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DESIGN OF ELECTRIC BIKE

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ABSTRACT

The main aim of the project is design an eco-friendly electric bike. We worked in this project is design and analysis the stress and bending moment. During the revolution for the eco-friendly technologies bikes were the most depended modes of transportation, along with this the consideration of the increase in fuel price and the environmental factors we must admit it is far more better to use a bike over a motor vehicle for short distance travelling. Imagine how useful would the bicycle be if even the small effort applied by man for climbing slopes and riding on rough terrain is reduced in it. We thought the same way to develop the basics of our project "The E-Bike". The unit developed by us is a combination of the standard geared bicycle with an electric power motor. The idea of mounting the motor assembly onto a geared bike was to reduce the effort to-be applied for extra little weight that the rider will have to take along with the bike. Detailed study about the gross weight has been mentioned inside the report. The unit has been designed in such a way that people of any age group can depend on it. Our idea of implementation of the project was mainly biased towards providing inter college transportation system and to reduce the use of automobiles inside the campus as a tribute to the "GREEN ENERGY".

INTRODUCTION

E-BIKE is a big craze to the Americans and Europeans. It is initially created in United States in 1950s and used as a way to pass spare time. Gradually it became a big hobby and other countries followed it. In India electric bike is getting ready to make waves. That means the engine in the electric bike is changed to motor and the batteries are used to supply current to the motor.

1. LITERATURE REVIEW

Su-Hau et al (2004) focused on the highly efficient energy usage of the battery energy and proposed an integrated management system for electric motor. This integrated management system includes the power-saving controller, energy management subsystem and some hardware protection strategies. The energy management system acts as a supervisor to manage all the events about the battery energy, including the residual capacity estimation and regenerative braking operation.

David and Sheng-Chung et al (2004) proposed new parallel-type hybrid-electric-power system comprises an engine's energy distribution and a torque-integrated mechanism (specifically including an engine, a motor/alternator, a CVT device, and PCM as well as a 3-helical gear set). To let the engine achieve maximum thermo-efficiency with minimum emissions, the servomotors adjust the diameter size of the pulley to control the engine output for the final power-output axle and the alternator. The system is applied with a stable engine-load to maximize operating performance. The vehicle is driven by the motor alone in the light-duty mode. Meanwhile, in the medium-duty mode, power comes from the engine, with extra energy being used for battery charging. Finally, in the heavy duty mode, both the engine and motor together power the vehicle. The engine output is fixed, but the motor output power can be controlled.

Wenguang et al (2005) presented an approach to control powertrain of series hybrid electric vehicles. A formulation of the system equations and controller design procedure were proposed by them. They also proposed a new switching algorithm for the power converter for motor torque and motor flux control. The sliding mode method is applied to excitation winding control in synchronous generator to achieve the desired current distribution in powertrain.

2. OBJECTIVE OF THE PROJECT

The design process of this single-person vehicle is iterative and based on several engineering and reverse engineering processes. Following are the major points which were considered for designing the vehicle

1. Endurance

2. Safety and Ergonomics

3. Market availability

4. Cost of the components

5. Standardization and Serviceability

6. Maneuverability

7. Safe engineering practices.

CHASSIS

In order to reduce the weight and cost. Most of the problem is faced during turning due to lack of differential. Hence it allows ease of turning. In bike no differential is used which means power is transmitted to the rear axle through chain drive, and rear wheels rotate with same speed and equal torque is transmitted to both of them. So while taking a turn the outer wheel of the bike must be able to loose traction and skid over the road surface. The frame is the main part of the chassis on which remaining parts of chassis are mounted. The frame should be extremely rigid and strong so that it can withstand shocks, twists, stresses and vibrations to which it is subjected while vehicle is moving on road. It is also called underbody.

TYPES OF SECTIONS USED IN FRAMES

Three types of steel sections are most commonly used for making frames:

(a) Channel section,

(b) Tubular section, and

(c) Box section

Fig 3.1

1. Front wheel assembly
2. Rear wheel assembly
3. Brake system
4. Driver seat
5. Motor
6. Batteries
7. Transmission systems

FRONT WHEEL ASSEMBLY

The front wheel assembly has a Front wheel hub and has a kingpin through which the front wheel is assembled in the vehicle. The front wheel has the bearing setup in their rims, so it just rolls with the acceleration of the rear wheel.

