

Design of an Electric Golf Cart with Batteries & Solar Panel for 6 Seat Capacity

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Abstract - In this paper, the Design & Calculations of an Electric golf cart is done, the calculations and process to develop a golf cart chassis, electrical transmission using transaxle, braking, steering geometry, suspension system, solar panel of a long six seat golf cart, battery calculations and controller which monitors the electric system is explained. All the materials used for the calculation are industry standard being utilized currently. Priority of this design was to keep the cart as light weight as possible and rigid with good performance on grassy and concrete terrain's used in either golf course or in hospitality / education sectors. A unique tubular chassis is used in the design comprising of circular & rectangular shape. The focus of design was long mileage from the batteries used and how much, adding solar panels can contribute to the existing range of the electric vehicle. A 23 miles range from the battery pack using 6 deep cycle lead acid batteries was achieved and increased 12 miles from the solar panels under the constraint assumed for its use (direct sunlight for over 6-7 hours)

Key words: electric golf cart, design, Electric vehicle, electric transmission, vehicle design, battery calculation, solar golf cart.

I. INTRODUCTION

This projects aim is to Design an electric golf cart with six people seating capacity which runs on batteries as well as solar panels. The need for a golf cart isn't just using it in the golf course and is nowadays being used in large universities spread over vast acres of land and in expensive hotels and latest complexes for comfort traveling i.e Hospitality industry and providing low cost of charging with increasing mileage as technology advances. In this project the aim has been to put forth an industry standard design for an electric golf cart that can run exceedingly well on grassy / muddy terrains as well as wet conditions and concrete roads. During designing all the constraints and cost considerations and all technical aspects were set according to the current advancements in the industry in a thoroughly researched manner. A battery electric vehicle (BEV) has far fewer moving parts than a conventional gasoline-powered vehicle. There's no need for a liquid fuels or oil changes. There's no transmission or timing belt to fail you. And most of the maintenance cost associated with ic engine are eliminated.



Figure 1 Golf cart solid works model

Customer needs and requirements are of prime importance regarding comfort, high reliability, economic and performance and long mileage with regards to an electric golf cart. A lot of engineering and reverse engineering tactics went into the designing phase and many alterations and modifications were done to improve the design. Solid works model of the entire vehicle has been made and assembly has been done of all important components. The solar panels greatly improve the vehicle's mileage performance and battery life in a certain way which is explained in full report.

II. DESIGN DEVELOPMENT

The design development stage is where the design considerations are outlined and according to which under certain parameters, criteria and market research the following design is developed.

a. The design considerations set are so the vehicle can meet all safety standards & ergonomics and performance during its entire life cycle

- Safety and ergonomics
- Standardization as per industry
- Maneuverability
- Safe engineering practices
- High quality cost ratio
- Power and performance
- Stability and long product life

b. The design development stages are divided in many chapters they are:

1. Chassis
2. Transmission
3. Brake
4. Steering
5. Suspension
6. Battery
7. Solar panel

1. CHASIS

A vehicle frame also known as its chassis is the main supporting structure of a motor vehicle, to which all other components are attached, comparable to the skeleton of an organism. Since the 1930s, virtually every car had a structural frame, separate from its body. This construction design is known as body on frame. Over time, nearly all passenger cars have migrated to uni-body construction meaning their chassis and bodywork have been integrated into one another. Nearly all trucks, buses, and most pickups continue to use a separate frame as their chassis.

▪ Structure and material selection

The chassis chosen for the design is a tubular space frame consisting of Round tubes and Rectangular tubes both for a sturdy construction. A space frame was chosen because of its simplicity of design and strong torsional rigidity and moderate difficulty in design and building practically in a cost effective manner. Round tubes used are primary 1.5 inch, secondary 1 inch with 0.112" thickness aisi 4130 chrome moly pipe and Rectangular one is primary 4x2x0.25 & secondary 3x2x0.25 chrome moly aisi 4130 rectangular tube. Different sizes of pipes and shapes are used to lighten the weight of the chassis and give it a modest look and not make it bulky and rectangular tubes are used only at the bottom of the chassis for better mountings of the standard components and its thicker and provides a strengthened base compared to round pipe as our vehicle is long.

AISI 4130 steel is used for its light weight and high strength ratio which suits best as compared to, for our cart even though it is high in cost, it is more durable and rigid. AISI 4130 is a low alloy steel containing molybdenum and chromium as strengthening agents. The carbon content is nominally 0.30% and with this relatively low carbon content the alloy is excellent from the fusion weldability standpoint.

Table 1 Chemical composition of material

Table 2 Mechanical Properties

AISI	4130
Carbon	0.28-0.33
Chromium	0.8-1.1
Manganese	0.7-0.9
Molybdenum	0.15-0.25
Phosphorus	0.035 max
Silicon	0.15-0.35
Sulphur	0.04 max

Properties AISI	Data
4130	
Density	7.7-8.03
Poisson's ratio	0.27-0.3
Elastic Modulus	190-210
Tensile strength	560.5
Yield strength	360.6
Elongation	28.2
Reduction in area	55.6
Hardness	156
Impact strength	61.7

The dimensions of our chassis according to six seat is ergonomically built with moderate aerodynamic considerations, as for a golf cart aerodynamics is not prime criteria.

The dimensions are selected,

3500 x 1200 x 1825 (LxBxH)

In which 3150 wheel base is found to be comfortable and 1200 is track width as per proper assemblies of transaxle and steering and all suspensions so the geometry provides with well spacious seating arrangement as well as some space on the back to keep storage.

