Static and Dynamic Analysis of All Terrain Eco-Green Vehicle (ATV)

Venu Gopal Prathimala, Prudhwee Raju, Roshan Puvvada, Sushanth Varma

Abstract: ***This paper provides an in-detail description of the design considerations, static & dynamic analysis and mathematical data involved in the design of an ELECTRIC MOTOR POWERED ALL TERRAIN VEHICLES (ATV). The main objective of this paper is to reduce the usage of organic fuel powered vehicles and to design a vehicle which works efficiently in the emerging electric vehicle sector. The fast moving metropolitan cities necessitates preplanning of transportation and mobile network. With the advancements in automobile industry, these problems has been tackled to a certain extend but has also brought a concern to vehicular pollution. Air pollution today being a major point of debate persuades engineers and scientists to think of something new and pose a solution to this ever growing issue. Going green seems an only feasible solution to this and that's the reason for choosing the electric motor as the main power source for this all terrain vehicle. In order to maintain the speed levels of the vehicle, seamless decision was made in motor selection.In today's world, Electric cars are gaining a great demand with increasingly new features established in them and rising demand of eco friendly status for each one of us. Electric cars which uses electricity to charge up their batteries; have replaced gasoline and diesel cars with features like high speed, less carbon emission, less maintenance, up to certain level with better mileage [1]. Hence the main focus has been laid on the simplicity of the design, high performance, easy maintenance and safety at a very affordable price. During the entire design process, consumer interest through innovative, inexpensive, and effective methods was always the priority. Most of the components have been chosen based on their easy availability and reliability. According to recognition of customer's need the vehicle is designed to be ergonomic, aerodynamic, highly engineered and easily manufactured. Hence, it makes the vehicle more efficient. This vehicle can navigate through almost all terrains, which ultimately is the main purpose behind the making of any all-terrain vehicle [7]. This report aimed at designing, analysing, fabrication and testing of steering, braking and power transmission for an eco-green all terrain vehicles (ATV) in a nutshell.***

Index Terms**:** ***All Terrain Vehicle, Electric Motor, Power Train, Steering, Braking System, Suspension, Eco friendly.***

Revised Version Manuscript Received on December 27, 2017.

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I. INTRODUCTION

The aim of the study is to determine the best design for the new age of eco green vehicles that would provide maximum efficiency in consideration of fuel utilization and to develop the All **-** Terrain Vehicle. There are many facets to an off-road vehicle, such as the roll cage, drive train, suspension, steering and braking, all of which require thorough design concentration [8].

A software model of the various component of steering, braking is prepared in Solid works software and Catia software. Later the design is tested against all modes of failure by conducting various simulations and stress analysis with the aid of Ansys Software (14). As weight is critical aspect in a vehicle which is powered by a small electric motor, a balance must be found between the strength and weight of the design.

The main objective of this work was to study the steering, braking and power train of an ATV by determining and analysing the dynamics of the vehicle when driving on an off road race track [4]. Though there are many parameters which affect the performance of the ATV, the scope of this work is limited to optimization, determination, design and analysis of steering, braking and power train systems and to integrate them into whole vehicle systems for best results.

II. STEERING SYSTEM

The designing of steering system places a crucial role in designing a vehicle, as it acts as an interface between the vehicle and the driver. The function of the steering system is to is to steer the front wheels in response to driver command inputs in order to provide overall directional control of the vehicle [2]. The driver turns the steering wheel which will rotate the steering column and give further movement in the steering rack. The motion is then transmitted to the wheels by the tie rods. In this design we intend to attain directional control of the vehicle, minimal steering effort to withstand high stress in off terrain conditions, and to provide good response from road to driver. The main consideration in design of the steering system is to produce pure rolling motion of the wheels while manoeuvring the tightest turns on the Dirt road tracks [9].

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A. Selection Criteria:

We prefer Rack and Pinion steering systems over other steering systems due to [3]:

- Low cost.
- Simple construction and working.
- Immediate response.
- Good maneuverability.
- Smooth recovery from turns and
- Minimum transmission of road shocks.

B. Steering Geometry:

The 'Ackermann geometry' is the easiest to implement and has been tested for all vehicles all over the globe and hence it was unanimous choice for the steering geometry over Davis steering geometry due to its ease of manufacture and simple geometry.

With Ackermann steering all four wheels of the vehicle pivot around the same point making sharp turns relatively sy to accomplish. This ensures that the vehicle tires do not slip during turns that are sudden and also assists in pure rolling.