Fig no.2 Front wheel assembly

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REAR WHEEL ASSEMBLY

The rear wheel is assembled on the rear axle shaft directly. There is no bearing setup in the rear wheels. So it runs with the power of the motor which is transmitted to rear axle by chain drive.

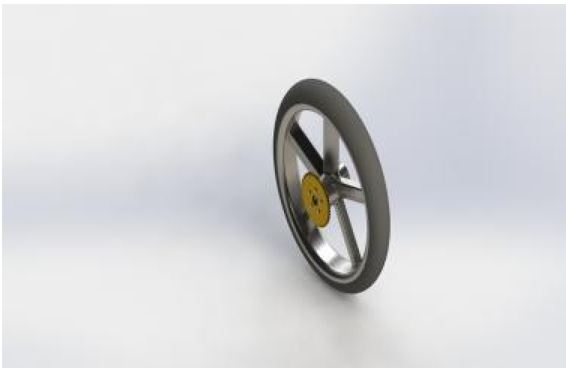


Fig no.3 Rear wheel assembly

TYRES

Tyres are mounted on the rims of wheels. They enclose a tube between rim and itself. Air is filled at a designated pressure inside the tube. The tyre remains inflated due to air pressure inside tube. The tyre carries the vehicle load and provides cushioning effect. It absorbs some of the vibrations generated due to vehicle's movement on uneven surfaces. It also resists the vehicle's tendency to over steer or turn which cornering. Tyre must generate minimum noise when vehicle takes turn on the road. It should provide good grip with the road surface under all conditions.

BRAKE SYSTEM

An excellent braking system is the most important safety feature of any land vehicle. We selecting the disc brake system. The main requirement of the vehicle's braking system is that it must be capable of locking all four wheels on a track.Ease of manufacturability, performance and simplicity are a few important criteria considered for the selection of the braking system.

MOTOR

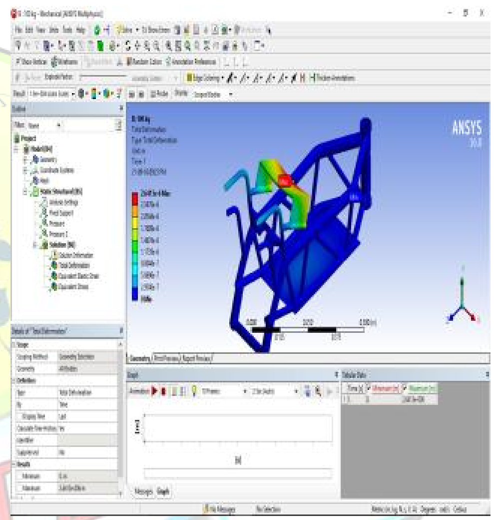
Motor is the heart of our vehicle. It is the drive unit. Motor is fixed rigidly with the help of motor bed with the frame. Smaller sprocket is connected with the motor shaft and power to the vehicle is given by the motor using chain drive. The brushed DC electric motor generates torque directly from DC power supplied to the motor by using internal commutation, stationary magnets (permanent or electromagnets), and rotating electrical magnets.

4. ANALYSIS

4.1 FRAME ANALYSIS

The material to be selected for frame should be feasible for manufacturing and strong enough to support all the components attached in frame. The Frame material should be less weight and highly efficient. The frame material should have less maintenance and corrosion resistance. The frame material should bear the impact and load.

ITERATION.NO 1: ALLOY STEEL 4130 DIA 21.3MM THICKNESS 1.65MM AND 100 KG Displacement



