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Design and Development of E-Kart

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Abstract: This paper represents the design and development of E-kart. This is Electric kart powered using batteries and motor and there won't be any harmful hazardous gases emissions with this kart this is the main objective behind development of the kart. In this kart we have used BLDC motor and lithium ion batteries. The motor used is 900 watt power and the batteries are 27 amp each we have use four batteries. The batteries are rechargeable. The chassis made of AISI 4130 material and it weighs around 18 kg and the whole kart is about 140 kg. This paper includes design and calculation of all components like batteries, motor, brake, etc. The maximum speed of the kart is 40 km/hr i.e. 11.11 m/s. It can go up to 45 km the motor is of 1.20 hp. International standards were followed for all the calculations.

I. INTRODUCTION

The first go-kart was invented by Art ingles in 1955, the go-karts are powered by fossil fuels which leads to emission of harmful hazardous gases like carbon monoxide, nitrogen oxides, hydrocarbons, etc. Whereas E-kart are powered by batteries and motors so there will not be any emission ans do not cause any harm to atmosphere. In this project we aimed to design and develop a electric kart and this paper represents the design and calculation of the E-kart. The material used for the chassis we used is AISI 4130 in this chassis has been welded through MIG welding. The material we used is flexible as well as tough so that it can sustain all the weight and shock. And we compared this AISI material with other materials and found that AISI 4130 is the best for the chassis. The chassis we used is three dimensional chassis. The material AISI 4130 is a low alloy steel containing molybdenum and chromium as a strengthen agent. The steering system is Ackerman steering system. Because Ackerman principal prevent tyres from sliding outward. The motor used is BLDC motor this motor has high speed and torque. And the battery used is lithium ion battery which is rechargeable battery. I. The braking system used is disc brakes these are the type of brake which uses callipers.

II. LITERATURE VIEW

"ELECTRIC VEHICLE TECHNOLOGY EXPLAINED" by J. Ma, John Wiley and John Lowry published by John Wiley and sons limited.
Year of publication : 2017
The manufactured Electric Vehicle is in line with reality, according to the test findings, and measuring the EV's energy consumption per kilometre, driving range, and equivalent emissions using official driving cycles results in huge relative mistakes.
Prof. Nirmal Chohaun has written report on "DESIGN AND FABRICATION OF ELECTRIC GO-KART" published on IRJET.
Year of publication : 15 SEP 2020
The goal of this report is to design and build a working model of an electric go-kart. The go-design kart's and construction are basic enough that even nonprofessional drivers can run it. The design is based on the vehicle's great strength, which allows it to carry more weight and deliver the best services at a low cost.
D. Karale, S. Thakre, M. Deshmukh "DESIGN AND ANALYSIS OF ELECTRIC VEHICLE", IJOI vol.4.
Year of publication : 2020
The EV's chassis was designed and developed based on assumptions about the vehicle's gross weight for carrying a suitable size of sprayer attachment, taking into account the agronomical requirements of the field crops available in the region, and validated using the Finite Element Method (FEM) with ANSIS software.
"ACKERMAN STEERING GEOMETRY" by Jonathan Vogel.
Year of publication : 2017

Steering geometry is one of the many options available to race car designers to guarantee that the car gets the most out of all of its tyres. The geometric setup that allows both front wheels to be steer at a definite angle to avoid tyre slippage is known as Ackermann Steering.

"Electric-Solar Vehicle Steering Mechanism Design and Analysis" M. A. Hmed, M. A. Ubin, A. Haik, M. A. Hmed, M. A. Ubin, M. A. Ubin, M. A. Ubin,

The purpose of this research is to construct and evaluate a steering system for an electric solar vehicle. The steering mechanism is taken into account using the Ackerman steering theory. The steering effort is made to the steering wheel to rotate the rack shaft, which is connected to a pinion gear that converts rotary motion into linear motion through the rack and pinion steering system, which aids in vehicle steering smoothness.

“Electrical Motors for Electric Vehicle – A Comparative Study” by P. Bhatt, Hemant Mehar, M. Sahajwani published on SSRN Electronic Journal.

Year of publication : 3 April 2019

A comparison of the most common kinds of electric motors in use across time is offered, along with their efficiency, power density, reliability, and size.

A article published by RATHEESH NAIR in Evreporter-e-magazine on.

