17057_AARYANS_Design Report

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1. INTRODUCTION

Drive excellence, efficient dynamics and effective steering performance are the major approaches followed during design.
For ensuring the drive excellence quality of vehicle we focused majorly on human, electric and hybrid drive, selection of proper braking system. For efficient dynamic performance we focused on quality of fabrication, aesthetics, ergonomics overall weight of vehicle, and suspension system. For effective steering performance we focused on the effort and less turning radius. In addition to all above factors we have also given major priority to safety of driver, occupants and vehicle itself in our design.

Specific features of vehicle can be listed as:

1) Aerodynamic design with polymer covering.

2) Ergonomically designed.

3) Employment of light weight system like frame, electrical system, seats, suspensions, in the vehicle.

4) Efficient transmission, braking, steering, system.

5) Innovations like pedal length adjustment without changing the length of chain, toe in and toe out adjustment of the front wheels, RFID locking system, speedometer, etc.

2. SELECTION & DESIGN OF SUB-SYSTEMS

{The detail design procedure and calculations should be show for each sub-system. If there are different options/types available for particular sub-system/s, then selection parameters should also be mentioned.}

2.1. DRIVETRAIN - *HUMAN*

Vehicle has tadpole structure with parallel sitting of drivers. In this Independent peddles (P1 & P2) of 40 teeth are provided to both drivers in order to transmit the power to the intermediate shaft by using chain drive. For normal driver 80 revolutions per minute are considered. The sprocket S1 and S2 are driven by peddle P1 and P2 respectively. Free wheel of 22 teeth is mounted on intermediate shaft to receive power from sprocket S1 and S2.28 teeth chain wheel on the shaft is connected to the 22 teeth sprocket on the left side of rear wheel.

Specifications Of Manual Transmission

1. 2 chain wheel with 40 no of teeth,

2. 2 sprockets (Freewheel) with 22 no of teeth,

3. American standard ANSI series, 11.5 mm pitch chain

4. For intermediate shaft support UCP bearings are used,

5. Mild Steel (Syt = 450 MPa) is use for intermediate shaft,

Calculations for Top Speed :

` :

Assumed paddle speed:	80rpm
Z1	No of teeth paddle chain wheel.(40)
z2	:No of teeth on freewheel(22)
z2'	: no. of teeth on chain wheel of auxiliary shaft (28)
Z3	No of teeth on wheel sprocket.(22)
N1	:Peddling rpm(80)
N2	:Speed of auxiliary shaft
N3	:Speed of rear wheel
d	:Diameter of rear wheel (28 inches = 0.711m)`

The product of speed and no of teeth on sprocket is constant.
Hence, we get two equations:
N1*z1= z2* N2

80*40 = 22*N2

N2= 145.45 rpm

Also,

N2*z2'=N3*z3

145.45*28=N3*22

N3=185.11 rpm

Hence maximum achievable velocity (Vw) = π**d*N3*3.6/60*
= π*185.11*0.711*3.6/60 =25 kmph

Mechanical Efficiency = 85%
25*85/100 = 21.25≈ 22 kmph
Hence
Top Speed with only human drive= 22 kmph

Calculations for Gradability

Human weight= 75 kg

Max Force Applied over pedal =500 N

Torque at sprocket 1= 500 x r = 750 x 0.16 = 85N m

Torque at Intermediate shaft = 85x 22/40=46.75 Nm

Torque at rear wheel = 46.75 x 22/28 = 36.73 N m

For two Drivers Torque = 36.73 x 2 = 73.46 N m

Coeff. of Rolling Resistance (Kr)=0.12

M= 280 kg

So,

Tw= (M x g x sinα + Kr x M) x Rw Therefore

Gradeabilty = α = 3.610 degs.

Acceleration by Human Power:

Torque at rear wheel = 73.46 Nm

Excess torque=73.46-12=61.46 Nm

Radius of rea wheel = 0.356 m

So, Force = Torque / Radius =172.64

a = F/m = 0.6165 m/s2

2.2. DRIVE TRAIN - *ELECTRIC*

Specification of Motor:

- Geared Motor with 6:1 Gear Ratio
- 48 volts and 400 watt
- BLDC type Motor

Battery:

- Battery Runtime: 2.5 Hr.
- 48 volts ,20 Ah.
- Lithium Ion Battery Pack

Controlling Devices:

BLDC controller 48 V control gives signal to windings of BLDC motor, basically converting Dc Power into AC according to position of pole which is determined by Hall Effect Sensor.

Max Speed by motor only:

Max speed of Motor Shaft (N1) = 500 RPM

Max Speed at rear wheel (N3) =225*42/14

=181RPM

=181*2*π*14*.0254*60/1000=**24.3741kmph**

Efficiency of Mechanical Equipment =0.9

Top Speed with only electric drive= 21.93 kmph

Gradeabilty of Electric Drive:

Motor power = 400 watt Speed = 3000 rpm

Torque = T =1.27 Nm

Reduction at Gear box = 6:1

Torque at motor shaft output = T x 6 =7.63Nm

Torque at rear wheel = 7.63*44/16 =21.008 Nm

So,

Tw= (M x g x sinα + Kr x M + Kax A x Vavg2) x Rw

Therefore **Gradeabilty = α = 0.53degres**

Acceleration by Electric Power:

Torque at rear wheel = 21Nm

Excess torque=21-11= 10 Nm

Radius of rea wheel = 0.356 m

So, Force = Torque / Radius = 28.089

a = F/m = 0.100 m/s^2

2.3. DRIVE TRAIN - *HYBRID*

Features:

1)Two sides of rear wheel are connected to human drive and electric drive respectively.

2)In case of failure of one system other system will be intact.

3)Ratchet mechanism at motor side of rear wheel will not increase load on the human drive.

6) 5 speed ratchet at the motor drive will give more speed for only hybrid dive.

Top speed of hybrid drive:

The max speed of the rear wheel will be the maximum of two sides of rear wheel.

As the speed of motor side is more than speed of manual side,max motor speed is max speed of rear wheel.

