INTERNATIONAL GO-KART CHAMPIONSHIP 2020

Organized by- LPU SAEINDIA COLLEGIATE CLUB

ENGINEERING DESIGN REPORT (EDR)

TEAM NAME – Automaestro

TEAM ID – M20007

COLLEGE NAME – Bangladesh University of Engineering and Technology (BUET)

Design Report

***Abstract*— Go-Kart Design Challenge is a contest formulated to enhance the approach of students practicing Engineering.This event is fundamentally all about designing and fabricating a Go-Kart at very low expenditure and manufacture Go-Karts yielding optimum performance. The objective of this report is to document and represent Go-kart designed by TEAM AUTOMAESTRO to compete in International Go-kart Championship 2020 successfully.**

I. Introduction

Team **"Auto maestro"** is a team of Bangladesh University of Engineering & Technology and facilitated by BUET AUTOMOBILE CLUB and consists of 25 members. This team has a bold interest in appliance of engineering knowledge over the automobile sector and firm on. The team aims at manufacturing a go-kart for competing in the **INTERNATIONAL GO-KART CHAMPIONSHIP 2020.**

We have divided our design task in the following sub-systems.

1) Frame design; 2) Steering system; 3) Transmission; 4) Tyres specification; 5) Braking System 6)Electric System; 7) Material and Manufacturing 8)Ergonomics; 9) Innovation;

IA. Technical Specification

A. Vehicle dimensions and data

Property	Value
Weight of vehicle (With driver approx.)	300kg max
Wheel Base	56 inch
Track Width	38 inch (smaller)
Ground Clearance	2.25 inches
Boundary Box Vol.(inch)	87*47.1*55

B. Material Specification

Type	201 Annealed stainless steel (SS)
Dimensions	1.75 inch outer diameter 2 mm thickness

C. Steering System

Property	Value
Steering System	*Pivot pin*
Steering Mechanism	Ackerman
Turning Radius	2.032 meter(80 inch)
Max. Steering Angle	45 degree
Inner Wheel Angle	42.5 degree
Outer Wheel Angle	29.5 degree

D. Engine Specification

Property	Value
Type	Single cylinder
Max Power	14.9 Bhp @9000 rpm
Swept Volume	149.5 cc
Torque	13.4 Nm@ 6000 rpm
Cooling	Air Cooling
Ignition System	Spark ignition

E. Transmission

Property	Value
Type	Manual
Clutch	4 Clutch
Gear Configuration	1 2 3 4 5
Gear Ratio (1st)	28.2:1
Gear Ratio (2nd)	19.17:1
Gear Ratio(3rd)	14.05:1
Gear Ratio(4th)	11.03:1
Gear Ratio(5th)	9.40:1
Top Speed	36.5 kmph @ 6000 rpm

F. Brake

Property	***Value***
Type	Hydraulic disc brake
Disc Diameter	200 mm
Master Cyl. Piston Diameter	16 mm
Calliper Piston Diameter	29 mm
Braking Effort	25 kgF
Stopping Distance	7.6 meter

G. Wheel

Front - 10x4.5-5 ; Rear- 11x7.1-5

IB. DESIGN METHODOLOGY

The frame is designed for a rear mounted engine and it has the minimum wheelbase of 56 inch, a proportional smaller front track width of 38 inch. The frame also accommodates the rear track width without increasing the width of the kart. The engine and the driver are in line, separated by a firewall. The design includes all the mounting features for the following: 1) Engine mounting, 2) Floor pan, 3) Body works, 4) Firewall, 5) Seat, 6) Rear axle mounting, 7) Steering mounting

II. CHASSIS DESIGN

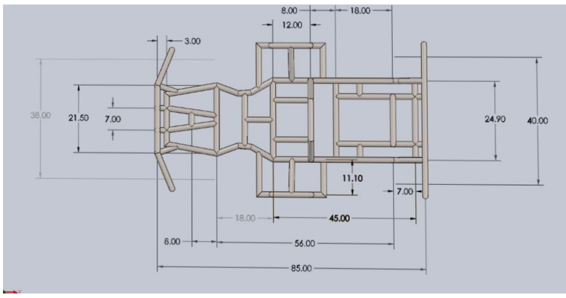
The design should have torsional rigidity, should resist bending, and sustain impact loading at low and high speeds.

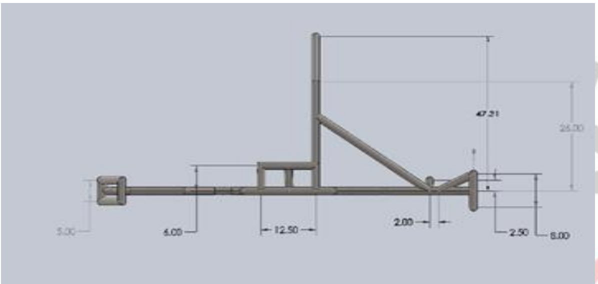
The Material 201 Annealed stainless steel (SS) will be used in frame design due to its low carbon content which increases its welding ability. Material specifications-1)Fe-72%; 2)C-.15%;3)Si-1%;4)Cr-16-18%;5)Ni-4.5%;6)Mn-6.5%;7)N-0.25%

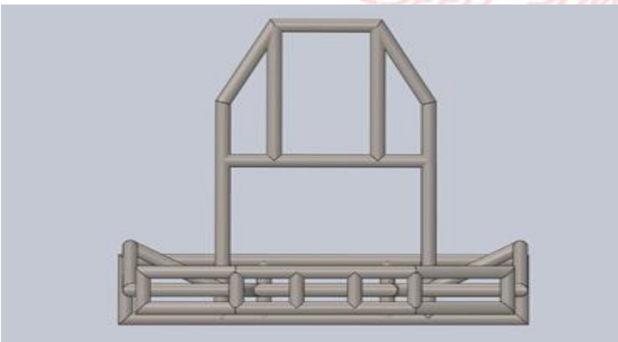
PROPERTY	*VALUE*
Yield Strength	292 MPa
Outer Diameter	1.75 inches
Wall Thickness	2 mm
Modulus of Elasticity	2.07e+11 N/m^2
Poisson’s Ratio	0.27 N/A
Mass Density	7860 kg/m^3

II.A. CAD DESIGN

According to the chassis design in solidworks the designs are being given as follows







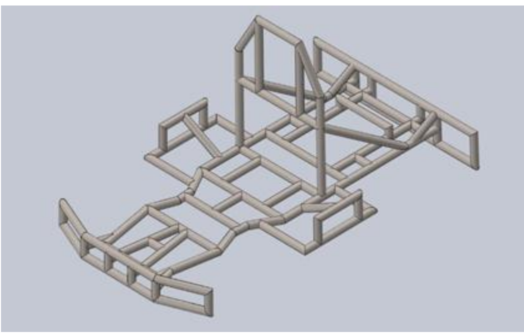


Fig .1,2,3,4(from the top): top, front, side, isometric view

II.B. CAE ANALYSIS

FEA is used for determining whether a member is in stress or in tension, The FEA (finite element analysis) is done on Ansys software. Using Mechanical APDL as the solver,these tests simulate conditions of high stress and evaluate the performance of the frame.