The steering geometry includes [4]:

1. Scrub radius
2. Steering axis inclination
3. Caster angle
4. Camber angle
5. Squirm

C. Scrub radius:

It is the distance between the king pin axis and centre of the contact patch of the wheel, where both would theoretically touch the road when viewed from the front.

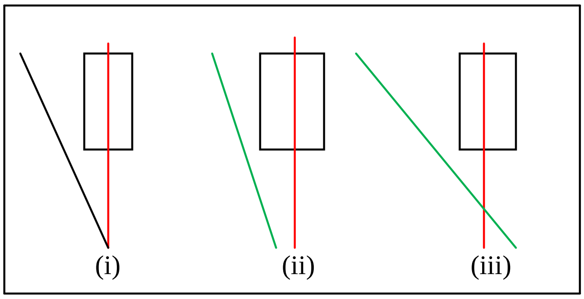

Figure 1: Scrub Radius

The above shown figures represent the zero, positive and negative scrub radius.

For many years, large positive scrub values of 4inches scrub radius were used. The benefit of using a small scrub radius is that the steering turns into less sensitive to braking inputs, in particular.

An advantage of negative scrub radius is that the geometry naturally compensates for split braking, or failure in one of the brake circuits. It also provides centre point steering in the event of tire inflation, which provides greater stability and steering control in this emergency.

D. Steering Axis Inclination:

Steering axis inclination is the angle between the centreline of the steering axis and vertical line from centre contact area of the tire (as viewed from the front). The inclination is normally kept 7-8 degrees [4].

E. Effects of Steering Axis Inclination:

SAI urges the wheels to a straight ahead position after a turns. Less positive caster is needed to maintain directional stability.

F. Caster Angle:

This is the angle between backward or forward tilting of the kingpin from the vertical axis at the top which is about 2 to 4 degrees. The backward tilt is called as positive caster and the forward tilt is called as negative caster.The castor angle has a great impact on the handling characteristics, depending on whether the vehicle is four wheel drive or rear wheel drive the castor angle can be chosen to be positive or negative.

Choosing a zero angle of castor is undesirable since it allows the external vertical forces to travel through one point of contact which introduces the instability in the vehicle.

For the off road vehicles in trailing positive castor angle can be set at the front and rear wheels, as all the ATVs are rear driven.

The +ve castor angle provides self centering force for the steering and makes the car easier to drive in a straight line.

A large castor angle is not recommended since it will make the steering much heavier and less responsive [6].

So incase of off road vehicles where no power steering is available the castor angle should be kept between 0 degrees to +5 degrees

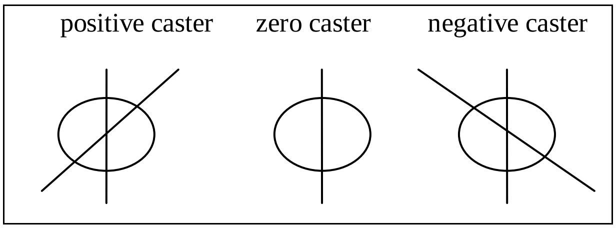

Figure 2: Caster Angles

The above shown figure represents the positive, zero and negative caster angles.

G. Camber Angle:

The angle between wheel axes to the vertical line at the top is called camber angle. It is approximately ½ to 2 degrees. In cars with double wishbone suspension, camber may be fixed or adjustable, but in Macpherson strut suspension it is normally fixed [4].

H. Positive Camber:

- If the top of the wheel is father out than the bottom then it's called +ve camber.
- Excessive +ve camber results in excessive uneven tire wear and impaired handling.
- Reduces the contact patch of the tire.
- Maximizes the amount of forces acting on the tire during cornering, results in undesirable tire wear pattern and affects the handling of the vehicle. To overcome this problem the vehicle is equipped with -ve camber [6].

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I. Negative camber:

- If the bottom of the wheel is father out than the top then it's called -ve camber
- Better cornering characteristics
- Improves grip,
- Reduces lateral load going through the control arms.
- Allows the vehicle to have overseer characteristics, which is desirable.
- This creates a force on the wheels called camber thrust.

NOTE: It is important to set the static camber angle to a small -ve value, in order to maintain the -ve camber throughout.

J. ZERO CAMBER:

The inside edge of the contact patch begins to lift off of the ground is the tyres have 0 camber.

- Reducing the area of contact patch

NOTE: The above condition is only true for the outside tire during the turn, the inside would benefit most from +ve camber.

So based on these considerations we have chosen -ve camber for our vehicle, which give more directional stability and better steer performances.