ITERATION.NO 2: ALLOY STEEL 4130DIA 26.7MM THICKNESS 2.11MM&150Kg

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Displacement

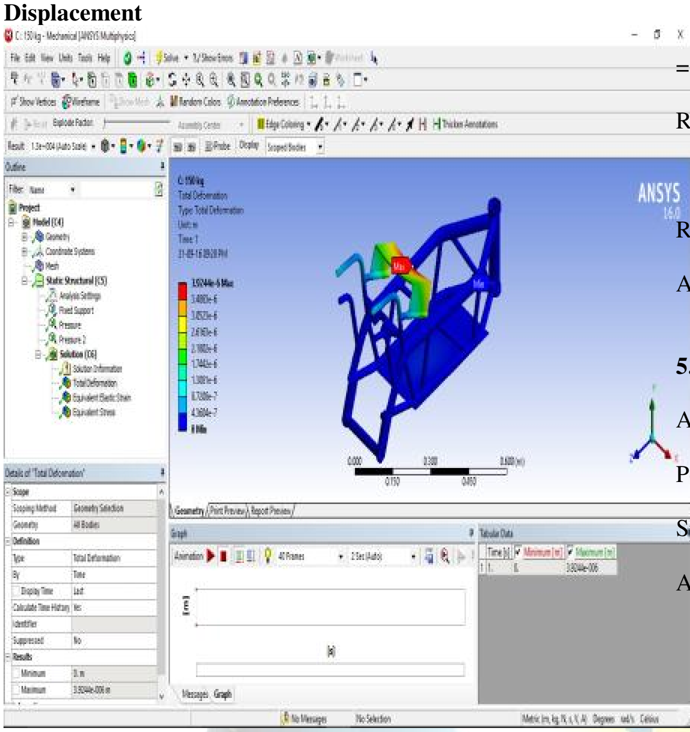

Fig 4.2

5. DESIGN CALCULATIONS

5.1 LOAD DISTRIBUTION:

Frame weight = 10kg

Seat and driver = 80kg

Motor = 10kg

Chain = 5kg

Batteries = 30kg

CALCULATIONS

1. REACTION FORCE AT WHEELS DUE TO FRAME, SEAT AND DRIVER

Total Load F= (30×9.81) N + (100×9.81) N =1275.3N

Distance from load to center of

Rear wheel L_R= 520mm

Front wheels L_F= 560mm

$F \times L_F - R_R \times (L_R+L_F) = 0$

$R_R = FL_F / (L_R+L_F) =$ 1275.3×560 / (1080)

= 661.266N (Reaction at rear wheels)

Reaction force front wheels $R_F = F-R_R$

= 418.7N

Reaction force at front wheel = 418.7N

Approx. weight of the vehicle=1177.2/9.8
=120 kg

5.2 CHAIN SELECTION:

Available data:

Power P=600W

Speed of driver N_1=480rpm

Approx. center distance a_0=190mm

1. Transmission ratio: i=2
2. No.of teeth on driver sprocket:Z_1=16
3. No.of teeth on driven sprocket:

$Z_2=iz_1=2\times16=32$

4. Standard pitch(p):

a = (30 to 50)p

Max. pitch, P_{max}=a/30=190/30=6.3

Min. pitch P_{min}=a/50=190/50=3.8

But the minimum standard pitch is 9.525mm. (PSGDB 7.74)

5. **06B-1/R957** is selected.
6. Total load on driving side(P_T):

   1. P_t=1020N/v N=0.25kw

$V=z_1pN_1/60\times1000$ =(12×9.525×3000)/(60×1000) =5.175m/s

P_t=(1020×0.25)/5.715 =44.62N

   2. $P_c=mv^2$ m=0.41kg/m

      $P_c=0.41\times5.715^2$ =13.4N

   3. P_s=k.w.a

k=1 for vertical drive

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$w=mg=0.41\times9.81=4.02N$

$a=0.15m$

$P_c=1\times4.02\times0.15=0.603N$

Total load, $P_T=P_t+P_c+P_s=44.62+13.4+0.603=58.623N$

7. Service factor (k_s): $K_s=k1k2k3k4k5k6$

$k_1=1$; $k_2=1.25$; $k_3=1$; $k_4=1.25$; $k_5=1.5$; $k_6=1$

$k_s=1\times1.25\times1\times1.25\times1.5\times1 =2.34$

8. Design load: $=P_T\times k_s=58.623\times2.34 =137.17N$
9. Factor of safety (FS_w): F_{Sw}= Breaking load/Design load 9100/137.17=66.34
10. Working Factor of safety is more than recommended (18 from PSGDB 7.77), So the **design is safe.**
11. Check for bearing stress: $\Sigma_{roller}=(P_t\times k_s)/A =(44.62\times2.34)22=4.75N/mm^2$

Induced stress is less than the allowable bearing stress (13.7N/mm² from PSGDB 7.77), So the **design is safe.**

12. Length of chain (L):

No.of links $l_p=2a_p+[(z_1+z_2)/2]+[(z_1-z_2)/2\pi]^2/a_p$

$a_p=a_0/p=190/9.525=19.95$

$l_p=(2\times19.95)+[(12+48)/2]+([36/2\pi]^2/19.95)$

$=39.9+30+1.646 =71.5$ rounding up to 72mm.