A. Front Impact Test

For impact analysis the force has been calculated by Impact force derivation formulae- **F*t=mv-mu**

Property	Value
Velocity	40 km per hour
Impact Time	0.5 sec
Mass	300 kg
Force	6666.66N
Max Deformation	0.72 mm
Max Stress	68.88 MPa
Factor Of Safety	4.24

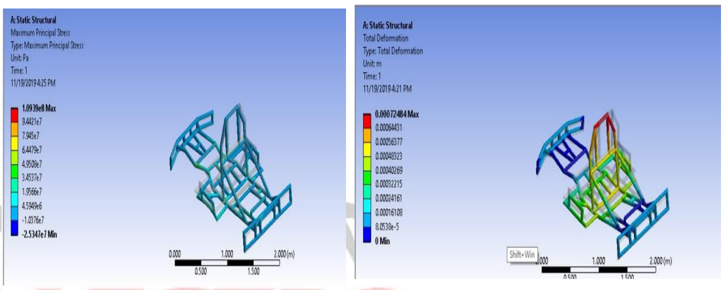


Fig 5,6: Total deformation, Max stress in front Impact test

B. Side Impact Test

The similar principle is applicable to side impact test. The force is applied on the side bumper.

Properties	Value
Velocity	40 km per hour
Impact Time	0.5 sec
Mass	300kg
Force	6666.66 N
Max Deformation	0.78 mm
Max Stress	109.3MPa
Factor Of Safety	2.67

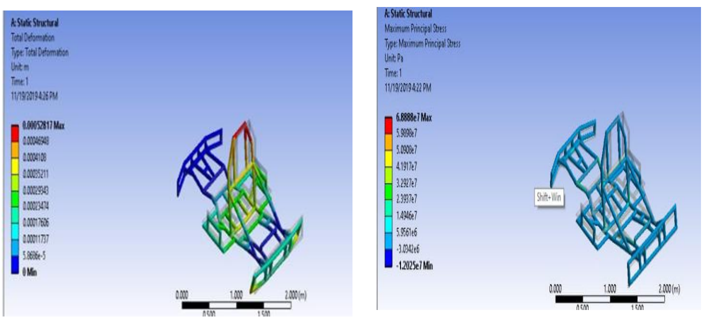
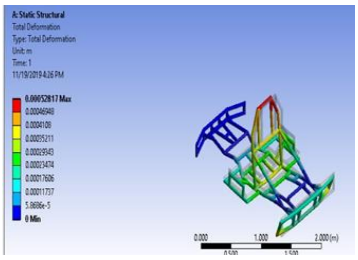


Fig6, 8: Total deformation, Max stress in side Impact test

C. Rear Impact Test

The similar concept is applicable

Properties	Value
Velocity	40 km per hour
Impact Time	0.5 sec
Mass	300 kg
Force	6666.66 N
Max Deformation	0.5mm
Max Stress	69.5MPa
Factor Of Safety	4.2



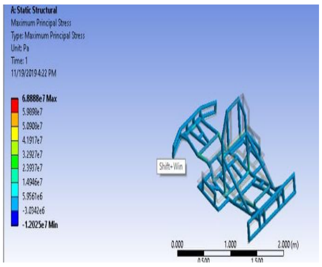


Fig9, 10: Total deformation, Max stress in rear Impact test

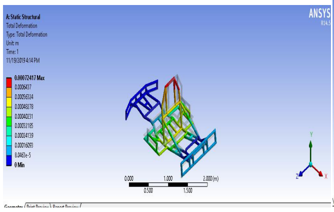
D. Static Load (static Bending)

In case of static load the total deformation and stress max is counted by following.

LOADS	MASS	FORCE
DRIVER	70kg	686N
ENGINE	60kg	588N
TOTAL	130Kg	1274N

Property	Value
Velocity	Static
Impact time	NA
Mass	130kg
Force	1274N
Max deformation	0.72mm
Max stress	68.84MPa
Factor of safety	4. 24

Hence the calculated value of static bending force was placed on the middle part of the frame while keeping the chassis part fixed. The analysis result is shown as-



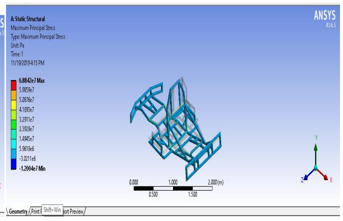


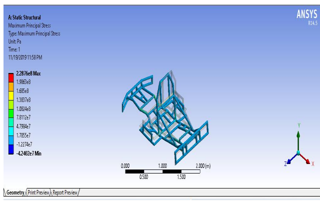
Fig -11: Total deformation in static bending. Fig 12: Max Stress in static bending

E. Rolling impact

The similar principle is applicable to rolling impact test. The force is applied on the roll bar.

PROPERTY	VALUE
VELOCITY	rolling
IMPACT TIME	NA
MASS	300kg
FORCE	6666.66 N
MAX DEFORMATION	0.5 mm
MAX STRESS	260 MPa
FACTOR OF SAFETY	1.12

Hence the calculated value of rolling force was placed on the roll bar of the frame while keeping the chassis part rolling. The analysis result is shown as-



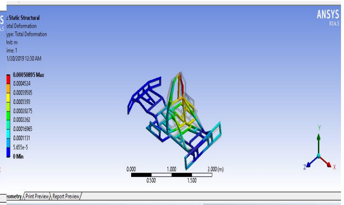


Fig 13- Max stress for rolling impact Fig 14- total deformation for rolling impact

III. STEERING SYSTEM

A. Design methodology

The steering system is designed to withstand the stress of safely maneuvering the vehicle in any type of possible condition. While designing the steering system the constraints that we had were centre alignment of steering system, track width, human effort at the steering wheel and the desired response of the steering system.

B. Calculation

The formulae's used for steering calculation are:

R = d/2+Lcosec (A/2+B/2)

% Ackerman = (A-B)/ ($\tan^{-1}$ (1/ (tan (B)-1))-B)

Where,

R=turning radius, L=wheelbase of the kart (56 inch)

A = outer wheel angle, B = inner wheel angle

d = smaller track width of the car (38 inch)

Raa = the Ackerman arm radius.