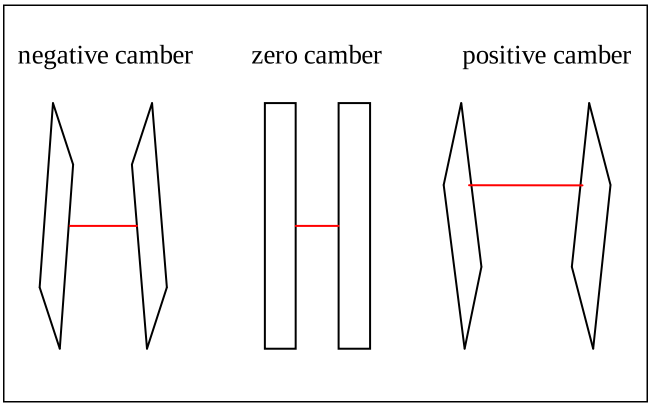

Figure 3: Camber Representation of the Wheels.

K. Squirm:

Squirm occurs when the scrub radius is at zero.

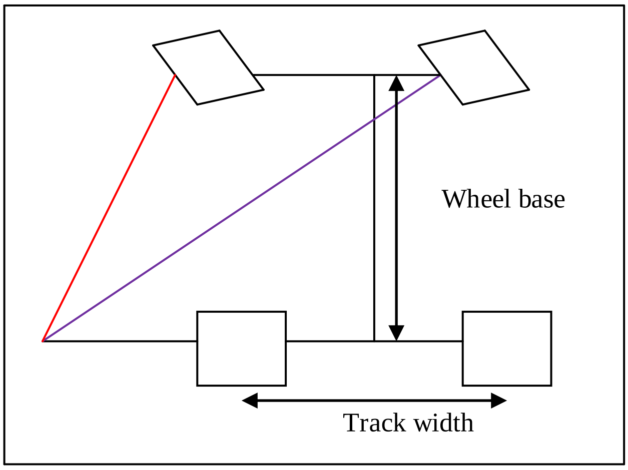

Figure 4: Akerman Mechanism.

Figure 5: Rack and Pinion.

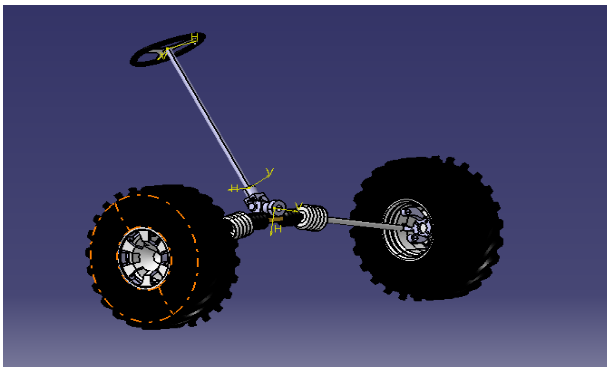

Figure 6: Steering Assembly.

III. STEERING SYSTEM CALCULATIONS

According to Ackerman's geometry, the Condition for perfect steering is given by [2]:

$$\boxed{\cot(\theta_o) - \cot(\theta_i) = \frac{T}{L}}$$

Where

T is track width,

L is wheel base,

θ_o is the outer wheel angle

θ_i is the inner wheel angle.

Substituting T=55" L=64" and θ_i =42.73

We get;

θ_o = 27.435 degrees.

Total angle= 42.73 0 +27.435 = 70.165177

Turning radius is given by the equation

$$\boxed{R_1 = \frac{L}{\mathrm{Tan}(\theta_i)} + \frac{T}{2}}$$

=64/tan (42.73) +54/2

=96.2833

= 2.44mt.

Now the Ackerman angle is given by the formula

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$$\mathrm{Tan}(\varphi) = \frac{L}{\dfrac{L}{\mathrm{Tan}(\theta_o)} - T_{Front}}$$

$$\text{Ackerman angle} = \mathrm{Tan}(\varphi) = \frac{L}{\dfrac{L}{\mathrm{Tan}(\theta_o)} - T}$$

= 16.12 degrees.

$$Ackerman_{Percentage} = \frac{\theta_i}{\varphi} * 100$$

(Or)

$$Ackerman_{Percentage} = \frac{\theta_i - \theta_o}{\theta_i (at 100\% ackerman)}$$

= 99.21%.

$$\text{Steering angle} = \varnothing = a\tan(\frac{L}{R_1})$$

= atan (64/101.98)
= 32.10 degrees.