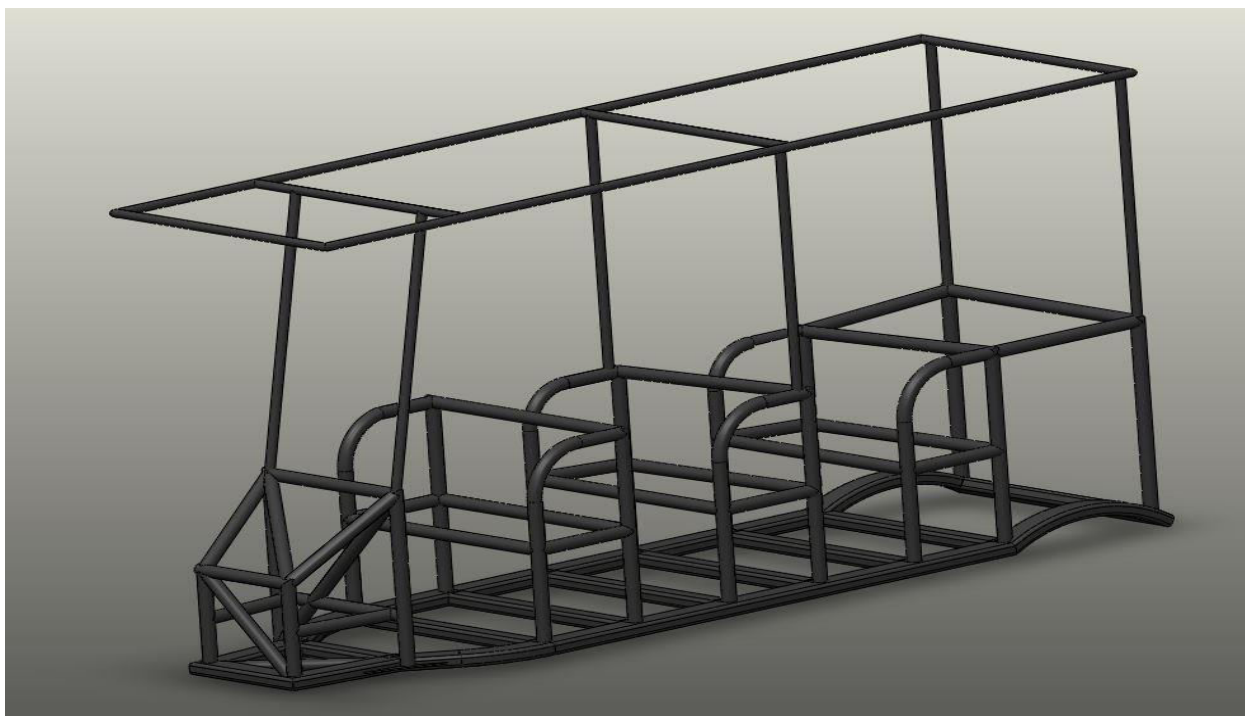


Figure 2 Chassis Solid works model

2. TRANSMISSION

A transmission is a system which transmits power and provides controlled application over that power. In layman terms a transmission usually consists of a set of a gearbox that uses gears and gear trains to provide speed and torque conversions from a rotating power source to the output.

The main components of our transmission assembly are Transaxle, Motor and its Controller

In an electric golf cart the need for using a separate gearbox is eliminated because of electric transmission and use of motor in place of an engine. Instead a single speed reduction Transaxle is used. A transaxle is an assembly of differential gear with one step reduction coupled with rear axle and hub which is used to directly mount on the chassis.

With market research there are various standard transaxles for electric golf cart with preset reduction gear. One such transaxle with a reduction ratio of **10.45** is chosen for our electric golf cart from various reduction ratios already available. The reason for choosing this ratio was it gave us the best speed for our cart with full loading condition and its cost effective.

Components & specifications of transmission

1. Motor – 4kw Dc brushed 3000 rpm (set rpm for transaxle ratio) 14 Nm torque
2. Transaxle with reduction ratio 10.45
3. Curtis controller

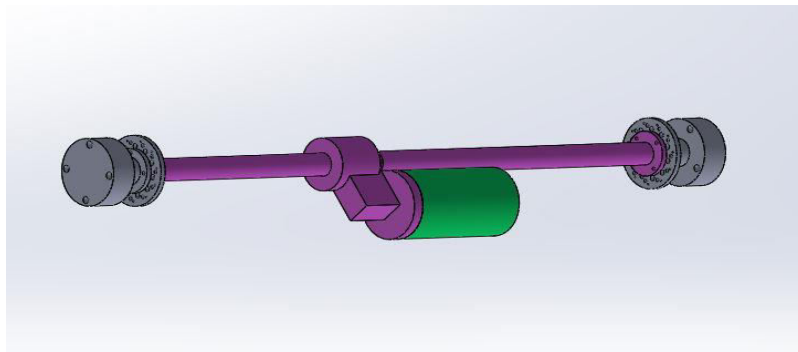


Figure 3 - Transaxle with motor disc brake & hub assembly

A. Transmission Calculations

Transmission calculations give the power transmitting values through the system in terms of Top Speed and Acceleration and the Time required to achieve it with the selected parameters for our electric golf cart and the total tractive effort.

➤ Parameters considered:

1. Motor 4kw with 3000 set rpm
2. Transaxle reduction ratio 10.45
3. Tire 18x8.5x8 (lxbxh)

1. Top speed

$$\begin{aligned} \text{Rpm of wheel} &= \frac{\text{Rpm of Motor}}{\text{Transaxle reduction}} \\ &= 3000/10.45 = 287.081 \text{ rpm} \quad \dots\dots (1) \end{aligned}$$

Rpm / 60 = Rps hence

$$\frac{287.081}{60} = 4.78 \text{ rps} \quad \dots\dots (2)$$

- [Speed = Rps x Circumference of the tire]

$$Speed = 4.78 \times \left(\pi \times 18 \times \frac{25.4}{1000} \times 0.95 \right)$$

Top Speed = 6.53 m/s or 23.5 km/hr.

2. Acceleration & Time for it

The total force is calculated first produced by the motor and Torque on the wheels and Resistance losses such as rolling resistance and aerodynamic drag are considered giving us a practical value for the cart.

➤ *Parameters:*

- Front area of the golf cart = (1825x1200 mm) height x breadth
- Motor torque - 14 Nm
- Reduction ratio - 10.45

Torque on wheel = Motor torque x Total gear reduction ratio

$$T = 14 \times 10.45 = \mathbf{146.3 \text{ Nm}} \quad \dots(3)$$

Now using formula

Torque on wheel = Force x Radius of tire

Where,

Radius of tire = $9 \times 0.95 \times \frac{25.4}{1000} = \mathbf{0.21717 \text{ m}}$

$$\text{Therefore, } F = \frac{T}{R} = \frac{146.3}{0.21717} = \mathbf{673.66 \text{ N}} \quad \dots (4)$$

This force acts on our vehicle for maximum speed at maximum set rpm

Now there are various resistances that occur in the real driving conditions such as rolling resistance and aerodynamic drag which needs to be considered for an optimum design

1. Rolling resistance = μmg
2. Aerodynamic drag = $\frac{1}{2} \rho Av^2 cd$

Where,

μ = coefficient of rolling resistance (0.04)

ρ = air density (1.2)

C = drag coefficient (0.4)

m = mass of the vehicle(950kgs)

A = Area of frontal impact (m^2) (2.16)

g = gravitational force (9.81)

V = flow velocity (vehicle)(m/s)

Total resistance losses = Rolling resistance + aerodynamic resistance

$$= \mu mg + 1/2 \rho Av^2 cd$$

$$= 0.04 \times 950 \times 9.81 + 0.5 \times 1.2 \times 2.16 \times 6.53^2 \times 0.4 = \mathbf{394.88 \text{ N}}$$

Now removing the resistance force from the Force obtained earlier,

$$\text{Force} = 673.66 - 394.88 = \mathbf{278.78 \text{ N}}$$

Now that we have force, we can use equations of motion,

Force = Mass x Acceleration, to find out the acceleration.