Year of publication : March 2017

Electric power trains are more efficient and environmentally friendlier than gasoline locomotives. Advanced motors and motor controllers are responsible for the performance and efficiency of electric power trains. BLDCM (Brush Less Direct Current Motor) or PMSM is used in the majority of electric power trains nowadays (Permanent Magnet Synchronous Motor). In comparison to traditional motors, these motors are highly light and efficient, and a motor controller is required for them to function.

“A FUTURE FOR SOLID STATE BATTERIES” by J. Janek, Wolfgang G. Zeier

Year of publication : 2016.

Solid-state batteries have recently attracted great interest as potentially safe and stable high-energy storage systems. However, key issues remain unsolved, hindering full-scale commercialization.

W.Salah, M Albreem “CONTROLLERS FOR ELECTRIC VEHICLE TECHNOLOGY” by
IJOPE,

Year of publication : 1 MAR 2019.

The use of digital signal controllers compared with conventional control systems minimizes the motor’s total harmonic distortion, lowers operating temperatures, and produces high efficiency and power factor ratings.

III. CALCULATIONS

A. Design and Calculation of Knuckle Joint

1) Design of Knuckle Joint

Knuckle joint has three main parts which are knuckle pin, eye end and fork (stub axle). It is used to connect front wheels with the chassis.

(All design and calculations are done as per design standards)

a) The diameter of rod, d= 17 mm

b) Thickness of eye end, t=1.2*d
= 20.4 mm

c) Outside diameter of eye end,
D1 = 2d
= 34 mm

d) Diameter of knuckle pin,
Dp = d= 17 mm

e) Diameter of knuckle pin head and collar,
D2 = 1.5*d
= 25.5 mm

f) Thickness of fork,
T2 = 0.75*d
= 12.75 mm

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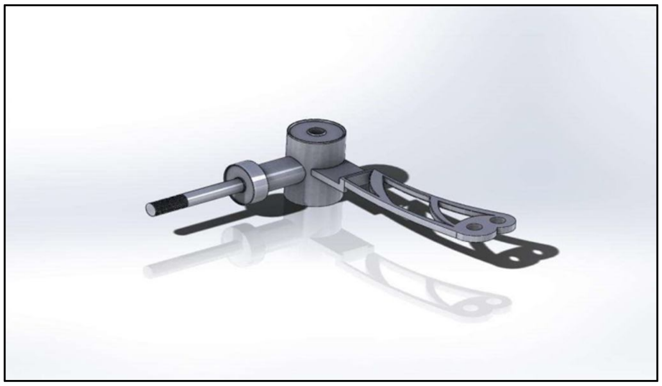
g) Thickness of knuckle pin head and collar,
T3 = 0.5*d
= 8.5 mm

h) Thickness of eye end,
T1 = 1.2*d
= 20.4 mm

i) Length of eye end,
L1 = 4.8*d
= 81.6 mm



3.1.1.a FIG: KNUCKLE JOINT

2) Analysis of Knuckle Joint

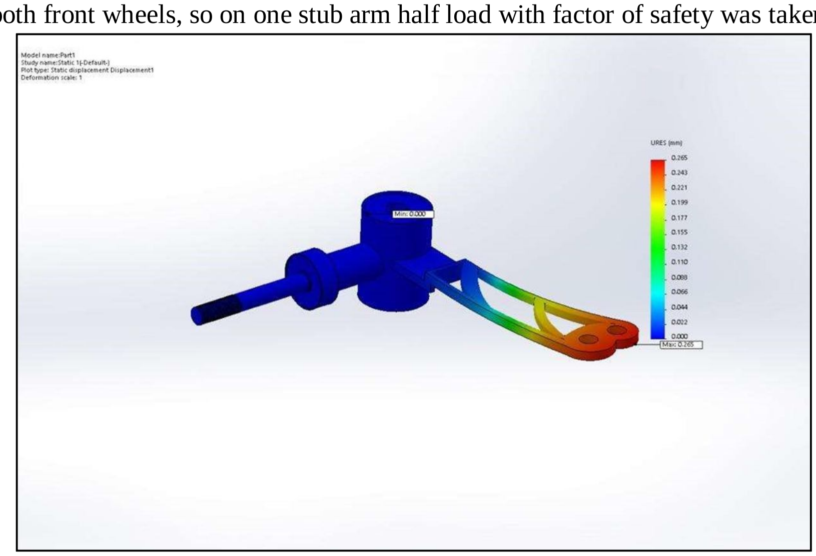
Structural Analysis Of Stub Arm Of Knuckle

Mass (in kg) on the front side of the vehicle is 56 kg, thus weight on one wheel stub axle would be half of the front weight.