Steering ratio:

Total angle turned by steering wheel=

$$= \frac{S}{2 * \pi * P_R}$$

Rack travel / 2*pi*pinion radius

= 7.99213/(2pi(1.515748/2)) = 604.210 0
=604.210 0 /69.97 0
=8.63:1.

Off-tracking distance is given by

$$\Delta = R_1 * (1 - \mathrm{Cos}(\frac{L}{R_1}))$$

= 0.01781 cm.

STEERING SPECIFICATION TABLE:

Steering mechanism	Ackerman steering mechanism	
Wheel Base	64 inches	
Track Width	Front track width	54 inches
	Rear track width	52 inches
Inner Angle	42.73 degrees	
Outer Angle	27.435 degrees	
Turning Radius	96.28inch (2.45mt)	
Steering Ratio	8.6:1	

Percentage Ackerman	87.23%
Length of Tie Rod	16 inches
Rack	14 inch
Steering Arm Length	3.48 inch
Ackerman Angle	42.71 deg

IV. BRAKING SYSTEM

The purpose of the braking system is to increase the safety and manoeuvrability of the vehicle by statically and dynamically locking all the four tires on both paved and unpaved surfaces.

The basic function of braking system is to convert the kinetic energy of motion of wheels into another energy source which could be dissipated as heat energy. As the mass and velocity associated with a vehicle increases; its kinetic energy naturally increases.

$$K.E = \frac{1}{2} MV^2 = Q_{Heatenergy}$$

So stopping a heavier vehicle moving at higher speeds require more force out of the brakes and would increase the stopping time. The braking system must be capable of locking all four wheels on a dry surface, improving performance along with reduction in weight, must provide an adequate braking power to the wheels and must have minimum wear and tear; heat loss also must improve reliability.

A. Types of Brakes:

1. Mechanical brakes
   - Disc brake
   - Drum brake
2. Hydraulic brakes
3. Power brakes
   - Air brakes
   - Air hydraulic brakes
   - Electric brakes
   - Vacuum brakes

B. Design:

The braking system is composed of both internal expanding drum brakes and disc brakes. But in case of drum braking there is a high possibility of mud and debris to gather in the space between the shoe and the drum. A similar problem is faced in mechanical disc brakes, but not in hydraulic disc brakes. Hydraulic brakes are found to be suitable for all type of terrain. So we have decided to use hydraulic disc brakes in the front and the rear. We decided to use single master cylinder operating both the front and rear braking systems. The hydraulic system is essentially of diagonal split system

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C. Brake Component Function:

Brake components are mainly divided into four sub-systems:

- Actuation sub system
- Foundation sub system
- Parking brake sub system
- BS & ESP (Electronic stability program sub system).

D. Actuation Sub System:

A. Brake Pedal:

After laying out a proper hydraulic layout, it then is necessary to make the system functioning with brake hoses and brake pedal.

B. Master Cylinder:

A tandem master cylinder or dual master cylinder is one of the most safety devices in a vehicle. It operates a divided or split hydraulic system so that if one circuit fails the other will still operate. Systems can be split so that one circuit is connected to the front brakes and the other to the rear, or diagonally between front left and rear right and vice versa.

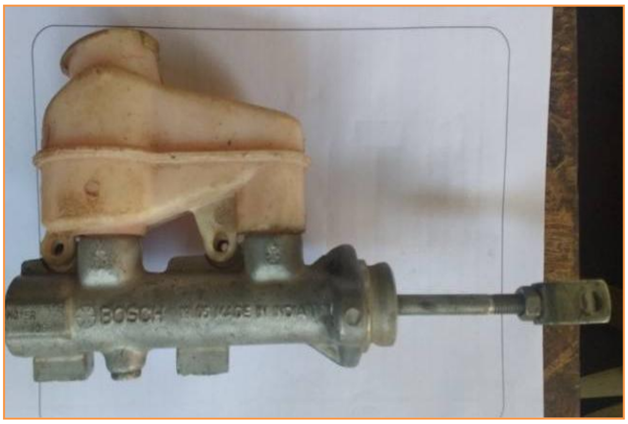

Figure 7: Master Cylinder

C. Proportioning Valves:

Many proportioning valves are integral to the master cylinder housing. This reduces weight and complexity of the hydraulic piping. It provides balanced braking conditions by reducing the hydraulic pressure to the rear wheels.

Figure 8: Proportioning valve.

D. Brake Lines:

Double wall steel tubing is industry standard are the most commonly used brake lines. The standard size is 3/16-inch outer diameter.