Therefore,

$$\text{Acceleration} = \text{Force} / \text{Mass}$$

$$= 278.78/950 = 0.295 \text{ m/s}^2$$

Now using $V = u + a.t$, u being initial speed is 0 hence can be taken out of the equation

Therefore,

$$t = v/a \quad \text{where, } v \text{ is final velocity}$$

u - initial velocity

a - is acceleration

$$t = 6.53/0.295 = 22.13s$$

Another equation is used to determine the distance it will take to achieve that acceleration in the time obtained above given as,

$$s = ut + \frac{1}{2}at^2$$

where, S is distance expressed in meters

Since initial velocity is 0,

$$s = \frac{1}{2}at^2$$

$$s = \frac{1}{2} \times 0.295 \times 22.13^2 = 72.23 \text{ m}$$

The transmissions Results obtained so far according to design are mentioned in a tabular form

Table -3 Results

Parameters	Results
Top speed of Golf cart	23.5 km/h
Force transmitting	278.78N
Acceleration	0.295 m/s ²
Time	22.13s
Distance	72.23m

Note: These values are calculated under full loading condition of the vehicle

3. Total tractive effort

The total tractive effort (TTE) is the sum of the forces of rolling resistance, grade resistance and acceleration force. When selecting drive wheel motors for mobile vehicles, a number of factors must be taken into account to determine the maximum torque required.

To choose motor capable of producing enough torque to propel the vehicle, it is necessary to determine the total tractive effort (TTE) requirement for the vehicle

But first we will need to determine all the resistances & force mentioned above

- Vehicle design criteria:
 - Gross vehicle weight (Gawk) - 950kg
 - Weight on each drive wheel 118kg/1165N
 - Radius of wheel r - 9inch
 - Top speed - 23.05km/h
 - Acceleration time - 22.13s
 - Inclination gradient -20%
 - Working surface - grass/mud

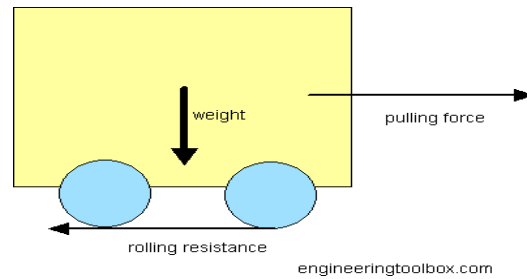


Figure 4 Force & Resistance Direction

a) Determining rolling resistance for TTE

Rolling Resistance is the opposing force that the vehicle has to overcome due to the rolling motion between the wheels and the surface of motion of the vehicle. The rolling resistance depends on the co-efficient of rolling friction which varies depending upon the material of tires and the roughness of the surface of motion. The Rolling resistance can be calculated as:

$$Rr = Gvw \times Cr$$

- Where, **Rr** - Rolling resistance
- **Gvw** – gross vehicle weight
- **Cr** – surface friction coefficient

Therefore,

$$Rr = 950 \times 0.05 = 47.5 \text{ kg} \times 9.81 = \mathbf{465.97N}$$

b) Determining grade resistance (Gr)

Contact Surface	C_{rr}
Concrete(good/fair/poor)	0.010/0.015/0.020
Asphalt(good/fair/poor)	0.012/0.017/0.022
Macadam(good/fair/poor)	0.015/0.022/0.037
Snow(2 inch/4 inch)	0.025/0.037
Dirt(smooth/sandy)	0.025/0.037
Mud(firm/medium/soft)	0.037/0.090/0.150
Grass(firm/soft)	0.055/0.075
Sand(firm/soft/dune)	0.060/0.150/0.300

Table 4 – Values of Coefficient of rolling resistance

Grade Resistance (GR) is the amount of force necessary to move a vehicle up a slope or grade. This calculation must be made using the maximum angle or grade the vehicle will be expected to climb in normal operation.

To convert incline angle, α , to grade resistance

$$Gr = Gvw \times \sin \alpha$$

where α is inclination considered 20% in this case

$$Gr = 950 \times \sin (20) = 324.91 \text{ kg} \times 9.81 = \mathbf{3187.45N}$$

c) Total tractive effort

The Total Tractive Effort (TTE) is the sum of the forces calculated in steps 1, 2, and 3

[Rolling resistance + Gradient Resistance + Acceleration force]

$$465.97 + 3187.45 + 280.25 = 3933.67N$$

$$\mathbf{TTE = 3933.67}$$

d) Determining the Torque required on the wheel (Tw)

To verify the vehicle will perform as designed in regards to tractive effort and acceleration, it is necessary to calculate the required wheel torque (Tw) based on the tractive effort.

$$T_w = TTE \times r \times R_f$$

Where, r is radius

Rf is Resistance factor (between 1.1-1.15) losses between caster wheel's axles and bearings etc

$$T_w = 3933.67 \times 0.229 \times 1.12 = 1008.97Nm$$

This torque can be obtained by directly mounting a motor with the torque value on the differential of the vehicle or by using a gearbox or chain drive to magnify a lesser torque to this value before it drives the wheel.

e) Reality check (Maximum tractive torque) (Tm)

When the required torque has been calculated it is necessary to check if the wheels of the vehicle are capable enough to transmit the required amount of torque for which the maximum torque that can be transmitted through the wheels need to be calculated. The maximum torque is given as:

Max Torque =

$$\mu \times G_{vw} \times r = 0.5 \times 950 \times 9.81 \times 0.229 = 1067.1Nm$$

$$1067.1 > 1008.97$$

Tmax > Tw, hence condition is satisfied.

3. CURTIS CONTROLLER

The 4kw BLDC motor drive is a high performance, cost-effective 3phase BLDC motor drive, which can provide power output Max 210 ah. The design is based on advanced DSP technology and feature high torque, low noise, low vibration, PID speed loop, PID current loop, over current protection, over load protection and a combined use of manual speed adjustment and automatic speed adjustment.

These motor controllers are used to control the speed output and rpm set of the motor.

It determines the amount of power to be transmitted and the rotational speed of the motor. Since motor is very powerful than its set state this controller provides a stable environment for it and it also controls the idling of the motor.

	Parameter	Values
1	Available Battery Voltage	48V, 60V,72V
2	Available Motor Power	1.2KW - 5KW
3	Phase Peak Current (Few Seconds)	210 A (The bigger, the higher torque)
4	Continue Current	40 A Peak 80A
5	Available Motor Angle	60 / 120
6	Working Frequency	15.6KHz
7	Working Efficiency	< 90 %
8	Control Method	Throttle; Potentiometer (Customized)
9	Default Throttle Working Voltage Range	0.8V - 3.8V, others have to customized
10	Mosfet's /Pieces	Domestic 24 Tubes
11	Case Material / Color	Aluminum /Sliver

Table 5 – Controller specification

4. BRAKE

Brakes are used to reduce the velocity of vehicle within less possible time with fewer disturbances in stability of vehicle. As per the design requirements; all wheels should be stopped at same time and should be equipped with hydraulic disc brakes

The disc brakes are chosen over drum because to provide a disk brake caliper mechanism that is easy to service, and that requires minimal disassembly to change the replaceable brake pads within the brake caliper assembly. A further object is to provide an extremely compact construction for a robust hydraulic disk brake assembly which is able to fit within the very confined space in the vicinity of the wheel hub and wheel rim of a small-size off-road vehicle such as a golf cart.