100% Ackerman angle is desired, A=29.5^o and B= 42.5^o was the best option. This gave a turning radius of **2.03m**.

This value is obtained from the numerical analysis of Ackerman percentage and turning radius followed by the spreadsheet bellow- https://docs.google.com/spreadsheets/d/1-hSla4aMpjlub8WMpMqMSvcdZhjHU_JeynfFfPV3BU8/edit?usp=drivesdk

Outer wheel angle- **Inside wheel angle-**

Tan A = L/(R-d/2) Tan B = L/(R+ d/2)

Tie rod length=Ackerman arm radius * Sin (Ackerman angle)

Technical data:

Inner Turning Angle	42.5^o
Outer Turning Angle	29.5°

Turning Radius	2.032 m(80in)

IV. TRANSMISSION SYSTEM

IV.A. Objectives

The purpose of the transmission system is to transmit power and torque from the engine to the wheels.

IV.B. Design Methodology

The transmission system of our kart is a chain-sprocket drive system attached with a 5-speed gearbox that comes as stock with the engine. The power from the engine will be transmitted to the left wheel of the driving axle only. Our design methodology includes:(1)Selecting a proper shaft material.(2)Selecting appropriate shaft diameter.

IV.C. Components of Transmission system

The main components of the braking system are:(1)Gearbox,(2) Chain-Sprocket Drive, (3) Drive Shaft.

We have selected AISI-1018 Mild steel as shaft material for availability and cheap price.

IV.D. Calculations

Assumptions:(1)Total vehicle weight is W=300kg. (2)The weight distribution ration is 60-40.(3)The weight is distributed equally on the two rear wheels.(4)The reaction acts on a single point along the middle line of the perimeter of the wheel.

Shear Force and Bending Moment Calculation on Vertical Plane:

Due to the assumptions we taken,

Vertical Bearing reaction=W_1=W_2=R=882N

Reaction force on each wheel=F_{w1}=F_{w2}=R=882N

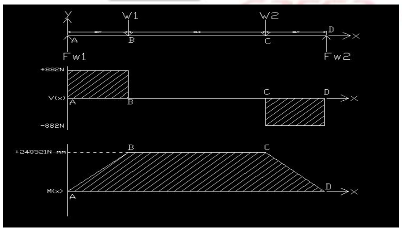


Fig 11: Free Body Diagram, Shear Force (V) and Bending Moment (M) diagram of drive shaft in vertical plane.

Lengths in the free body diagram:

LAB=281.77 mm, LBC=632.46 mm, LCD=281.77 mm

From the Bending Moment Diagram (BMD), a maximum bending moment of Mv=248,521 Nmm is found in BC section.

Shear Force and Bending Moment Calculation on Vertical Plane:

We will consider the power wheel under locked condition for maximum possible horizontal bending analysis.

Friction force on the locked wheel,

Fs1=μp×R=0.9×882=794.61N

Friction force on the free wheel,

Fs2=μf×R=0.8×882=706.32N

Force due to brake rotors,

$$B1 = B2 = \frac{1}{2}\left(\frac{\text{Fs1×Tire radius}}{\text{Effective radius of the brake disc}}\right)$$

Now, Tire radius = (0.2794/2) m and Effective radius of the brake = 0.08019m (From Braking System Analysis).

Caster Angle	7°
Camber Angle	0°
Tie Rod Length	25.59 inch
King Pin Inclination	15°
Ackerman angle	13.55 degree
Ackerman arm radius	3 inch

$$\therefore\ B1=B2=\frac{1}{2}\left(\frac{794.61\times0.2794}{2\times0.08019}\right) = 692.15\ \text{N}$$

Rz1 and Rz2 are horizontal Bearing Reaction forces.

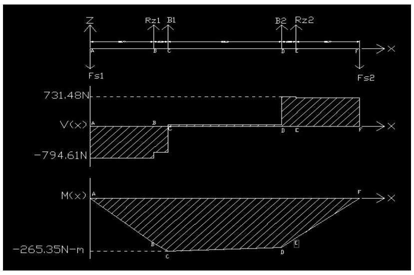


Fig 12: Free Body Diagram, Shear Force (V) and Bending Moment (M) diagram of drive shaft in Horizontal plane.

Lengths in free body diagram:

LAB=LEF=281.77 mm, LBC=LDE=63.50 mm, LCD= 505.46 mm

Taking moment about B on horizontal plane:

Fs1×LAB+B1×LBC+B2×LBD+Rz2×LBE-Fs2×LBF=0

∴ Rz2= -25.16 N

Taking summation of forces in positive z-axis:

Rz1=Fs1-B1-B2-Rz2+Fs1=141.794 N

From the Bending Moment Diagram, Maximum horizontal Bending moment, Mh = 265.35 Nm at point C.

Maximum and Minimum Bending Moments:

Mv = 248,521 Nmm = 248.52 Nmm

Mh = 265.35 Nm

$$\text{Mmax} = \sqrt{\text{Mv}^2 + \text{Mh}^2}$$

$$=\sqrt{(248.52)^2 + (265.35)^2}$$

= 363.55 Nm= 363550 Nmm

Mmin=Mv= 248,521 Nmm

(As, in case of minimum Bending moment, Mh=0)

Minimum shaft diameter required to support maximum bending:

$$\sigma b = Sy = \frac{Mc}{I} = \frac{32M}{\pi d^3}$$

$$\therefore\ d = \sqrt[3]{\frac{32M}{\pi Sy}} = \sqrt[3]{\frac{32\times363550}{\pi\times220}} = 25.628\text{mm} = 1.009\text{ inch}$$

Taking 1.25 inch = 31.75 mm dia solid circular shaft for fatigue safety factor analysis.

V.Tyre specification & selection parameter

Tyres of Go kart for International Go-karting championship are standard go kart tyres . These tyres are typically of two types- Standard wet tyre(striped) ,Standard dry tyre(flat)

Dimensions for dry tyres:

Tyre	**Tyre outer diameter**	**Tyre width**	**Rim size**
Front	10 inch	4.5 inch	5 inch
Rear	11 inch	7.1 inch	5 inch

Dimensions for wet tyres:

Tyre	Tyre outer diameter	Tyre width	Rim size
Front	10 inch	4.5 inch	5 inch
Rear	11 inch	6 inch	5 inch

VI. BRAKING SYSTEM

A. Design Methodology

The brake system of our kart is dual system. The brake system design includes two discs at the rear axle to stop the vehicle. Master cylinder is used at the front near the brake pedal providing the occupant to easily accessible space. our objectives were i) List design criteria or requirements, ii)Assign maximum deceleration of the vehicle iii)To calculate target stopping distance and target braking force Select optimum brake parts which in combination help achieve target iv) Optimize brake design.

