheel travelling is decreased
- It is simple and cheap

GEOMETRY	VALUES
Caster Angle	12 degrees
Camber Angle	0 degrees
King pin Inclination	10 degrees
Combined Angle	10 degrees
Toe-in	5 mm
Scrub Radius	9 mm
Minimum Turning Radius	1.12 m
Maximum Turning Radius	2.59m

XIII. CALCULATIONS

Inner lock angle (θ) = (total steering wheel rotation * 360) / steering ratio = 40 degrees

Outer lock angle(ϕ)= $\cot\phi - \cot\theta = w / l$ = 25.57degrees

Ackerman angle calculation: $\tan\alpha = (\sin\phi - \sin\theta) / (\cos\phi + \cos\theta - 2)$ = 32.46 degrees

Ackerman inside angle: $\Psi = \tan^{-1}(WB / (WB / \tan\phi - TW)) - \phi$ = 13.36 degrees

Ackerman percentage: %Ackerman = ((inside angle - outside angle) / (Inside 100% Ackerman)) * 100% = 99.97%

Turning Radius(R_{max}) Calculation

R_{min} = length of wheel base / tan θ = 1.12 m

$R_{max}^2 = [R_{min} + \text{Wheel track width}]^2 + \text{Length of wheel base}^2] =$ 2.59 m

XIV. BRAKING SYSTEM

The **braking system** has to provide enough braking force to completely **lock the wheels** at the end of a specified acceleration run, it also proved to be cost effective. The braking system was designed by determining parameters necessary to produce a given deceleration, and comparing to the deceleration that a known braking system would produce.

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Considerations for braking system selection:

Discs, calipers and master cylinders which were used for considering suitable vehicle after market survey

Sr. No.	Disc	Outer diameter in mm	Thickness in mm
1	Pulsar 150	**240**	**6**
2	Pulsar 220	**230**	**8**
3	**Apache RTR 180**	**200**	**6**

Reasons for selection for apache RTR 180 disc

- Thickness (6mm) is not too high
- Outer diameter is 200mm which is in accordance with our required design

We have used floating type calliper in our design and finalised Apache RTR 180 calliper

XV. BRAKE FLUIDS

We have decided to use DOT 3 Brake fluid

- Economical
- Easily available
- Compatible

XVI. WEIGHT DISTRIBUTION

Gross weight of the kart = 160kgs
Front: Rear = 2:3

Type	Specification
Rear disc	OD – 200 mm
Master cylinder Dia.	10 mm
Caliper piston diameter	25.4 mm
Brake Pedal Lever ratio	4:1
Stopping distance	2.237 m

XVII. CALCULATIONS

1. Gross weight of the vehicle

W = weight of the vehicle (with load conditions) in kgs * 9.81= 160*9.81=1569.6N

2. Brake line pressure:

p = force on the brakes / area of master cylinders
(as pedal ratio is 4:1)
(Assume the normal force applied on the pedal: 350n)
= pedal ratio *force on the pedal / area of master cylinder
$= 4*350/(\pi/4)*(0.01)^2$

= 17.8343Mpa

3. Clamping force (CF):

Cf= brake line pressure *(area of caliper piston*2)
$= 17.8343*((\pi / 4) * (25.4 * 10^{-3})^{2} * 2)$
= 18064.6825N

4. Rotating force:

RF = CF* number of caliper pistons * coefficient friction of brake pads
=18064.4334*0.3*2
=10838.6825N

5. Braking torque (tn) = rotating force* effective disc radius

08=138.6825*0.09
=975.48 N-m
(torque available at the two tires of the rear shaft)

6. Braking force=(braking torque /tire radius)*0.8

=6982.6677*0.8
=5586.0820N

7. Deceralation:

f=-ma(-ve sign indicates force in opposite direction)
a=-B.f/m=5586.0820/160

=-34.913m/s

8. Stopping distance:

$v^2 - u^2 = 2*a*d_s$ (v=0,u=12.5m/s)
Stopping Distance =2.2370meters

XVIII. ENGINE AND TRANSMISSION

The engine in deployed in our Kart is BRIGGS and STRATTON 3.5HP 127 CC it has 14kg weight with single cylinder.