After a thorough market research certain parameters and industry standard components were selected to meet required comfortable braking distance and ride quality assuring safety standards

➤ *Parameters*

1. Pedal force - 17kgs ~ 170N
2. Pedal Ratio - 6:1
3. Tmc diameter - 19mm
4. Caliper diameter - 40 mm
5. Brake disc diameter - 190 mm
6. Wheel size - 18 inch
7. Mass of vehicle - 850 kg

a) Design of brake: Calculations -

According to the seating in golf cart an optimum pedal force is selected with 6:1 ratio

Pedal force x pedal ratio = 170 x 6 = **1020N** Force applied to the brakes

Now,

$$1. \text{ Pressure} = \frac{\text{Force}}{\text{area of Tmc dia}} = \frac{1020}{\left(\frac{3.14}{4}\right) \times 19^2} = 3.59 \frac{N}{mm^2} \quad \dots 1$$

$$3. \text{ Pressure} = \text{Force} / (\text{area of piston caliper}) = 3.59 = F / ((\pi/4) \times [40]^2)$$

$$4. \text{ Force on one piston} = 4511.32N$$

$$5. \text{ Force on dual Pistons} = 4511.32 \times 2 = 9022.65N$$

$$6. \text{ Brake force} = \mu \times F(\text{piston force}) = 0.4 \times 9022.65 = 3609.06N$$

$$7. \text{ Brake torque} = F \times R_{\text{eff}} = 3609.06 \times \frac{90 + 70}{2} = 288724.8 \text{ Nmm}$$

Where R effective is given as (R+r) / 2 for disc brake diameter

- Total force =

$$\frac{\text{BrakeTorque}}{\text{radius of tire}} = \frac{288724.8}{9 \times 25.4 \times 0.95}$$

$$= 1329.48N$$

- Total force = 1329.48 x 4 (pistons) = 5317.84N

Now, adding rolling resistance + aerodynamic drag to the total force for de-acceleration

$$\mu mg + 1/2 \rho A v^2 cd = 0.04 \times 950 \times 9.81 + 0.5 \times 1.2 \times 2.16 \times [6.53]^2 \times 0.4 = 394.88N$$

Adding the total force + total resistance

$$F = 5317.84 + 394.88 = 5712.82N$$

Using equations of motion, we get ,

$$\text{Deacceleration} = \text{Force/mass} = 5712.82/950 = 6.013 \text{ m/s}^2$$

And,

$$V = u + at =$$

$$\text{time} = \frac{v}{a} = \frac{6.53}{6.013} = 1.1s$$

$$\text{Distance } s = \frac{1}{2}at^2 = 0.5 \times 6.53 \times 1.1^2 = 3.61m$$

According to the design of hydraulic disc braking the vehicle comes to stop in 1.1s with a de-acceleration speed of 6.013 m/s² in distance 3.61 meter

Table - 6: Results

Design	Results
De acceleration	6.01m/s ²
Brake time	1.1s
Braking distance	3.6m
Total force	5712.82N
Resistance force	394N

5. STEERING

The steering system is the key interface between the driver and the vehicle. The main requirement is that the steering should be precise, with no play. In addition, the steering system should be smooth, compact and light. It must also provide the driver with a perfect feel for the road surface and help the wheels return to the straight-ahead position. Among two basic steering mechanisms i.e rack and pinion and recirculating ball mechanism, rack and pinon method was chosen considering various parameters such as availability, widely used and cost .

In the rack and pinion system, a pinion gear is attached to the steering shaft, i.e. turning the steering wheel turns the pinion gear which then moves the rack. The rack and pinion gear is enclosed in a metal tube, with each end of the rack protruding from the tube. It does two things:

- It converts the rotational motion of the steering wheel into the linear motion needed to turn the wheels.
- It provides a gear reduction, making it easier to turn the wheels.

A tie rod at each end of the rack connects via the swivel ball joint to the steering arm which finally moves the wheel. The specific advantage of the rack and pinion design is a good feedback and a direct steering feel.

➤ Design parameters

- 1 Wheel base – 3150 mm
- 2 Track width C = 1200 (refer diagram)
- 3 Assumed inside angle as per geometry
- 4 Should be less than 5m as per industry standard of golf carts

PERFECT STEERING CONDITION

$$\cot \phi - \cot \theta = c/b$$

θ = angle turned by inner wheel

ϕ = angle turned by outer wheel

b = wheel base

c = distance between pivot points

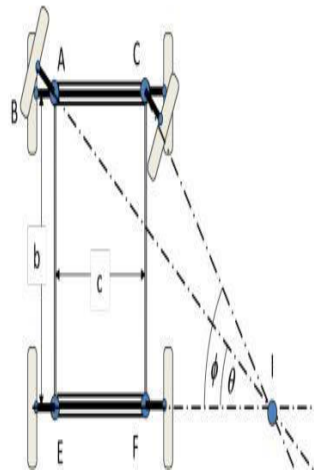


Figure 5 – Psc

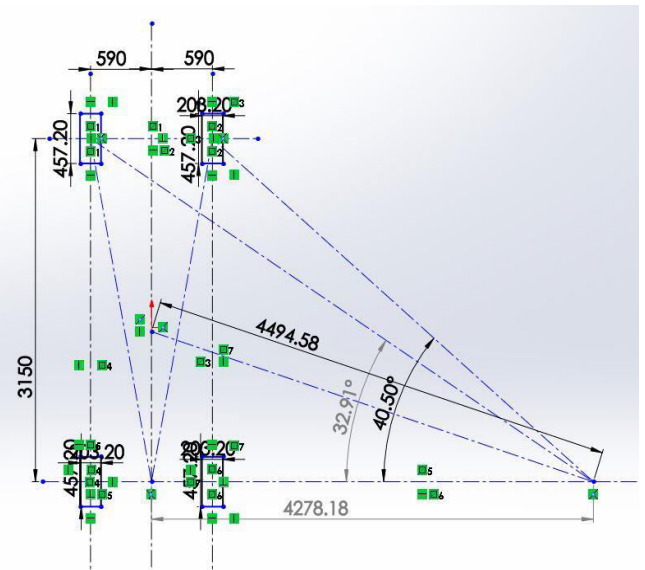


Figure 6 – Solidworks Ackerman Geometry

a) Design calculations

$\theta = 40.5$ degree assumed

According to perfect steering condition of ackerman geometry

$$\cot \theta - \cot 40.5 = \frac{1200}{3150}$$

$\theta = 32.91$ deg outside angle

Ackerman inside angle

$$(\beta) = \tan^{-1} \left(\frac{[Base]}{\left[\frac{base}{\tan \beta} - width \right]} \right)$$

$$\tan^{-1} \left(\frac{[3150]}{\left[\frac{3150}{0.706} - 1200 \right]} \right) = 44.0$$

Now, ackerman % =

$$\frac{(inside\ angle\ \theta)}{ackerman\ inside\ angle\ \beta} \times 100$$

$$= \frac{40.5}{44} \times 100 = 92.04\%$$

This shows our geometry is 92% close to being true ackerman.