Motor speed=500 rpm=67.029 kmph

No of teeth on the motor sprocket=16

No of teeth on 1^{st} sprocket =44

No of teeth on 3^{rd} sprocket =22

No of teeth on 5^{th} sprocket=18

Speed on 1^{st} sprocket = 24.37 kmph

Speed on 3^{rd} sprocket =48.74 kmph

Speed on 5^{th} sprocket=59.58 kmph

Efficiency=90%

Maximum speed of hybrid drive=53.62 kmph

Greadeability for Hybrid Drive:

Torque on rear wheel by Human power = 73.76Nm
Torque by Electric Drive = 21 Nm

Total Torque = 94.76Nm

Tw= (M x g x sinα + Kr x M) x Rw

Therefore, **Gradeabilty = α =4.9degs.**

Acceleration by Hybrid Power:

Torque at rear wheel = 94.76 Nm

Excess Torque=94.76-11=83.76Nm

Radius of rea wheel = 0.356 m

So, Force = Torque / Radius = 235.54N

a = F/m **a = 0.84 m/s^2**

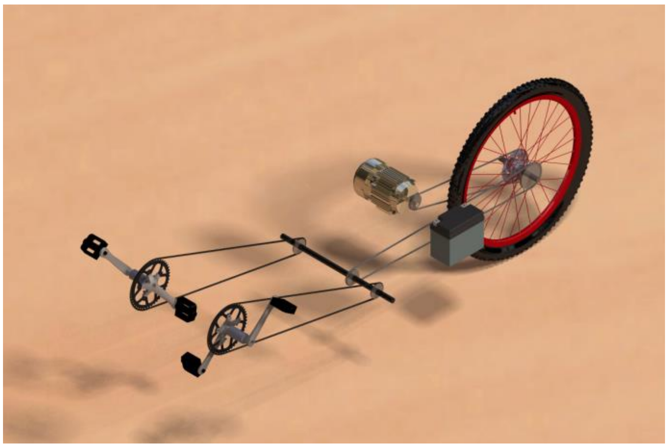


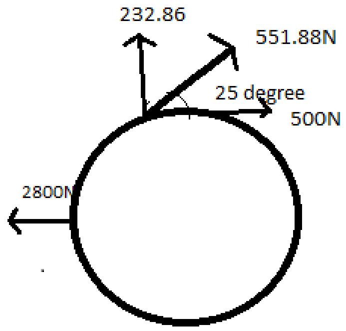
Figure: 2.3.1: Cad Model of Hybrid Drive

2.4. SHAFT DESIGN:

The intermediate shaft is used to transfer power from the paddle chain wheel to the rear wheel by chain sprocket mechanism. The paddle is driven by driver at approx. torque of 30 Nm.

Radius of chain wheel=8 cm
Radius of sprocket on rear wheel=4cm
Tensile force on the chain=30/0.08=375N
Tensile force on slag side=375/$e^{0.24*\pi}$ =176.88N
Total force acting on shaft by front chain=375+176.88=551N
Angle of front chain with horizontal=25^0
Torque on rear wheel=76Nm
Force on tension side=1900N

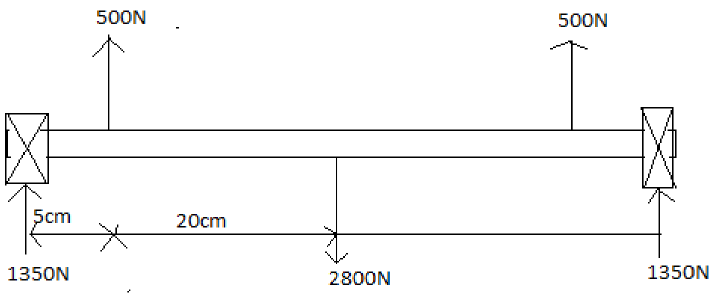
Force on slag side=896.22N
Total force on shaft in backward direction=2796.22N

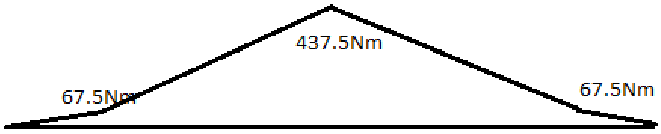


There are both torsional and bending moments acting on the shaft .Also the bending moments are acting in horizontal and vertical direction.

Horizontal Direction:
Due to two different horizontal forces acting on the shaft,The shaft undergoes bending load and the force diagram and bending moment diagram are as shown:

Horzontal forces:



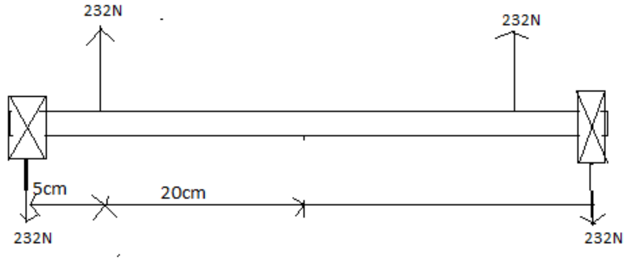


Bending Moment diagram

From the diagram above the maximum bending moment is acting at the center of shaft.
Horizontal bending moment at center=M_{bh}=437.5Nm

Vertical Direction:
In vertical direction only the vertical components of forward chains are acting.Their force and bending moment diagram is as shown:

Vertical Forces:



Bending Moment Diagram

From the bending moment diagram the maximum bending moment in vertical direction is at center.
Vertical Bending moment at center=M_{bv}=11.6Nm

Torsional Moment:

Torque on chain wheel front=30Nm
Torque on shaft sprocket=30*22/40=16.5Nm
Total torque on shaft=2*16.5=33Nm
Torsional Moment on shaft=M_t=33Nm

Design :

The shaft is under combined loading condition.
It is made of mild steel (S_{yt}=650MPa).
Total bending moment=(M_{bh}^2+M_{bv}^2)^(1/2)
M_b =437.6 Nm

By Maximum sheer Stress Theory:

$S_{yt}/2=(16*(M_b^2+M_t^2)^{1/2})/\pi*D^3$

Hence, D=19.01mm
D=20mm

2.5. STEERING

The goal of a steering system is that it should work with minimum applied force and it should be of the low cost, less manufacturability, and low weight.
For this front under seat type of steering configuration is used. This steering uses a tie rod and a ball joint, the links joining the handlebar of the steering and ball joint so that vertical plane motion is converted into the required projections.

Ackerman steering geometry is employed in the vehicle.