XIX. ENGINE SPECIFICATIONS

Configuration	**Value**
Engine Technology	single cylinder , 4-stroke,air cooled , OHV(Over Head Valve)
Maximum Power	3.5 HP@3600 R.P.M
Gross Torque	7.5 N-M@2600 R.P.M
Bore*Stroke	62mm*42mm
Displacement	127cc
Dry Weight	14 Kg
Fuel Capacity	1.8L
Length	261mm
Width	347mm
Height	326mm

Assuming transmission efficiency = 80%
Gross weight of the Kart = 160kgs
Number of teeth on CVT output = 54
Number of teeth on rear shaft sprocket = 36
Ratio = 0.66:1

SPEED (RPM)	CVT RATIO	SPROCKET RATIO	FINAL RATIO
1850	16	0.66	10.56
2750	10	0.66	6.6
3600	6.4	0.66	4.224

XX. CALCULATIONS

Speed = (circumference of the wheel * rear shaft rpm) / (60*1000) m/s

= (∏*11*25.4*852.27)/(60*1000)

= 12.468 m/s

= **44.86 km/hr**.

Drive torque = engine torque * reduction * efficiency

=7.5*16*0.66*0.8 = **63.36 Nm**

Drive force = drive torque/radius of wheel

= 63.36*1000/5.5*25.4 = **453.54 N**

Acceleration = Drive Force/mass

= 453.54/160= **2.83** m/s^2

XXI. BODY WORKS

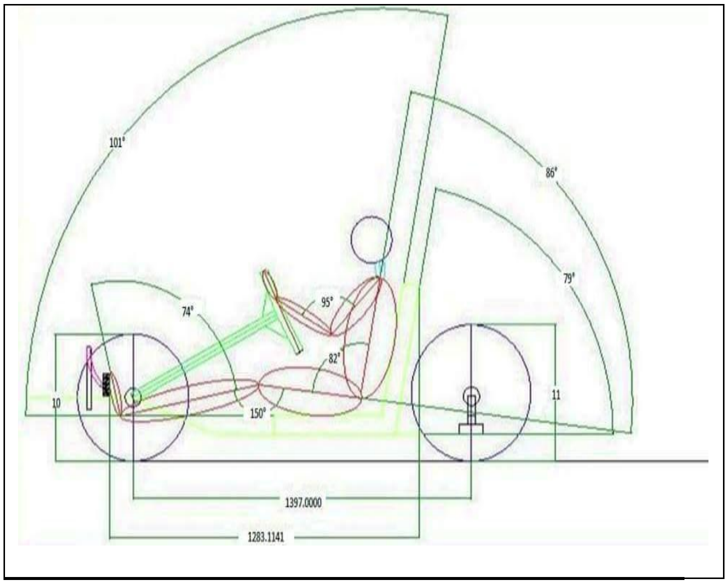
External appearance of the vehicle depends upon bodyworks. It is an important part of the vehicle design. It also dominates sale and marketing of the vehicle.

We have selected fiber on the basis of market survey because of its

- light weight
- good electrical insulator

XXII. SAFETY AND ERGONOMICS

Safety is the most important concern for our Go-kart. Bumper is provided for safety. In addition fire extinguishers and kill switches will also be used in case of emergency. Ergonomics are designed perfectly for the comfort of the driver.



SPECIFICATIONS	VALUES
KNEE ANGLE	**150 DEGREES**
ELBOW ANGLE	**95 DEGREES**

XXIII. ELECTRICALS

12V DC Battery will be used to power all the electrical components.

XXIV. CONCLUSION:

The design and construction for GO-KART DESIGN CHALLENGE 2015 has become more challenging due to the increased participation. Our team is participating for the first time in this event so a detailed study of various automotive systems is taken as our approach. Thus this report provides a clear insight in design and analysis of our vehicle

The making of this report has helped us in learning of various software. We want to give a vote of Thank you in this regard as GKDC competition has given as this opportunity to learn many things which will also help us in leading a bright future.

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