$L=l_p\times p=72\times9.525=685.8mm$

13. Center distance (a):

$a=\{[e+\sqrt{(e^2-8M)}]/4\}\times p$

$e=l_p-(z_1+z_2)/2=72-(12+48)/2=42$

$M=[(z_1-z_2)/2\pi]^2=[(48-12)/2\pi]^2=32.8$

$a=\{[42+\sqrt{(42^2-8\times32.8)}]/4\}\times9.525 =192.2mm$

Decrease in center distance for an initial sag=0.01a =0.01×192.2=1.92mm

Exact Center distance =192.2-1.92=190.2mm

14. Smaller sprocket diameter:

$D_1=p/[\sin(180/z_1)]=9.525/[\sin(180/12)]$

=36.8mmOutside diameter $d_{01}=d_1+0.8d_r$

$D_r=6.35$ from PSGDB7.74

$D_{01}=36.8+0.8\times6.35=41.8mm$

Larger sprocket diameter:

=145.63mm

Outside diameter $d_{02}=d_2+0.8d_r=145.63+0.8\times6.35$ =150.71mm

5.3 SPROCKET DIMENSIONS:

DRIVING SPROCKET DIMENSIONS:

Diameter d1 **=** 36.8mm Roller seating radius (ri)

(ri)max = (0.505dr+0.0.069(dr)⅓

= (0.505×6.35+0.069 ×(6.35)⅓) = 3.33mm

(ri)min = 0.505dr = 0.505×6.35= 3.21mm

(ri) = (3.33+3.21)/2= 3.27mm

Roof diameter (Df)

Df = D-2r= 36.8-2×3.27= 30.26mm

Toooth flank radius (re)

(re)max = 0.008×dr (Z^2+180) = 0.008×6.35(12^2+180) = 16.45mm

re)min = 0.12 dr(z+2) = 0.12×6.35(12+2) = 10.67mm

re = (16.45+10.67)/2 = 13.56mm

Tooth side radius (rx) (rx) min = p= 9.525mm

Tooth width (bf)m bf = 0.93 b1 $p \le 12.7mm$

width b/w inner platesbf1= 0.93×5.90= 5.487mm

Tooth side relief (ba) ba = 0.1p to 0.15p

= (0.1×9.25) to (0.15×9.525) = 0.9525to 1.4287

ba = (0.9525+1.4287)/2= 1.19mm

DRIVEN SPROCKET DIMENSIONS:

Diameter d2 = 145.63mm

Roller seating radius (r1) (ri) = 3.27mm

Root diameter (Dt) (Dt) = D-2r= 145.63-2×3.27= 139.09mm

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Tooth flank radius (re)

(re) max = 0.008 dr (z^2+180) = 0.008×6.35(48^2+180) = 126.18mm

(re)min = 0.12 dr (z+2) = 0.12×6.35 (48+2) = 38.1mm

Re = (126.18+38.1)/2= 82.14mm

Tooth side radius rx = p = 9.525mmTooth width bt = 5.487mm

Tooth side relief ba = 1.91mm

5.4 SPEED CALCULATIONS:

No. of teeth in smaller sprocket = 16

No. of teeth in bigger sprocket = 32

Speed ratio = 3

Max speed of smaller sprocket = 480rpm

Max speed of bigger sprocket = 480 / 3 = 160rpm

Wheel circumference = πD = π × 0.254 = 0.798m

Distance travelled per rotation = 0.798m

Distance travelled per minute = 0.798×480
= 283.04m/min

Max speed of the vehicle = 28.3km/hr

5.5 BATTERY CALCULATION:

MOTOR SPECIFICATION

Volt = 36 V

Amps =21.3 A

Watts = 600 W

Type =Permanent magnet 36V Dc motor

BATTERY SPECIFICATION

Volt = 12 V

Amps = 21.3 A

Amp per hour = 24 Ah

BATTERY CHARGING CALCULATION

Charger 12V , 6 A

Charging time =2×B_{Ah}/ C = 2×21.3/6= 21mins.

Battery will be full charged in 21 mins.