Formulae and calculations :

Generally, ratio track arm length to wheel base is 0 .1
Let pin to pin distance =w;
arm length = x;
Ackerman angle =α
Tie rod length = w– (2*x*sin(α))
Here,
w=860mm; and x=120mm; and α=25degree…….(From sketch – Appendix 2)
Tie rod length = 860-(2*120*sin(25)) =758mm

Calculations for inner and outer wheel angles:

Let wheel base = l; turning radius= r;
For inner wheel (θ) =$\tan^{-1} l/r$;
Outer angle (ϕ)=$\tan^{-1}\frac{l}{r+w}$;
By putting values r= 2000mm; l=1270mm; w=860mm;

We get inner angle = 32 degrees;
Outer wheel angle = 24 degrees;
These are extreme values though the turning radius is not 2000mm, it is more than this but we have considered the worst case.

Thus for 2500mm turning radius,
inner angle = 27°; outer angle = 21°;
Now we check for correct steering action by applying fundamental equation of correct steering.0
$\cot\varphi - \cot\theta =$ w/l;
$\cot(24°) - \cot(32°) = 0.6457;$
w/l = 860/1270 = 0.677;
Thus it follows fundamental equation of correct steering.

Critical Velocity to avoid toppling of vehicle:

Limiting case when a vehicle just begins to overturn on 0a plain horizontal road:-

Let, Mg = weight

N = Total normal reaction = $N_1 + N_2$

F = Total friction force

v = velocity

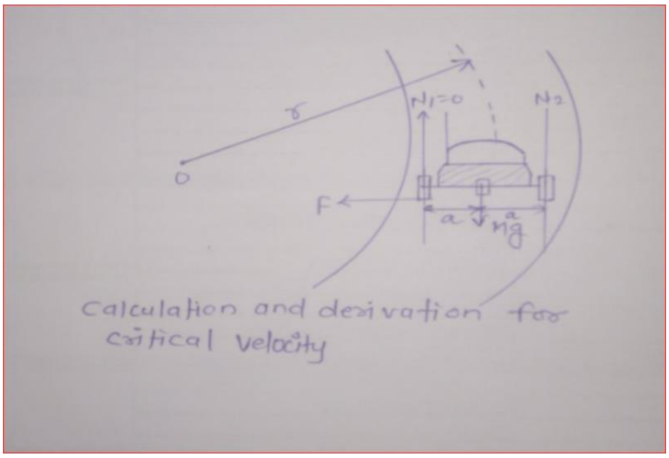
r = min. turning radius,

2a = distance between inner and outer wheels.

G = Center of gravity.

h = height of center of gravity from Earth.

Frictional force F will provide the necessary centripetal force.



Therefore, $F = mv^2/r$ .... (i)

When the vehicle just begins to overturn, the inner wheels will just begin to lift off from the ground. Their pressure on ground will become zero, so the reaction N_1 on the inner wheels will become zero.

Thus, $N_1 = 0$

$N_1 + N_2 = N_2 = mg$ ……………………………….(ii)

Let us take moments about G

$N_2 \times a = F \times h$ ……………………………….(iii)

Putting (i) and (ii) in (iii) gives,

$mg \times a = mv^2/r \times h$,

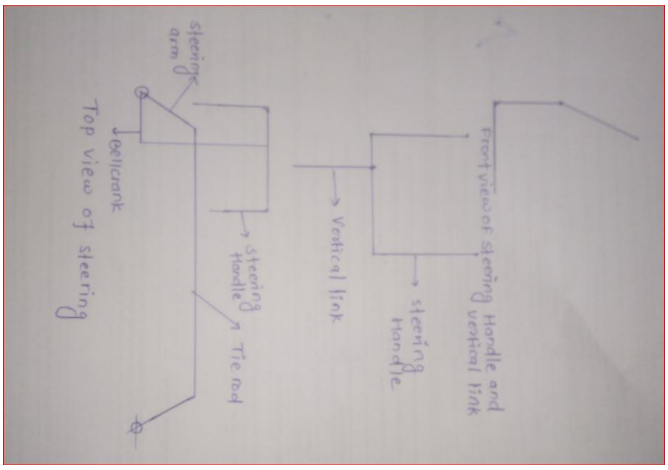
$v_{max} = \sqrt{rga/h}$

Here, r= 2500 mm= 2.5 m, g= 9.81 m/s^2,

2a= 47.5 inches= 1.206 m, h= 17 inches= .432 m

$v_{max} = \sqrt{2.5*9.81*0.603/.43}$ = 21.65 m/sec.

Steering Efforts:



1. **Moment due to vertical loading:**

$M_V = -(F_{21} + F_{2R}) * \sin(\gamma) * d * \sin(\delta) + (F_{21} - F_{2R}) * d * \sin(\Psi) * \cos(\delta)$

F_{21} = vertical load (left wheel) = 613.125 N

F_{2R} = vertical load (right wheel) = 613.125 N

d = lateral offset on ground = 0.1 m

γ = lateral inclination = 3^0

δ = steer angle = 30^0

Ψ = camber angle = 3^0

$M_V = -(613.125 + 613.125) * \sin(3) * 0.1 * \sin(30) + (613.125 - 613.125 * 0.1 * \sin(3) * \cos(30))$

M_V = - 3.209 N-m

2. Moment due to lateral force

$M_L = F_{Y1} + F_{YR} * r * \tan(\Psi)$

r = tyre radius = 0.2794 m

$F_{Y1} = F_{YR}$ = (CF * mass) / 4

CF = cornering force (where according to FSAE standard CF = 1.25 g)

$F_{Y1} = F_{YR}$ = (1.25 * 9.81 * 62.5) / 4 = 191.6 N

M_L = -191.6 + 191.6 * 0.2794 * tan(3)

M_L = - 5.61 N-m

3. Moment due to Tractive force

$M_T = (F_{XL} – F_{XR}) * d$

d = lateral offset distance from ground

F_{XL} = tractive force on left wheel

F_{XR} = tractive force on right wheel

Due to symmetry $F_{XL} = F_{XR}$

So **M_T = 0 N-m**

4. Moment due to application of force by the driver

L_1 = length of steering handle = 0.4 m

L_2 = length of vertical link = 0.25 m

L_3 = length of drag arm = 0.33 m

L_4 = length of bell crank = 0.12 m

F_1 = force on end of the bell crank

F_2 = force on joint between drag arm and vertical link

F_3 = steering effort

Applying momentum conservation about steering axis

Sum of moment due to M_V, M_L, M_T, M_D = 0

Therefore, $M_V + M_L + M_T + M_D = 0$

-3.209 – 5.61 + 0 + (F_1*0.12) = 0

F_1 * 0.12 = 8.819

F_1 = 73.49 N

$F_1 = F_2 * \cos(5)$

F_2 = 73.77 N

$F_2 * 0.25 = F_3 * 0.4$

F_3 = (73.77 * 0.25) / 0.4

F_3 = 46.1 N

Therefore, steering effort is **46.1 N**

Generally, pulling force by hand when area of application is in front of human is 0.1 times of his weight. Here weight of driver is 65 kg therefore he can apply a force of 65 * 9.81 * 0.1 = 63.765 N.