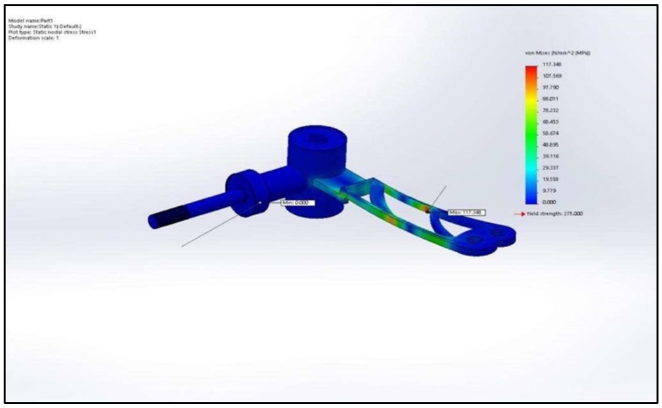
a) Normal force on stub arm,
N= m*g
= 56*9.81
= 549.36 N

b) Lateral force on stub arm,
F= mv2/r
= 2160.06 N

Since, the load applied on both front wheels, so on one stub arm half load with factor of safety was taken.



3.1.2.a FIG: TOTAL DEFORMATION



3.1.2.b FIG: VON-MISSES STRESSES

Lateral force of 1080.03 N is applied on a stub arm, thus 0.265 mm of deformation is occurred. Internal stresses generated due to load is 117.348 MPa.

B. Design of Chassis

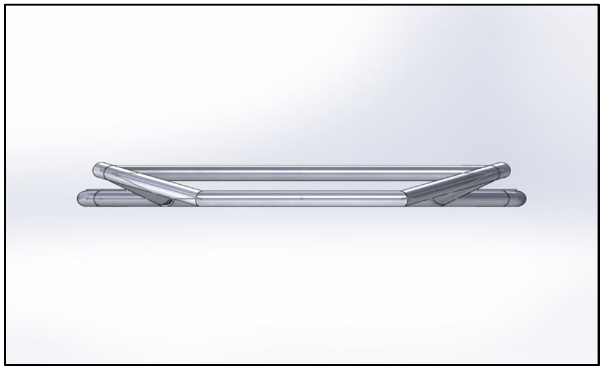
Chassis is the main component of any vehicle on which mounting of parts are placed. It should have high strength and stiffness, stability, torsional rigidity as well as it should resist the thermal stresses generated due to load. The following factors are considered to select the material for chassis

1) Purpose of utility
2) Strength
3) Weight
4) Cost
5) Size

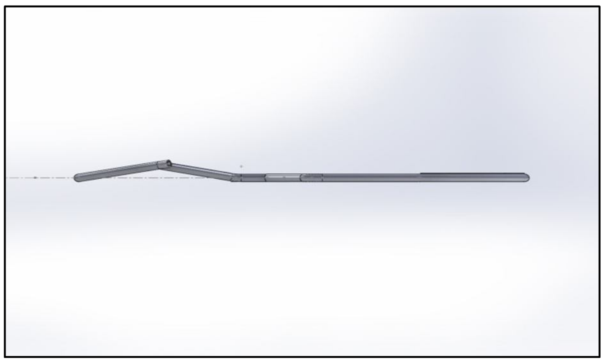
The material selected for design of chassis is AISI4130 an alloy steel. The total weight of the chassis is 18.016 kg.
The properties of material for selection of chassis are as follows:

PROPERTIES	AISI 4130
MODULUS OF ELASTICITY	190 G Pa
SHEAR STRENGTH	340 G Pa
FATIGUE STRENGTH	320 M Pa
TENSILE YIELD STRENGTH	460 M Pa
POISSON RATIO	0.29
DENSITY	7.8 g/cm3

