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Design And Fabrication Of Environment Friendly Kart

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ABSTRACT
In this paper, design and fabrication of an environment friendly, battery powered, single passenger Eco-kart is discussed. Kart was designed following the fundamentals of design and using 3ANSYS software. Various systems of the kart were designed and fabricated e.g. chassis, brakes, steering, and drive train. Further, adequacy and performance of each designed component/system was checked by ANSYS simulation. To insure safety of the passenger a factor of safety equal to 6.2 and 3.2 respectively, was used for intermediate and final drive shaft.
Keywords: ANSYS 12.1 software, Design, drive shaft, multiple gear set.

I. Introduction

There is growing demand for fossil fuel like diesel and petrol to power the automotives and cater other needs of human. Fossil fuels are being depleted because of their excessive use and limited stocks. Further the use of fossil fuels is polluting the environment. In metro cities like Delhi, Beijing, level of pollution from vehicles, during peak hour is dangerous. Because of this people are fragile to wear mask for filtering the polluted air for respiration. Further, there are frequent traffic jams on the road due to this there is wastage of fuel and time. All these factors are responsible for various problems in human such as headache, stress, reduced performance etc.

To minimise all these problems and to keep our earth free from pollution and human health and fitness, there is an urgent need to explore alternative in place of fossil fuel powered vehicles. Efforts are being put 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, small distance travelled in single charging, small range of speed in comparison to conventional automotives short battery life etc. In order to overcome above mentioned problems an attempt has been made to design and fabricate environment friendly, battery powered, single passenger Eco-kart.

II. Design And Fabrication

The main aim of this work is to reduce the usage of organic fuel powered Vehicle. Therefore, kart is operated by an AC or DC motor, running on electricity, produced by chargeable battery. Battery can be charged by electricity or solar energy. The use of solar energy for powering the kart will not only save the fossil fuel but also help us to keep earth green and pollution free. It favours Go Green Concept. To increase the number of kilometre travelled in single charging it is must to reduce the weight of the kart so we made our eco-kart of IS3604 grade steel [2] which is lightest possible material for frame. Four bar link steering system which is simple and light in weight [3] is used for steering the kart in minimum possible radius. To stop the kart in minimum possible distance hydraulic disc brake is used on rear shaft. For transferring the power from driver motor to rear wheels power train with two drive shaft and multi teeth gear is used. By this, we have increased starting torque of vehicle and maximum speed.

The components/ system are design and simulated using ANSYS software. Optimally design component were either purchased from the market or fabricated in the workshop. Chassis was fabricated using hollow bar of 24 mm outer diameter and 2 mm wall thickness. Chassis was fabricated by arc welding. Figure 1 shows designed kart.

Fig.1: Different view of designed kart

III. Vehicle Specifications

Overall Dimension:-

Ø Wheel Base = 1350mm
Ø Wheel Track = Front-1130mm
Rear-1080mm

Target Weight:-

Ø Kerb weight of vechile = 104 Kg
Ø Weight distribution = 40:60

Power Train:-

Ø 800W BLDC Motor
Ø Chain Drive

Brakes:-

Ø Type- Rear - Disc Brakes
Ø Tires SIZE:- Front –76.2/R10
Rear – 76.2/R10

Electrical:-

Ø 800 W BLDC Motor
Ø Gel Lead Acid Battery
Ø Battery Limit:- 12x4 = 48V
Ø Current Outsource = 24 AH
Ø Charger Voltage:- 220V 50HZ

Steering:-

Ø TYPE :- Two Wheel Steering
Ø GEOMETRY :- Ackermann
(Centre Point Steering)

IV. Target Performance:

Ø Top Speed:-43 Kmph
Ø Max. acceleration- 1.10m/s²
Ø Stopping distance- 8.91m
Ø Turning radius - 2.46m
Ø Ground clearance- 77.8mm
Ø Height of CG- 228.6 mm

V. Design And Simulation

5.1 Front Crash Analysis

To insure the safety of passenger and vehicle crash test are performed using simulated models. Fig. 2 shows the results of different crash test.