B. Calculations

Assumption in the design of Braking System:

(1)The maximum force the driver can apply in the pedal is 25 kgf.(2)The sudden application of the brake arrests the rotational kinetic energy.(3)The translational kinetic energy is reduced by the action of friction during skidding due the absence of rolling motion.

Rules and regulations:

For braking Test given by IGC Rule book (section k1.1)-i)The vehicle must be accelerated to 35 km/h before application of brake in a 100ft (30.48m) track .ii) After the application of brake, the vehicle must stop in 26.25 ft. (8 m)

By work energy principle,

KE of the vehicle = Work done by friction. But, sudden application of the brake will cause the instant locking of the rear axle. Thus, the brake will contribute in arresting the rotation of the shaft and preventing the rotation of the shaft. The vehicle is stopped by the skidding of the rear wheels. Torque required to lock the breaking wheel-Assuming time required to lock the wheel is t =1s and second moment of inertia of wheel-axle=0.8 kgm². For v = 40 kmph = 11.11 ms^{-1}

$$\omega_i = \frac{v}{r} = 85.67\, {}^{rad}\!/_{\sec}$$

$$\alpha = \frac{\omega_i - \omega_o}{t} = \frac{85.67 - 0}{1} = 85.67\, {}^{rad}\!/_{s^2}$$

$$\theta = \frac{\omega_i^{\,2}}{2\alpha} = \frac{(85.67)^2}{2 \times 85.67} = 42.835 rad$$

Work energy principle gives:

$$T\theta = \frac{1}{2} I \omega_i^{\,2}$$

$$\therefore T = \frac{{}^1\!/_2\, I\omega_i^{\,2}}{\theta} = \frac{0.5 \times 0.8 \times (85.67)^2}{42.835} = 68.536 Nm$$

Now, calculating maximum torque available in the hydraulic braking system,

The pedal ratio (i.e. the leverage) is = X: Y=5:1

Force on the Master cylinder piston

$$F_{mc} = (25 \times 9.8) \times 5 = 1225 \text{ N}$$

Area of the Master piston

$$A_{mc} = \frac{\pi}{4} \times 0.016^2 = 2.01 \times 10^{-4}\ m^2$$

Pressure developed in the system

$$P_{mc} = \frac{F_{mc}}{A_{mc}} = \frac{1225}{2.01 \times 10^{-4}} = 6092650.165 \text{ Pa}$$

Area of the Pads

$$A_p = \frac{\pi}{4} \times (0.029)^2 = 6.6052 \times 10^{-4} m^2$$

Actuation force by the caliper pads on the rotor

$$F = P_{mc} \times A_p = 4024.316 N$$

Effective rotor radius is calculated by uniform pressure theory-

Assuming "Uniform Pressure",

$$R_e = \frac{2}{3}\frac{(R_0^{\,3} - R_i^{\,3})}{(R_0^{\,2} - R_i^{\,2})} = \frac{2}{3} \times \frac{(95)^3 - (63.25)^3}{(95)^2 - (63.25)^2} m$$

= 80.19 mm [eqn 16-39]

$$\mu_{road} = 0.35$$

Braking Torque available-

$$M_t = \mu \times F \times R_e = 112.94 Nm \quad \text{[eqn 16-31]}$$

The required torque is less than the available torque.

$$M_t \rangle T$$

Available torque is greater than our locking torque. So one brake will be able to safely lock the braking wheel which validates our assumption.

Braking Distance

Brake will be applied on a single wheel and all the other wheels will be allowed to rotate freely. We can determine the braking distance for a v=40 kmph to stand still by Energy method- ${}^1\!/_2\, mv^2 = \sum \mu RgS$

Where,

m= vehicle mass,

v = 40 kmph = $\frac{40}{3.6} ms^{-1} = 11.11 ms^{-1}$

$\mu_l = 0.9$ (For locked wheels)(Assuming)

$\mu_f = 0.8$ (For free wheels) (Assuming)

R_l = Reaction on locked wheel (30%of total vehicle mass) = 0.3m (Assuming a 60-40 weight distribution and equal reaction on each rear wheels)

$\sum R_u$ = Summation of Reaction on free wheels (70 % total vehicle mass) =0.7 m, g=9.81 ms$^{-1,}$

S = braking distance

Thus, $\frac{1}{2} mv^2 = \mu_l \times 0.3 \times gS + \mu_f \times 0.7 \times gS$

$$\Rightarrow \frac{1}{2} v^2 = gS \times (0.3\mu_l + 0.7\mu_f)$$

$$\Rightarrow S = \frac{v^2}{2g(0.3\mu_l + 0.7\mu_f)}$$

$$= \frac{(11.11)^2}{2 \times 9.80 \times (0.3 \times 0.9 + 0.7 \times 0.8)}$$

$$\therefore S = 7.59m \approx 7.6m \langle 8m)$$

Thus, just the friction of the wheels can stop our vehicle in less than 8m from 40 kmph which indicates that our vehicle should pass the brake test.

VII.ELECTRICAL SYSTEM

A. Design Methodology

The electrical system of the cart comprises of the battery, electrical components (electric start/ignition switch, engine kill switch, brake over travel switch, brake lights, external switch, and cockpit switch), wires and fuses. The components are wired with battery which carries electric current, and fuses which protect the components as well as the wirings. Our objective is to provide power to the starting motor (via ignition switch) for self-starting the vehicle, to start-up the vehicle and to keep it running and to ensure the safety of the cart and the driver.

***2) Starter Circuit*:** The main components are 12V battery, starter switch, the solenoid and motor, connecting cables and the earth, or ground return circuit. When the ignition switch is closed, power is supplied to the motor through the solenoid for self-starting the engine.

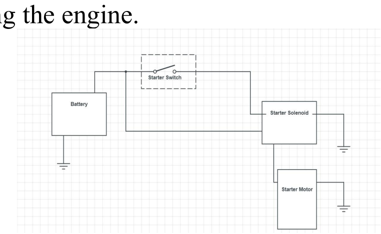


Fig 13: Starter Circuit

2) Brake Light: The figure shows when the break paddle is pushed, the light switch completes the circuit and turn the brake lights on.

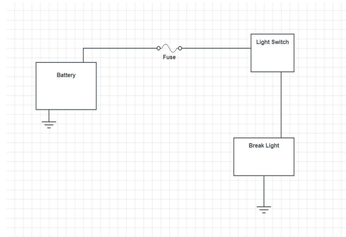


Fig 14: Brake Light Circuit

3) Engine Kill Switch: Two kill switches are provided to stop the vehicle safely in case of any emergency. However, they do not deactivate the brake light.