Calculating “m” which is the distance between the icr of ackerman and the bottom left tire centre with the help of outside angle

$$\tan \theta = \frac{b}{width + m} = \tan 32.91 = \frac{3150}{1200 + m}$$

$$m = 3667.29\ mm$$

Now calculating R, The turning radius with formula,

$$R^2 = (0.4b)^2 + \left(\frac{w}{2} + m \right)^2 \text{ putting above values, } R = 4449.42\ mm$$

Rack length is assumed to be 35 cm.

1) Rack travel:

$$\frac{\theta}{360} 2 \times \pi \times r$$

Travel = $73.41/360 \times 2 \times \pi \times 6'' = 7.68''$ rack travel

2) Steering ratio = (lock to lock steering angle / $\theta + \emptyset$)

$$\text{Steering ratio} = 360/73.41 = 4.90:1$$

Rack travel = $\pi \times d \times \text{turns of steering wheel}$

$$7.68 = \pi \times d \times 1$$

$$d = 2.44'' = 6.198 \text{ cm}$$

d is the diameter of pinion

6. SUSPENSION DESIGN

Suspensions are very important in any automobile. We know that any system used in vehicle to reduce road shocks and vibration known as suspension system. Springs, shock absorber torsion bar etc. are components of suspension system. so the Suspension Springs are used to separate the Wheel of the vehicle to the body so when the vehicle feel those vibration it transfer to the spring and the spring start oscillating without transmitting this vibration to the vehicle body.

In our electric golf cart we had the option to choose independent or dependent suspension systems but we chose independent ones for various benefits of it. We have used two types of suspension systems for front and rear

The front suspension system is double wishbone and rear one is leaf spring. As the vehicle is long and heavy at rear leaf spring at the rear was considered necessary according to design and also the market research revealed that.

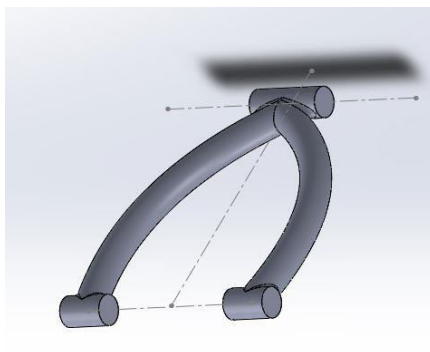


Figure 7 Upper wishbone

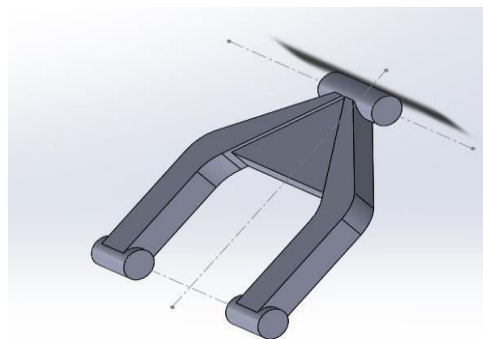


Figure 8 Lower wishbone

a) Design of front Suspension

Double wishbone system was chosen for the front suspension for rough and tidy road where having unequal surfaces, water and many other obstacles. To clear these types of obstacles with ease and without damage to the vehicle systems for long period suspension system should be that much of rigid and effective to sustain at any adverse condition. For this suspension system is designed by considering 1500 kg which is more than the actual weight of vehicle.

double wishbone type of suspension has been selected because of its high load handling capacity and rigid support to the wheel geometry. It is ease to control Camber angle, Castor angle and King Pin inclination angle.

According to the geometry created shown below with real design values the angles and length of all our components like upper wish bone, lower wishbone and strut was determined, the knuckle was also designed entirely new way to support the newly designed geometry. Many modifications were also necessary during the solid works modeling and checking the assembly of files and they have been changed accordingly.

With the help of our geometry the length of upper and lower wishbones was determined to be

Upper wish bone - 230.66 mm and lower wishbone -252.59 mm keeping their lengths dissimilar to obtain better control on camber and other geometries of suspension.

The wishbones are made out of the same material used for the chassis 4130 chrome-moly to provide its strength benefits and easy weld assembly and component match.

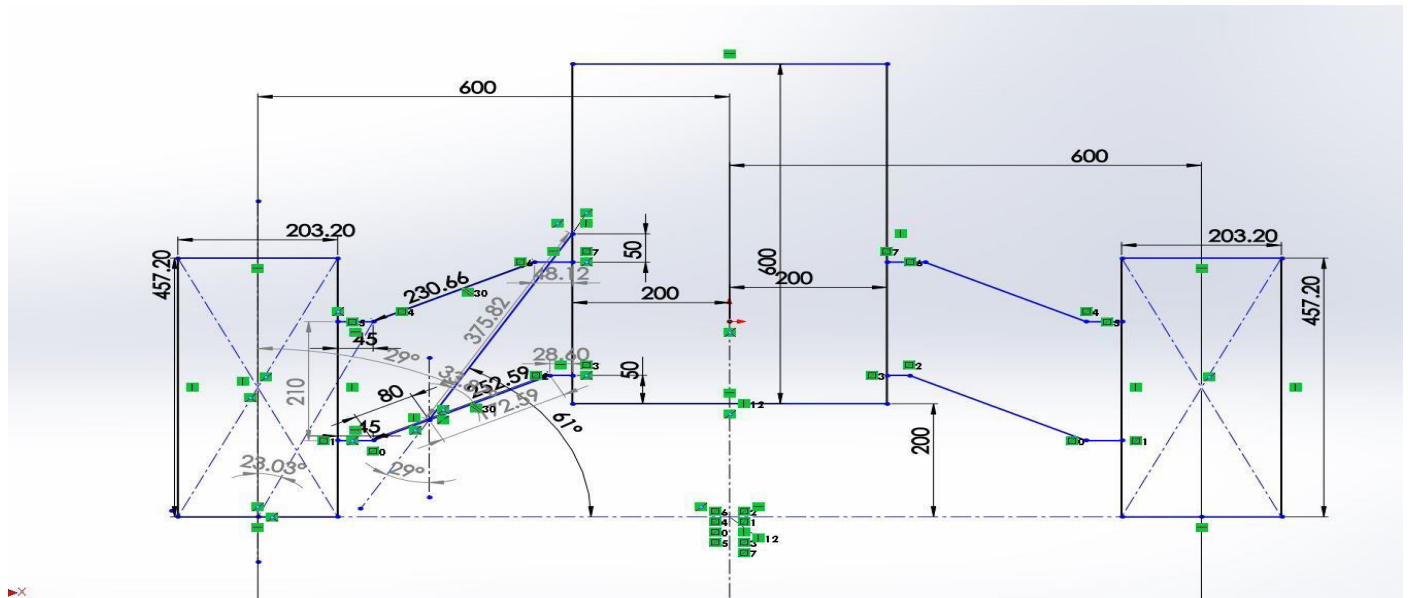


Figure 9 - Suspension geometry

b) Strut Design

Strut assembly consist of spring and damper shock absorber system. After a lot of market research and types of struts to be used in a golf cart a component was selected from a EIBACH SPRINGS CATLOG 2012 code no. 1200 250 0550S as it matched our needs and criteria. our design load conditions and dimensions.