A. Front Crash Analysis

Fig.5.1 Front Crash Analysis

- Force applied= 6,500 N
- Factor of safety =2.15
- Deformation= 1.1mm

B. Rear Crash Analysis

Fig. 5.2 Rear Crash Analysis

- Force Applied= 6,500N
- Factor of safety= 2.20
- Deformation=1.7mm

C. Side Crash Analysis

Fig.5.3 Side Crash Analysis

- Force Applied= 6,000 N
- Factor of safety=2.26
- Deformation=6.0mm

D. Dead Weight Analysis

Fig. 5.4 Dead Weight Analysis

- Force applied= 1,250 N
- Factor of safety= 4.80
- Deformation=1.4mm

5.2 Transmission System

Transmission system is the way to transmit the power and torque of motor to the wheel. We are using the chain drive mechanism. The two chains are using to transmit the power and two drive shaft. One is intermediate shaft and final drive shaft. This is done to increase the gear ratio. [5]

A. Motor specification

- Max. power = 800 W
- Max. torque = 12.8 Nm at 2400 rpm
- Max. rpm = 3600

B. Chain drive calculation-

- First chain # 35 roller chain having pitch of 9.5mm and second one is #45 roller chain having a pitch of 13mm
- Teeth on smaller sprocket = 12
- Teeth on bigger sprocket = 66
- Gear ratio: Max. =16.5:1
  Min=5.5:1
- Length of chain #35 = 108.15mm
- Length of chain #45 =1095.24mm

C. *Calculation of vehicle dynamics*-[6]

Assumptions:

- Gross vehicle weight = 170 kg
- Coefficient of rolling friction = 0.015
- Coefficient of static friction = 0.6
- Gravity $g=9.81m/s^2$

1) Tractive force (at static condition):

- FT=μmg = 0.6*170*9.81
  = 1000.62 N

2) Starting torque:

- T = FT*R, T = 1000.62*0.1778
- = 177.91=178 N-m

Where R= radius of wheel
Ft = Total tractive force (at static condition)

Since our source torque is less due to which we are using two chains so our gear ratio will increase. Therefore we can increase our output torque. It is sufficient to start the vehicle.

3) Max possible acceleration of vehicle:

- A_{max} = (Tractive effort – Tractive force) /mass of vehicle
  A_{max} =1.10 m/s^2

4) Air drag

- FD=1/2*□ *A*Cw*V^2

Where, □ = Air density, A=frontal area
C_w = Coefficient of air drag
V = Max velocity of vehicle

FD = 0.5*1.19*0.39*0.35*43.2*43.2*5*5)/18*18
FD = 11.6 N

5) Max possible speed of vehicle

- S_{max} =(ωmax * Coeff)/Gts [5]
- S_{max} =Max speed of vehicle
- Coeff =Distance travelled per axel revolution
- ω_{max} =revolution per hour
- Gts = minimum gear ratio
- ω_{max} =144000
- Coeff =1.116
- Gt s=5.5
- S_{max} =43.2 kmph

D. Drive shaft analysis

1) Intermediate shaft:

- Ultimate tensile strength = 450 MPa
- Yield tensile strength = 310 MPa
- Modulus of elasticity = 200 GPa
- Max. shear stress = 210 MPa
- According to torsion equation T/J=τ/R [2]

Where, T= twisting force on shaft
J= second moment of inertia
τ= max shear strees on shaft
r= radius of shaft
d=diameter of shaft

- T=70.4 N-m
- R=22mm
- J=16T/πd^3
- τ_{max} = 33.68 MPa
- τ = 210 MPa (Used shaft material)
- So factor of safety = τ/τmax = 6.2

2) For final drive shaft:

- T=211 N-m(used shaft material)
- d=25.4mm
- τ_{max} = 65.61 MPa
- τ = 210 MPa (Drive shaft material)
- So Factor of safety= 210/65.61= 3.2

5.3 Steering System

The Steering System allows the driver to control the direction of vehicle travel. This is made possible by the linkages that connect the steering wheels to the steerable wheels and tires. We use four bar steering system in our kart since it is very simple and it does not include any gears; it consists of two tie rods connected to the spindle arm of a spindle that rotates about a vertical shaft, which is also connected to the steering column. Welded on the steering column are two pieces of metals that the tie rod ends are connected to, so that when the steering turns, it moves the tie rods left or right hence moving the wheels.

- Wheel base, l = 135cm
- Front wheel track, w = 113cm
- Steering ratio = 6.5 : 1
- Number of turns of steering wheel lock to lock = 1.28
- Inner lock angle (θ) = (total steering wheel rotation X 360^0) /steering ratio)
  θ = (0.64/6.5) X 360^0
  θ = 35.46^0
- Outer Lock angle (Ø)-

By using correct steering equation
Cot Ø – cot □ = w/l= Wheel track/ wheelbase)
Cot Ø – Cot 35.460 = 113/135
Cot Ø = 113/135 + cot Ø
Ø = 24.06^0

1) Ackerman angle calculation:

Tanα = (sin Ø – sin θ)/(cos Ø+cos θ -2)
Tanα(sin24.06^0- sin35.46^0)/ (cos24.06^0 + cos35.46^0-2)
α = 32.34^0