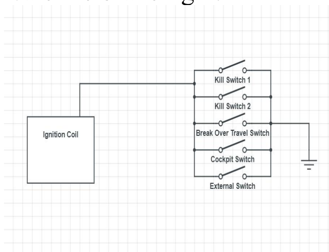


Fig 15: Kill switch, Brake over Travel Switch, Cockpit Switch, External Switch

4) Brake over travel switch: Brake over travel switch acts as a kill switch in case of brake failure. It is fitted behind the brake pedal in pulled up condition.

5) External Switch: External Switch is mounted on the side of the driver so that the volunteers can easily reach it and turn the engine off in case of emergency.

VIII. INNOVATION

<u>Use of Jute Fiber Reinforced Polymer Composites (JFRPC) on Go-Kart Body Panels</u>

A. Abstract

During last few years, the interest in using natural fibers as reinforcement in polymers has increased dramatically. Natural fibers are not only strong and lightweight but also relatively very cheap. In this report, a theory based study alongside data carried out from some highly cited papers has been used to validate the feasibility of using JFRP in our Go-Kart. Jute a natural fiber abundantly available in Bangladesh. We are avid using **bidirectional jute fiber mat as reinforcement and epoxy resin** as matrix material because of its impeccable performance, re-usability and environment-friendliness.

1. Introduction

Environmental concerns are increasing day by day and the demand of replacing the existing synthetic fibers with the biodegradable, renewable and low-cost natural fibers for fabrication of composite materials increases. The use of natural fiber like jute not only help us in ecological balance but can also provide employment to the rural people in countries like Bangladesh where jute is abundantly available.

2. Experimental Details

2.1. Materials and Method: Bidirectional jute fiber mat has been obtained from the local sources as a reinforcing material as well as the Epoxy resin and the corresponding hardener. The polymers composites are fabricated by hand lay-up technique.

2.2. Physical and Mechanical Characterization: Hardness measurement is done using a Rockwell-hardness tester equipped with a steel ball indenter. Tensile test is performed as per ASTM D 3039-76 test standards using universal testing machine Instron 1195. Impact strength of the composites is evaluated by a low velocity impact tests conducted in an impact tester as ASTM D256 test standard.

3. Result and conclusion

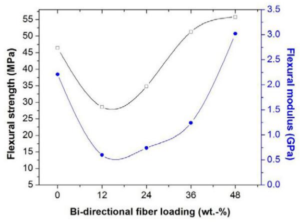


Fig 3. Effect of fiber loading on flexural strength and modulus of composites

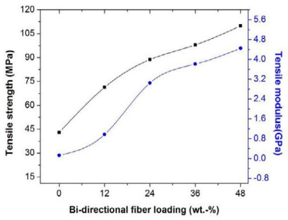


Fig 2. Effect of fiber loading on tensile strength and modulus of composites

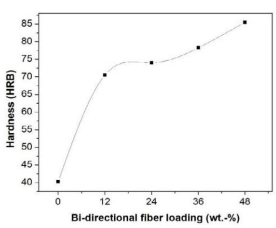


Fig 1. Effect of fiber loading on hardness of composites

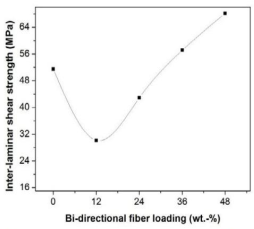


Fig 4. Effect of fiber loading on inter-laminar shear strength of composites

1. Successful fabrication of the bidirectional jute fiber reinforced epoxy composite has been done by the hand lay- up technique.

2. The hardness, tensile properties and impact strength of the jute-epoxy composites increases with the increase in fiber loading.

3. The properties like **flexural strength, inter-laminar sear strength** are greatly influenced by the void content of the composites. It has been found that these properties reduced from 0-12 wt% fiber loading and with the **reduction in the void content from12-48wt% the properties are improved**.

SMART SEATBELT WARNING SYSTEM WITH ON-DEMAND FUEL CUT-OFF

Abstract

Most modern cars come equipped with SBR that provides the rider an audible or visual reminder if they forget to buckle up when the car is moving. The system is quite benign and can be easily overridden by inserting the buckling chip inside the latch without actually fastening up. The system we have here works minding all the aforementioned factors and to some degree compels the rider to fasten up properly.

Introduction

The system works with fairly basic electronic components controlled by an on-board Arduino chip and a very simple algorithm that makes sure if the belt is fastened properly.

List of the components used:

1. Arduino UNO ATMega-328 Microprocessor
2. Force Sensitive Resistor SEN-00097
3. LEDs
4. I2C LCD Display 16x2 DIS-00089
5. Load resistors
6. Relay

The force sensitive resistors placed on the inner webbing of the belt itself will read the amount of force applied and send it to an algorithm that'll determine if the belt is properly worn. On the other hand, force sensitive resistors placed on the outer webbing of the belt will theoretically read 0 to confirm the belt is not being overridden. Based on the data obtained from either of the sensors, if the algorithm senses that the system is being overridden, it'll automatically send instructions to the fuel system relay to restrain fuel supply to the engine(on-demand)

2. Experimental Details

2.1. Method

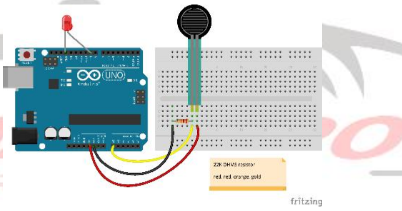


Fig: Sensor Interfacing

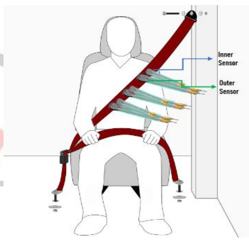
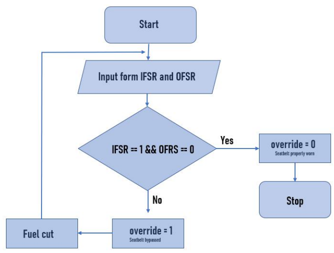


Fig: Sensor Positioning

2.2. Algorithm



3. Result and Discussion:

The system consists of very basic electronics and can be hooked up easily with the seabelt and in later revisions may be updated with self- learning capability in order to assis the rider to properly wear his seatbelt.