1200.250.0500S	12.00	305	2.50	64	500	88	5.49	139	6.51	165	3,252	14,466	6.06	2.75
1200.250.0525S	12.00	305	2.50	64	525	92	5.65	144	6.35	161	3,334	14,830	6.37	2.89
1200.250.0550S	12.00	305	2.50	64	550	96	5.83	148	6.17	157	3,390	15,079	6.72	3.05
1200.250.0600S	12.00	305	2.50	64	600	105	5.82	148	6.18	157	3,706	16,485	6.83	3.10
1200.250.0650S	12.00	305	2.50	64	650	114	5.81	148	6.19	157	4,025	17,904	6.94	3.15
1200.250.0700S	12.00	305	2.50	64	700	122	5.85	148	6.15	156	4,207	18,168	7.12	3.22

Table - 7 Spring data

- Parameters of the spring are :
 - Rod diameter - 9.5 mm
 - 12 inch length / 305 mm spring
 - 2.5 inch internal diameter
 - Spring rate - 96 N/mm
 - Block height - 148 mm
 - Spring travel 20-70 % ratio - 6.17 inch
 - Load capacity - 15097 N
 - Weight 3.05 kg

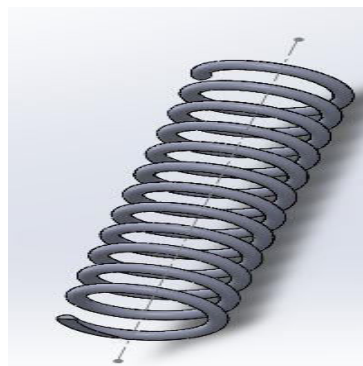


Figure 10 - Spring

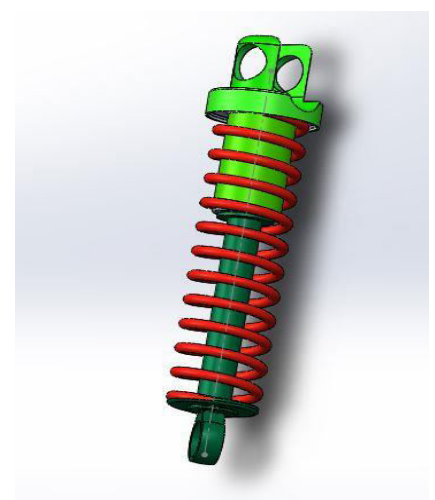


Figure 11 - Strut

c) Designing to find wheel rate and suspension frequency & ride quality

1. Motion ratio front = $d1/d2 = \frac{172.59}{297.59} = 0.5799$ mm
2. Motion ratio rear beam axle = $d3/d4 = \frac{600}{900} = 0.66$ mm
3. Angle correction factor – 29 degrees
4. Wheel rate = [Motion ratio front ^2 x spring rate x acf angle]
 $= 0.5799^2 \times 96 \times 29 = 926.89$ N/mm

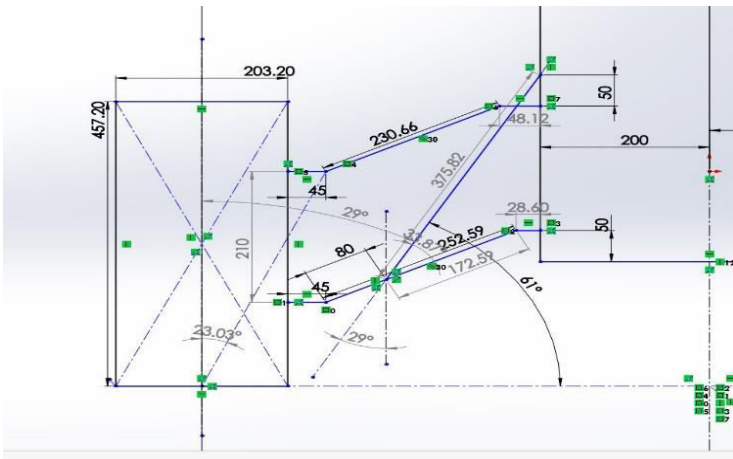


Figure 12 – Dimensions/Angles of Arms and Suspension

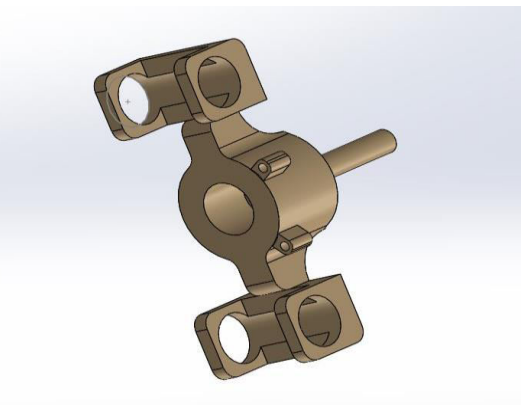


Figure 13 – Front Knuckle

These values of d1, d2, d3, d4 and acf angle was found according to the suspension geometry to calculate wheel-rate and ride quality.

