Design and Simulation of an Electric Vehicle

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Abstract

In this paper, design and fabrication of an battery powered, single passenger E-GO KART is discussed. An Electric Go-kart is a small four wheeled vehicle without suspension and differential. E-Go kart is designed and meant for racing only (though in some countries it is used for fun and personal transportation). It is a miniature of a racing car. This E-Go kart is designed and fabricated for participation at the E-NKRC {Electric National Kart Racing Championship}. Modelling of chassis is performed in solid works. Kart chassis is different from Car chassis. Kart chassis is made from hollow circular pipes/rods. Strength and Light weight are the basic consideration for choosing the chassis material. AISI1018 is the suitable material to be used for E-GO kart chassis which is medium carbon steel having high tensile strength and offers good balance of toughness and ductility.

INTRODUCTION

There is a growing demand for fossil fuel like diesel and petrol to power the automotive and cater other needs of human. Fossil fuels are being deflected because of their excessive use limited stocks. Further the use of fossil fuels is polluting the environment. Because of this people are fragile to wear mask for filtering the polluted air for respiration. To minimize all these problems and to keep our earth free from pollution, there is an urgent need to explore alternative in place of fossil fuel powered vehicles. Efforts are being out to develop vehicle powered by solar energy, hydrogen, biodiesel and batteries. Battery powered vehicle are not so popular in India because they need frequent charging. In order to overcome above mentioned problems an attempt has been made to design and fabricate environment friendly, battery powered, single passenger E-GO kart. **Forces on vehicle and power calculations**

Total friction (f) = F(rolling) + F(Gradient) + F(Aerodynamics)

1. Rolling Resistance: -

Rolling Resistance = Rolling density * mass * acceleration

Rolling Resistance = 0.01 * 410 * 9.81

Rolling Resistance= 40.221N

2. Gradient Resistance: -

Gradient resistance = mass * Acceleration * sine angle

Gradient resistance = 401 * 9.81 * sine38

Gradient resistance = 1192.024N

3. Aerodynamic Drag: -

Aerodynamic Drag = ½ * Density *Area * (velocity) ^2 *Ad

Aerodynamic Drag = ½ * 1.23 * 2.231 8 * (22.22) ^2 * 1.5

Aerodynamic Drag = 1016.14N

POWER REQUIRED TO OVER COME TOTAL FRICTION: -

Total Power (p) = Power (PR) + Power (GR) + Power (AD)

1. Rolling Resistance: -
   Power (PR) = F(Rolling) * V
   Power (PR) = 40.221 * 80 * 3.6
   Power (PR) = 893.3 watt
   Power (PR) = 0.8933kwatt

2. Gradient Resistance: -
   Power (GR) = F(Gradient) * V
   Power (GR) = 1192.024 * 22.22
   Power (GR) = 26486.77watt **Power (GR) = 26.486Kwatt**

3. Aerodynamic Drag: -
   Power (AD) = F(Aerodynamics) * V
   Power (AD) = 1016.14 * 22.22
   Power (AD) = 22578.6watt
   Power (AD) = 22.57Kwatt

Total friction (f) = F(rolling) + F(Gradient) + F(Aerodynamics)

Total friction (f) = 40.221 + 1192.024 + 1016.14

Total friction (f) = 2248.38N

Total Power (p) = Power (PR) + Power (GR) + Power (AD)

Total Power (p) = 0.8933 + 26.482 +22.57

Total Power (p) = 49.94Kwatt

Steering system

Weight :- Keeping the frame as light as possible is a main priority when power is limited, vehicle weight is a larger factor in vehicle performance. The frame is one of the largest and heavy component of the vehicle. Hence we must employ the correct material in the Frame and steering system to minimize power.

Accomplish those goals, FEA aided the material decision making process. FEA specifically helped to determined whether a member was under high or low stress, we are using **AISI- 1018** for frame and its thickness **1.5mm** ,outer diameter **1.5 inch**. the final weight of the vehicle along with driver is estimated to be **410kg.**

Steering system:- For the simplicity and safety of the main design specification for vehicle steering system , we are drawing steering geometry in solid works and calculate following points. Most suitable we have choosen the **Ackermann steering design with rack and pinion type. Calculations :-**

Outer turning Angle:- 30.06 deg.
Inner turning Angle:-22.29 deg.
Turning Radius :-3000 mm
Steering wheel Diameter : - 3040mm
Tie rod length: -330.22mm (from manufacturer)
Actual Turning Radius:- 3776.96mm

Turning radius=3000mm
wheel base(d)=1025mm
wheel track(L)=1440mm
Outer angle :-Tan(A)=(L/(R-d/2))=0.579