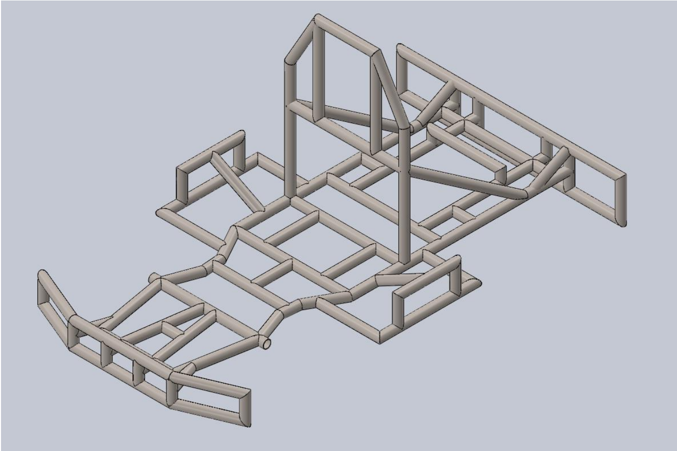


Fig 16: Isometric View of the Chassis

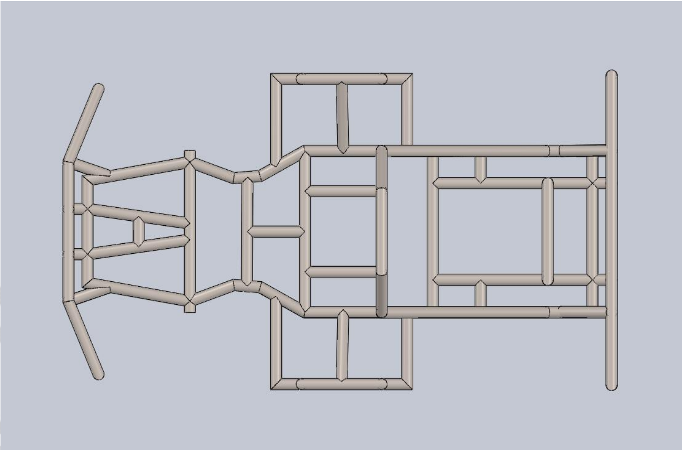


Fig 17: Top View of the Chassis

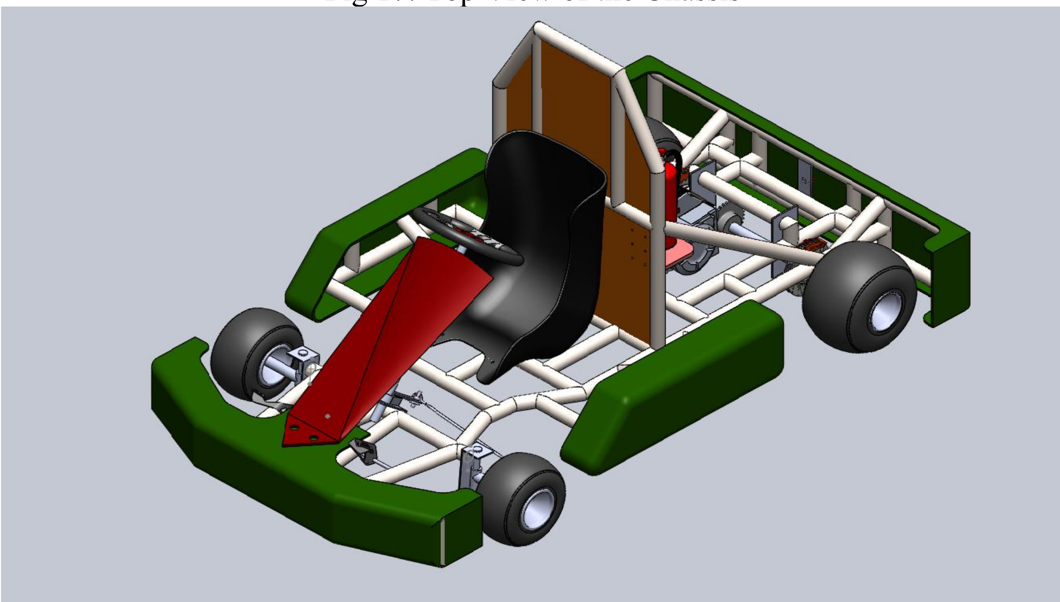


Fig 18: Isometric view of Go-Kart

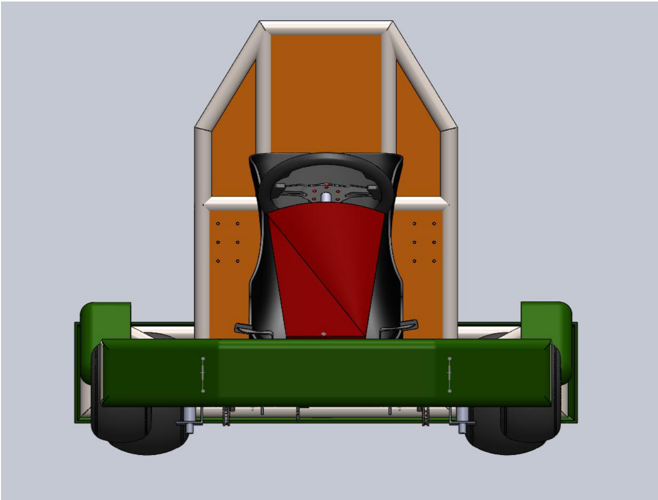


Fig 19: Front View of Go-Kart

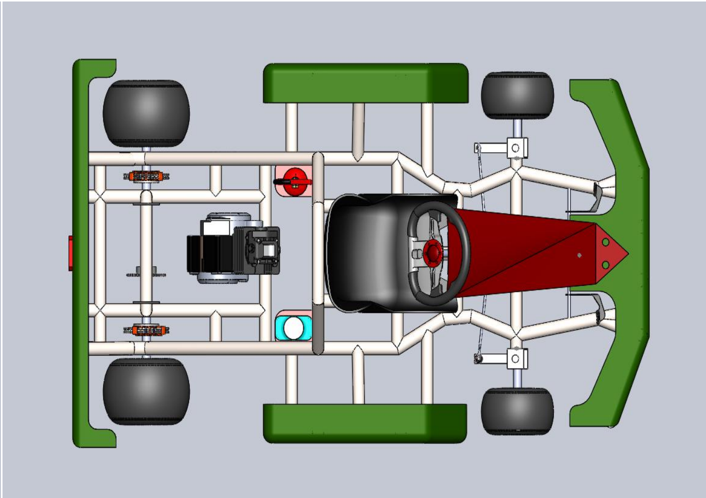


Fig 20: Front View of Go-Kart

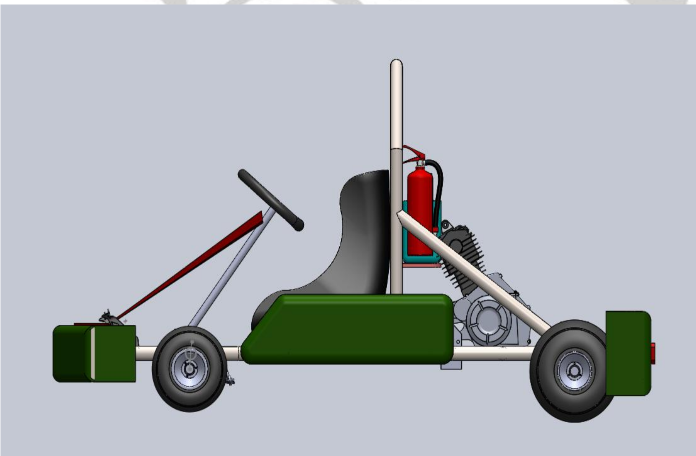


Fig 21: Right View of Go-Kart

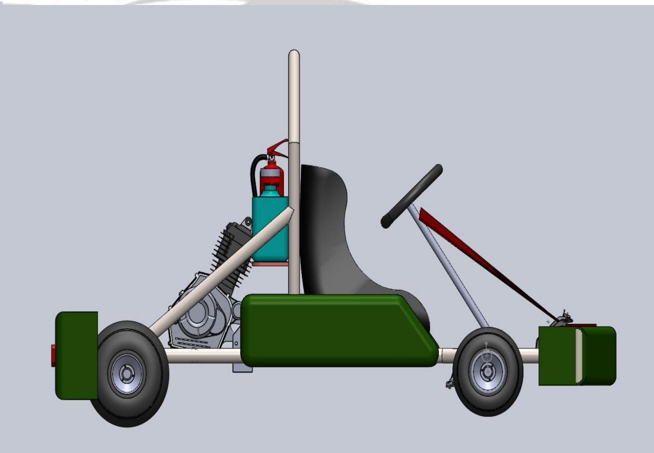


Fig 22: Left view of Go-Kart

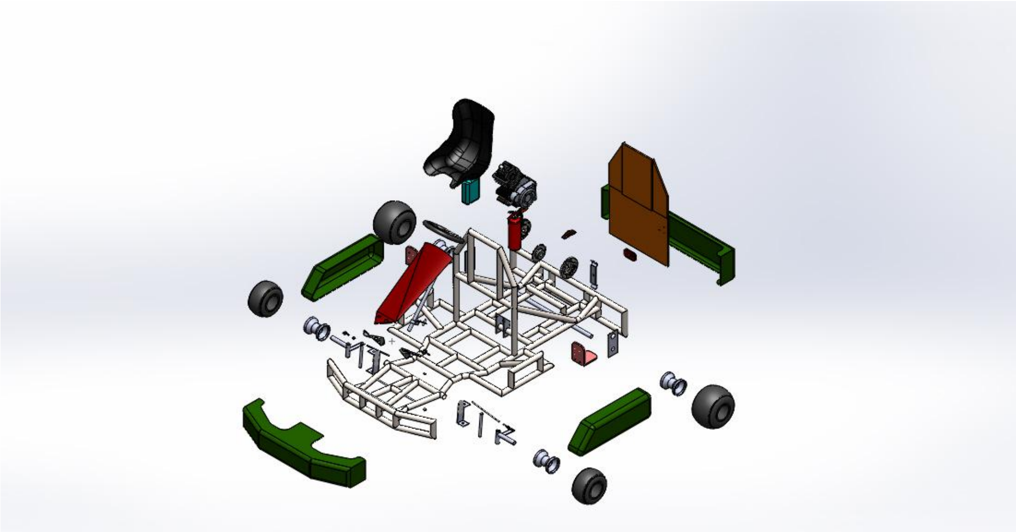


Fig 23: Exploded View of Go-Kart

Fig 24: Go-Kart Under Construction

DESIGN VALIDATION PLAN (DVP)

Test Category	Test Name	Test Procedure	Test Target