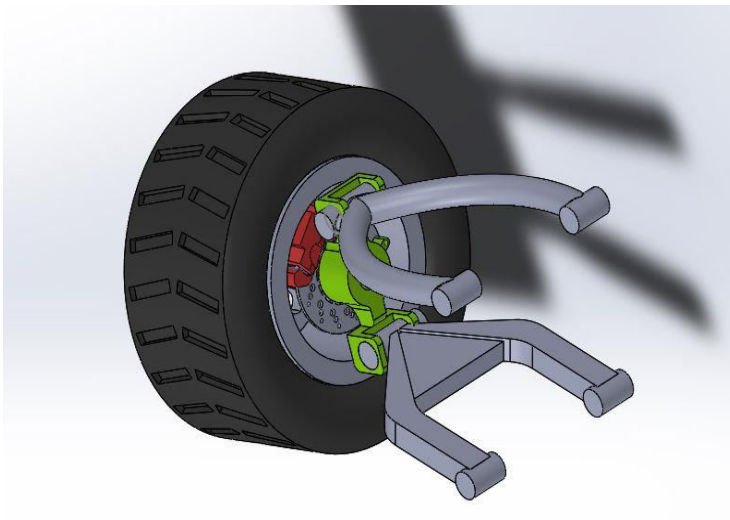


Figure 14 – Wheel Assembly with wishbones

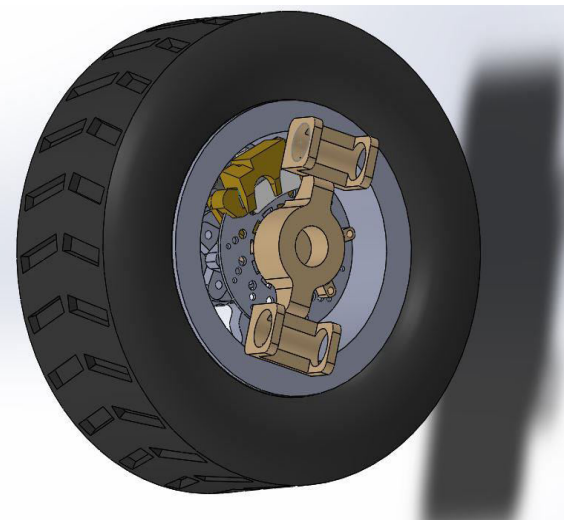


Figure 15 – Wheel Knuckle assembly

d) Rear Suspension Design

Rear suspension was chosen to be leaf spring, Due to the sheer amount of metal layered together, leaf springs offer a large amount of support between the wheels, axles and the car's chassis. They can take huge vertical loads being applied to them due to their tight-knit structure, hence why heavy duty industries still use them. Vertical loading is also distributed throughout the length of the leaf spring rather than acutely through a small spring and damper, which can potentially create a concentrated force too large for the suspension to handle.

The material chosen for rear leaf spring is **chromium steel aisi 5150** , this was chosen for its excellent yield stress and its strength criteria.

Young's modulus - 210 gpa

Yield stress - 1520 MN/M²

Again market research was done to find optimum leaf spring parameters and sizing and certain criteria were selected and designed

Parameters selected

1. No of full length leaf - 1
2. No of graduated leaf - 2
3. U bolt length - 5 inch to fit across the rectangular tube holding the leaf
4. Total length 21 inch x 48 mm x 7 cm LXBXH

➤ *Design parameters*

1. Upsprung weight = 0.15 x vehicle net weight = 0.15 x 550 = 82.5kg
2. Sprung weight = gross weight - uppsprung weight = 550 - 82.5 = 467.5 kg
3. Load on each spring = ¼ x weight = 320 kg full loading
4. Allowable stress = yield stress /factor of safety = 1520 / 2 = 760 Mn/m²
5. Max loading capacity of selected material leaf spring is 400kg more than required hence the design is safe.
6. Suspension frequency obtained is same

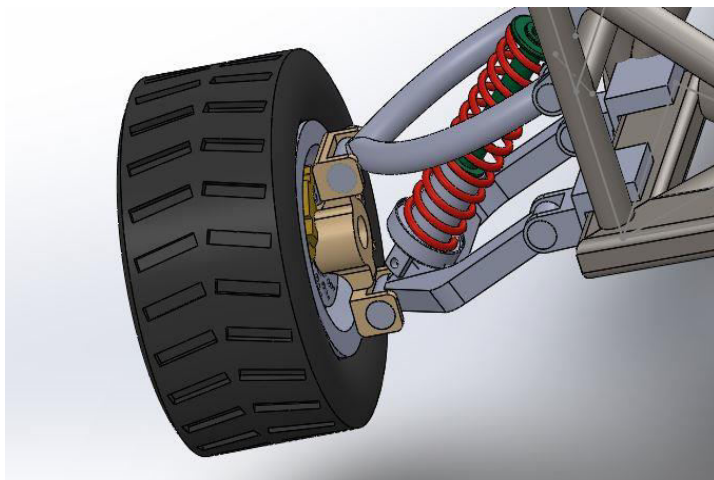


Figure 16 – Front Right wheel mount Assembly

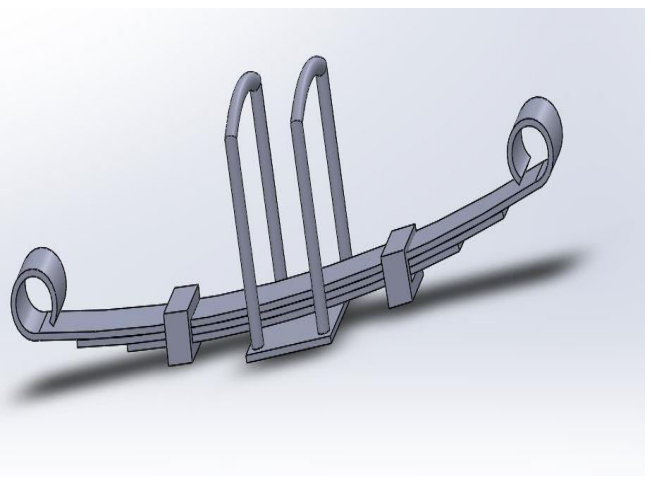


Figure 17 - Leaf Spring model solid works

Table 8 - Suspension System data for both front and rear in a tabular form

Data	Results
Length of coil spring	12inch
Wire diameter	12 mm
Coil spring ID	2.5 inch
No of active turns	12
Travel of coil	6.17 inch
Length of leaf spring	21 inch
Thickness of leaf	5mm
Spring rate	96 N/mm
Wheel rate	926.87 N/mm
Front camber	-2 degree
Castor angle front	+5 degree
Toe in front	-1 degree
Rear camber	0
Rear castor	0
Rear toe in	0

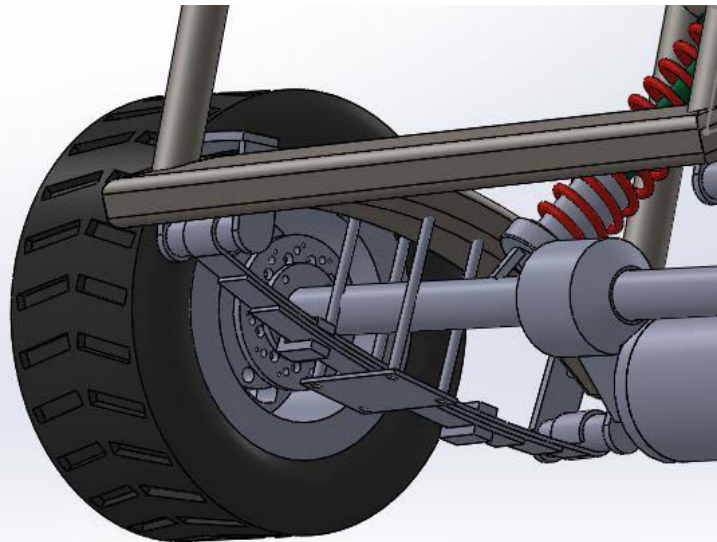


Figure 18 – Rear wheel assembly with leaf spring

7. BATTERY

The battery which uses sponge lead and lead peroxide for the conversion of the chemical energy into electrical power, such type of battery is called a lead acid battery. The lead acid battery is most commonly used in the power stations, and substations etc because it has higher cell voltage and lower cost.