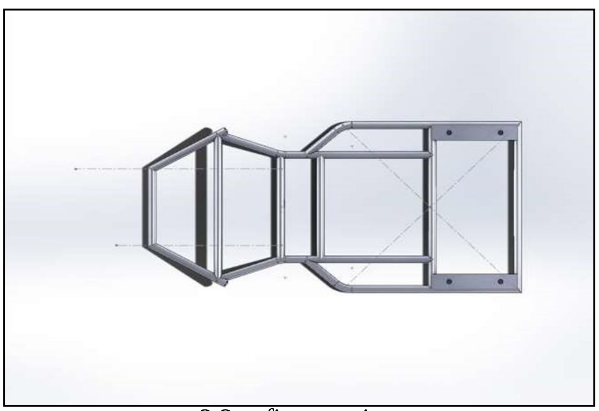
C. Cad Design Of Chassis



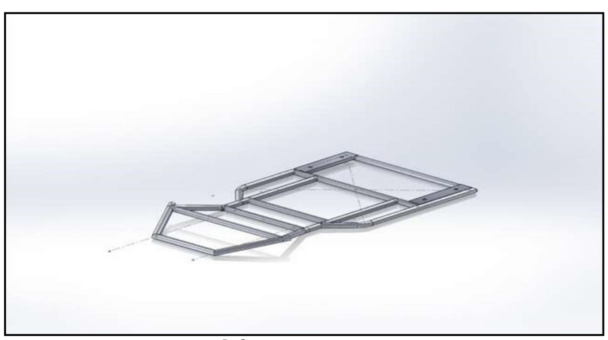
3.3.a Fig: Front View



3.3 b fig: side view



3.3 c fig: top view



3.3 d fig: iso-metric view

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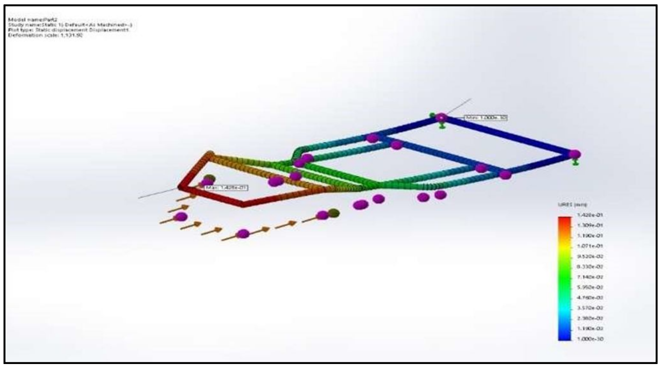
1) Calculation of Front Impact Test

For the front impact during collision, the force is acting on frontal members of chassis and rear end is kept fixed. Front impact was calculated for maximum speed of 40 km/h of the vehicle. Time of impact is considered as 0.8 sec. From impulse momentum equation,

F x t = m x (Vi – Vf)

F x 0.8 = 140 x (11.11 – 0)

F = 1.944 KN



3.3.e FIG: Deformational Analysis of Chassis

2) Rear Impact Test

Consider the worst case collision for rear impact test, the force is calculated similar to front impact test. Force was calculated in rear impact for optimum speed of 40 km/h and time of 0.8 sec.

Using impulse momentum equation,

F x t = m x (Vi – Vf)

F x 0.8 = 140 x (11.11 – 0)

F = 1.944 KN

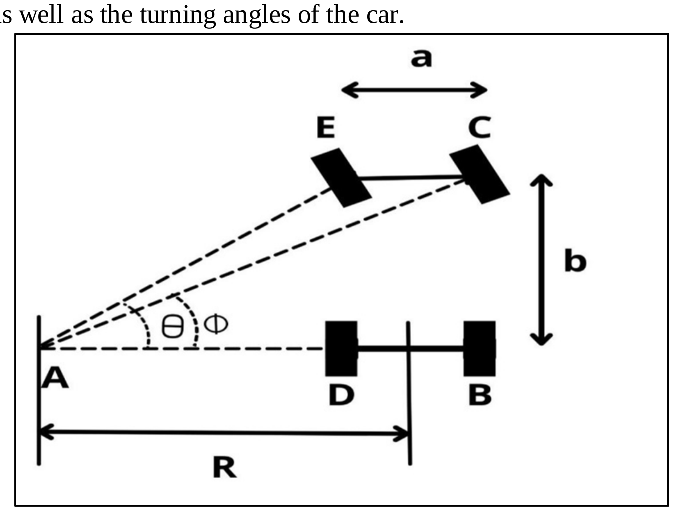
D. Design of Steering

In our E-kart project we are using Ackermann steering system . This system is used because turning time of vehicles Ackermann steering geometry prevents the tyre from sliding outwards. Ackermann steering system uses shorter tyre rods and with the help of shorter tyre rods the capacity to control the vehicle increases.