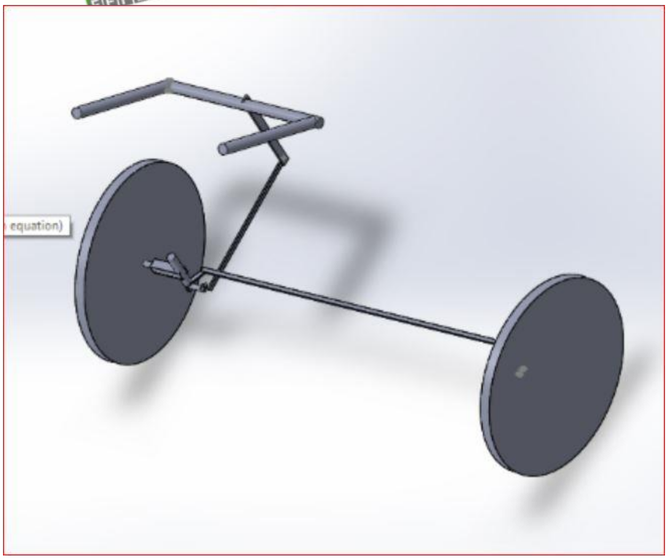


Figure: Steering CAD Model.

2.6. SUSPENSION

REAR SWING ARM SUSPENSION DESIGN:

Weight of vehicle is assumed to be 270 kg = 2648.7 N
Location Of C. G. of Vehicle is as shown in figure 2.5.1 by point **G**

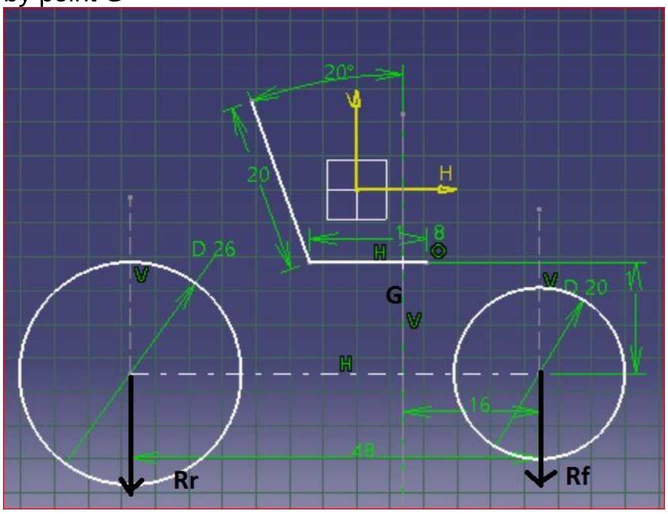


Figure: 2.5.1

Now,
Sum of moments about front wheels centre = 0
Σ M at Rf = 0
Rr * 48 = 2648.7 * 16
Rr = 882.9 N
Therefore, reaction on rear wheel ,Rr= 882.9 N

After fixing the pivot point, suspension geometry was drawn and then spring was designed.

We are using a mono suspension, that is single strut-spring assembly for rear wheel.

Now, Rear wheel reaction = 882.9 N. (At 1G).
This force is acting at wheel center. But as spring is not mounted at wheel center, leverage action will be there and more force will act on spring.

Optimized pivot point is selected for suspension. Also point where suspension is pivoted to frame is decided.

This is shown schematically in below figure.

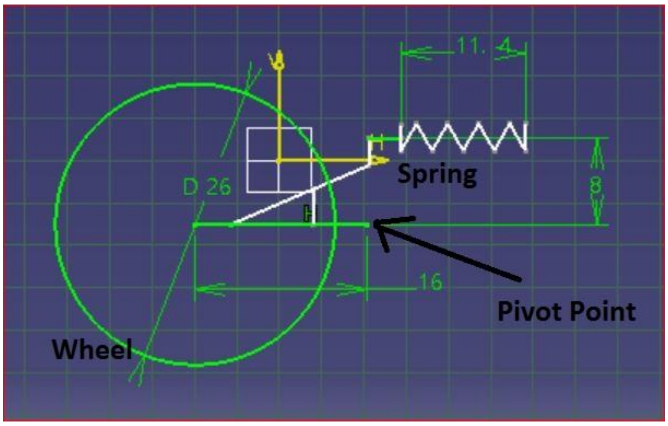


Figure: 2.5.2

Now, moment about the pivot point = 0
F * 8 = 882.9 * 16
Therefore, load on spring, F = 1765.8N
Assume load factor to be 1.5
So, Load on spring considering load factor =1.5*1765.8 **=** 2648.7 N

For rear wheels, deflection absorbed by wheels= 3 cm
P= 2648.7 N, G= 200 GPa for steel.
we consider Active turns (Na)= 10,

$\delta = (8PD^3N_a)/(Gd^4)$
So, we calculate ratio of D^3/d^4
$D^3/d^4 = (0.03*200*10^9) / (8*2648.7*10)$
$= 28315\ m^{-1}$

We have selected mono single strut suspension whose D=66 mm and d=10 mm.
For this;
$D^3/d^4 = (0.066^3 / 0.010^4) = 28749.6\ m^{-1}$.

So, by calculating the value for deflection for this available standard spring, we get,

$\delta = (8*2648.7*0.066^3*10) / (0.010^4*200*10^9) = 3.1$ cm

So this value is considerably near to the calculated one. So, chosen suspension for rear wheel is suitable.