2) *Ackerman percentage calculation*:

$\square_{inside}$ = tan-1{135/ (135/tan24.06)-113}-24.06°
$\square_{inside}$ = 11.42°

3) *Ackerman percentage:*

- % Ackerman = (inside-outside) *100/ inside100 % Ackerman
  Inside = inside steer
  Outside = outside steer
- %Ackerman = (35.46-24.06)*100/11.42°
- %Ackerman = 99.8%

4) Turning radius calculation:

- $R_{(min)} = l/\tan\theta = 135/\tan 35.46°$
- $=189.54$cm
- $R_{(max)}^2 = \{R(min)+w\}^2 + L^2$
- $= [189.54+113]^2 + (135)^2$
- $R_{(max)} = 331.29$cm

5) Overturning couple Generated:

- $T_c = (mv^2/r) \times G$
- $T_c = 170\times (3.80)^2/2.46\times 0.228$
- $T_c = 227.51$N-m

5.4 Braking System

- Gross weight of vehicle (w) = 170kg × 10 = 1700N
- Weight of front axle without the application of Brake static R-WR = 680N
- Weight of rear axle without the application of brake static R-WR = 1020N
- Height of C.G, h = 22.86 or 23cm
- Wheel base, l = 135cm
- Coefficient of Friction b/w road and tyre, μ =0.6
- Calliper piston diameter =25.91mm
- Master cylinder diameter = 19.05mm (Maruti 800)
  - Disc diameter = 19cm (7.5inch)
  - Tire Radius = 7inch =17.7cm
  - Deceleration Rate, a = 0.8g m/s^2

1) Dynamic weight transfer

- For front wheel

- WF brake = WF + W × a/g × h/l
  = 680 + 1700 ×0.8g/g × 23/135
  = 680+231.70 =911.70 app.
- For rear wheels WR brake = W - WF brake
  WR brake = 1700 – 911.70
  = 788.30 N approx.

- Retarding brake force on rear wheels
  Friction Force = μ × WR brake
  Friction Force = 0.6 × 788.30 =472.98
- Brake torque required
  TR = brake force × tyre radius
  TR = FR brake × tyre radius
  = 472.98 × 0.1778
  = 84.09 NM
- Brake line pressure
  P = Force of pedal ratio/Area of master cylinder
  $= (P.R \times mass \times g)/ \pi/4 \times (11.94)^2$
  $= (6 \times 10 \times 10)/ \pi/4 \times (19.05)^2$
  = 2.10 MPa

1) Clamping force

C.F = brake line pressure × (area of calliper × 2) × 2(floating calliper.)
= 2.10× (π/4 × 24 × 24 × 2×2)
C.F = 4426.736 Nm

2) Available torque

TA = C.F × disc radius
= 4426.736 × 0.095
= 420.540 Nm

3) Stopping distance

S = V2/2a
V = 43 × 1000/3600
= 11.944N m/s
S = (11.944)2 / 2 ×10 × 0.8
= 8.91 m

4) Braking efficiency

η_b = (v2/2gs) ×100
= {(11.944)2 / 2× 10 ×8.91} × 100
η_b = 0.8006 × 100
= 80.06%

- Force of rolling friction = μmg
  =0.015×170×9.8
  =24.99N =25N

5.5 Electrical System

- Air drag $=1/2 CA\rho V^2$
  $= 1/2*0.35*0.39*1.2*(11.9)^2$
  = 11.68N =12N

ET = 25+12
=37N

- Power required to maintain the constant speed of 41 Kmph

P = FT*V
P = 37*12
= 444 watt (say 500 watt)
Let the vehicle travel a distance of 50 km in one charging
Hence time for discharge= (50Km)/ (43Km/h)
= 1.162 hr
= 1 hr 9 min.
Current consumed to generate a constant power of 500 watt
IDC = 500/48
= 10.41 amps

Let the DOD of battery be 80%
DOD factor= 1.25
Let the temp. of battery performance = 10%
Temp. Factor = 1.1
Design factor = 1.1

- Final rating = IDC*K factor*discharge factor*T-factor*Design margin
  = 10.416*1.25*1.5*1.2*1.1
  = 23.63 amph
- Required ampere hours = 24 Ah

VI. Conclusion

Finally, we succeed to design and fabricate battery powered efficient, single passenger Eco- kart. All the designed components/ systems are safe and performing their intended functions satisfactorily. The Kart travelled a distance of 110 kilometres at a speed of 40 Km/ hour in single charging of four hours.

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