BATTERY DISCHARGING CALCULATION

Motor consumption at initial pick up = 13 A × 60

= 780 Ah

Battery delivery = 24 Ah

Motor running = 24/780=0.03hrs= 3.07mins

Motor consumption at running = 2 A × 60

= 120 Ah

Motor running = 24×4/120=2 hrs =120 mins.

Motor runs for 120 mins with full charge.

5.6.DISTANCE TRAVEL BY THE VEHICLE AT FULL CHARGE

Max speed of motor = 480rpm

Max speed of rear axle shaft = 480 / 3 = 160rpm

Wheel circumference = πD = π × 0.254= 0.798m

Distance travelled per rotation = 0.798m

Distance travelled per minute = 0.798×1000= 79.88m/min

Max speed of the vehicle = 35.91km/hr

Distance travel by our vehicle at full charge = 598.5m/min × 21 min= 12.568 Km

Vehicle runs 12.568 km at full charge

5.7 DRIVE WHEEL TORQUE CALCULATIONS

Gross vehicle weight (GVW) =100kg

Weight on each drive wheel (WW) =63.2kg

Radius of wheel/tire (Rw) =0.254m

Desired top speed (Vmax) =35km/hr =9.97m/s

Desired acceleration time (ta) =10sec

Maximum incline angle (α) =5^0

Working surface =concrete (good)

TTE [N] = RR [N] + GR [N] + FA [N]

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Where:

TTE = total tractive effort [N]

RR = force necessary to overcome rolling resistance [N]

GR = force required to climb a grade [N]

FA = force required to accelerate to final velocity [N]

The components of this equation will be determined

in the following steps.

Step 1. RR = GVW xCrr

where: Crr = surface friction (0.01 for good congreat)

RR =105 ×0.01=1 N

Step 2. GR = GVW x sin(α) =100 × sin (5) =8.71 N

Step 3. FA = GVW x Vmax / (g x ta) =(100×9.87) / (9.18×10)

=10.06 N

Step 4. TTE= RR + GR + FA=1.00 + 10.06 +8.71

=19.77 N

Step 5. Tw = TTE x Rw x RF

Where: RF = "resistance" factor(1.1)

T_w = 19.77 ×0.254 = 5.021 N

Step 6. MTT= Ww x μ x Rw

Where: μ = friction coefficient between the wheel and the

ground (~0.4 for plastic on concrete)

MTT= 63.2 × 0.4 × 0.254= 6.42 N-m

T_w< MTT × 2

The total wheel torque calculated in Step Five is less

than the sum of the Maximum Tractive Torques

for all drive wheels, So **slipping will not occur.**

MIN. BRAKING TIME AND DISTANCE:

Mass of the vehicle m=120 kg

Speed of vehicle v= 9.97 m/s

Rotational speed of rotor n_r=480rpm = 12.5 rev/s

Kinetic energy of vehicle KE = ½ mv^2 =1/2 ×100 ×9.97^2

=9.62KJ

P = F_R (2πr_r) n_r=5643.67 (2π × 0.075) 12.5= 33.2 KW

Braking time t = KE / P = 9.62 / 33.2=0.29 sec

Decelerationa= v / t =9.97 / 0.29=34.4 m/s^2

Stopping distance s= ½ at^2 = ½ × 39.88 × 0.29^2

=1.44 m (values under optimum conditions)

6. RESULTS AND ADVANTAGES

RESULT OF ANALYSIS

So Alloy steel 4130AISI is feasible for manufacturing and maintenance free and weight less and we can proceed further for frequency analysis and upcoming design calculations with pipe of Diameter= 26 mm and thickness= 1.5 mm & 2.5 mm.

ADVANTAGES OF E-BIKE

- No emission and it is environment friendly.
- Designed E-Bike is highly sturdy and with stand impacts.
- Maintenance free and highly endurance.
- Easy serviceability and safety ergonomics.

7. CONCLUSION

The Electric bike which is designed and analyzed will be less weight, more strength, highly efficient, Eco-friendly, comfortable, low maintenance, high speed and torque.This project aims at the development of Electric vehicles. This study come up with the development of Electric vehicles.Best alternative fuel vehicle are developed with the best outcome of efficient and good eco-friendly. The future development of our project are undergrown until it fulfills the needs of users.

8. FUTURE SCOPE

This design should be developed more to make the Electric vehicles more efficient and to manufacture this design highly efficient and feasible manner. This electric vehicle should be designed as a hybrid Electric vehicle in future developments.

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