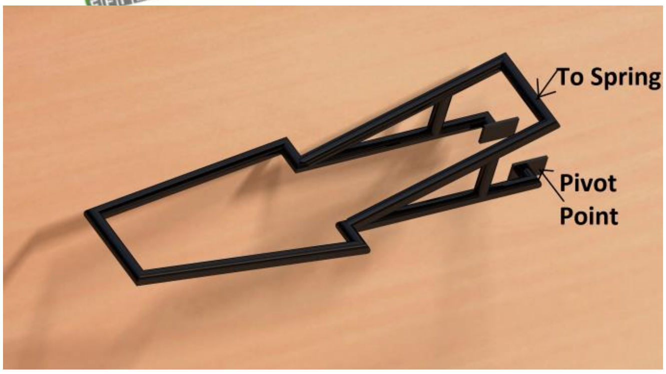


Figure: 2.5.3: Cad Model of Rear Suspension Support

FRONT SUSPENSION DESIGN:

Weight of vehicle is assumed to be 270 kg = 2648.7 N
Location Of C. G. of Vehicle is as shown in figure 2.5.1 by point **G**
Now,
Sum of moments about front wheels centre = 0
Σ M at Rr = 0
Rf * 48 = 2648.7 * 32
Rf = 1765.8 N
Therefore, reaction on front wheel ,Rf= 1765.8 N
So, Each wheel takes 1765.8/2= 882.9 N

After fixing the pivot point, suspension geometry was drawn and then spring was designed.

We are using a mono suspension, that is Mountain Bike Shock Absorber for front wheel.

Now, Front wheel reaction = 882.9 N. (At 1G).
This force is acting at wheel center. But as spring is not mounted at wheel center, leverage action will be there and more force will act on spring.

This is shown schematically in below figure.

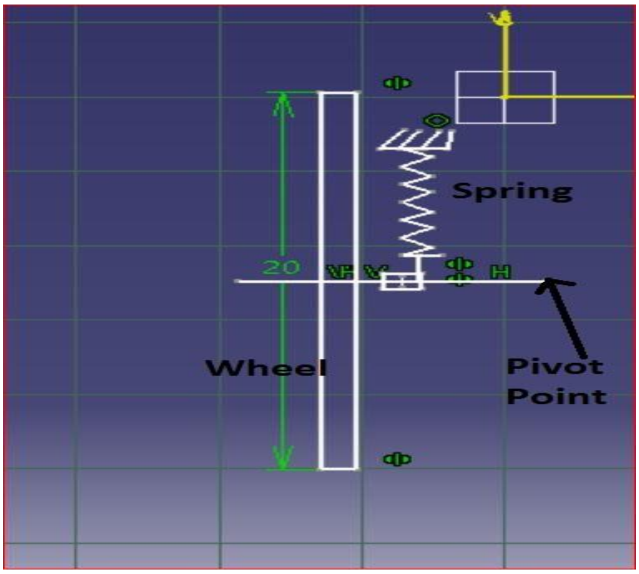


Figure: 2.5.4: Cad Model of Rear Suspension Support

Now, moment about the pivot point = 0
F * 3.5 = 882.9 * 4
Therefore, load on spring, F = 1177.2 N
Assume load factor to be 1.5
So, Load on spring considering load factor =1.5*1177.2
= 1765.8 N

For front wheels, deflection absorbed by wheels= 2 cm
P= 1765.8 N, G= 200 GPa for steel.
Available space is of 15 cm for shock absorber. So to avoid any interference of shock absorber with moving parts, we consider Active turns (Na)= 5.

$\delta = (8PD^3N_a)/(Gd^4)$
So, we calculate ratio of D^3/d^4
$D^3/d^4 = (0.02*200*10^9) / (8*1765.8*5)$
$= 56631.5\ m^{-1}$

We have selected Mountain Bike suspension whose D=50 mm and d=7 mm.
For this;
$D^3/d^4 = (0.050^3 / 0.007^4) = 52061\ m^{-1}$.

So, by calculating the value for deflection for this available standard spring, we get,

$\delta = (8*1765.8*0.05^3*5) / (0.007^4*200*10^9) = 1.84$ cm

So this value is considerably near to the calculated one. So, chosen suspension for front wheels is suitable.

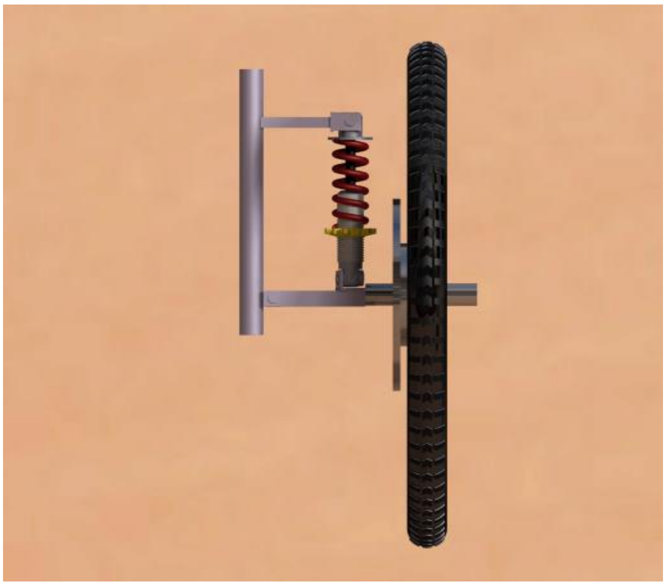


Figure: 2.5.3: Cad Model of Front Suspension Support

2.7. WHEELS & TYRES

Rear wheel have to carry load equal to the 30% of total load. Hence we have decided to keep it more heavy

duty. Available size for heavy duty rim is 640mm so we selected it.
A front wheel carries 70% of the total load and there are two wheels in front to carry this. Hence each wheel carries 35% of the total load. Hence strength of wheel is primary concern. But also width of vehicle depends upon front wheel dimensions. Hence to keep the width minimum and to get the required strength, the front wheel size should be optimum. Hence we have selected 440mm diameter rim for front.

Tire specification:
Front wheel -

- Material – Nylon
- Dimension – 18 in * 2in

Rear wheel:

- Material – Cotton and Nylon
- Dimension – 26 in * 2 in

Wheel specification:
Front wheel:

- Rim diameter – 381 mm

Rear wheel:

- Rim diameter – 585 mm

2.8. BRAKING SYSTEM

1) Mechanical braking system is employed for the vehicle.
2) disc brakes are chosen for two front wheels, and for rear wheel V-Brake is chosen.
3) The control of all the three brakes are given to main driver through levers. The levers apply brakes by actuating brake calipers and torsional springs in V-Brakes.