Steering system calculation are as follows :-

While the designing of Ackermann steering system the satisfaction of the driver and also the safety of driver is considered. So , the following parameter are set up , a = Wheel track width =1105mm b = Wheel base =1050mm

By using geometric relation , for validation of result more analytical approach is applied. Ackermann steering calculation are performed to find the turn radius as well as the turning angles of the car.



3.4.a Fig. Ackermann Steering Geometry

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Let , A -centre of rotation R -turning radius a -track width b -wheel base θ – angle of inside lock

Φ – angle of outside lock

So , the turning radius R = track width/2 + wheel base /average steer rate

R = a/2 + b/sin30

=1105/2+1050/sin

R =2652.5mm approximately 2653 mm

Therefor Turning radius R= 2.653m

Inner angle (θ) :-

For θ , consider triangle ADE where angle D = 90°

Therefor tan θ =DE/AD

From figure , DE =CB = b =1050 mm ----- (1)

AD = R- DB / 2

= 2652.5-552.5

AD = 2100 mm ----(2)

Therefor , tan θ = b/AD = 1050/2100 = 0.5

θ = tan ^-1(0.5)

$$\boxed{\theta = 26°}$$

Outer angle (Φ) :-

For Φ , consider triangle ABC where angle B =90°

Therefor tan Φ = CB / AB

= b / (R +a/2) from equation (1) CB = b

= 1050 / (2652.5+1105/2) = 1050/3205 tan Φ = 0.327

Φ = tan^-1(0.327)

$$\boxed{\Phi = 18.1336°}$$

From figure , AE - Inner radius of front wheel

AC – Outer radius of front wheel

AD – Inner radius of rear wheel

AB – outer radius of rear wheel

Therefor , AE =√AD^2+DE^2

= √2100^2+1050^2 -- From equation 1 and 2

<u>AE = 2347.87 mm</u>

AC =√AB^2+BC^2

=√(2652.5+552.5)^2+1050^

<u>AC = 3372.61 mm</u>

And AD = 2100 mm --- from equation (2)

<u>AB =3205 mm</u>

Ackermann Angle (α) :-

Tan α = c/2b

Where , c – distance between pivots of the front axle = 900 mm

b – wheel base

Therefor , Tan α = 900/2*1050

α = tan ^-1(0.4285)

$$\boxed{\alpha = 23.19°}$$

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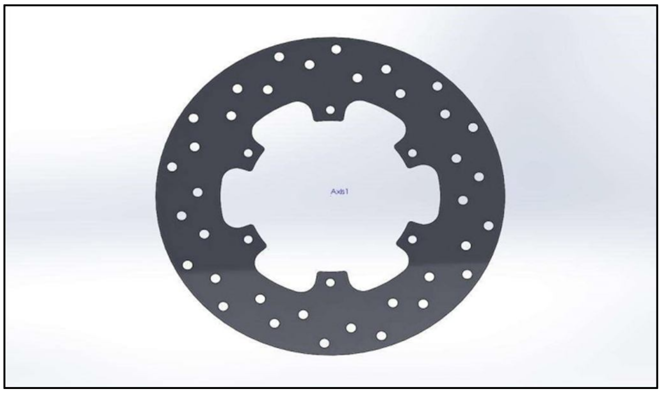
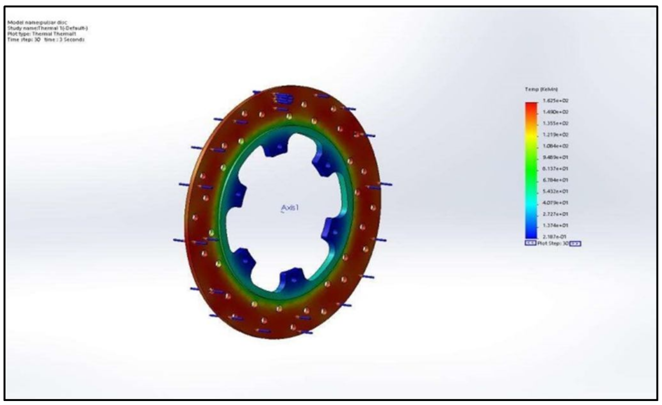
E. Braking System Design

The braking system used is Disc brake as it is very efficient.