Design	Material Test	Universal Tensile Testing Machine, Solid Mechanics Lab	Strength to be equal or more than AISI 1018 (>375MPa)
Design	Welding Test	Welding shop	Welded joint to have strength close to base metal
Braking system	Brake Test	Local test track having specified runway	Vehicle to stop within 8 metre after pressing the brake pedal
Steering System	Steering Test	Local test track	Vehicle to turn a circular track with radius 2.5 metre
Transmission	Acceleration Test	Local test track having specified runway	Vehicle to obtain handsome point calculated as described in the rulebook
Endurance	Mock Endurance	Local test circuit track	Vehicle to run 1.5 hours continuously

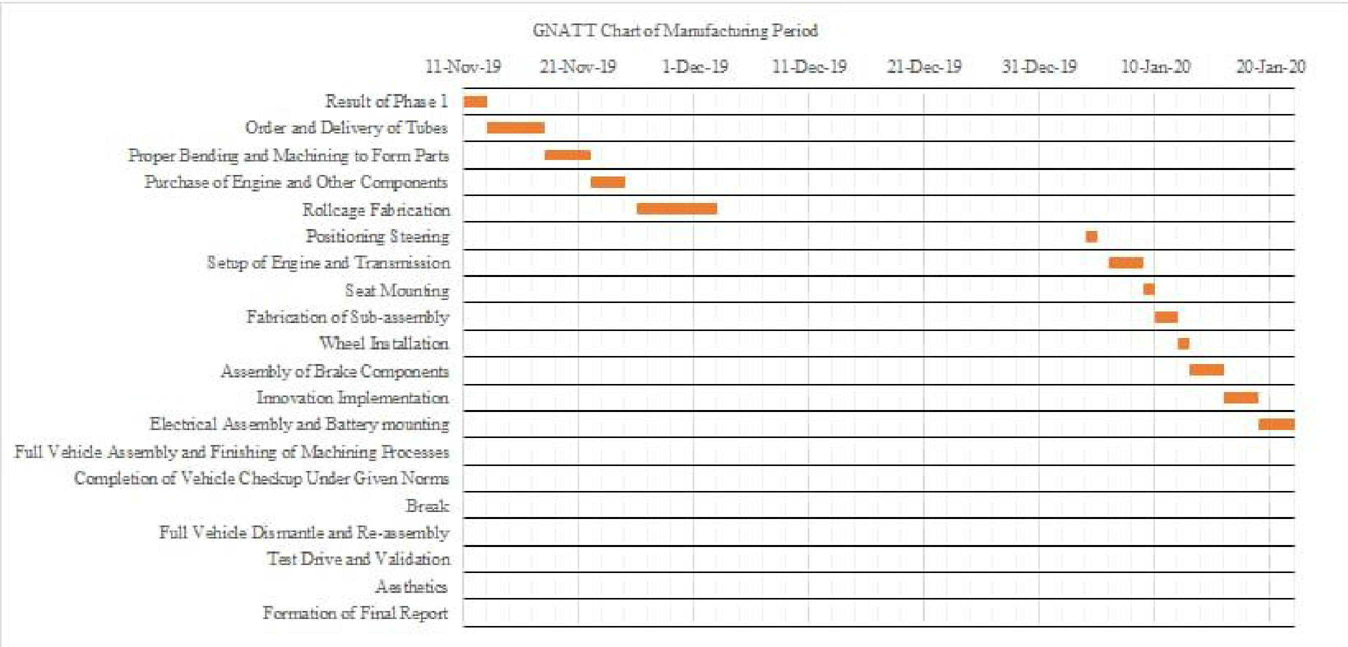
Design Failure Mode Effects Analysis (DFMEA)