Initial velocity = 0 m/s
Accelerating distance = 50 m.
Accelerating time = 15 sec.
So the acceleration of vehicle = 0.45 m/s2. ……….(S = u*t + 0.5*a*t2)
Hence the velocity before braking = 6.7 m/s……….
(v2-u2 =2*a*s)
Required braking distance = 4 m.
Mass of vehicle=110 kg.
Mass of two riders = 75 + 75 = 150 kg.
Mass of utility box = 20kg
Total mass of vehicle in loaded condition = 280 kg.
Required retardation = 5.61m/s2 ……………(v2-u2 =2*a*s)

Calculations for front wheel:

Now let us consider that, the front wheel carries the 70% mass of vehicle, hence mass on front wheel is 0.7*267 = 186.9 kg.

Let us assume that the small circle indicates the disc of disc brake and large circle indicates the front wheel.
Radius of disc, r = 0.08 m
Radius of wheel, R = 0.254 m
Braking force = mass * retardation
= 186.9 * 5.61
= 1048.509 N

This force get equally divided into two wheels.
Hence braking force on each wheel = 1048.509/2
=524.2545 N
For easy calculation we consider a single front wheel,
Hence mass on each wheel = 93.45 N
Weight force = 93.45 * 9.81
=916.7445 N
Similarly taking moment about axle to find D,
D = (Fb*0.254) / 0.08
= 1664.508038 N
As weight assist the braking, net braking force is,
Σ Fy= 1664.508038 – 916.7445
= 747.7635375 N
Σ Fx = 524.2545 N
Resultant force applied =√ (Σ Fy)2 + (Σ Fx)2
R = 913.2322206 N
This force is acting at angle α which is given by,
α = tan-1 (Σ Fy/ Σ Fx)
= 54.96583202deg
The net force which causing the stopping of vehicle is given by,
Hence the net force applied = R * sin (54.96583202)
= 747.7635375N
This is the minimum force applied but for our safety we use the two and half of it,
Hence force applied = 1869.408844 N
Torque applied = 1869.408844 * 0.08
= 149.5527075 Nm
Now considering kinetic energy of front wheel we have,
K.E = 0.5 * mass on wheel * (velocity of vehicle)^2 + 0.5 * mass of wheel * [(initial angular velocity)^2
-(final angular velocity)^2]* (radius of gyration)^2=
2097.48525+59.50413=2156.98938J.
Also angular displacement is β = S/R
= 15.7480315 rad
Now we know that,
Torque * angular displacement = K.E
Torque required = 136.9688256 N. m.
Hence torque required is less than torque applied.

Hence design is safe

Specification of Brakes

1) Diameter of disc = 160mm.

2) One disc brake on each front wheel

3) Control- lever operated calliper.

Calculations for rear wheel:

Maximum avg. force can be applied by hand = 70N
Now let us consider that, the rear wheel carries the 30% mass of vehicle,
hence mass on rear wheel is 0.3*267 = 80.1 kg.
Now,
Leverage at brake lever = 0.06m
Leverage at brake pad lever = 0.035m
Coefficient of friction between pad and rim = 0.4
Effectiveness of wire (in %) = 82%
Radius of gyration of rear wheel = 0.3302m
Mass of rear wheel = 3Kg
Change in Kinetic Energy rear wheel = 0.5 * mass of wheel * (radius of gyration)^2 * [(Initial angular velocity)^2 – (final angular velocity)^2] =
$0.5*3*0.3302^2*[(20.29073289)^2 – 0^2] = 67.335J$
Change in Kinetic Energy of vehicle (considering 30% weight on wheel) = 0.5 * mass on wheel * velocity^2 =
$0.5*80.1*6.7^2 = 1797.8445J$
Total Change in Kinetic Energy = 1865.1795J
Angle rotated by rear wheel = Braking Zone/ Radius of rare wheel = 4/0.3302 = 12.11387^o
Braking radius of rear wheel = Radius of rear wheel/2 = 0.3302/2 = 0.1651m
Braking force on rear wheel = T.C.I.K.E/ (2*Coefficient of friction*Braking radius of rear wheel*angle rotated by rear wheel) =1865.1795/ (2*0.4*0.1651*12.11387) = 1165.737188N
Actual breaking force = Braking force on rear wheel/Effectiveness of wire=1165.737188*100/84=1387.782366N0
Force should be applied on brake lever = Actual break force/ (Leverage at brake lever *leverage at brake Pad lever) =1387.782366/ (6*3.5) =**66.08487457N**

Hence Force which should be applied on brake lever is less than Maximum avg. force can be applied by hand.
Hence design is safe.

2.9. SEATS

- Seating configuration - Adjacent seats

- Adjustment of Seats -
  Optimum seat configuration is provided with considering different heights of driver. Removable seats are provided, so as to change the shape and size after the long use of them. Adjustment seats are not employed because they tend to increase vehicle weight with little use. Head restrains, back support and thigh support are made separate so that one can have the adjustment of seat as per need.

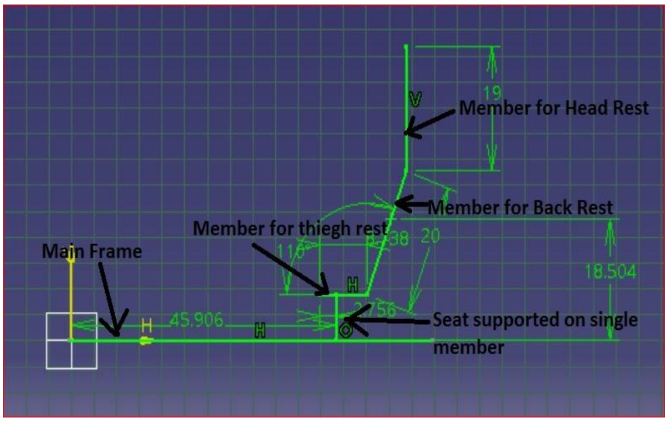
- Head restrictions -
  Head restraints are provided for better comfort of drivers during riding.