Electric golf cart batteries operate purely on the power of deep cycle batteries. Here are the primary features of a deep cycle golf cart battery to understand.

- Designed with thicker plates, higher density active paste material and thicker separators to withstand the rigors of deep discharging and recharging cycles.
- Twin terminal design for simplified connection for primary power and accessories
- Convenient handle for easy installation

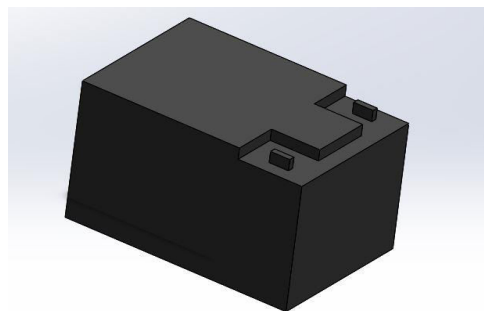


Figure 19 – Battery

Advanced AGM and gel batteries have a spill proof design with casing engineered to withstand heat and vibration elements. A deep cycle Lead acid battery is chosen because of availability and cost considerations as its competitive lithium ion batteries are very costly but deep cycle lead acid batteries syncs correctly with our nominal day to day usage as per usage statistics obtained from market survey.

We are using 6 batteries of 8 Volt sizes with 190 amp hour rating. The batteries are placed in three groups of two below every seat and it was found optimum according to chassis design and it helped keep the over all cg of vehicle low and improved stability. The size of the battery is 259 x 183 x 340 mm weighing 25 kg each.

A. Battery Calculations

- 1 batteries of 8 volt 190 amp hour
- 2 Hence pack voltage size - $6 \times 8 = 48v$

Pack size of the battery - $48 \times 190 = 9.12kwh$

We cannot use the entire pack size of the battery, we should not ever go below 20% so only 80% of pack-size is actually practical otherwise we will kill our batteries extremely fast

In addition because an electric vehicle will discharge the batteries faster than the manufacture tested and rated we get an effect called peukert's effect. Therefore our calculations need to be considered that effect because its considerable effect in lead acid batteries where we only get to use about 60% of the power in the battery. But since these are deep cycle we can give a little more allowance say about 65 - 75%

Hence,

Usable pack = $9.12 \times 0.8 \times 0.7 = 5.10 kwh$, this pack size we are using to calculate is for longest life of the battery. The battery can be drained more for more mileage or in emergency but this is the safest side usable for daily use and long durability.

Now under full loading condition our golf cart weighs 950 kg about 2094 pounds

Considering rolling and aerodynamic drag load on the vehicle the force which we obtained earlier can be added. The force in newton shall be converted into weight

Earlier we obtained 394.88N as the rolling + aerodynamic drag, therefore

$$394.88/9.81 = 40.25kg$$

This will be converted to pounds and added to the vehicle weight to calculate the estimated usage of power as thumb rule. Hence total mass now is 2182.19 pounds.

For design purpose this value needs to be divided by 10 in order to get the wh/mile usage of the vehicle under full loading condition including resistances.

Therefore **$2182.19/10 = 218.21 wh/mile$** .

Now calculating range,

Range:

$$\frac{\text{usable capacity}}{\frac{wh}{mile}} =$$

$$\frac{5102 \text{ watt hr}}{218.18} = 23 \text{ miles or } 37 \text{ km}$$

Is the true range of our designed golf cart under full loading condition which is perfect result because as per the market research an average use of a golf cart is not more than 15-18 miles per-day. A standard charger is provided for to charge the batteries which take about 8 hours to charge completely.

8. SOLAR PANEL

Solar golf carts are golf carts powered by mounting a photovoltaic (PV) or thin film panel on top of the existing roof or using a PV panel as the roof itself. A controller converts the sun's energy to charge the golf cart's 36-volt or 48-volt battery bank. Not only does the solar power take the cart off the electric grid, it also increases the driving distance and extends the life of the batteries.

Solar panels on golf carts have received a lot of attention and are used when needed for long range like maintenance workers, golf cart fleets, luxurious hotels, college campuses etc.

A lot of market research was put into finding the right type of golf cart panels for our vehicle and the type of impact it will have on our vehicle & batteries. Since the roof our cart is long as it is for 6 seating capacity naturally we have more space exposed and we found that we can fit two panels easily available in the market on top of it and its economical as well.

- Components are:

- 2 panels of 250 watt
- Mppt charge controller with only 2 wire installation to the battery
- Panel size 1600 x 990 mm (fits on top of the roof of chassis)

A. Calculations

- 2 panels of 250 watt is 500 watts in total.
 - Considering 6 hours of direct sunlight since morning hours we get,
 - $500 \times 6 = 3000$ watt hr or 3kwh
 - Considering 90 % efficiency of solar panels, we get usable pack of 2.7kwh from them
- Now, the usable pack size in our battery is 5.10 kwh and the solar panel usable pack will be added into that which gives us $5.10 + 2.7 = 7.8$ kwh total usable pack size overall

Now, calculating the range with the new usable pack size we get,

$$\frac{7800}{218.18} = 35.75 \text{ miles}$$

35 miles - 23 miles = 12 miles.

So the 2 solar panels give an extended range of 12 miles / 19kms

Giving a total of 56 km range.

The solar panels help in longing the battery life by constantly charging them and limiting their depletion below the standards mentioned above. Thus addition of solar panel is desired.



Figure 20 – Solidworks Scenic display of Golf cart



Figure 21 - Solidworks Scenic display of Golf cart

III. CONCLUSION

- 1) A lot of market research went into the design of the electric golf cart and desirable results were achieved.
- 2) A new design for knuckle was made.
- 3) Finalizing the design of the chassis and trying to innovate with different pipe sizes and shapes gave a unique looking cart with considerable strength and balance.
- 4) An excellent range of mileage of 37km is obtained for such a heavy vehicle with lead acid batteries which cannot be discharged below 50% for long term of quality usage which was further increased by 12 miles with two solar panel additions
- 5) The performance of overall vehicle including its speed, braking distance and turning radius were achieved under the pre-set assumed values.
- 6) The suspension design was selected carefully to give the best performance in overloading conditions as well and the leaf spring to provide more support in case of heavy luggage/materials kept behind.
- 7) The transaxle ratio of 10.45 gives the most optimum output with the coupled motor and the cart is able to climb fully loaded in grassy/muddy terrain with 20% inclination as per design.
- 8) The batteries being heavy and placed below the seats makes the majority of the weight carried by the cart at low height even after passenger is seated which gives the entire cart a very low roll center which reduces the chance of cart rolling during uneven terrains

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V. BIOGRAPHY



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