S.No	FUNCTION	POTENTIAL FAILURE MODE	EFFECT OF FAILURE	POTENTIAL CAUSES OR MECHANISM FAILURE	S	O	D	RPN	ACTION TAKEN	O	S	D	RPN
1	Upright assembly	Imbalanced forces and over-steering	Loss of control	Imbalanced forces and over-steering	8	4	2	64	Design review and change in geometry	6	2	2	24
2	Frame	Improper welding, weak joints, corrosion, improper meshing, huge impact force on joints.	Breaking, bending (Permanent/ Structural damage)	Improper welding, weak joints, corrosion, improper meshing, huge impact force on joints.	9	5	7	315	Choosing proper weld material, proper welding on joints, corrosion resistant paint, high factor of safety)	6	5	4	120
3	Engine	Leakage in fuel tank, defective seals, improper mounting	Fire hazard, poor engine performance and poor fuel economy	Leakage in fuel tank, defective seals, improper mounting	9	5	3	150	Proper handling and mounting of engine, proper welding of fuel-tank and leakage check of fuel line.	6	3	3	54
4	Transmission Assembly	Improper/faulty chain-sprocket design, axle breaking due to fatigue	Power transmission inefficiency, Possibility of accident.	Improper/faulty chain-sprocket design, axle breaking due to fatigue	7	4	3	84	Exact coupling, proper lubrication, fatigue safety analysis of axle.	4	3	2	24
5	Steering System	Tie rod bending/ breaking, improper alignment, excessive load on tie rod	Possibility of losing control	Tie rod bending/ breaking, improper alignment, excessive load on tie rod	5	5	2	50	Regular checking of steering connection and meshing etc. Replacing the tie-rod in regular interval.	4	5	2	40
6	Braking System	Accidental damage of breaking line, leakage of breaking fluid	Brake failure, damage of the vehicle, severe injury of the driver	Accidental damage of breaking line, leakage of breaking fluid	10	5	2	100	Installation of brake linings above the frame, rigid mounting of the master cylinders, using of a "Parallel Dual-braking system"	7	3	2	42
7	Upright assembly	Imbalanced forces and over-steering	Loss of control	Imbalanced forces and over-steering	8	4	2	64	Design review and change in geometry	6	3	2	18
8	Electrical System	Faulty electrical circuit, faulty connection	Circuit break-down, short-circuit, fire hazards	Faulty electrical circuit, faulty connection	5	5	2	50	Proper circuit design, using high quality cables, regular checking of connections.	3	3	18	50
9	Drive Shaft	Twisting and Bending	Failure of Transmission System	Lower thickness,lower diameter,Lower material quality	7	8	6	336	Increased Diameter of shaft	4	5	6	120

PROCESS FAILURE MODE EFFECTS ANALYSIS (PFMEA)

S.No	FUNCTION	POTENTIAL FAILURE MODE	EFFECT OF FAILURE	POTENTIAL CAUSES OR MECHANISM FAILURE	O	S	D	RPN	ACTION TAKEN	O	S	D	RPN
1	Tig Welding	Thermal failure	Change in material property	Excessive heat generation	5	4	5	100	Interval between welding two joints is increased and welded joints are given enough time to cool down	5	2	5	50
2	Cutting operation	Material deformation	Rough cut and improper finishing	Too much heating of material due to holding the grinder for long time	6	4	5	120	periodically operated cutting and proper surface finish	4	4	3	48
3	FRP manufacturing	Crack formation	Crack in body panel, weakening of front, side and rear bumpers	Air pocket between layers	5	3	5	75	proper folding of material and rolling of layers	4	3	3	36
4	Axle mounting	Misalignment	Vibration, Uneven traction, excess wear of axle tire	Eccentricity in lathe machining operation of the axle. Vertical misalignment of bearings. Improper bearing mounting	6	5	5	150	Careful centering in lathe machining of axle. Mounting bearings in properly aligned bearing houses	5	3	4	60
5	Braking system	Improper air bleeding	Compressible air bubbles causes reduction in braking effect	Air bubbles stuck in brake fluid	6	4	5	120	Checking emission of air bubbles in brake fluid chamber by immersing with some excess fluid	4	3	4	48

Gantt Chart of Manufactueing Period



TEAM MEMBER ROLES:

TEAM MEMBER ROLL	TEAM MEMBER NAME
Team Captain	Sakib Sadat Shondhi
Vice Captain	Fahmid Hasan Dipta
Technical Head	Nazmul Haque Nafisa Raihana
Logistics and Sponsors	Sakib Sadat Shondhi Ahmed Sabul Masani Susmit Das Gupta
Go Kart Driver	Rahat Chowdhury Ishat Raihan Jamil
Marketing Sector	Sakib Sadat Shondhi Wahid Tausif Tahmid Kalam Ramim
Financial Sector	Sakib Shadat Shondhi Fahmid Hasan Dipta Nasim Mahmud Fuad
Frame Design & Analysis	Anutam Bairagi Subah Mubassira Umaeer Al Hammad Nudrat Nawal Rashed Nizam
Steering System	Shoummya Shouvik Fardin Ishtiaq Araf Tousif Sadman Nabil
Transmission	Nazmul Haque Irtesam Nasrat
Braking System	Nazmul Haque Md Fuad Amin Jarif
Electric System	Wahid Tausif A N M Fuhad Ul Islam Munkasir Ahnaf
Material Testing	Nafisa Raihana Shoumya Shouvik Mehrab Hossen Siam
Ergonomics	Sadib Fardin
Innovation	Nazmul Haque Nafisa Raihana Irtesam Nasrat