- Seat height -
  23 inches (for both riders) from ground.

- Seat back angle – 20 deg.

- Seat design -
  Considering strength and weight of the seat required, seat material is selected. The size of seat and seatback angle is determined by fixing the paddle assembly and changing the position of seats. Depending upon comfort of driver and some theoretical considerations, seat angle and lengths are decided.

- Fabrication -
  Seat is cushioned by using sponge and leather material. Single pipe support is given for each of the three parts of the seats. Sketch of the model provided below with dimensions.



3. SAFETY FEATURES OF VEHICLE

- **Mechanical safety:**

Mechanical safety for drivers consist of safety from collision and safety during all static and dynamic operation of vehicle.

1) To ensure the safety of driver and occupants front rear side impact, and rollover suitable protection members are provided.

2) To provide protection of driver from running chains, chain guard are provided.

3) Intermediate shaft is below driver's seat so it will be safe for drivers.

4) Side protection members are placed in such a way that they prevent body parts to come outside vehicle frame during dynamic condition so it adds to safety of driver and occupants.

5) Fairing is also acts as a safety features of vehicle as it prevents entry of foreign particle to vehicle during dynamic condition.

6) All projected part of vehicle is within vehicle periphery so it adds to safety.

- **Electrical Safety:-**

Electrical safety for drivers consist of safety from overloading of motor during all static and dynamic operation of vehicle.

1) Parking assistance module for safety of driver during parking and during high traffic conditions.

2) Overload detectors which detect overload and alarms driver

3) Fuses and kill switches to kill supply in special cases like overload and some emergency.

- **SECURITY FEATURES OF VEHICLE**

Anti-theft RFID locking system is provided to ensure the security.

- **OTHER FEATURES OF VEHICLE**

Speedometer is provided so that driver can have quick judgement of speed while driving vehicle.

4. ERGONOMICS & COMFORT FEATURES

1) Adjustable paddle positions without changing chain-length, so that leg height problem and paddling force problem is solved.

2) Seatback angle is maintained in such a way that it provides maximum comfort to the driver and sufficient leg room, head room for both drivers.

3) Under seat steering system is used in this vehicle -
   a) Its design makes it easy to master.
   b) Provide comfortable support to the arms.
   c) Gives the rider support during high G turn
   d) Precludes the need of lateral support.
   e) Clearance between driver and co-driver is maintained during all dynamic conditions.

4) Steering handlebar is adjustable so that steering control is effortless and comfortable to the rider.

5) All levers and switches are provided within in reach of driver's and co-driver's hand.

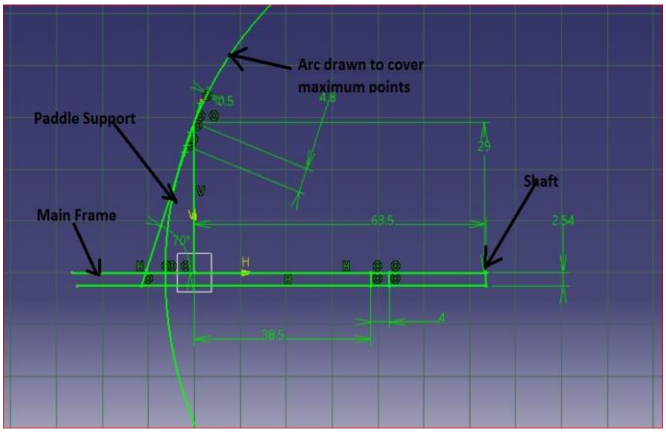
6) Seat geometry is designed so that the rider experiences minimum fatigue due to peddling.

5. INNOVATIONS

Paddle Height Adjustor without change in length of chain: -

While paddling, as the height of the man varies, there will be the change in the requirement of the paddle center to seat distance which will then change the length of the chain. So to avoid all these complications and vehicle be able to drive for people with different height of age group above 18, we have introduced paddle length adjustor without changing the length of the chain between chain wheel and shaft.

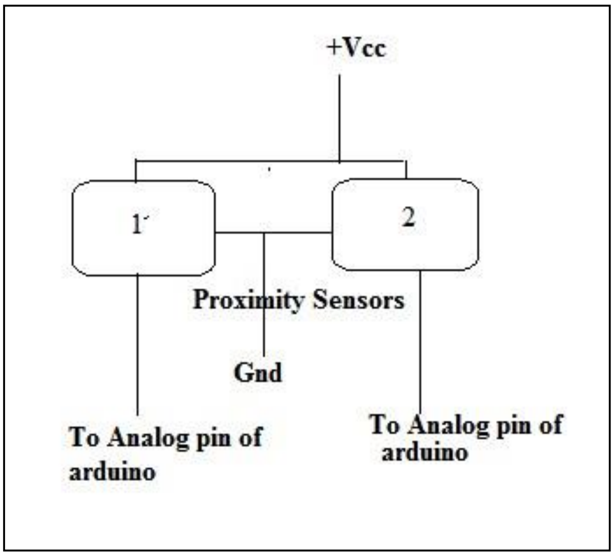
So, to form this we have drawn an arc having center at shaft and this arc passes through maximum extension of paddle points. A line making an angle of 20 degrees with the horizontal main frame base is drawn, so as to pass the paddle support point from maximum points. We have fixed three positions according to different height groups of people. The diagram is drawn below:

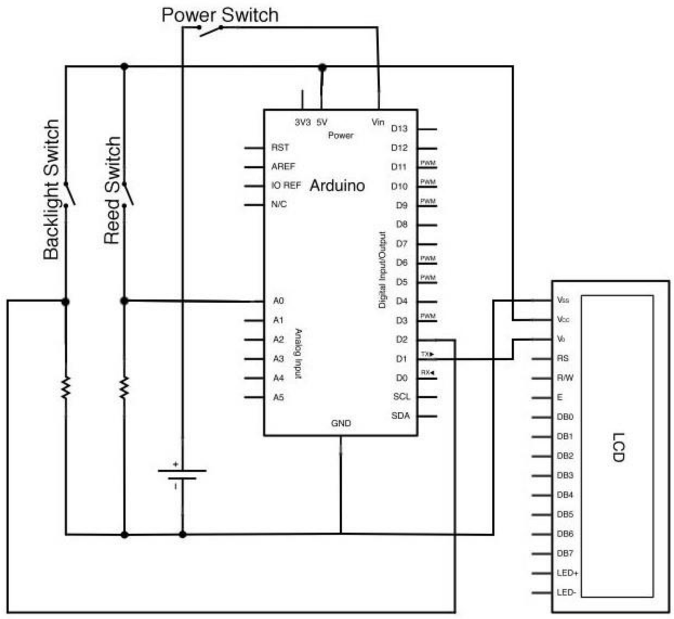


Parking assistance module: -

While parking vehicle it is not easy to look back side as well as controlling whole vehicle, so we came up with parking assistance module which helps driver while parking his vehicle by giving an alarm if a obstacle comes in any of the proximity sensors installed and to understand how far it is we will show the distance on LCD display. But while stuck in a traffic to avoid frontal collisions, proximity sensors are installed in front of vehicle as well with proper alarming system. Every proximity sensor has three pins in which two are supply pins and supply is given +5v from Arduino and its third pin is connected to analog pin of Arduino to display distance on LCD screen.

Schematic diagram: -





Theft Protection: -

It is based on RFID tag recognition. Each RFID tag has a unique identification number which is magnetically engraved on each card. The RFID tag is read by a Arduino microcontroller which has a written code programmed in it. If the card is recognized the controller operates the motor and opens the lock which is fixed at the rear wheel.

If the card is not recognized the motor won't operate and the lock attached to the rear wheel stays fixed.

Digital Speedometer: -

Using reed switch at every cycle it gives a pulse against a counter and it is monitor by the Arduino microcontroller. By taking into account the diameter of the wheel and the time taken by each pulse of reed switch the speed of the vehicle can be calculated.

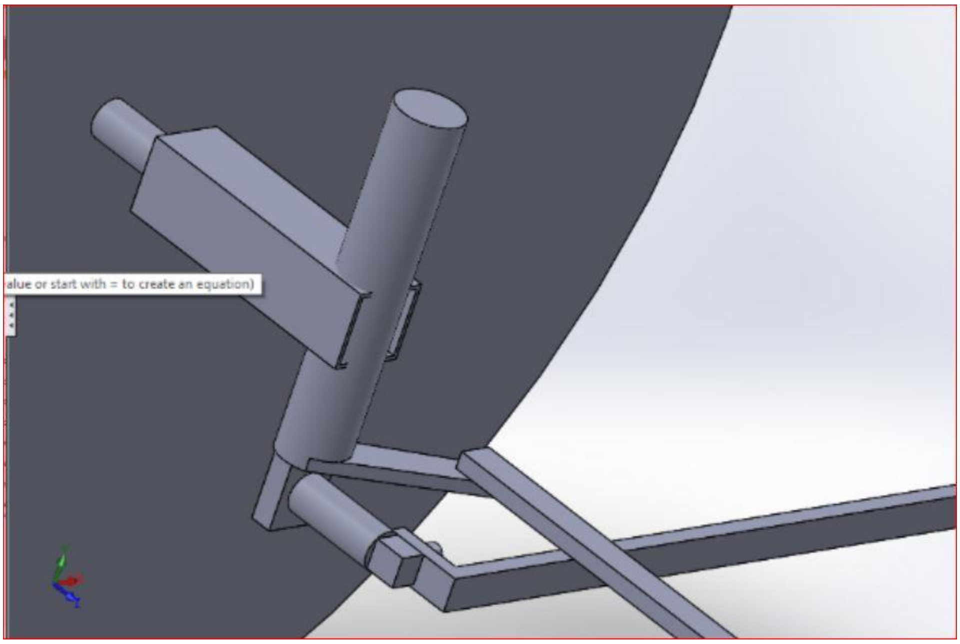
Schematic diagram: -

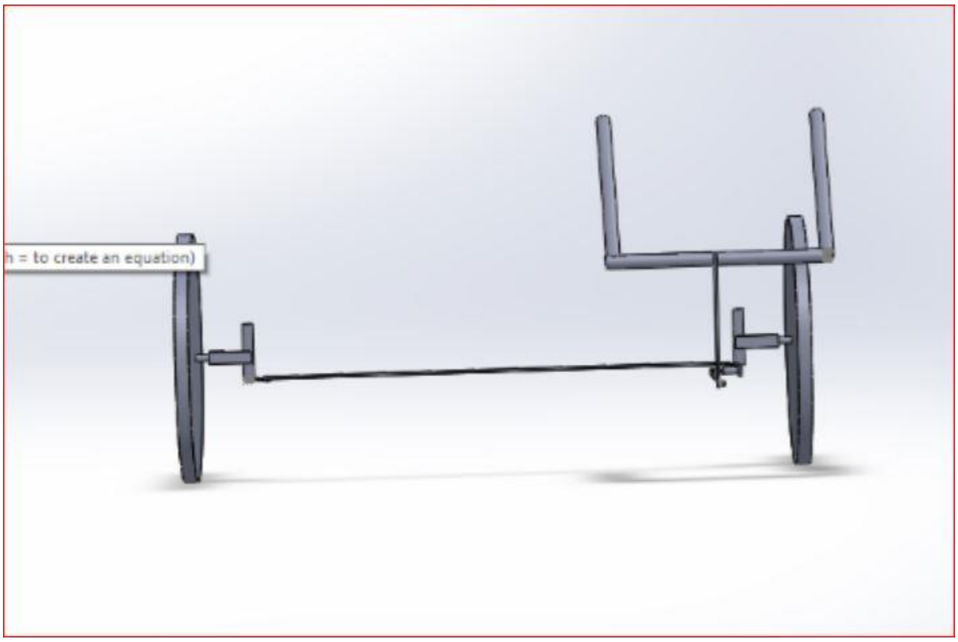
Speed limit indicator : -

It consists of a transmitter and and a receiver. Transmitter transmits the limit of speed of particular area to the receiver on the tricycle. Driver will get an indication of speed limit. If speed of tricycle have crossed maximum speed limit driver supposed to slow down the speed of tricycle.
E.g. When Tricycle is passing through School it is advisable to drive vehicle in certain limit so this indicator will catch attention of Driver towards maximum speed limit.

APPENDIX-1 : PICTORIAL PRESENTATIONS

1. **CAD Model of Steering**





APPENDIX-2 : CALCULATIONS & ANALYSIS

1. **Steering Geometry Determination**