A=30.06

Inner angle:-Tan(B)= (L/(R+d/2))=0.4099

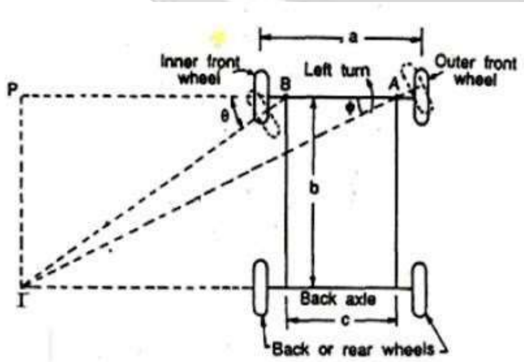
B=22.29

Actual Turning Radius =d/2 + L cosec(A/2+B/2)=3776.96mm

Ackermann condition =cot(Θ)-cot(Φ)= d / L

Length of Tie rod= 330.22mm (13 inches) Height of sleeving column=25 inches

- Steady-state cornering is defined as cornering along a path of constant curvature, with constant speed and sideslip angle.
- Low speed cornering ;- (Turning)
  At Low speed the tires need not to develop lateral forces (No slip)
  Tires roll with no slip angle centre of turn must lie off projection of rear axle and the perpendicular of front wheels passes through same turn centric.



- High speed cornering:-Tires must develop significant lateral force to counteract the lateral acceleration forces.
  Cornering equations :- cornering force at fornt wheel(Ff) + cornering force at rear(Fr) wheel=mass of vehicle*velocity^2/(radius of turn)

- Steering Effort = mass of vehicle * acceleration due to gravity* weight transfer ratio at front wheels=410*9.81*0.45

  = 1809.945N

- Forces on Steering arm= coefficient of friction * normal reaction force

  F=0.18*1809.945

  F=325.79N

- Velocity of vehicle(V)= 16.66mls
- Torque on steering wheel =force on tie rod *0.0889=325.79*0.0889

  T =28.96N-m

- Force on steering wheel=torque/steering wheel diameter

  =28.96/0.304

  = 95.2N

- Steering Effort() =95.2/9.81

  =9.704Kg

- Let Front , rear be denoted by f and r respectively

  Slip angles : - α =F/Cα

  =MV^2/(R Cα)

  =wV^2/(CαRg)

  δ=57.3*L/R +((w/ Cα)front- (w/ Cα)rear))V^2/(gR)

  (From design data book)

  Neutral steer:- The neutral steer point is the point along the car centerline where a sideways force would produce equal slip angles on both wheels. If the neutral steer point is behind the center of mass, the car has understeer. If the neutral steer point is at the center of mass, the car has neutral steer.

  (w/ Cα)front= (w/ Cα)rear)

  Understeer:-understeer occurs when a car steers less than the amount commanded by the driver.

  (w/ Cα)front->(w/ Cα)rear)

  Oversteer:- Oversteer is what occurs when a car turns (steers) by more than the amount commanded by the driver.

  (w/ Cα)front< (w/ Cα)rear)

Ackermann=tan^-1(Wheel base/(Wheel base/tan Ɵ -Trandc front))

Ackermann%= (Ɵinside/Ackermann)*100

Braking System

The main focus while designing the brakes of the vehicle was not only on its efficiency but also on the braking efficiency. Following are the design consideration kept forward while designing & assembly of braking system.

- *Effective braking in all condition.*
- *Simple and reliable brake system*
- *Less driver fatigue*
- *Adequate braking force*

A hydraulic circuit is designed in accordance with the vehicle weight, vehicle length and its top speed. For good handling brake should be consistent, use only a moderate amount of pedal force and should be free of fade in its hardest use. The work a disc brake has to do and the heat it has to dissipate during braking, are directly proportional to the weight of the vehicle and its speed. The hardest working conditions occur with maximum load and maximum speed. The brakes are installed on the rear wheel of the vehicle on both the wheel to increase the braking force so that stopping distance can be reduced.

The innovative part in the braking system is "BRAKE OVER TRAVEL KILL SWITCH" which automatically cuts off the current supply in motor in case the braking system fails and there is an over travel in brake pedal. This causes the vehicle to decelerate even if brakes don't work.
Braking calculations were done at a velocity of 85kmph considering the vehicle weight as 450kg and results are shown in the Table.

PARAMETER	VALUE
Brake Pedal Force	1860N
Pedal Ratio	6:1
Fluid Pressure	
Braking Force	7080.23N-M
Braking Torque	2403.74N-M
Stopping Distance	17.72m
Brake fluid	DOT 4
Max. deceleration	15.73m/s2

The thermal analysis of the disc has been carried out, to ensure the reliable performance of braking system.

Reference:

1- ***Wikipedia***

2- ***Engineering automobile by kripal singh***

3- ***Vehicle dynamics – Vimal ojha***