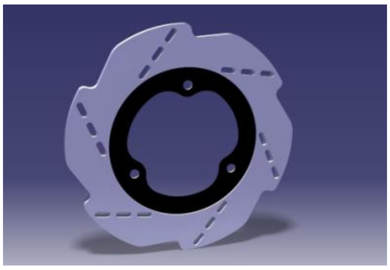

Figure 9: Brake disc.

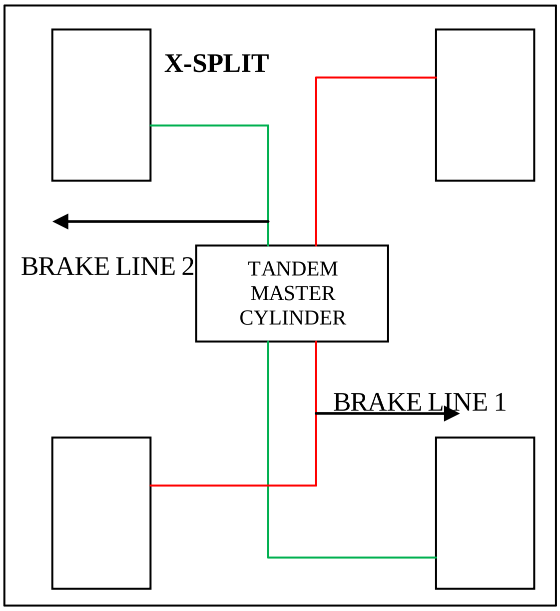

Figure 10: Brake Circuit Diagram.

E. Braking Calculations:

Gross vehicle weight (GVW) = mass of the vehicle * A_g

$GVW = Mass * A_g$	(i)

Where

M is the mass of the vehicle,

A_g is the acceleration due to gravity.

=320*9.81=3136 N.

- **static load s on level ground:**
  When the vehicle sits statically on level ground, the load distribution on front and rear axles are given by [2]:

$W_{fs} = W * (\frac{C}{L})$	(ii)

$W_{rs} = W * (\frac{B}{L})$	(iii)

Where

Wfs is the static weight on front axle,

Wrs is the static weight on the rear wheels,

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- Weight on front axle distribution (W_{fs})

=total weight* (static weight distribution onto front/100)
= (3136*40)/100=1254.4N

- Weight on each front wheel(W_{fw})

$W_{fw} = \frac{W_{fs}}{2}$	(iv)

W_{fw} = 627.2 N.

- weight on rear axle distribution(W_{rs})

= total weight * (static weight distribution onto rear/100)
= (3136*60)/100=1881.6 N

- weight on each rear wheel (W_{rw})

$W_{rw} = \frac{W_{rs}}{2}$	(v)

W_{rw} = 940.8 N.

- To calculate dynamic weight transfer(W_{dt})

$W_{dt} = \frac{W}{g} * \frac{h}{L} * a$	(vi)

- h=centre of gravity = 457.2mm
- L = wheel base of the vehicle
- a = deceleration = µg,

Where
µ = coefficient of static friction between the road and tire =0.6
Thus, a = 0.6*9.81 = 5.886m/s^2.
Now weight transfer is given by

$$W_{dt} = \frac{W}{g} * \frac{h}{L} * a$$

= [(3136/9.81)*(457.2/1625.6)*5.886]

=529.2N.

Weight on the front axle after dynamic weight transfer is given by

$W_{fd} = W_{fs} + W_{dt}$	(vii)

Where
W_{fd} is the weight on front axle after dynamic weight transfer,
W_{fs} is the static weight on the front axle,
W_{dt} is the dynamic weight transfer .
Hence weight on front axle after dynamic weight transfer is
= 1254.4 + 529.2
=1783.6N.

Weight on the rear axle after dynamic weight transfer is given by

$W_{rd} = W_{rs} - W_{dt}$	(viii)

Hence Weight on rear axle after dynamic weight transfer is
= 1881.6 – 529.2
= 1352.4N.

Now the weight on each front wheel after dynamic weight transfer is given by (W_{FD})

$W_{FD} = \frac{W_{fd}}{2}$	(ix)

= 1783.6/2
= 891.8N.

Now the weight on each rear wheel after dynamic weight transfer is given by (W_{RD})

$W_{RD} = \frac{W_{rd}}{2}$	(x)

=1352.4/2
= 676.2N.

Dynamic weight distribution ratio $= W_{FD} : W_{RD}$

= 56.87:43.12.