3.5.a Fig. Disc Brake



3.5.b Fig. Analysis of disc brake

1) 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: 350 N)= pedal ratio *force on the pedal / area of master cylinder = $4*350/(\pi/4)*(14)^2$

= 9.09 MPa

2) Clamping force (CF)

Type	Specification
Rear disc	180mm
Master cylinder diameter	14 mm
Calliper piston diameter	20.1 mm
Brake pedal ratio	4:1

CF= brake line pressue *(area of calliper piston*2)

$= 9.09*((\pi/ 4) * (20.1)^2 * 2)$

=5768.67 N

3) Rotating Force

RF= CF* number of caliper pistons *coefficient friction of brake pads
= 5768.67 *0.3*2
=3461.16 N

4) Braking Torque

BT= rotating force* effective disc radius
= 3461.16*0.09
=311.50 N-m

5) Braking Force

= (braking torque /tire radius) * 0.8
= (311.50/0. 14) * 0.8
=1780 N

3.5.6 Deacceleration f=-rna(-ve sign indic ates force opposite direction)
a=-BF/m= -1780/140 = -12.7 m/s

3.5.7 Stopping distance:

$v^2 - u^2 = 2as$ (v=0, u=11.11m/s) Stopping Distance = 4.85 meters.

F. Motor

The motor used is 48 volt and 900 watt these specification motor used according to the calculation given below.

Consider the total force acting on E-Kart are,

Ftotal = Frolling + Faerodynamic drag

Hence,

$F_{rolling} = C_{rr} * m * g$ where,

C_{rr} = coefficient of rolling resistance = 0.01

m = total mass of kart = 140 kg

$\boxed{F_{rolling}} \boxed{= 13.734\ N}$

$F_{aerodynamic\ drag} = Cd * 0.5 * ʄair * A$ where, Cd = 0.8

ʄair = 1.2 kg/m3

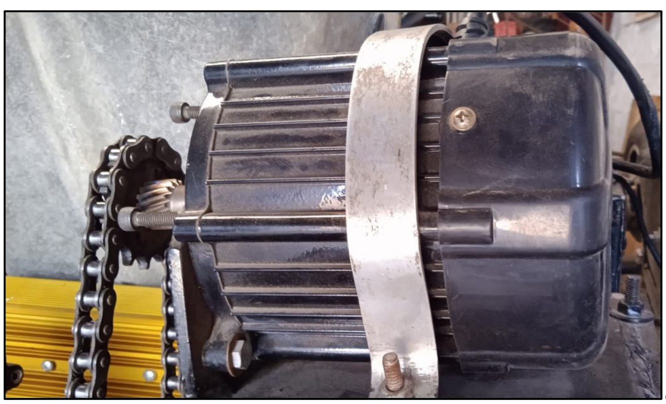
$\boxed{F_{aero} = 71.41\ N}$

Therefore, Power required to drive the vehicle,

Pr = Fr * v = 13.734 * 11.11 = 152.58 W

Pd = Fd * v = 71.41 * 11.11 = 793.41 W

Power, P = Pr + Pd = 945.39 W



3.6.1 fig: motor

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G. Battery

For power transmission battery is used the specifications of battery are it is 48volt and 27 amp. Such four batteries are used for transmission. The calculation of the battery is given below

Consider, the vehicle runs at 40 km/hr for the range of 45 km.

Travel Factor = Total Range / Total Speed

= 1.125

V * AH = 900 ,   Where V is volt of motor

48 * AH = 900

$\boxed{AH = 18.75\ A}$

Take accl. current loss = 5%

18.75 * 1.05 = 19.687 A

Power = V * I = 945 W

Power * Travel Factor = 1063.125 W   Consider Li-ion batteries have charging and discharging efficiency is 85%.

So,

The battery pack capacity required = 1.063/0.85

=1.25 kwh

Also consider the motor have efficiency of 85%.

Hence,

Total battery capacity required = 1.25/0.85

= 1.47 kwh

IV. RESULT

A. Analysis of Chassis

The maximum load is applied on the front part of chassis, thus 0.1428 mm of displacement occurred.

SR. NO	IMPACT	FORCE (KN)	TOTAL DEFORMATION (mm)	EQUIVALENT VON MISSES STRESSES (MPa)
1.	FRONT	1.944	0.1428	1.676

B. Steering System

The steering is design in such a way that the turning radius of vehicle will be minimum, also the safety of driver kept in mind and over steering condition is reduced.