Now,
Let the max velocity for our vehicle be
45 Kmph =12.5 m/s
Therefore,

$V^2 - U^2 = 2 * a * s$	(xi)

Where
V is the final velocity
U is the initial velocity
a is the acceleration due to gravity
s is the distance travelled.
12.5^2=2*5.886*s
S = 13.33 mt.

Time taken by the vehicle to come to rest state is

$V = U + (a * t)$	(xii)

0=12.5+5.886*t
T = 2.12sec.

The braking force is given by the equation ($B.F_{avg}$)

$B.F_{avg} = \frac{M * U^2}{2 * S}$	(xiii)

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$B.F_{avg}$ = 320*12.5*12.5/(2*13.33)

= 1875.46 N.

Retarding brake force on front wheels is given by

$$B.F_{avg-on-fw} = \frac{M_{fd} * U^2}{2 * S} \tag{xiv}$$

= 181.81*12.5*12.5/2*13.33

= 1065.585 N.

Similarly retarding brake force on the rear wheels is given by

$$B.F_{avg-on-rw} = \frac{M_{rd} * U^2}{2 * S} \tag{xv}$$

= 137.85*12.512.5/2*13.33

= 807.97 N.

Brake torque required on front wheels is given by

$$B.T_{on-fw} = B.F_{on-fw} * R_{fw} \tag{xvi}$$

Where R_{fw} is the radius of front wheels = 23/2 inch

= (1065.585*23*0.0254)/2

= 311.257 N-m.

Brake torque required on the rear wheels is given by

$$B.T_{on-rw} = B.F_{on-rw} * R_{rw} \tag{xvii}$$

= (807.97*23*0.0254)/2

= 232.29 N-m.

But we are using the tandem master cylinder of bore diameter 13.05 mm. Hence the braking force developed by the master cylinder is,

For this let us assume that a force of 200 N is applied by human on the brake pedal in his driving position.

Pedal ratio (r) = 6: 1

Now brake force = (force applied by driver on brake pedal * pedal ratio).

= 200*6

= 1200 N.

This brake force is equally distributed to the rear and front wheels.

Thus the force from the driver side to the wheels is greater than the braking forces developed on the wheels when the vehicle is stopped at a speed of 45 Kmph.

Brake line pressure P = (force on the pedal*pedal ratio/area of master cylinder)

$$BRAKE_{LinePressure} = \frac{F_{Pedal} * r}{A_{Cylinder}} \tag{xviii}$$

Area of master cylinder is given by

$$A_{Cylinder} = \frac{\pi}{4} * d^2$$

= 200*6*4/ (3.14*13.05*13.05)

$BRAKE_{LinePressure}$ = 8.97 N/mm^2.

V. POWER TRANSMISSION

The main objective of the drive train is to transmit power from the motor to the wheels. The velocity, acceleration and torque of the vehicle are all concerned with the power train [7]. An important component that is an integral part of all electric vehicles is the motor. The amount of torque that the driving motor delivers is what plays a decisive role in determining the speed, acceleration and performance of an electric vehicle. The following work aims at simplifying the calculations required to decide the capacity of the motor that should be used to drive a vehicle of particular specifications.

It is the torque that forms the part of the force to drive the wheels and set the vehicle in motion. In simple terms the torque can be defined the turning power of the motor.

When selecting drive motor for the electric vehicle, a number of factors must be taken into account to determine the maximum torque required. These factors are [4]:

1. Rolling resistance
2. Grade resistance
3. Acceleration force

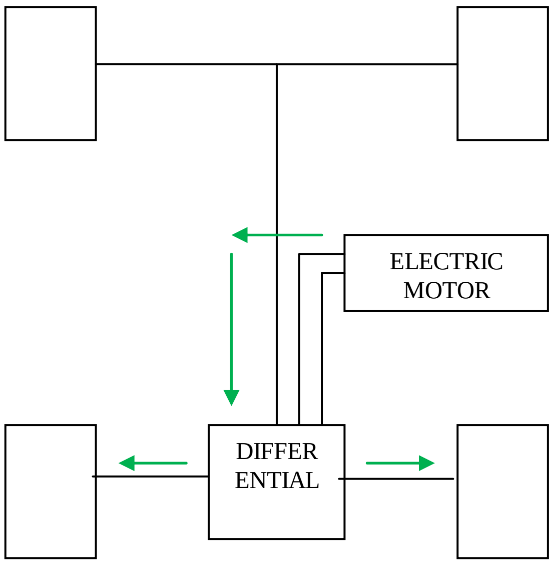

Figure 11: Schematic Representation of Power Transmission

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A. Calculating the Rolling Resistance:

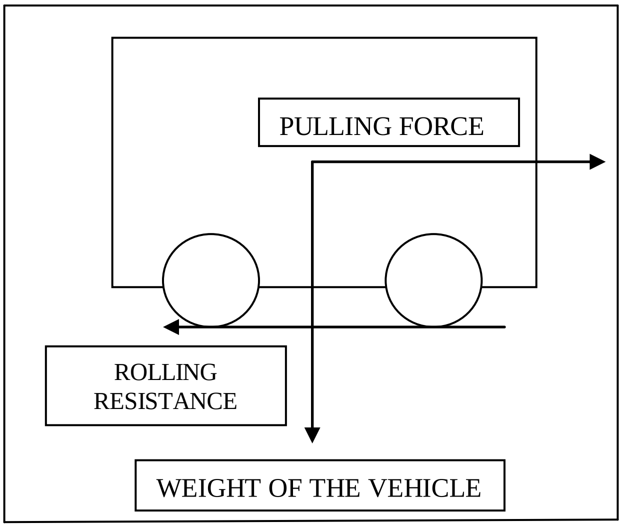
Rolling Resistance is the opposing force that the vehicle has to overcome due to the rolling motion between the wheels and the surface of motion of the vehicle. The rolling resistance depends on the co-efficient of rolling friction which varies depending upon the material of tyres and the roughness of the surface of motion [4].

The Rolling resistance can be calculated as:

$$R.R = GVW * C_{rr} \quad \text{(i)}$$

RR= Rolling Resistance
GVW= Gross Vehicle Weight
Crr = coefficient of Rolling Resistance.

Contact Surface	Crr
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



B. Calculating the Grade Resistance:

Grade resistance is a form of gravitational force that tends to pull the vehicle back when it is climbing an inclined surface. The grade resistance acting on the vehicle can be calculated as:

$$GR = GVW * \mathrm{Sin}(\theta) \quad \text{(ii)}$$

GR= Grade Resistance
ϴ=Grade or inclination angle.

C. Calculating the Acceleration Force:

Acceleration force is the force that helps the vehicle to reach a predefined speed form rest in a specified period of time. The motor torque bears a direct relationship with the acceleration force. Better the torque, lesser the time required by the vehicle to reach a given speed [5].

The acceleration force is a function of the mass of the vehicle. Acceleration force is calculated as:

$$Acceleration_{Force} = M * a \quad \text{(iii)}$$

$$M = \frac{GVW}{g}$$

M=mass of the vehicle
G=acceleration due to gravity
A=required acceleration

D. Finding the Total Tractive Effort (TTE):

The Total Tractive Effort is the total force required to move the vehicle with the desired characteristics and is the sum of the forces calculated in above three sections. Therefore, the Total Tractive Effort can be calculated as:

Total tractive effort

$$TTE = RR + GR + FA \quad \text{(iv)}$$

E. Torque Required on the Drive Wheel

The torque that is required on the drive wheel will be the one that the drive motor requires to produce so as to obtain the desired drive characteristics.

The torque is

$$TORQUE = R_f * TTE * R_w \quad \text{(v)}$$

R_f =Friction factor that account for frictional losses between bearings, axles etc.
TTE=Total tractive effort
R_w =Radius of the wheel

F. Selection of Transmission System:

We are using the differential of gear ratio 10 for our transmission which is directly coupled to the motor. The number of teeth at the motor and the differential may be varied depending upon the required torque and top speed.

Gear ratio=10

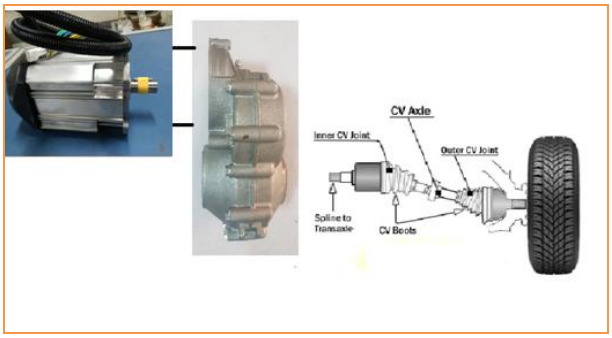


Figure 12: Motor and Differential Assembly.