SR. NO	TURNING RADIUS (m)	ACTUAL TURNING RADIUS (m)	ACKERMANN ANGLE (° C)
1.	2.6	4.5	34.68

V. CONCLUSION

When compared to AISI 1020, AISI 4130 is a better material in terms of strength, reliability, and performance. It can also be used in large-scale manufacturing. The chassis of a CAD modal was successfully analysed using SOLIDWORKS to determine equivalent stresses and total deformation results. The electric motor, battery and drive train have been chosen to provide maximum performance in terms of both speed and battery efficiency. When approaching a turn, the steering system designed for the kart requires less effort.

After all of the calculations and analysis, it has been determined that this go-kart is safe to fabricate under healthy engineering conditions. Practices are followed, and the performance goals are met.

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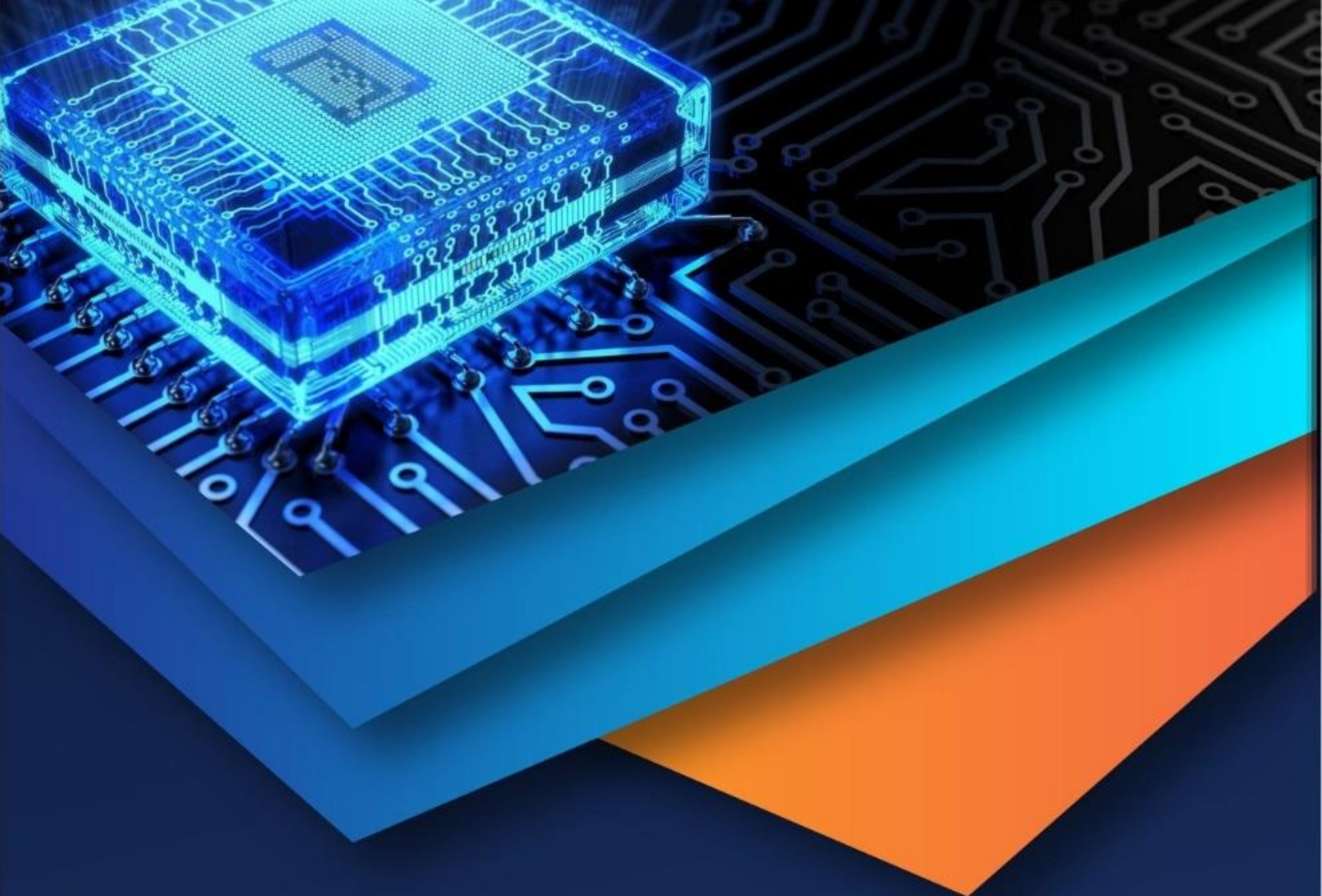

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