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G. Power Train Calculations:

1. Part 1(Basic parameters):

Wheel Base (L) = 64inch =1625.6mm
Track width:
Front = 54inch =1371.6mm
Rear=52inch =1320.8mm
Kerb weight of the vehicle = 240Kg (*approx)*
Brushless DC motor = 6Kw (kilowatt)
Battery = Lithium ion battery
Battery discharge rate = 150Amp-h (ampere per hour)
Maximum voltage = 48volts
Maximum allowable Torque = 35N-m
Maximum RPM = 6000RPM
Differential ratio =10.00: 1

2. Part 2(Calculation of vehicle dynamics):

Gross vehicle weight =320Kg
Coefficient of Rolling Friction = 0.015
Coefficient of Static Friction (C_{sf}) = 0.6
Gravity = =9.81m/s^2

$$TractiveForce(FT) = C_{sf} * M * g \quad \text{(vii)}$$

= 0.6*320*9.81
= 1883.52N

$$StartingTorque = FT * R_w \quad \text{(viii)}$$

(Where R_w is the radius of the wheel)
=1883.52*((23*0.0254)/2)
=550.176N-m.

3. Maximum possible Acceleration of the vehicle

A_{max} = (Tractive effort-Tractive force)/Mass of the vehicle

$$A_{max} = \frac{TTE - FT}{M} \quad \text{(ix)}$$

Total Tractive Effort=Rolling Resistance +Grade Resistance +Acceleration Force
Rolling Resistance=Gross Vehicle weight * Coefficient of rolling friction

=320*9.8*0.015=47.88N
Grade Resistance=Gross Vehicle Weight*sin (ϴ)
(ϴ=Gradability Angle)

$$\%Gradability = (\frac{FT}{W} - C_{rr})*100 \quad \text{(x)}$$

= ((1883.52/320*9.81)-0.15)*100
= 45%.

$$GradabilityAngle(\theta) = \mathrm{Tan}^{-1}(\frac{\%Gradability}{100}) \quad \text{(xi)}$$

= tan$^-$(45/100)=24° (degrees).

Grade resistance=320*9.81* sin 24.2 =1286.83N
Acceleration Force = M*a =320*a
Net gear Reduction = (Gear ratio)*(Clutch gear reduction)*(sprocket reduction)*(differential gear ratio)
By neglecting clutch gear ratio, gear ratio and sprocket reduction.
Assuming Transmission efficiency of 85%
Differential ratio=10.00:1
Net gear reduction= 0.85*10 = 8.5

Required Torque = Max Torque of motor*Net Gear reduction
=35*8.5
=297.5 N-m

$$Acceleration(a) = \left[\frac{T}{M * R_w}\right] \quad \text{(xii)}$$

=297.5/ (320*0.292)
= 3.182 m/s^2.

$$Acceleration_{Force} = M * a$$

=320*3.182
=1018.24N.

Total Tractive Effort =47.88+1286.83+1018.24
=2352.95N.
Max possible Acceleration = (tractive effort - tractive force)/Mass.

$$A_{max} = \frac{TTE - FT}{M}$$

= (2352.95-1883.52)/320
= 1.46 m/s^2.

$$\frac{N_1}{N_2} = \frac{T_2}{T_1} \text{ [8]}$$

N_2=6000*(35/297.5) = 706
Considering the Efficiency of Transmission to be 85%
Wheel RPM will be 600 RPM
Now,

$$Velocity = \frac{\pi DN}{60}$$

=3.14*23*0.0254*600/60
=18.34m/s= 66.03 Kmph.
Maximum velocity = 66.03Kmph.
Time required to gain max velocity is

$$V = U + a * t$$

66.03*5/18 =1.47*T
T=12.47 sec

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Distance required gaining max velocity

$$\boxed{v^2 - u^2 = 2 * a * s}$$

18.34^2=2*1.47*S

S=114.40 Meters.

VI. CONCLUSION

The project aimed at designing, analysing fabrication and testing of steering, braking and power train systems for an all terrain vehicle. The primary objective of this project was to identify and determine the design parameters of a vehicle with a proper study of the vehicle dynamics. This project helped us to study and analyse the procedure of vehicle steering, braking and drive train systems and to identify the Performance affecting parameters. It also helped us to understand and overcome the theoretical difficulties of the vehicle design. The entire designing and manufacturing period was a great experience for the entire team as we were introduced into the amazing world of automobile engineering.

The selection of motor to achieve the required starting torque and top speed was a challenging task. Finally an effective design for the ATV is developed which can outperform the existing ATV'S and also in the upcoming era of electric automobile vehicles.

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